

APLICACIONES INDUSTRIALES DE LA TRANSFERENCIA

DE CALOR

INTRODUCCION

La Transferencia de Calor, como varias otras ramas de la ingeniería, ha sido un campo de estudio que se ha ido extendiendo y no puede ya considerarse como parte de ningún tipo de ingeniería en particular.

Cronológicamente, se puede decir que esta area de estudio se creó como una parte de la ingeniería química, debido a la gran importancia que tienen los fenómenos térmicos en la ingeniería de procesos. Más tarde, al ponerse en uso nuevos tipos de equipo tales como generadores de vapor, sistemas de refrigeración y aire acondicionado, motores de combustión interna, etc., en los que las limitaciones térmicas son de gran importancia, la transferencia de calor pasó a ser una de las disciplinas principales en la educación del ingeniero mecánico.

Aunque a la fecha, casi todos los especialistas en transferencia de calor tienen de una forma u otra relación con la ingeniería mecánica; este campo se ha desarrollado hasta el punto de abarcar una gran cantidad de areas de aplicación. Como ejemplos, además de la ingeniería química y la ingeniería mecánica, se pueden mencionar la industria nuclear, bioquímica, tecnología de alimentos, ingeniería biomédica, etc. Como consecuencia, la Transmisión de Calor se ha convertido en una disciplina interdepartmental que está comprendida y forma parte de varios campos de la ingeniería.

las correlaciones de forma más general y hacer uso de modelos para estudiar fenómenos más complicados.

Finalmente, los procesos de transmisión de calor por radiación se estudiará en general por separado ya que la naturaleza de estos procesos es totalmente distinta de los procesos de conducción y convección. En los procesos de radiación las dificultades fundamentales, desde un punto de vista práctico, son causadas por la sensibilidad del proceso a las condiciones de las superficies radiantes, en el caso de radiación en el vacío, y las propiedades radiativas de los medios transparentes y semi-transparentes en caso de radiación en este tipo de medios.

En resumen, la transferencia de calor se puede clasificar como una rama de estudio que requiere gran cantidad de trabajo experimental y se puede decir también que uno de los requisitos necesarios para el especialista en esta área es la habilidad de simplificar los problemas reales para poder aplicar los conocimientos disponibles hasta la fecha, o bien poder diseñar experimentos que le proporcionen la información necesaria con un esfuerzo razonable.

*En los últimos años, gracias a la presencia de computadores digitales, se ha hecho posible utilizar las ecuaciones fundamentales para obtener soluciones a problemas complicados de mecánica de fluidos y transmisión de calor. Esto tiende a reducir la naturaleza empírica de este campo, aunque todavía es necesario usar conceptos tales como la difusividad de eddy y la distancia característica de turbulencia.

CAMPOS ESPECIALES DE LA TRANSFERENCIA DE CALOR

En procesos de transmisión de calor a fluidos, muchas veces se presenta el caso de cambio de fase ya sea evaporación de un líquido, condensación o solidificación. La presencia de estos fenómenos hace necesario el uso de nuevas técnicas de estudio que no se cubren en los procesos "convencionales."

Los fenómenos de solidificación son de gran interés, entre otros, para la industria metalúrgica, sobre todo para la fabricación de piezas de fundición, ya que una solidificación no controlada puede causar huacos en la pieza final, o bien debido a fenómenos de difusión se pueden causar concentraciones excesivas de impurezas que finalmente pueden hacer que el producto quede defectuoso.

Los procesos de evaporación y condensación presentan características de transmisión de calor bastante distintas de los procesos con una sola fase; lo que hace que tengan una gama adicional de campos de aplicación.

Por una parte, se encuentran equipos industriales, cuya base de operación requiere la evaporación y condensación del fluido principal. Como ejemplos se pueden mencionar las plantas termoeléctricas funcionando con el ciclo Rankine, y plantas de destilación. Para el análisis y diseño de estos equipos es necesario tener un buen conocimiento de las características termo-hidráulicas de los procesos con cambio de fase. Hay que calcular los coeficientes de transmisión de calor para poder estimar las temperaturas de operación y dimensiones de las superficies de calefacción. Hay también que estimar cantidades tales como las caídas de presión en

las tuberías del equipo y analizar la posibilidad de inestabilidades y vibraciones causadas por los flujos de dos fases, que pueden hacer que los equipos no funcionen adecuada y económicamente.

Por otra parte, es deseable hacer uso de los procesos de evaporación y condensación en otro tipo de equipos, debido a los altos coeficientes de transmisión de calor característicos de estos procesos. Entre las aplicaciones prácticas dentro de este campo están la refrigeración de componentes electrónicas y controles de temperatura.

Finalmente, otra rama que puede considerarse como parte de los campos especiales de la transferencia de calor, es la transferencia de masa por difusión. Al analizar los procesos de transferencia de masa se encuentra que las ecuaciones fundamentales que describen los procesos son idénticas a las de transferencia de calor* y de hecho muchas de las correlaciones empíricas son también idénticas siempre y cuando se haga uso de analogías apropiadas.

Entre los campos recientes de aplicación de este tipo de procesos están la ingeniería biomédica, en la que los oxigenadores de sangre (pulmón artificial) y riñones artificiales son equipos básicos de transferencia de masa; y la ingeniería bioquímica en la que los fermentadores para cultivo de levaduras requieren un sistema eficiente para transmisión de oxígeno.

*Siempre y cuando los términos de generación de energía por disipación viscosa no sean importantes y se desprecien en la ecuación de energía. En las ecuaciones de transferencia de masa no hay términos equivalentes.

CAPITULO I

Principios Básicos de Transmisión de Calor

Introducción

La Transmisión de Calor es una parte del campo más general de los fenómenos de transporte. Estos fenómenos se caracterizan por medio de fenómenos de difusión que tienden hacia un estado de equilibrio.

Como procesos de transporte se pueden mencionar además de la transmisión de calor, el flujo de corriente eléctrica y la transferencia de masa, en los que el estado de equilibrio son uniformidad de temperatura, de potencial eléctrico y de concentración respectivamente.

Desde un punto de vista fenomenológico (macroscópico) se puede describir este tipo de procesos como el flujo de una propiedad (energía, masa, cantidad de movimiento) causada por una fuerza motriz o gradiente de potencial. Matemáticamente esto se expresa por medio de la ecuación general de difusión,

$$\frac{R}{A} = -D \frac{\partial P}{\partial y} \quad \dots (1)$$

P Es alguna propiedad por unidad de volumen

(R/A) Es el flujo por unidad de area de la misma propiedad

D Es el coeficiente de difusión

En el caso de transmisión de calor estamos interesados en el flujo de energía. El flujo de energía se puede representar como un flujo de calor (q/A) , y la energía por unidad de volumen sera ρe . La difusividad térmica se representa generalmente por α . Haciendo uso de estas definiciones en la ecuación general de difusión se obtiene:

$$\dot{q}/A = -\alpha \frac{\partial(\rho e)}{\partial y}$$

$$\dot{q}/A = -\alpha \rho C_p \frac{\partial T}{\partial y}$$

Definiendo la difusividad termica $\alpha = \frac{k}{\rho C_p}$
y substituyendo en la ecuacion anterior:

$$\dot{q}/A = -k \frac{\partial T}{\partial y}$$

que es la Ley de Fourier para conducción de calor.

De la misma forma, se pueden generalizar la ecuacion de Newton para viscosidad y la ley de difusión de Fick para expresarlas por medio de la ecuacion general de difusion.

*Se consideró que la densidad del medio es constante.

La siguiente tabla* condensa los resultados de expresar las tres ecuaciones empíricas (Fourier, Newton y Fick) en la forma de la ecuación de difusión.

<u>Proceso de Transporte</u>	<u>Propiedad Transferida</u>	<u>Flujo por Unidad de Área</u>	<u>Difusividad</u>
Masa	C_m	$\dot{W}/A$	D
Energía	Q	$\dot{Q}/A$	α
Cantidad de Movimiento	ρU_x	τ	ν

Hay que hacer notar que en la tabla anterior se consideró que el esfuerzo de fricción τ es equivalente a un flujo de cantidad de movimiento por unidad de área. Esto está de acuerdo con el enfoque clásico de la mecánica de fluidos con la única diferencia de la convención de signos, lo que no afecta ninguno de los principios fundamentales.

Además de la ecuación general de difusión, que caracteriza los fenómenos de transporte, otra ley fundamental es necesaria para describir los fenómenos físicos. Esta ley o hipótesis es la bien conocida ley de Conservación. En Mecánica de Fluidos estamos acostumbrados a usar la ley de conservación de masa para plantear la ecuación de continuidad; en Termodinámica tenemos la primera ley, que se puede considerar como una ley de conservación de energía; y finalmente en Mecánica estamos familiarizados con la conservación de cantidad de movimiento.

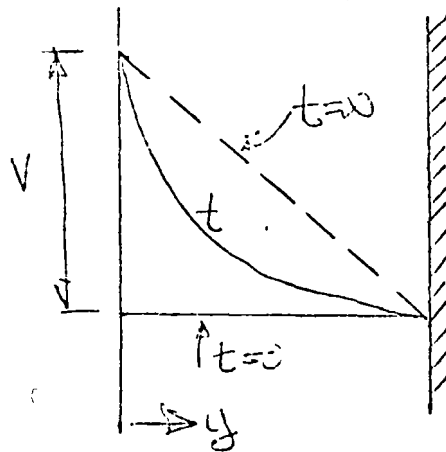
*Tomada de Rohsenow y Choi (2).

De forma general, las leyes de conservación, combinadas con la ecuación de difusión se pueden expresar como :

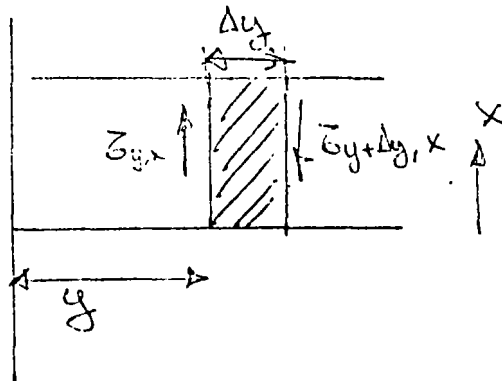
$$\frac{\partial}{\partial y} \left(\rho \frac{\partial P}{\partial y} \right) = \frac{\partial P}{\partial t}$$

La forma de obtener la ecuación de conservación se ilustra a continuación para el caso de la ecuación de cantidad de movimiento. Como ejercicio sería conveniente obtener la ecuación general de conservación para los casos de transferencia de masa y calor.

Consideremos el caso de un fluido comprendido entre dos placas planas. Supongamos que en un tiempo dado una de las placas se pone en movimiento con una velocidad V . El perfil de velocidades a un tiempo t será como el indicado en la figura.



Consideremos un elemento de fluido en una posición (y, x)



El flujo neto de cantidad de movimiento (o la fuerza neta por unidad de area) estará dada

$$A (\tau_{y,x} - \tau_{y+\Delta y, x})$$

o bien

$$- \frac{\partial \tau_{y,x}}{\partial y} A \Delta y$$

Usando ahora el principio de la conservación de cantidad de movimiento, el flujo neto hacia el elemento considerado será igual al aumento de cantidad de movimiento del elemento mismo.

$$\frac{\partial (\rho \bar{u}_x)}{\partial t} A \Delta y$$

o sea

$$- \frac{\partial \tau_{y,x}}{\partial y} = \frac{\partial (\rho \bar{u}_x)}{\partial t}$$

La ecuación general de difusión, aplicada a la transferencia de cantidad de movimiento se puede escribir como:

$$\tau_{y,x} = - \eta \frac{\partial (\rho \bar{u}_x)}{\partial y}$$

$$\frac{\partial}{\partial y} \left(\eta \frac{\partial (\rho \bar{u}_x)}{\partial y} \right) = \frac{\partial (\rho \bar{u}_x)}{\partial t}$$

La similaridad entre los fenómenos de transporte mencionados hasta ahora sugiere que se pueden aplicar los mismos métodos de análisis a los diferentes procesos. Por otra parte, en la mayoría de los casos se tienen varios procesos al mismo tiempo: Transmisión de calor y transmisión de cantidad de movimiento simultáneas; transmisión de calor y transferencia de masa, y finalmente transferencia de cantidad de movimiento y transferencia de masa simultáneas.

Las velocidades de difusión relativas entre los diferentes procesos estarán dadas por las magnitudes relativas de los coeficientes de difusión respectivas. Siguiendo esta línea de razonamiento el cociente de la viscosidad cinemática entre la difusividad térmica nos indicará la velocidad relativo-de propagación de la cantidad de movimiento y energía, que será un parametro de gran importancia en los procesos de convección.

$$Pr = \frac{\nu}{\alpha} = \frac{c \mu}{k} \quad (\text{Número de Prandtl})$$

$$Le = \frac{\alpha}{D} = \frac{k}{\rho c D} \quad (\text{Número de Lewis})$$

$$Sc = \frac{\nu}{D} = \frac{\mu}{\rho D} \quad (\text{Número de Schmidt})$$

Las similitudes y analogías presentadas hasta ahora son válidas en gran cantidad de casos, pero hay que tener cuidado en no extenderlas mas allá de sus límites de aplicación.

Por ejemplo, la ecuación general de difusión dió como resultado la ecuación de Fourier para conducción, pero en el proceso se hicieron varias suposiciones tales como la de densidad constante. Otros

factores tales como la presencia de fuerzas gravitacionales, disipación viscosa de energía, reacciones químicas, etc., introducen términos en las ecuaciones fundamentales que no tienen análogos en otros procesos.

CAPITULO II

Conducción de Calor

El problema de transferencia de calor por conducción es posiblemente de los que más se prestan a un tratamiento analítico. Esto no quiere decir que siempre sea posible obtener soluciones exactas a los problemas que se presentan, y, de hecho, la mayoría de los problemas prácticos en conducción son bastante complicados y nos tenemos que conformar con soluciones numéricas aproximadas. Sin embargo, la presencia de computadoras digitales ha puesto al alcance del ingeniero soluciones que de otra forma habrían requerido un equipo de especialistas.

Las áreas de aplicación de la transmisión de calor por conducción pueden encontrarse en cualquier tipo de industria. Es cierto que en muchos casos los procesos de conducción no tienen importancia decisiva en el diseño de equipos, y lo único que se hace al respecto son unos cálculos sumamente aproximados. Sin embargo, hay áreas específicas en las que unos cálculos más detallados de los procesos de conducción son necesarios.

Como ejemplos se pueden mencionar los cambiadores de calor de superficies extendidas, en las que la conducción de calor a través de las superficies sólidas es una de las limitaciones de diseño. Los procesos de conducción en las varillas de combustible de reactores nucleares tienen que analizarse para determinar la

temperatura máxima de operación del combustible. Finalmente, como otro ejemplo introductorio se puede mencionar el problema de los esfuerzos térmicos, que se pueden presentar en piezas sólidas que operan a temperaturas no uniformes.

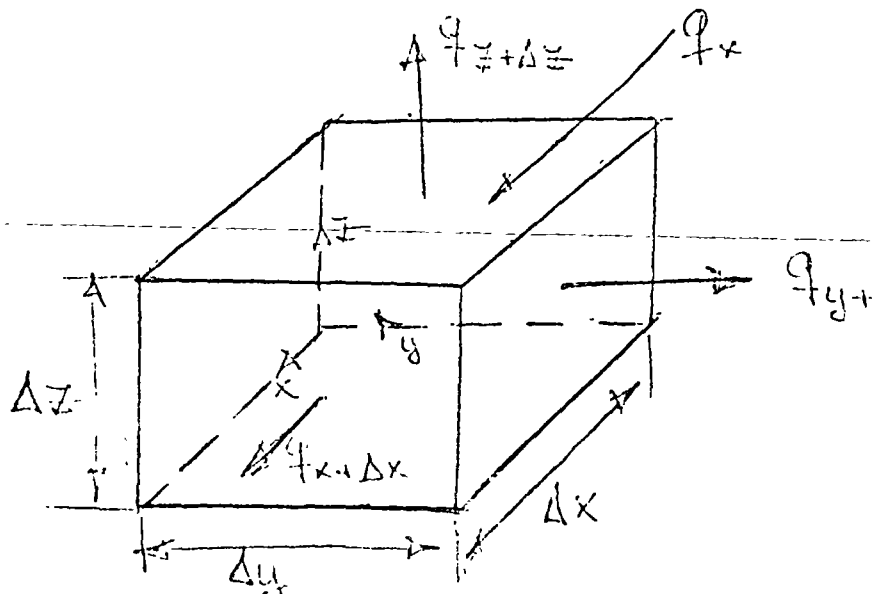
El material que se va a presentar a continuación no es un tratamiento completo de la Transferencia de Calor por Conducción ni mucho menos. Se presentarán las ecuaciones fundamentales y soluciones para algunas casos sencillos de interés. Mencionaremos también los métodos mas usados en aplicaciones prácticas así como algunas soluciones aproximadas que son útiles para cálculos rápidos y estimaciones de las condiciones de operación de algunos tipos de equipo.

1.--Ecuación de Conducción

La ecuación fundamental de conducción es la ley de Fourier

$$q_n = -k A \frac{\partial T}{\partial n} \quad \text{--- (1)}$$

Si consideramos un elemento de material de dimensiones Δx , Δy , Δz como un volumen de control, podemos hacer un balance de energías de la forma siguiente:



$$\begin{aligned} & (\text{Energía que entra}) + (\text{Energía Generada}) - (\text{Energía que sale}) = \\ & = (\text{Aumento de energía almacenada}) \end{aligned}$$

$$\begin{aligned} & (q_x + q_y + q_z) + W_i (\Delta x \Delta y \Delta z) - (q_{x+\Delta x} + q_{y+\Delta y} + q_{z+\Delta z}) = \\ & = \frac{\partial}{\partial t} (\rho e) (\Delta x \Delta y \Delta z) \quad (2) \end{aligned}$$

Considerando que $q_{x+\Delta x} = q_x + \frac{\partial q_x}{\partial x} \Delta x$

$$\begin{aligned} & - \frac{\partial q_x}{\partial x} \Delta x - \frac{\partial q_y}{\partial y} \Delta y - \frac{\partial q_z}{\partial z} \Delta z + W_i \Delta x \Delta y \Delta z = \\ & = \frac{\partial}{\partial t} (\rho e) \Delta x \Delta y \Delta z \quad (3) \end{aligned}$$

Substituyendo ahora la ecuación de Fourier

$$\begin{aligned} & \frac{\partial}{\partial x} \left(k_x \frac{\partial T}{\partial x} \right) \Delta x \Delta y \Delta z + \frac{\partial}{\partial y} \left(k_y \frac{\partial T}{\partial y} \right) \Delta x \Delta y \Delta z + \\ & + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) \Delta x \Delta y \Delta z + W_i \Delta x \Delta y \Delta z = \\ & = \frac{\partial}{\partial t} (\rho e) \Delta x \Delta y \Delta z \quad (4) \end{aligned}$$

Si suponemos que el medio es isotrópico (las propiedades físicas son independientes de la dirección considerada) y además, que la conductividad térmica es constante, la ecuación anterior se reduce a:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{W_i}{k} = \frac{\rho C_p}{k} \frac{\partial T}{\partial t} \quad (5)$$

o bien

$$\nabla^2 T + \frac{W_i}{k} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (6)$$

Para poder obtener una solución a la ecuación anterior, hay que especificar seis condiciones de frontera y una condición inicial.

La condición inicial es en general una distribución de temperaturas determinada

$$[T]_{t=0} = f(x, y, z) \quad (7)$$

Las condiciones de frontera pueden ser de varios tipos. Los más comunes son los siguientes:

- a. El cuerpo está aislado térmicamente en una de sus fronteras.

Digamos en $z=0$

$$\text{Para } z=0 \quad \frac{\partial T}{\partial z} = 0 \quad (8)$$

- b. La temperatura en una de las fronteras es conocida

$$[T]_{z=0} = a \quad \text{ó bien} \quad [T]_{z=0} = f(x, y, t) \quad (9)$$

- c. Una de las superficies del cuerpo está en contacto con un fluido que tiene un coeficiente de transmisión de calor h

$$k \left(\frac{\partial T}{\partial z} \right)_{z=0} = -h (T_{z=0} - T_f) \quad (10)$$

En general los problemas de transmisión de calor por conducción se reducen a resolver la ecuación diferencial anterior con diferentes condiciones de frontera. Las técnicas matemáticas adecuadas dependen del problema en particular y de la naturaleza de las condiciones de frontera. En algunos casos la Transformación de Laplace simplifica el problema, y en otros, métodos tales como el de separación de variables son más convenientes.

En el libro de Carslaw y Jaegggers se presenta un gran número de soluciones conocidas a problemas específicos. Se recomienda que antes de intentar una solución nueva se consulte esta referencia (a menos que a la persona le gusten los ejercicios matemáticos). En lo que se refiere a técnicas de solución, la referencia que se recomienda es el libro por Arpacci mencionado al final del capítulo.

Si alguien tiene interés y se dedica a resolver algunos problemas de conducción, pronto se dará cuenta que aun casos aparentemente sencillos, tales como problemas monodimensionales, se vuelven complicados. En la vida real, la situación puede ser mucho peor, pues en general nos encontramos con cuerpos de formas complicadas que hacen imposible un tratamiento matemático exacto.

Para estos casos, la alternativa disponible es plantear la ecuación de conducción en forma de diferencias finitas y resolverlas

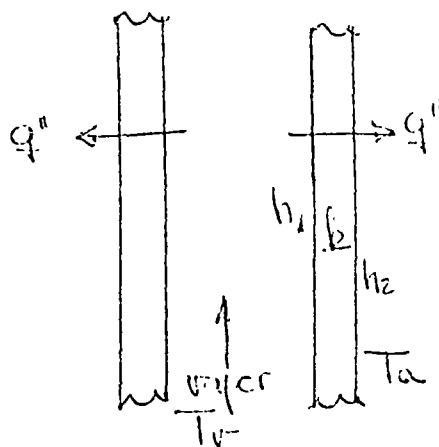
por medio de computadora.

Dejando para el final los métodos de solución numérica, pasemos a analizar algunos casos sencillos de aplicación práctica.

2.--Análisis de Superficies Extendidas

En muchos equipos industriales hay que transferir energía (calor) a medios cuyas propiedades térmicas dejan algo que desear. En estos casos la resistencia total a la transmisión de calor es causada por un bajo coeficiente de transmisión de calor entre una superficie sólida y el medio refrigerante (o calefactor).

Como ejemplo se pueden mencionar los sistemas de calefacción a base de vapor (unos tipos operan no solo por convección, sino también por radiación). En este caso, el coeficiente de transmisión de calor entre el vapor y el tubo es bastante alto, y la resistencia a través de las paredes del tubo es también despreciable.



Sin embargo, el coeficiente de transmisión de calor entre las paredes del tubo y el aire es bajo y de hecho sería el valor que dominaría el proceso.

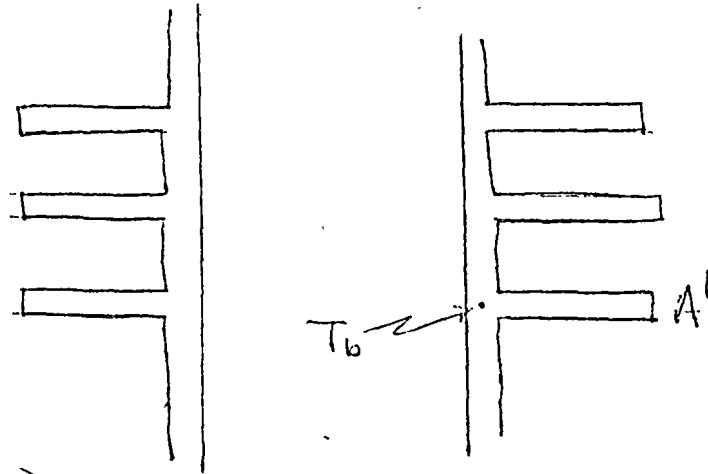
$$\Delta T = T_v - T_a \quad (11)$$

$$q \cong h_2 A \Delta T \quad (12)$$

En general, ΔT está fija (o bien puede variarse dentro de límites restringidos). h_2 podría cambiarse provocando una circulación forzada del aire, pero en general al aumento que puede

*Véase nota en la página siguiente.

lograrse es reducido y tiene limitaciones. La otra alternativa es aumentar el área de la superficie de transmisión de calor.



Si el material sólido tuviera una conductividad infinita, la superficie estaría a una temperatura uniforme y la cantidad de calor transferida sería:

$$q \cong h_2 A' \Delta T \quad (13)$$

Sin embargo, este resultado no es realista pues habría que transmitir el calor a lo largo de la aleta, lo que causará una caída de temperatura. De manera cualitativa se puede representar

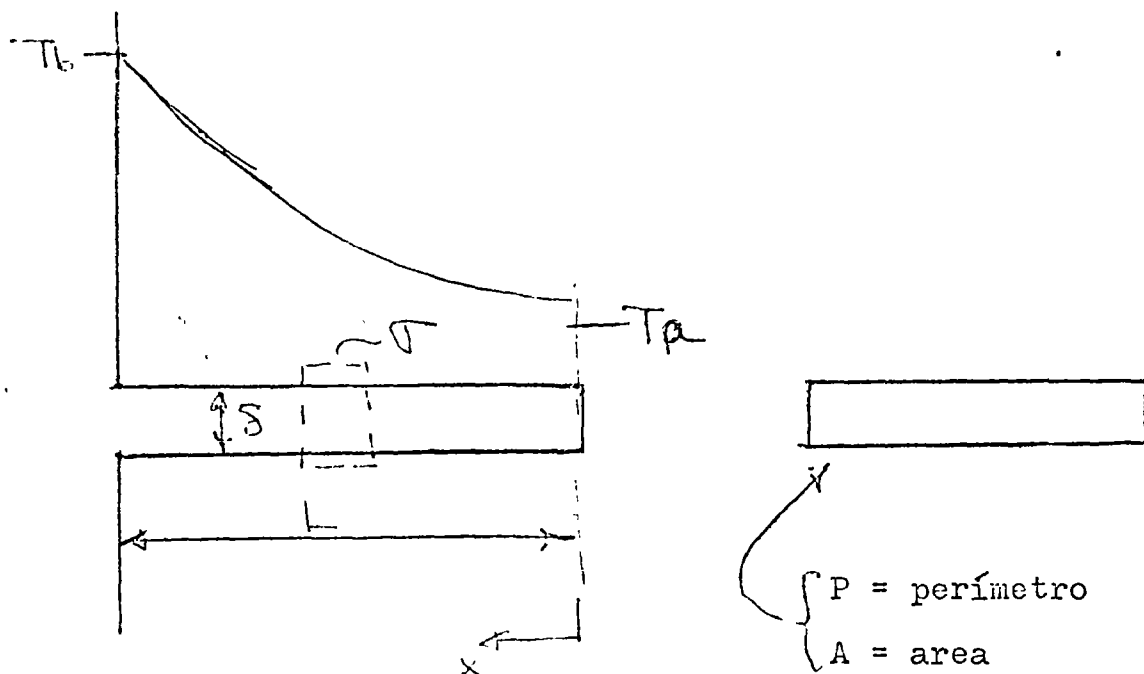
La solución real sería.

$$q = \frac{1}{\frac{1}{h_1} + \frac{x}{k} + \frac{1}{h_2}} A \Delta T ;$$

$$\text{Si } h_1 \rightarrow \infty \text{ y } \frac{x}{k} \rightarrow 0$$

$$q = h_2 A \Delta T$$

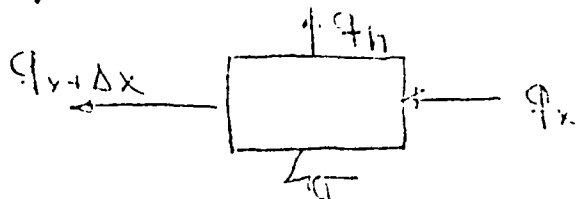
en la figura siguiente:



Si la cantidad $\frac{\delta}{L}$ es pequeña, se puede suponer que la temperatura es solo una función de x . Bajo estas condiciones la ecuación de conducción se puede plantear de forma sencilla y se pueden obtener soluciones aproximadas.

Supongamos además que la transmisión de calor en la punta de la aleta es despreciable y puede considerarse aislada.

Si consideramos el volumen de control δ , indicado en la figura, la ecuación de conservación de energía se puede plantear:



(Energía que entra)-(Energía que sale)=(Aumento de energía almacenada)

$$q_x - q_{x+\Delta x} - q_h = 0 \quad (14)$$

(El aumento de energía almacenada será cero en operación continua)

$$q_x - q_{x+\Delta x} = - \frac{\partial q_x}{\partial x} \Delta x \quad (15)$$

Usando la ley de Fourier

$$- \frac{\partial q_x}{\partial x} \Delta x = - \frac{\partial}{\partial x} \left(- k \frac{\partial T}{\partial x} \right) \Delta x \quad (16)$$

El calor disipado por convección será:

$$q_h = hP \Delta x (T - T_a) \quad (17)$$

Como la única variable independiente es x , las derivadas parciales son derivadas ordinarias.

$$A k \frac{d^2 T}{dx^2} - hP (T - T_a) = 0 \quad (18)$$

$$\frac{d^2 T}{dx^2} = \frac{hP}{kA} (T - T_a) \quad (19)$$

Definiendo $m^2 = \frac{hP}{kA}$ y tomando en cuenta que T_a es una constante se tiene

$$\frac{d^2 T}{dx^2} = \frac{d^2 (T - T_a)}{dx^2}; \quad \theta \triangleq T - T_a; \quad (20)$$

y la ecuación se puede escribir

$$\frac{d^2 \theta}{dx^2} - m^2 \theta = 0 \quad (21)$$

La solución general de esta ecuación es:

$$\Theta(x) = C_1 \cosh(mx) + C_2 \sinh(mx) \quad (22)$$

$$\frac{d\Theta}{dx} = C_1 m \sinh(mx) + C_2 m \cosh(mx) \quad (23)$$

Para $x=0$ $\frac{d\Theta}{dx} = 0 \Rightarrow C_2 = 0$

Para $x=L$ $\Theta = \Theta_b = (T_b - T_a)$

$$\Theta_b = C_1 \cosh(mL) \quad (24)$$

$$\therefore \Theta(x) = \Theta_b \frac{\cosh(mx)}{\cosh(mL)} \quad (25)$$

La cantidad de calor disipada en la aleta puede ahora calcularse de dos formas.

$$q_T = \int_0^L h P (T - T_a) dx \quad (26)$$

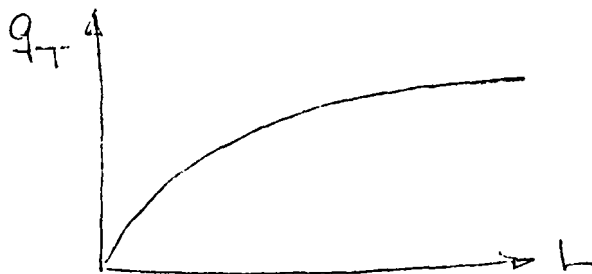
o bien, de manera más sencilla, considerando que el calor disipado tiene que ser transmitido por conducción en la base de la aleta

$$q_T = kA \left(\frac{\partial T}{\partial x} \right)_{x=L} \quad (27)$$

$$q_T = \Theta_b kA m \tanh(mL) \quad (28)$$

$$q_T = \Theta_b (hP kA)^{1/2} \tanh(mL) \quad (29)$$

La ecuación anterior nos daría una gráfica de la forma siguiente



donde claramente se ve la famosa ley económica de los rendimientos de crecientes.

Para terminar con el caso de superficies extendidas mencionemos los conceptos de eficiencia. Se define como la relación de calor transferido por la aleta entre el calor que se transmitiría con un material de conductividad infinita.

$$\eta = \frac{q_b (h P k A)^{1/2} \tanh(mL)}{q_b h P L} \quad (30)$$

$$\eta = \frac{\tanh(mL)}{mL} \quad (31)$$

El ejemplo de superficies extendidas que se presentó ahora ilustra la forma de plantear las ecuaciones de transmisión de calor por conducción y la forma de obtener una solución. Cabe hacer notar que fue posible obtener la solución debido a las simplificaciones que se hicieron al plantear el problema.

Se supuso:

- La temperatura fue función de x solamente.
- La punta de la aleta estaba aislada.

Sin estas suposiciones el problema hubiera sido

bastante más complicado.

Para gente interesada en el diseño de superficies extendidas se les recomiendan las referencias 3, 4 y 5, donde se presentan cálculos de eficiencias para diferentes tipos de aletas. Sin embargo, desde un punto de vista práctico hay que tener en cuenta que un diseño exótico de superficies extendidas nos podrá dar aletas muy eficientes, pero lo más probable es que los costos de manufactura resulten prohibitivos.

3.--Métodos Numéricos Para Solución de Problemas de Transferencia de Calor

El ejemplo anterior sirvió para ilustrar casos en los que es posible integrar la ecuación de conducción con facilidad relativa. Una de las limitaciones más frecuentes al resolver problemas de conducción es la forma misma de los cuerpos involucrados. En casos sencillos tales como cilindros, es suficiente plantear las ecuaciones en coordenadas cilíndricas o esféricas y proceder con la integración. Sin embargo, si el cuerpo a analizar es el aspa de una turbina de gas por ejemplo, lo más probable es que no sea posible encontrar un sistema de coordenadas en el que las condiciones de frontera se puedan representar adecuadamente.

Para todos estos casos, lo más probable es que no sea posible encontrar una solución exacta; o bien puede que, para encontrarla, haya que ser un buen especialista en técnicas de solución de ecuaciones diferenciales. De cualquier forma, el método más usado en ingeniería para estos problemas es el método de diferencias finitas. Hay también otros medios que se mencionarán más adelante.

Método de Diferencias Finitas

El principio fundamental de este método consiste en reemplazar la ecuación diferencial de conducción y las condiciones de frontera por un sistema de ecuaciones algebraicas.

Aceptemos sin demostración que

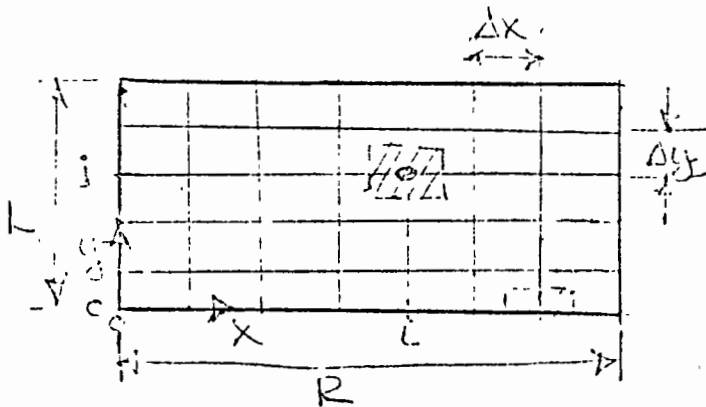
$$\lim_{\Delta y \rightarrow 0} \frac{\Delta T}{\Delta y} = \frac{\partial T}{\partial y} \quad (32)$$

Por consiguiente, la ecuación de Fourier podría aproximarse como:

$$q_n = -kA \frac{\Delta T}{\Delta n} \quad (33)$$

Teniendo en cuenta que la aproximación será mejor a medida que Δn se reduce.

Consideremos ahora un cuerpo rectangular como el indicado en la figura:



Las condiciones de frontera están dadas por

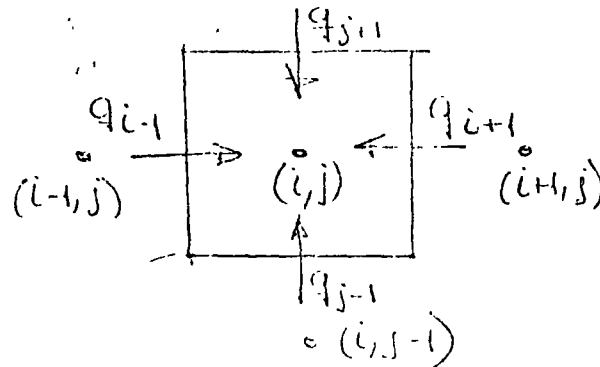
$$\begin{aligned} T &= T_1 & y=0 & \text{ y } y=H \\ T &= T_2 & x=0 & \text{ y } x=L \end{aligned}$$

Dividamos ahora la superficie del cuerpo en una red cuadrículada, y consideremos que cada vértice de la red es el centro de un elemento que se puede caracterizar con una temperatura T_{ij} . Por el momento supongamos que el cuerpo fue de dimensiones tales

que $R/\Delta x = n$ y $L/\Delta y = m$ donde m y n son números enteros.

Observando la figura notamos que hay basicamente dos tipos de elementos (nodos): internos y localizados en la frontera de la superficie.

Consideremos el elemento interno (i, j)



Si hacemos un balance de energías en el elemento (i, j) , para un caso de operación permanente:

$$q_{i-1} + q_{j-1} + q_{i+1} + q_{j+1} + u'' \Delta x \Delta y = 0 \quad (34)$$

(u'' es la generación de energía por unidad de superficie)

Usando la ecuación de Fourier podemos escribir:

$$q_{i-1} = -k \Delta y (T_{i,j} - T_{i-1,j}) / \Delta x \quad (35)$$

$$q_{i+1} = -k \Delta y (T_{i,j} - T_{i+1,j}) / \Delta x \quad (36)$$

$$q_{j-1} = -k \Delta x (T_{i,j} - T_{i,j-1}) / \Delta y \quad (37)$$

$$q_{j+1} = -k \Delta x (T_{i,j} - T_{i,j+1}) / \Delta y \quad (38)$$

Suponiendo que $\Delta x = \Delta y = \Delta l$ y substituyendo estos valores en la ecuación 34 se obtiene:

$$T_{L-1,j} + T_{L,j-1} + T_{L+1,j} + T_{L,j+1} - 4T_{L,j} + \frac{u''(\Delta l)^2}{2} = 0$$

Esta ecuación es válida para:

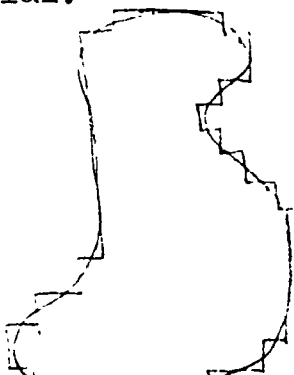
$$\left. \begin{array}{l} 2 \leq L \leq n \\ 2 \leq j \leq m \end{array} \right\} \quad \text{ó sea todos los nodos internos.}$$

Tenemos que escribir ahora las ecuaciones para los nodos situados en las fronteras. Para el caso que estamos analizando, las ecuaciones tomarán la siguiente forma:

$$\begin{array}{ll} T_{1,j} = T(1) & \text{para } j = 1 \rightarrow m+1 \\ T_{n+1,j} = T(1) & \text{para } j = 1 \rightarrow m+1 \\ T_{L,1} = T(2) & \text{para } L = 1 \rightarrow n+1 \\ T_{L,m+1} = T(2) & \text{para } L = 1 \rightarrow n+1 \end{array}$$

Con estas condiciones de frontera tenemos ya un sistema de $(m+1)(n+1)$ ecuaciones simultáneas que podemos resolver por cualquiera de los métodos convencionales. Como ejemplo, se pueden ordenar las ecuaciones en forma de matrices y resolver después el sistema por inversión.

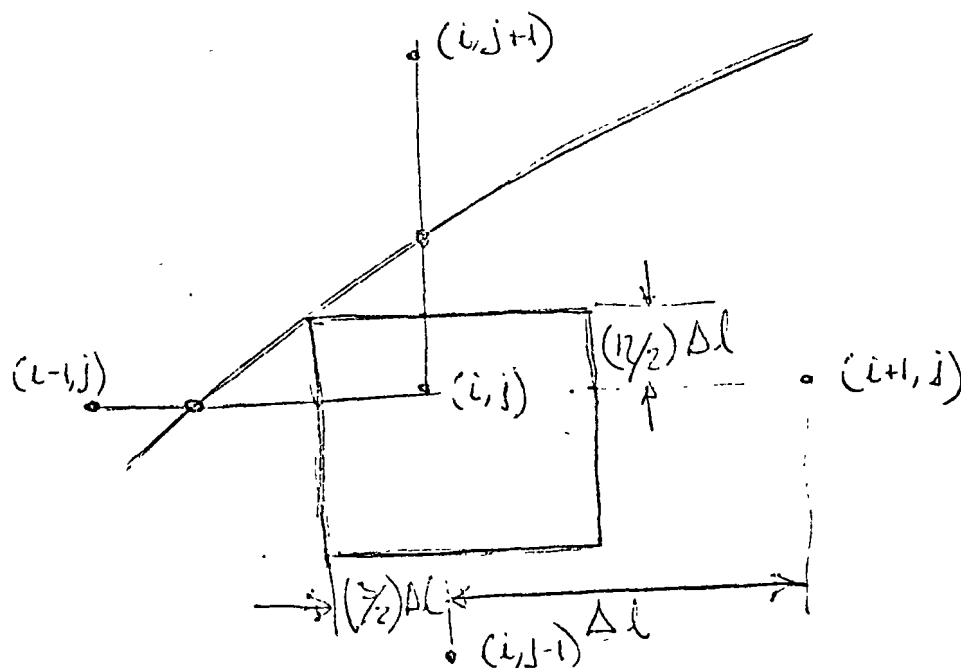
El caso de fronteras de forma arbitraria requiere atención especial.



Una de las formas de tratarlas es substituyendo la forma real de la frontera por una aproximación, de forma que se pueda acomodar en un sistema cartesiano.

Es posible que esta aproximación no sea suficiente, y será entonces necesario plantear las ecuaciones de otra forma.

Consideremos la siguiente figura:



Si se aplica el principio de conservación de energía al elemento mostrado en la figura obtendremos:

$$\frac{2}{1+s} T_{i+1,j} + \frac{2}{1+r} T_{i,j-1} + \frac{2}{s(s+1)} T_{i-s,j} +$$

$$+ \frac{2}{r(1+r)} T_{i,j+1} - 2\left(\frac{1}{r} + \frac{1}{s}\right) T_{i,j} + \frac{u''' (\Delta l)^2}{k} = 0$$

Como puede verse, este caso es algo más complicado que aquel de fronteras regulares, pero la dificultad es mínima si se toma en cuenta que una solución exacta sería imposible.

El tratamiento de fronteras irregulares en problemas de diferencias finitas es un tema que se ha estudiado bastante en el área de métodos numéricos. Para ver métodos aplicables a otros

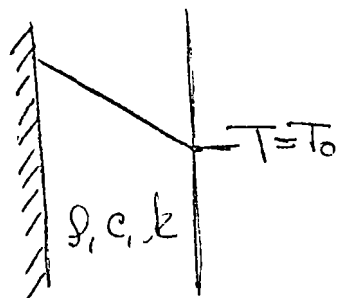
tipos de condiciones de frontera, (en el caso anterior se supuso que las temperaturas en las fronteras eran conocidas) será necesario referirse a ~~un~~ libro sobre métodos numéricos. Una buena referencia para métodos matemáticos aplicados a problemas de ingeniería es el libro de Crandall (6) que también puede usarse como una fuente de referencias para estudios más avanzados.

Con respecto a la exactitud de estos métodos, bastará con decir que la solución se aproximará más a la solución exacta a medida que se disminuye el tamaño de los elementos considerados. Sin embargo, al aumentar el número de elementos considerados, el tiempo de computación necesario aumenta en proporción. Desgraciadamente no hay criterios establecidos para la selección adecuada del número de elementos, ya que dependerá de las condiciones de cada problema en particular. La sugestión es empezar una solución con un número relativamente bajo y compararla con otra solución obtenida con un mayor número de elementos. Si las dos soluciones son cercanas, es indicación de que la subdivisión es adecuada.

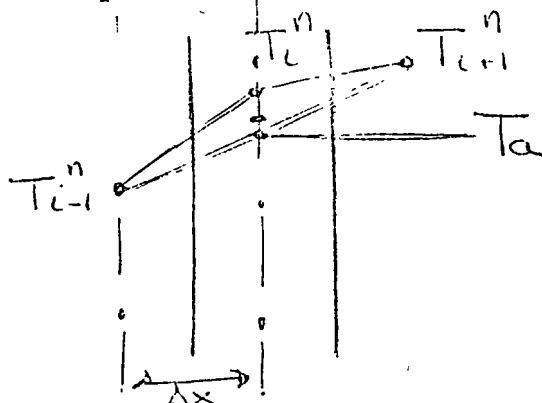
Hay que mencionar algunos problemas especiales que se presentan al resolver problemas de conducción en estado transitorio. El método para plantear la ecuación fundamental en diferencias finitas es el mismo que se usó para el caso de operación constante; pero en este caso tenemos libertad para escoger no solo el número de elementos, sino también el intervalo de tiempo que se va a considerar. Una selección equivocada de las magnitudes de estos intervalos causará inestabilidades numéricas en la solución.

Considérese un caso sencillo. Una placa de material, que en un instante determinado tiene una distribución de temperaturas

como se indica en la figura.



Si dividimos la placa en elementos de espesor Δx , uno de los elementos típicos sería:



Donde la ley de la conservación de energía queda:

$$\rho c \Delta x \frac{T_i^{n+1} - T_i^n}{\Delta t} = k \frac{T_{i-1}^n - T_i^n}{\Delta x} + k \frac{T_{i+1}^n - T_i^n}{\Delta x}$$

$$T_i^{n+1} - T_i^n = \frac{k \Delta t}{\rho c (\Delta x)^2} [T_{i+1}^n - 2T_i^n + T_{i-1}^n]$$

Analicemos un poco más detenidamente la forma en que se planteó la ecuación anterior. Para calcular el cambio de T_i en un intervalo Δt , se supuso que las temperaturas T_{i-1} y T_{i+1} podían considerarse constantes. De acuerdo con esta suposición, el

máximo cambio que podría ocurrir en T_i sería hasta alcanzar el valor indicado por T_a , ya que en este caso, debido a la pendiente constante del perfil de temperaturas, se tendría el mismo flujo de calor entrando y saliendo del volumen de control. Un valor menor de T_a estaría en violación de la segunda ley de Termodinámica.

Volviendo ahora a la ecuación, observamos que para valores de $\frac{k\Delta t}{\rho C (\Delta x)^2} > \frac{1}{2}$, T_i^{n+1} sería menor que T_a . Si se procede adelante con la solución de esta forma, se encontraría que se presentan inestabilidades numéricas, y los resultados calculados oscilarán alrededor de la solución real, con las oscilaciones aumentando a medida que la solución progresa.

Hay demostraciones formales para establecer los criterios de estabilidad para este tipo de problemas. Sin embargo, analizando las ecuaciones resultantes de la forma que se hizo antes, los criterios de estabilidad pueden establecerse.

Un método de plantear las ecuaciones de diferencias finitas que no presenta problemas de estabilidad es el método implícito. Básicamente el método consiste en plantear las ecuaciones en función de las temperaturas a un tiempo $(n+1)$.

$$T_i^{n+1} - T_i^n = \frac{k\Delta t}{\rho C (\Delta x)^2} \left[T_{i+1}^{n+1} - 2T_i^{n+1} + T_{i-1}^{n+1} \right]$$

El principio de formulación es exactamente el mismo que se usó antes. Sin embargo, en este caso solo se conoce T_i^n en cada ecuación, y para encontrar los otros valores (T_e^{n+1}) hay que resolver todo el conjunto de ecuaciones simultáneas. Obsérvese que en el caso anterior, las ecuaciones eran explícitas, y en cada una de ellas se conocían todas las variables menos una.

La decisión sobre cual método usar queda en manos de la persona que va a resolver el problema. El método implícito, definitivamente requiere el uso de una computadora, y aun así se necesita más tiempo de máquina que con el método explícito. Por otra parte, no hay que preocuparse por establecer los criterios de estabilidad.

Con esto terminamos el campo de solución numérica de problemas de conducción. Hay otros métodos para obtener soluciones numéricas, tales como el método de Montecarlo, que son interesantes y en algunos casos muy elegantes. Sin embargo, son más limitados en su campo de aplicación y en general terminan siendo más complicados.

Otros métodos que se usan para solución de problemas de conducción son los métodos gráficos y las analogías eléctricas.

Los métodos gráficos han sido desplazados por las computadoras, y en opinión del autor no presentan ninguna ventaja sobre los métodos numéricos.

Las analogías eléctricas, o método de solución por medio de computadoras analógicas tienen su campo especial de aplicación. El principio básico consiste en plantear las ecuaciones de conducción (~~generalmente en forma de diferencias finitas~~) y diseñar un circuito en el que las corrientes y los voltajes obedecen las mismas ecuaciones que los flujos de calor y las temperaturas.

El problema no es tan complicado como parece, pues en la literatura sobre métodos análogos pueden encontrarse cantidad de circuitos ya diseñados para representar diferentes funciones matemáticas.

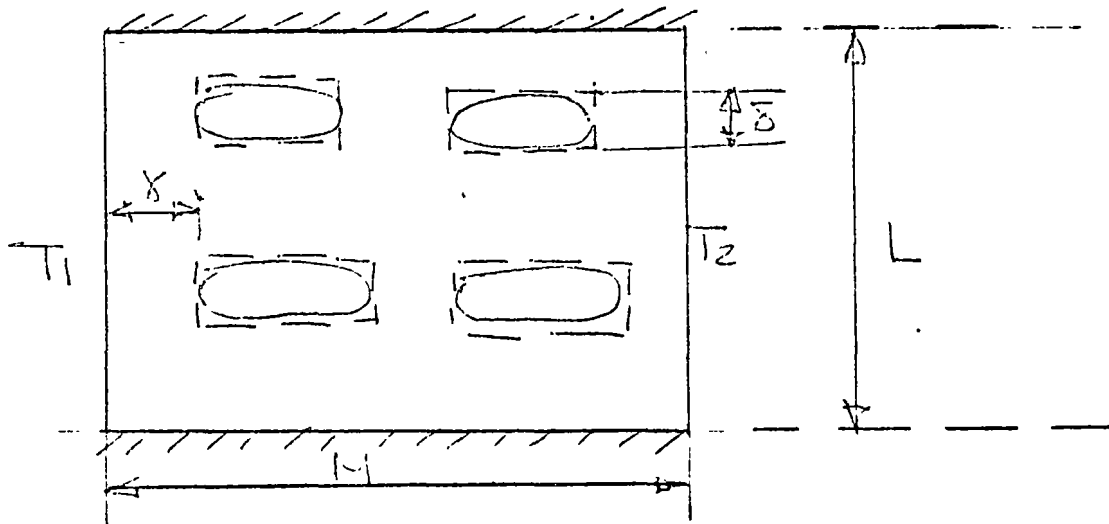
Aunque el diseño y montaje del circuito análogo (que requiere conocimiento de las ecuaciones fundamentales) es más complicado que preparar un programa de computadora para resolver las ecuaciones de diferencias finitas, una vez que se ha construido se pueden realizar cantidad de estudios paramétricos con un esfuerzo adicional mínimo.

4.--Métodos Aproximados Usados en Transferencia de Calor por Conducción

En algunos casos prácticos no es necesario obtener una solución exacta a un problema de conducción y basta con obtener una idea aproximada de las condiciones de operación. En todos estos métodos es necesario hacer suposiciones bastante radicales con respecto al comportamiento de los sistemas y la exactitud o validez de los resultados dependerá en las suposiciones mismas.

Mencionaremos ^{uno} ~~dos~~ de los métodos más comunes y sencillos.

Consiste en dividir los cuerpos en diferentes secciones, dentro de los cuales la ecuación de conducción puede escribirse en forma algebraica. Ilustremos el método por medio de algunos ejemplos.



Se quiere determinar el flujo de calor que ocurrirá debido a la diferencia de temperaturas $(T_1 - T_2)$. La solución exacta no parece sencilla debido a las irregularidades internas del cuerpo.

Podemos aproximar la forma de los huecos por medio de rectángulos. A continuación procedemos a evaluar el flujo de

calor por dos formas.

Consideremos primero que el flujo de calor en la dirección vertical es despreciable, y por consiguiente el material comprendido entre los huecos no participa en la conducción de calor.

El flujo de calor puede entonces calcularse como:

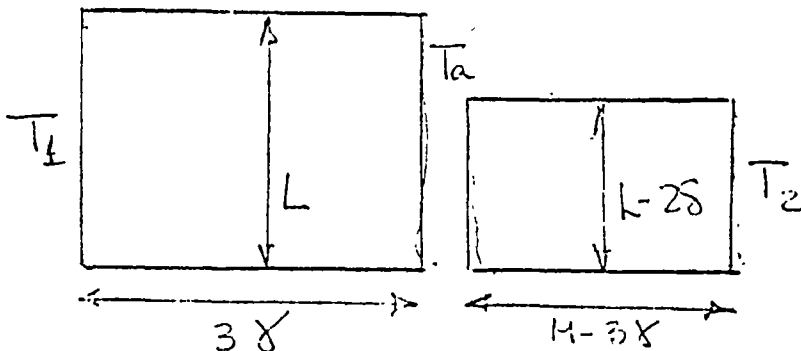
$$Q_{1-2} = k(l-2s) \frac{T_1 - T_2}{M} e$$

Donde e es el espesor del material.

Analizando las suposiciones hechas nos damos cuenta que el flujo de calor calculado será menor que el flujo real, ya que la resistencia a la transmisión de calor en las partes donde no hay huecos será menor que la que usamos en la solución. El valor Q_{1-2} puede considerarse entonces como el límite inferior del problema.

Con igual facilidad podemos determinar ahora un límite superior. Para esto, supongamos que la conducción en la dirección vertical es infinita (esto equivale a suponer que la temperatura es únicamente función de la posición en el sentido horizontal).

Para resolver el problema de esta forma podemos agrupar todas las secciones sólidas (sin huecos) en un bloque y formar otro bloque de menores dimensiones con el material restante:



El flujo de calor pasa por los dos bloques y por consiguiente podemos plantear las ecuaciones de la siguiente forma.

$$Q_{1-2} = k(1-28) \frac{T_1 - T_2}{14-28}$$

$$Q_{1-2} = k l \frac{T_1 - T_a}{28}$$

Entre las dos ecuaciones podemos eliminar T_a y obtener la solución para el valor Q_{1-2} .

BIBLIOGRAFIA , INTRODUCCION Y CAPITULO I

Lista de Textos Básicos de transferencia de calor y flujo de dos fases.

- 1.- Kreith , Principles of Heat Transfer, International Textbook Company. 1965.
- 2.- Rohsenow, W.M.; Choi H. "Heat, Mass and Momentum Transfer". Prentice Hall
- 3.- Rohsenow, W.M. editor "Developments in Heat Transfer" the MIT Press.
- 4.- Tong, L.S. "Boiling Heat Transfer and Two Phase Flow". Wiley
- 5.- Wallis, G.B. "One Dimensional Two Phase Flow". Mc Graw-Hill.

Textos Recomendados

- 1.- Arpaci, V "Conduction Heat Transfer". Addison Wesley
- 2.- Carslaw, H.S. Jaeger, J. C. "Conduction of Heat in -- Solids". Oxford: Clarendon Press.
- 3.- Schmidt, E. "Die Wärmeübertragung durch Rippen". Zeitschr. VDF, 70, 885.
- 4.- Harper, W.B. y Brown, D.R. "Mathematical Equations for Heat Conduction in the Fins of Air Cooled Engines". Reporte NACA, # 158,679 (1922).
- 5.- Gardner, K. A., "Efficiency of Extended Surfaces". Trans, ASME, 67, 621. (1945).
- 6.- Crandall, S. H. "Engineering Analysis" Mc Graw Hill.
- 7.- Haji - Sheikh, A.; Sparrow, E. M. "The Solution of Heat Conduction Problems by Probability Methods". Journal of Heat Transfer. Mayo 1967.

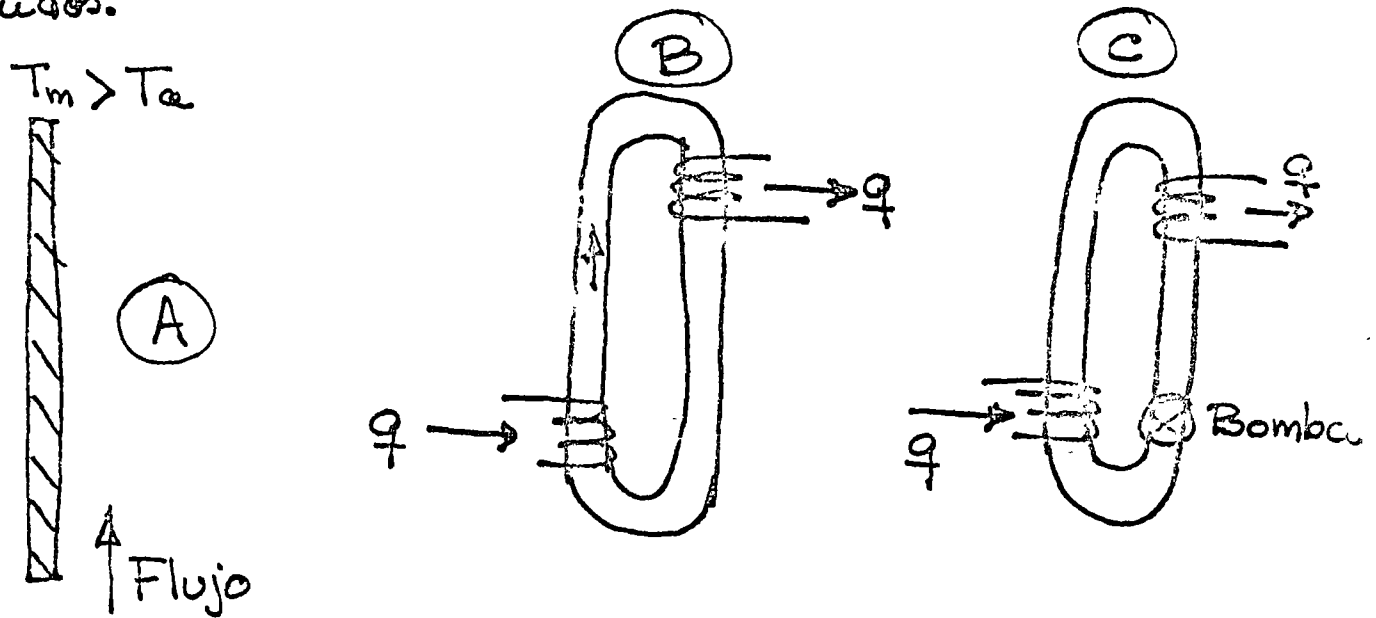
Transferencia de Calor por Convección

Por convección se entiende la transmisión de calor entre una superficie sólida y un fluido en movimiento. Estrictamente hablando, la convección no es una forma independiente de transmisión de calor, ya que la energía es transferida por un proceso de conducción. Sin embargo, el movimiento mismo del fluido alrededor de la superficie sólida, requiere, para la solución completa de los problemas, la combinación de las ecuaciones de mecánica de fluidos con las ecuaciones de conducción.

En aplicaciones industriales los procesos de convección tienen una gran importancia. Los fluidos se pueden usar para transportar energía térmica de un lugar a otro, en ingeniería de procesos o ingeniería química es a veces necesario calentar o enfriar los fluidos para poder llevar a cabo los procesos, etc. Los campos de aplicación son tan extensos, que la convección en general se estudia por separado, como una rama especial de la ingeniería de procesos.

Tipos de Convección

Los procesos de convección se clasifican de acuerdo a las fuerzas que causan el movimiento de los fluidos.



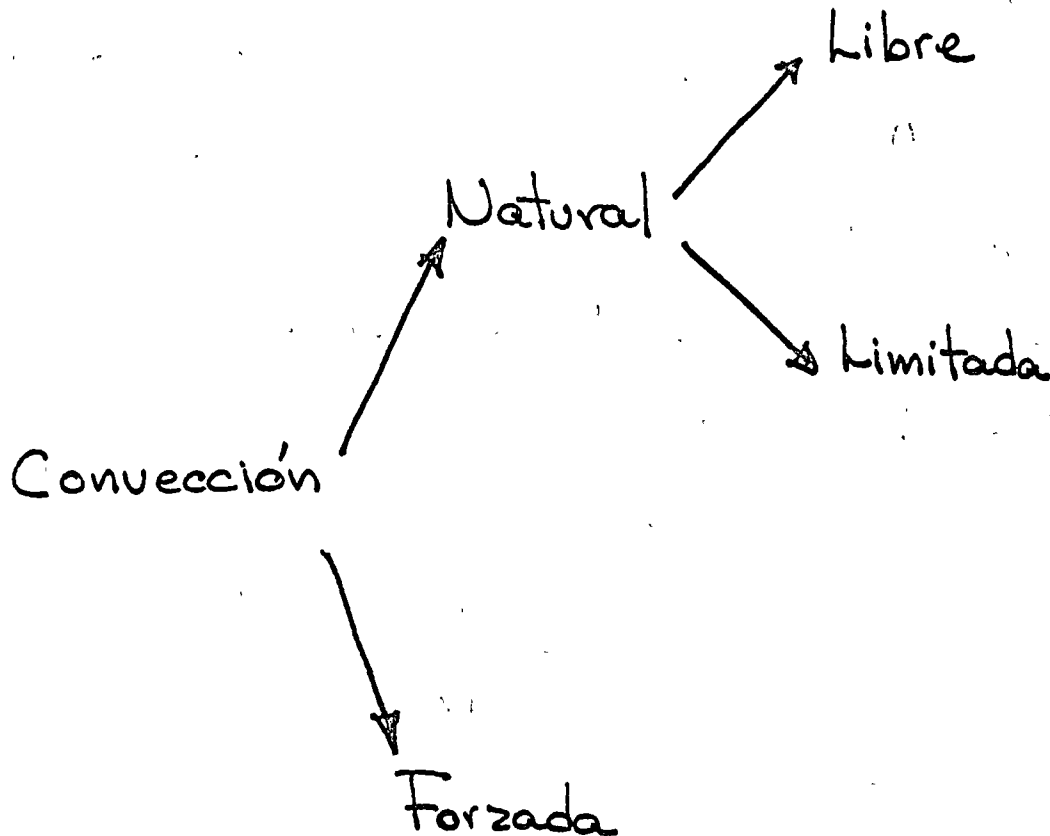
La figura (A) ilustra el caso de convección natural libre. La placa indicada está a una temperatura mayor que la ambiente. El fluido cercano a la placa estará también a mayor temperatura y por consiguiente tendrá una densidad menor que el resto. Esta diferencia de densidades causará un flujo vertical a lo largo de la placa.

La figura (B) ilustra la convección natural

limitada. El flujo tambien es causado por la diferencia de densidades en las ramas del circuito, pero está confinado por las tuberías.

Finalmente, la figura © ilustra la convección forzada. En este caso, la circulación del fluido es causada por medios mecánicos externos (bomba).

En Resumen:



El estudio que vamos a presentar a continuación lo vamos a restringir a casos en los que los fluidos se pueden considerar incompresibles. Esta restricción simplifica grandemente las ecuaciones generales y no es una limitación seria en lo que se refiere a los campos de aplicación.

Por fluidos incompresibles vamos a definir aquellos cuya densidad no cambia con cambios de presión. Esta suposición es practicamente cierta cuando tratamos con líquidos, aun cuando la presión cambie apreciablemente. En el caso de gases, esta suposición no es tan obvia. Sin embargo, se puede demostrar que los efectos de compresibilidad en las ecuaciones de mecánica de fluidos solo se vuelven importantes cuando la velocidad del flujo es del mismo orden de magnitud que la velocidad del sonido en ese fluido.

Estas velocidades no se alcanzan en procesos de convección natural. En procesos de convección forzada es posible que se obtengan velocidades de esta magnitud en restricciones del circuito

(orificios, válvulas, etc.), pero en general, en los equipos de transferencia de calor las velocidades serán menores.

Para cualquier fluido en general, la velocidad del sonido está dada por

$$c = \left(\frac{\partial P}{\partial \rho} \right)^{1/2}$$

y para gases perfectos, substituyendo las ecuaciones apropiadas se tiene

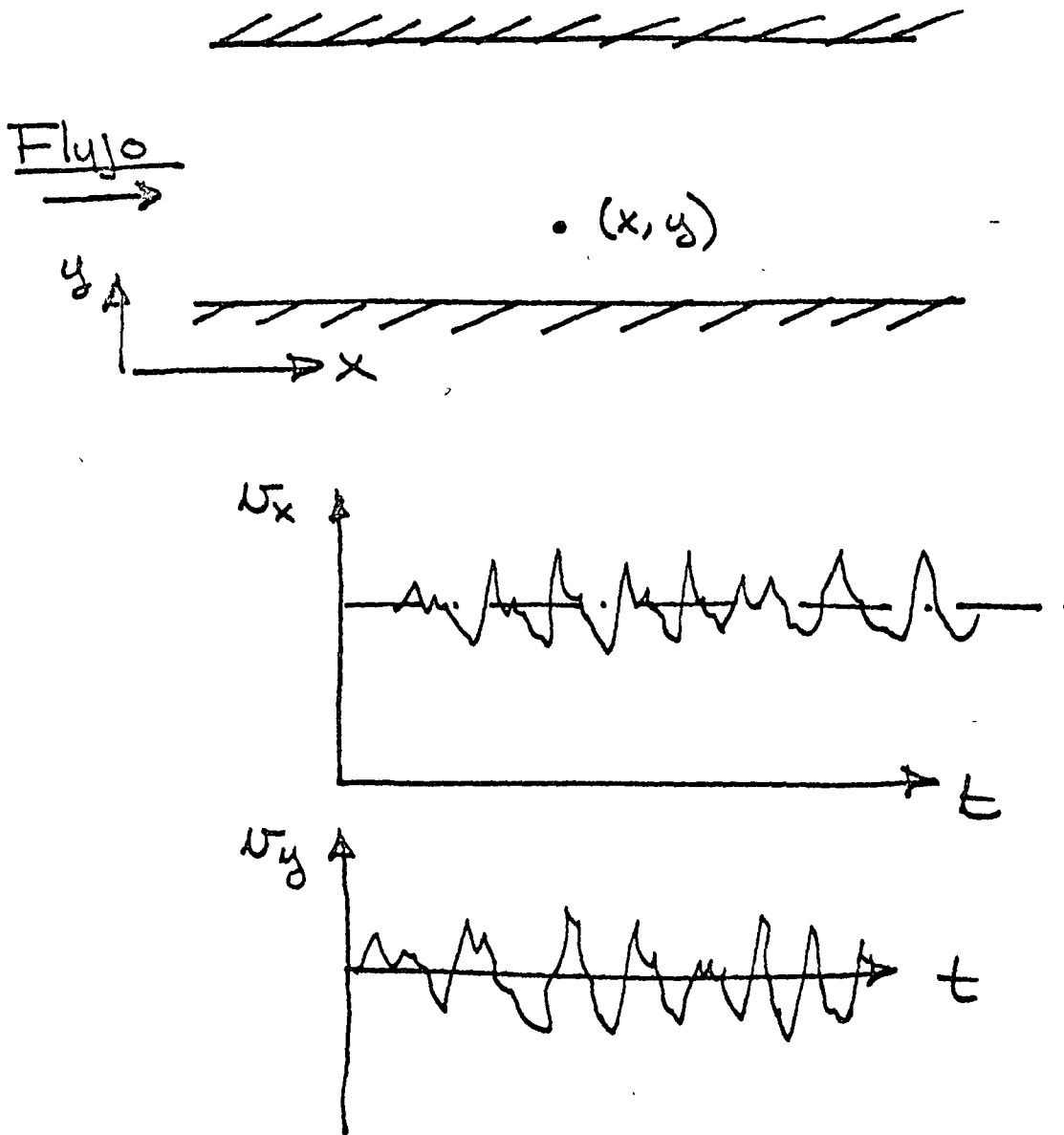
$$c = \sqrt{\gamma R T} \quad \gamma = C_p / C_v \quad *$$

Con estas bases en mente empezaremos el tema de convección con una revisión de las características de Mecánica de Fluidos, y después usaremos estos resultados para el análisis de la transferencia de calor.

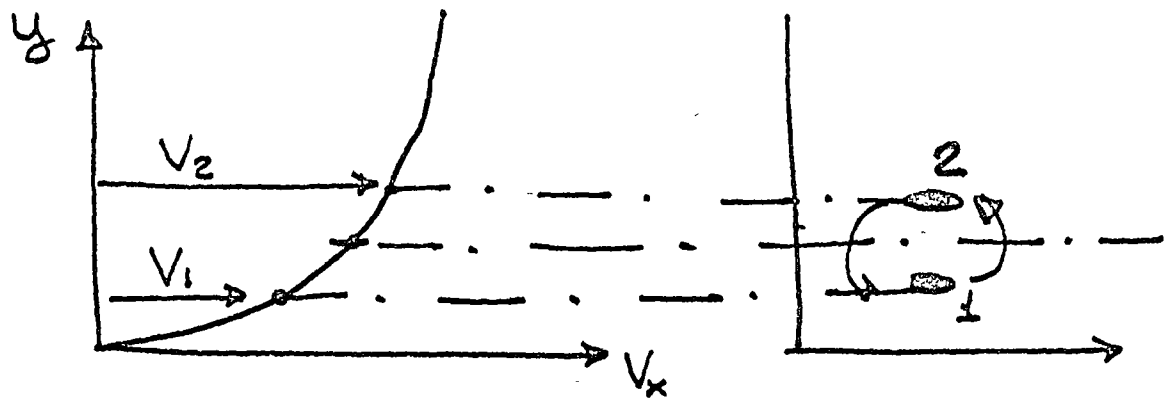
* Derivaciones de estas ecuaciones se pueden encontrar en cualquier libro de Termodinámica. Por ejemplo "Thermodynamics" G. van Wylen.

Turbulencia . Mecanismos de Transporte

Conceptos Generales



Transferencia de Cantidad de Movimiento



Flujo laminar

$$\frac{\tau}{\rho} = -\nu \frac{\partial \bar{U}_x}{\partial y}$$

Flujo Turbulento

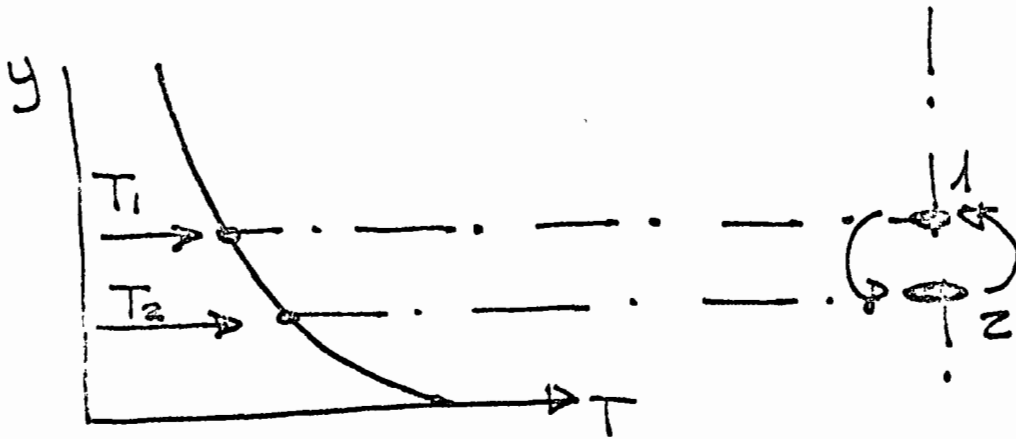
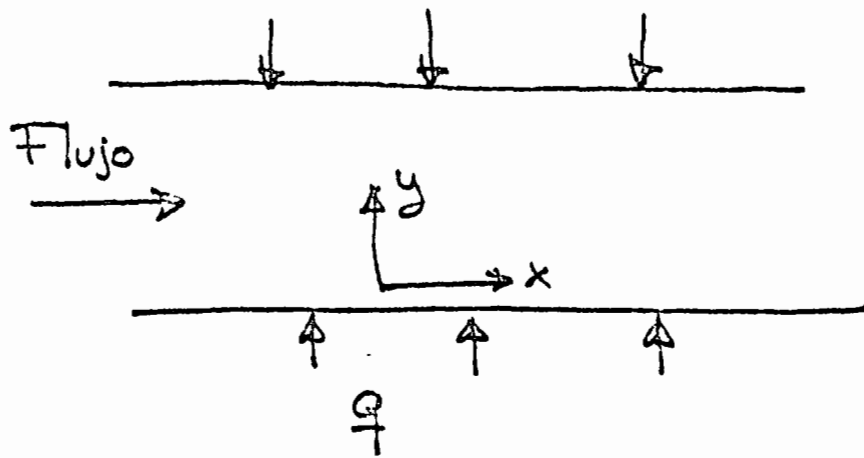
$$\frac{\tau}{\rho} = -(\nu + \epsilon_m) \frac{\partial \bar{U}_x}{\partial y}$$

Para flujos muy turbulentos:

$$\nu \ll \epsilon_m$$

$$\Rightarrow \frac{\tau}{\rho} = -\epsilon_m \frac{\partial \bar{U}_x}{\partial y}$$

Transferencia de Calor



$$\frac{Q/A}{\rho C_p} = -\alpha \frac{\partial T}{\partial y} \quad \text{Flujo laminar}$$

$$\frac{Q/A}{\rho C_p} = -(\alpha + \epsilon_H) \frac{\partial T}{\partial y} \quad \text{Flujo Turbulento}$$

$$\alpha \ll \epsilon_H$$

$$\frac{Q/A}{\rho C_p} = -\epsilon_H \frac{\partial T}{\partial y}$$

Resumen de Formulas Empíricas.

1.- Caidas de Presión en Tubos

Coeficiente de Fricción:

$$f = \frac{\Delta P}{4 (L/D) \left(\frac{\rho V^2}{2} \right)}$$

$$f = \frac{16}{Re} \quad Re < 2300$$

(flujo laminar)

$$f = \frac{0.046}{Re^{0.2}} \quad 10^4 < Re < 10^7$$

$$f = \frac{0.0791}{Re^{0.25}} \quad Re < 10^4$$

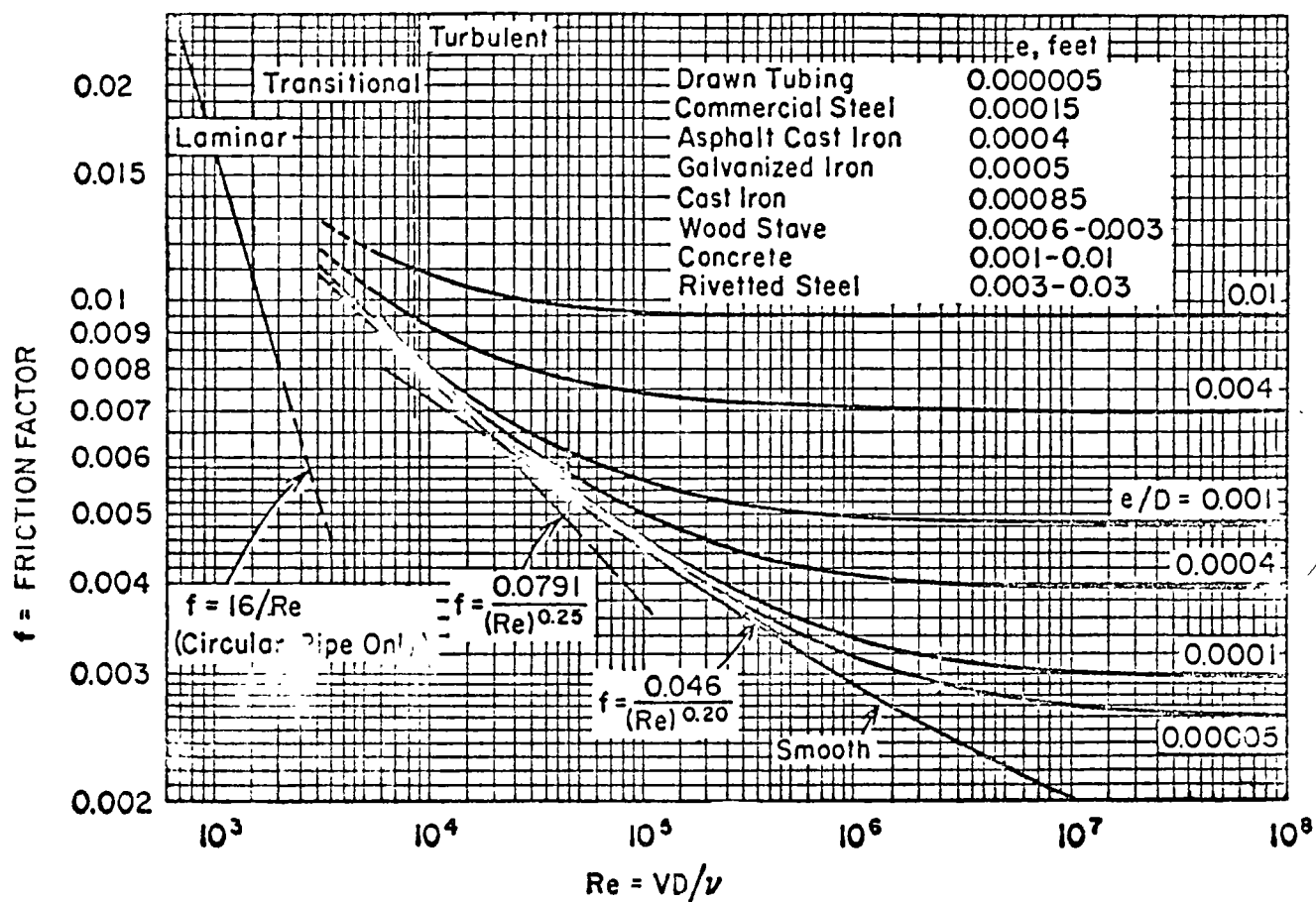


Fig. 4.1. Friction factor for flow in circular pipes. [Moody (5)].

Tuberías Curvas (Serpentines) Helicoidales

$$\frac{f_s}{f_T} = \left[Re \left(\frac{D}{D_H} \right)^2 \right]^{1/20}$$

$$Re (D/D_H)^2 > 6$$

D = diámetro de la tubería

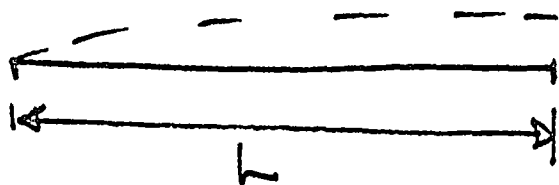
D_H = diámetro del Serpentin.

Secciones no circulares

Para flujo turbulento pueden usarse las mismas formulas empleadas en tubos.

Flujo sobre una placa plana

→ V_∞



$$C_{f,x} = \frac{\tau_0}{\frac{1}{2} \rho V_{\infty}^2}$$

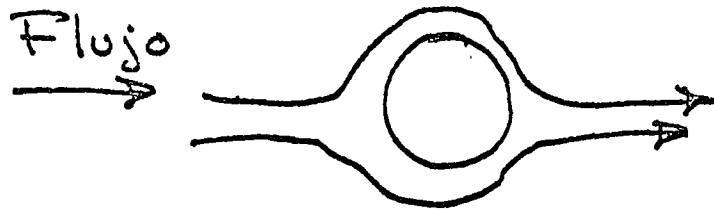
$$C_{f,x} = 0.0576 \left(\frac{V}{V_{\infty x}} \right)^{1/5}$$

$$C_f = 0.072 \left(\frac{V}{V_{\infty h}} \right)^{1/5} \quad (\text{Total})$$

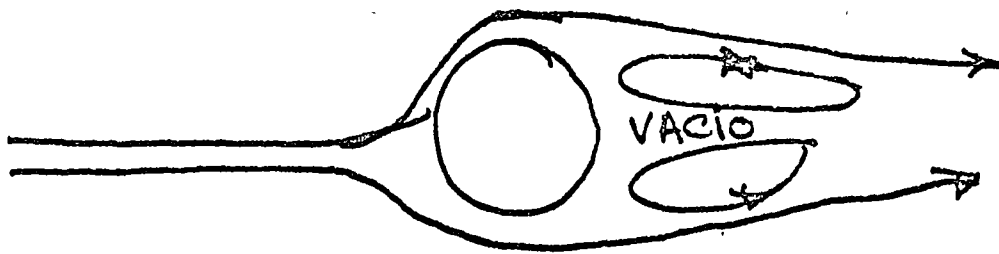
Flujo a Través de cuerpos sumergidos

Fuerza ejercida por el fluido =

Fricción + Fuerza hidrodinámica.



Situación, sin
separación



Separación

$$F_D/A = C_D \frac{\rho V^2}{2}$$

Area = Area proyectada sobre un plano perpendicular al flujo.

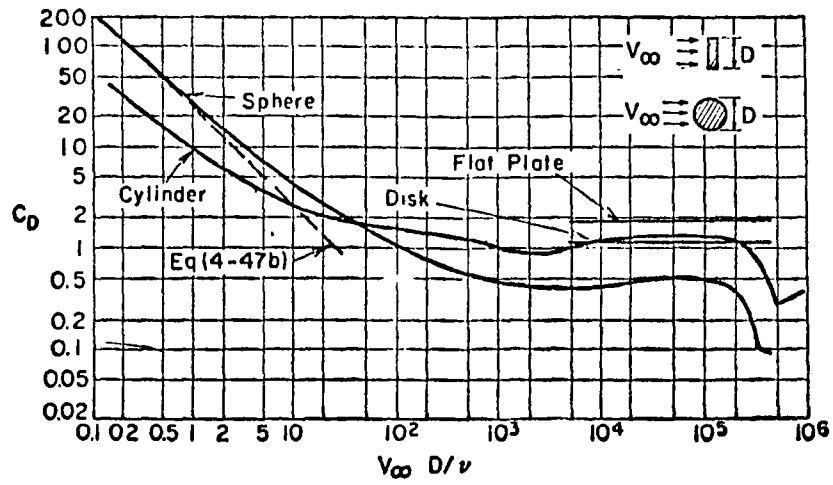
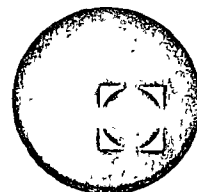


Fig. 4.17. Drag coefficient for some common shapes [Eisner (17)].



centro de educación continua

facultad de ingeniería, unam



DIRECTORIO DE PROFESORES DEL CURSO APLICACIONES DE LA TRANSFERENCIA DE CALOR EN PROCESOS IND.

Dr. José Ma. González Santaló
General Electric Co. M/C 583
Nuclear Energy Division
175 Curther Ave.
San José, Cal. 95125

Ing. Martiniano Aguilar Rodríguez
Comisión Federal de Electricidad
entrada por Atoyac 10 Piso
México, D.F.

Ing. Isidro Becerril Salinas
Comisión Federal de Electricidad
entrada por Atoyac 10 Piso
México, D.F.

Ing. Julián Castellanos Fernández
Instituto Mexicano del Petróleo
Jefe del Depto. de Desarrollo de
Ing. Básica
Av. de los 100 Metros # 152
México, D.F.

Ing. Julio Sta. María Esteves
Jefe de Grupo de Proceso
Bufete Industrial Diseños
y Proyectos, S.A.
Tolstoi No. 22
México, D.F.

Ing. Carlos Castillo Cruz
Jefe del Depto. de Seguridad
de Reactores
Centro Nuclear de Salazar
Inst. Nal. de Energía Nuclear
Salazar, Edo. de Méx.

Ing. Anton Baron G.

EL MATERIAL ADJUNTO ES PROPIEDAD EXCLUSIVA DEL
INSTITUTO TECNOLOGICO DE MASSACHUSETTS.
REPRODUCCION PARCIAL O TOTAL DEL MISMO ESTA
ABSOLUTAMENTE PROHIBIDA, SIN LA AUTORIZACION DE:

P. GRIFFITH
DEPT. OF MECHANICAL ENGINEERING
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
CAMBRIDGE, MASS.

OVERALL CALCULATION METHODS FOR TWO PHASE FLOW

Peter Griffith

List of Symbols

A	Area of the cross section
A_{in}	Area in
A_{out}	Area out
A_w	Wall area
C	Empirical constant used to correlate fitting pressure drop data
D	Diameter
f	Friction factor
F_x	Force in the "x" direction
G	Mass velocity
g	Acceleration of gravity
g_0	Gravitational constant
K_1	Empirical constant used to correlate fitting pressure drop data
K_2	Empirical constant used to correlate fitting pressure drop data
L	Length
m	Exponent relating friction factor to Reynolds number, Table III
n	Exponent relating friction factor to Reynolds number, Table III
N_{Reg}	Reynolds number of gas phase flowing alone in the pipe
N_{Ref}	Reynolds number of liquid phase flowing alone in the pipe
P	Pressure
ΔP	Pressure difference

ΔP_f	Pressure difference due to friction
ΔP_g	Pressure difference due to gravity
ΔP_m	Pressure difference due to momentum changes
ΔP_o	Pressure drop with liquid flowing alone at mixture mass velocity
ΔP_{tp}	Pressure drop with two phases flowing
ΔP_{fh}	Pressure drop due to friction in a heated pipe
ΔP_{gh}	Pressure drop due to gravity in a heated pipe
r_f	Dimensionless friction multiplier for an unheated pipe
r_{fh}	Dimensionless friction multiplier for a heated pipe
r_g	Dimensionless gravity multiplier for an unheated pipe
r_{gh}	Dimensionless gravity multiplier for a heated pipe
V	Velocity
V_f	Superficial liquid velocity
V_g	Superficial gas velocity
V_x	Velocity in the x direction
v	Specific volume
v_l	Specific volume at inlet
v_i	Specific volume at discharge
v_f	Liquid specific volume
v_{fg}	Change in specific going from liquid to vapor
w_f	Liquid flow rate
w_g	vapor flow rate
X	Parameter for calculating the pressure drop using Martinelli method (see Table II)
x	Quality

α	Void fraction
μ_f	Liquid viscosity
μ_g	Gas phase viscosity
ϕ	Pressure drop multiplier
ϕ_f	Pressure drop multiplier for liquid, see equation (I-29)
ϕ_{ftt}	Pressure drop multiplier, see equation (I-29)
ρ	Density
ρ_f	Liquid density
ρ_g	Gas phase density
θ	Angle measured from horizontal
τ_w	Wall shear stress

INTRODUCTION

Two phase flow is only one branch of fluid mechanics. It is treated separately, however, because the number of dimensionless variables which is needed to characterize the flow is so large. If one looks, for instance, at the friction pressure drop for a fully developed single phase flow in a smooth pipe, the dimensionless wall shear stress, the friction factor, is a function of a single variable, the Reynolds number. If a comparable analysis is done for a two phase flow in the same pipe, the dimensionless pressure gradient becomes a function of six dimensionless groups. These could be a volume flow ratio, a density ratio, a Reynolds number, a viscosity ratio, a Froude number and a Webber number. Clearly, the empirical methods which have proven effective in single phase flows will have to be altered if they are to apply to two phase flows. It is impossible to handle in a rigorous way any system which is a function of six independent variables. For this reason, we have a hierarchy of methods for handling two phase flow problems starting with the simplest which can only give approximate answers up to very complex techniques which give precise answers to a very narrow range of conditions. In these notes three levels of precision will be touched on. First, the homogeneous model will be described. It is the least accurate, most general and the easiest to use. It possesses many attributes characteristic of a real two phase flow. Next, the Thom method will be described. It is generally more accurate and only a little more complex in application than the homogeneous method. Finally, the methods based on particular flow regimes will be described. They apply

to only a narrow range of conditions and must rely on a considerable amount of empirical information. They do describe the flow quite completely.

Before going into the details of these methods, it is worthwhile to make some general remarks about the state of the art in the two phase flow fields. A perfectly stupendous amount of work has been done. Three reasonably recent literature surveys [1], [2], [3] cover most of this work and it is recommended that copies of these surveys be secured if any serious work is to be done in the field. In addition to these surveys, three books on two phase flow have recently been published. The first [4] is essentially a handbook on two phase flow and heat transfer as applied to reactor design. The second is a translation of a Russian book which is very useful when trying to read Russian literature. It provides the common basis upon which the Russian literature is based. Reference [6] is a synthesis of what is known about two phase flow and should be purchased as a reference book by every one of you. References [19] and [20] are books addressed to special classes of problems which may be useful for reference or design in certain areas.

BASIC RELATIONS

Whatever model is used to calculate a two phase flow, continuity on each phase requires a certain relationship between the quality, average phase velocities and the void fraction must be satisfied. The relationship is particularly useful if one is interested in computing the velocity ratio of the phases from void fraction data.

If a two phase flow in a pipe is examined, the following continuity relationships for the phases must be satisfied.

$$w_f = (1 - \alpha) V_f A \rho_f \quad (I-1)$$

$$w_g = \alpha V_g A \rho_g \quad (I-2)$$

In addition, the definition of quality gives the following equation

$$x = \frac{w_g}{w_f + w_g} \quad (I-3)$$

Equations (I-1), (I-2) and (I-3) can be combined to give equation (I-4)

$$\frac{1 - \alpha}{\alpha} = \left(\frac{V_g}{V_f} \right) \left(\frac{\rho_g}{\rho_f} \right) \left(\frac{1 - x}{x} \right) \quad (I-4)$$

The pressure drop equation, including all the terms for a two phase flow will be derived next. The manner of evaluating these terms differs for the different models.

Let us consider a steady flow in a uniform diameter pipe with small enough changes in conditions along it so that we can speak of an average wall shear stress and an average density. If the x direction momentum equation, (I-5), is as follows

$$\Sigma F_x = \int_{A_{out}} \rho v_x^2 dA - \int_{A_{in}} \rho v_x^2 dA \quad (I-5)$$

the substitution for the gravity, shear and pressure forces results in equation (I-6).

$$(P_1 - P_2)A - \tau_w A_w - \rho \frac{g}{g_o} AL \sin \theta = \int_{A_{out}} \rho v_x^2 dA - \int_{A_{in}} \rho v_x^2 dA \quad (I-6)$$

Solving for the pressure difference and collecting terms on the right we obtain

$$(P_1 - P_2) = \frac{4L}{D} \tau_w + \rho L \left(\frac{g}{g_o} \right) \sin \theta + \frac{1}{A} \left[\int_{A_{out}} \rho v^2 dA - \int_{A_{in}} \rho v^2 dA \right] \quad (I-7)$$

or

$$\Delta P = \Delta P_f + \Delta P_g + \Delta P_m \quad (I-8)$$

In equation (I-8) the terms can be expressed as follows

$$\Delta P_f = \frac{4L}{D} \tau_w \quad (I-9)$$

$$\Delta P_g = \rho L \frac{g}{g_o} \sin \theta \quad (I-10)$$

$$\Delta P_m = \frac{1}{A} \left[\int_{A_{out}} \rho v^2 dA - \int_{A_{in}} \rho v^2 dA \right] \quad (I-11)$$

In order to determine values for each of these terms it is necessary to run some experiments. Let us now turn our attention, briefly, to how these experiments are run.

PRESSURE DROP AND VOID EXPERIMENTS

Only one kind of experiment can give the friction pressure drop in a two phase flow. It is necessary to use a horizontal pipe to eliminate the effects of gravity. The flow must be both fully developed and experience a low enough drop in absolute pressure so that the momentum changes are negligible. A practical difficulty which makes experimentation difficult arises from the fact that the pressure drop is not steady but fluctuates with a magnitude larger than the average drop. Unless special care is taken to average the pressure drops properly, erroneous reading result. Because there are so many ways to go astray in two phase measurement, there is considerable scatter in the friction pressure drop data and disagreement from experimenter to experimenter.

Very often measurements have to be run in vertical tubes as this is the geometry of interest. In order to extract the friction component of the pressure drop, it is necessary to measure the void fraction. This is most easily accomplished with quick acting valves which isolate a section of the test section. A gamma ray attenuation scheme can be used,

too. Both techniques measure the density directly. Momentum pressure drop measurements will be described in the section on homogeneous flow. It is clear, however, that unless these terms are small or evaluated correctly, the calculated friction pressure drop component for the two phase flow will be in error.

THE HOMOGENEOUS MODEL

The homogeneous model is the simplest two phase flow model there is, yet it displays many of the properties of a real two phase flow. For this reason, it is worthwhile to spend a lot of time on it. We shall begin by describing the model, then continue by evaluation its constants and range of validity. Finally, we shall use it as the basis of the recommendations for calculating the pressure in fittings of various kinds.

The premise upon which the homogeneous model is based is that the real two phase flow can be replaced by a single phase flow with the density as defined in equation (I-12)

$$v = v_f + x v_{fg} \quad (I-12)$$

Using this specific volume, it is possible to go ahead and evaluate the various terms in the pressure drop equation (I-8).

The friction pressure drop can be evaluated by means of a homogeneous friction factor. From the definition of the friction factor for fully

developed flow in a circular pipe,

$$\Delta P_f = \frac{4L}{D} \tau_w = 4 f \left(\frac{L}{D} \right) \frac{G^2 v}{2g_o} \quad (I-13)$$

The evaluation of the friction factor should be discussed. Having defined a fictitious homogeneous fluid for calculation purposes, it is not possible to go back to your assumptions and see clearly what should be used. For this reason, it is proposed to choose a constant friction factor, independent of quality or mass velocity, on the basis of convenience. We bypass, in this way, such questions as what is the mixture viscosity or what is the Reynolds number for the mixture. It is felt these questions are diversions. The friction factor is an empirical quantity for a single phase flow and it is to be expected that it is an empirical quantity for a two-phase flow also. The best friction factor has the right behavior at the limits and gives the best answer in the region of its validity.

For this reason, it is suggested that a friction factor evaluated as though saturated liquid were flowing in the system at the mixture mass velocity be used in equation (I-13). This gives a smooth approach to the pure liquid solution but gives rise to a step at the 100% quality limit. For a single, constant value of the friction factor, this is probably about the best that can be done. Let us now compare this calculation technique to some data.

Figure (I-1), Reference [7] is some friction pressure gradient data for steam and water at different mass velocities as a function of quality.

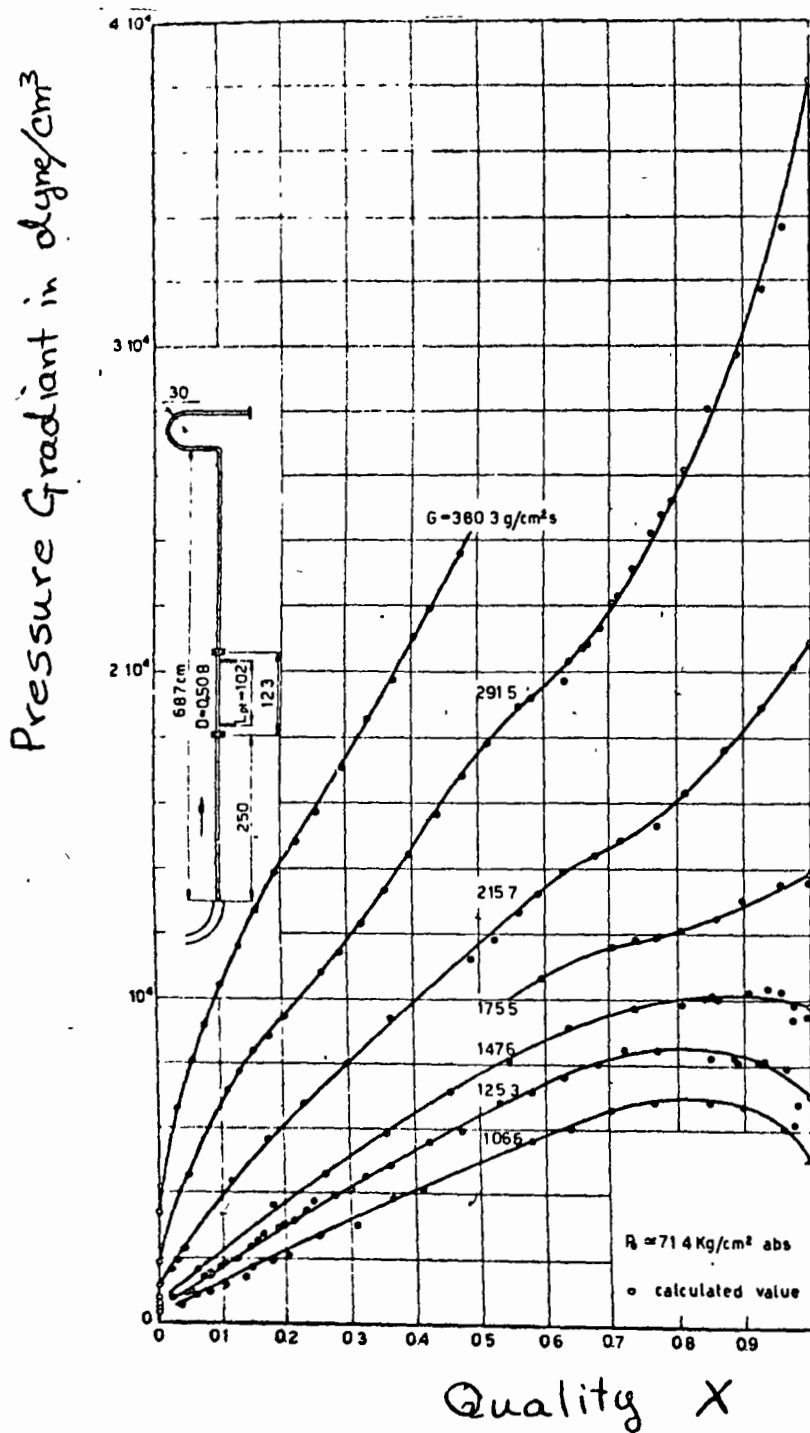


Figure I-1 Pressure gradient versus quality for steam and water at approximately 1,000 psia. The test section is sketched at the left and the tube dimensions shown. Reference (7).

It is clear that the shape of the curves changes as the mass velocity changes. For the lowest mass velocity, where the amount of entrainment is small, there is a pronounced peak in the pressure gradient at a high quality. When the mass velocity is large enough to ensure a large amount of entrainment, (and an almost homogeneous flow) the pressure gradient is almost linear with quality.

If we substitute equation (I-12) into (I-13), it is evident that the friction pressure gradient should be a linear function of quality and the model is quite good. For lower mass velocities there is an annular film on the wall and the curves of friction pressure gradient versus quality drop with increasing quality at high quality. In a rough way, the step in the friction factor at 100% quality which exists in this model approximates this behavior.

Figure (I-2) shows a comparison between theory and experiment for pressure gradients evaluated for water at several pressures by Owens [8]. Again, the correlation is reasonably good.

The other terms in the pressure drop equation need evaluation also. The gravity pressure drop can be evaluated directly from the definition of specific volume.

$$\rho = \frac{1}{v} \quad (I-14)$$

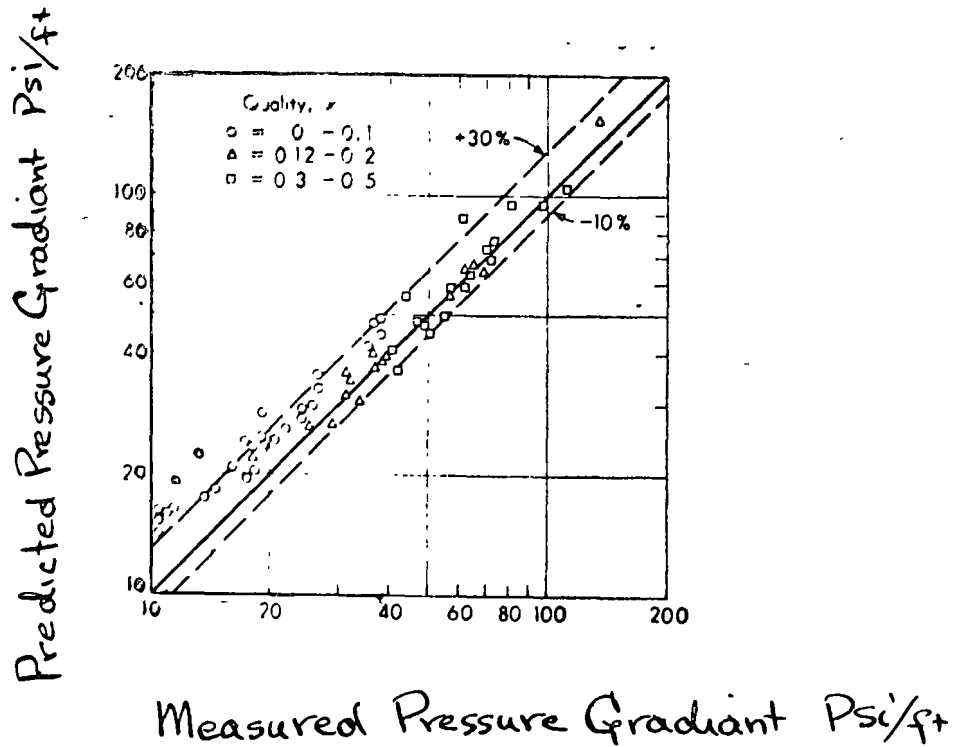


Figure I-2 Predicted versus measured pressure gradient for the homogeneous theory for water ranging in pressure from 71 through 374 psia and qualities from 4 to 50 percent. The tube is .118 in diameter and 30 inches long.

And with this definition for density, the gravity pressure drop becomes

$$\Delta P_g = \rho \left(\frac{g}{g_0} \right) L \quad (I-15)$$

Generally, the prediction of the gravity pressure drop from the homogeneous theory is quite poor. If the gravity pressure drop is very large, the homogeneous model will do a poor job of predicting the pressure drop. A comparison of this, the homogeneous, no slip model with data is shown on the velocity ratio curve of the section on the Thom-Martinelli calculation method.

The momentum pressure drop can also be evaluated from the homogeneous model. The equation for evaluating this is

$$\Delta P_m = \frac{G^2}{g_o} (v_2 - v_1) . \quad (I-16)$$

In this case, the predictions are quite good. Reference [9] reports a comparison of the separated (Thom-Martinelli) model and the homogeneous model with some air water data for several pressures near one atmosphere. Figure (I-3) which is typical of this data shows a comparison between the homogeneous model and experiment. As can be seen the homogeneous model does the best job. Before proceeding, let us look more closely into the experiments reported in Reference [9].

The momentum flux is measured directly by turning a stream through a known angle, in this case, 90°. The difference between the inlet and outlet momentum multiplier is what one multiplies the mass velocity squared by in order to obtain the momentum pressure drop. The reason the homogeneous model does a good job is not that it really describes the flow but rather that it has compensating errors.

It is shown in Reference [9] that the velocity of the liquid phase is appreciably lower than that of the gas phase. At the same time, however, the flow is quite unsteady. Because of this unsteadiness, because the average momentum flux of the liquid depends on the velocity squared, the average force experienced by the turning tee is increased by waves. The waves are always present in a two phase flow and can have a very dramatic effect on the momentum flux. Thus, what we have in the real flow is slip tending

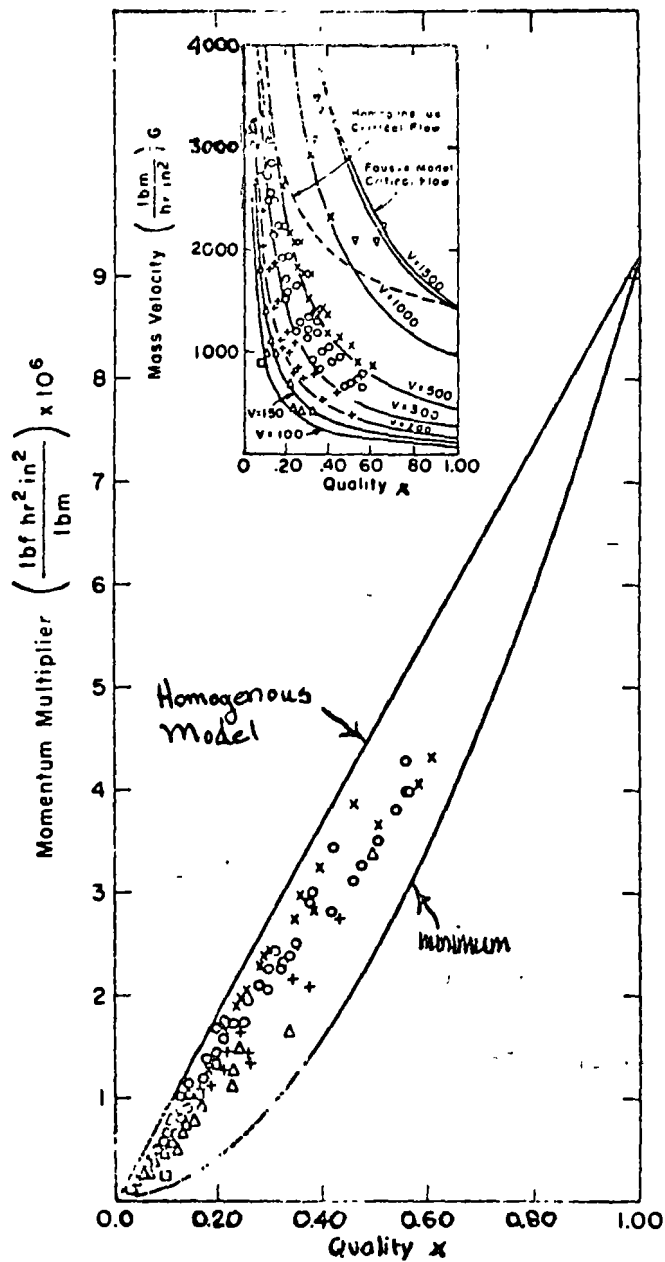


Figure I-3 Momentum multiplier as a function of quality for air and water at about one atmosphere pressure in a one inch pipe. The momentum multiplier difference multiplied by the mass velocity squared gives the momentum pressure difference. Reference 9.

to decrease the momentum flux from that of the homogeneous model and unsteadiness tending to raise it. The sum of the two effects just cancels out.

HEATED PIPES - HOMOGENEOUS METHOD

There is no reason why one cannot integrate the local contributions of the friction, gravity and momentum pressure drops and obtain a pressure profile for a heated pipe. In practice this becomes very clumsy because the algebraic expressions are so inconvenient in form. For this reason, it is suggested that dividing the heated pipe up into a number of short sections and using average values for velocity and density in each usually turns out to be easier. This method also has the merit of allowing one to eliminate the unimportant terms in the equation and trace the drop in static pressure so that appropriate values of the specific volume and enthalpy can be used.

FITTINGS

All piping systems contain fittings and for many, the major part of the pressure drop occurs in them. For this reason, it is important to have some way of calculating their pressure drop. Most of the material contained in this section is culled from Reference [10]. Data from other sources is included, however. Let us start by describing how two phase pressure drop

in fittings is determined, then alter the homogeneous theory to correlate the data and finally present a tabulation of recommendations for use in design.

Typically, two phase pressure drop data for fittings is measured in a test setup in which a long approach pipe and a long departure pipe are attached onto the fitting in question. They are both instrumented with a large number of manometers. The manometer readings are extrapolated back to the fitting in question and the resulting step in pressure is the pressure drop ascribed to the fitting. This is the pressure drop which is correlated. In order to insure that the flow in both the approach and departure pipe has become fully developed, it is necessary to have enough taps to see that there is a straight line pressure distribution. Figure (I-4) illustrates this while Figure (I-5) shows some actual pressure length data. (Reference [11]).

One of the most striking things about the data of Figure (I-5) is the fact that the lengths that it takes to fully develop the flow are much longer for a two phase flow than a single phase flow. As a result of this fact, much of the two phase pressure drop data which appears in the literature is flawed. There was simply not enough approach and departure length provided.

For high pressure two phase pressure drop data, it is customary to precede the test section such as illustrated in Figure (I-5) by a preheater. The liquid flow rate is set and the preheater turned on to give the desired quality. With this setup it is very easy to form the ratio of pressure drops in the fitting to those that would exist at the same mass velocity

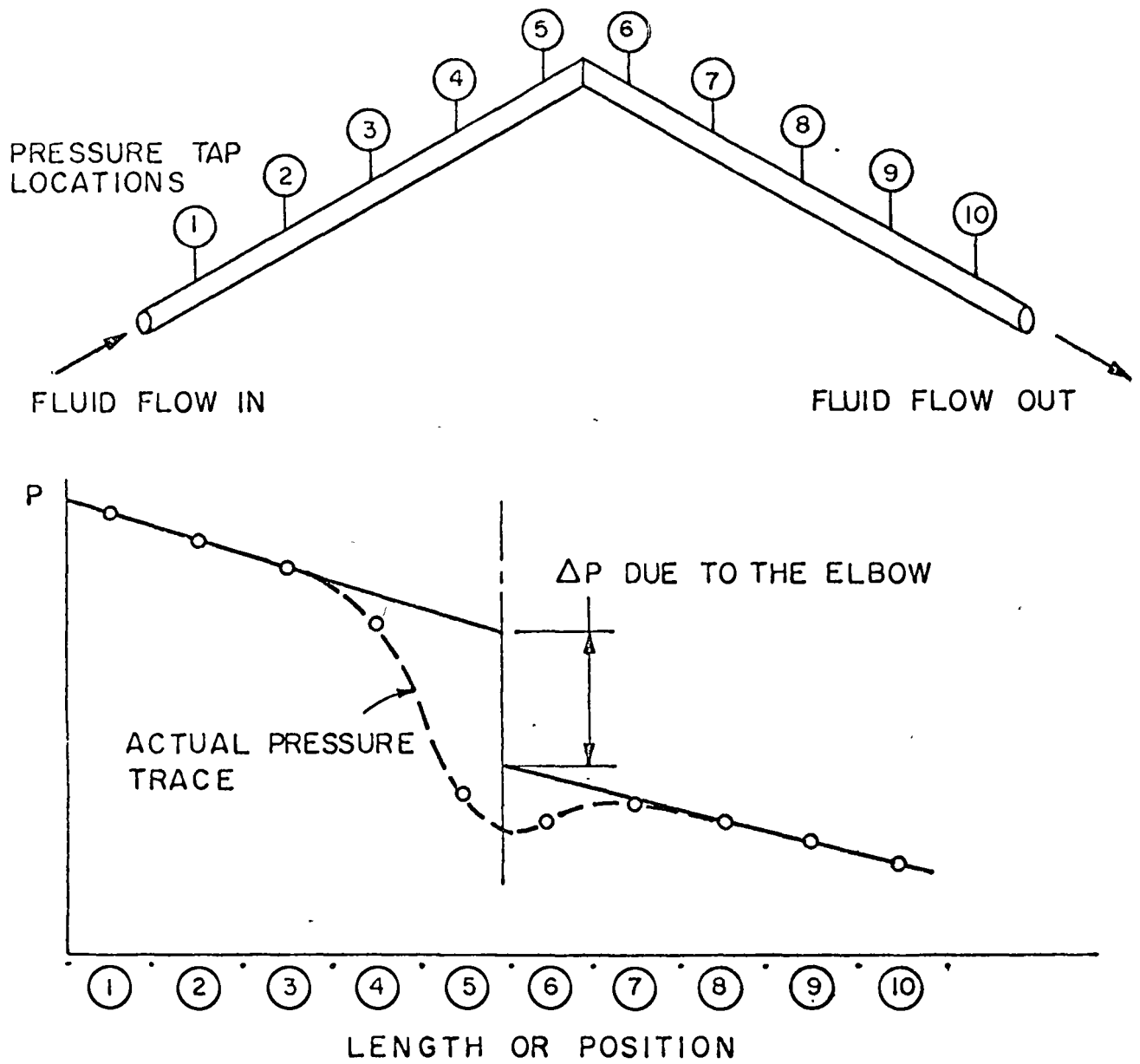


Figure I-4 Pressure drop due to a fitting defined experimentally.

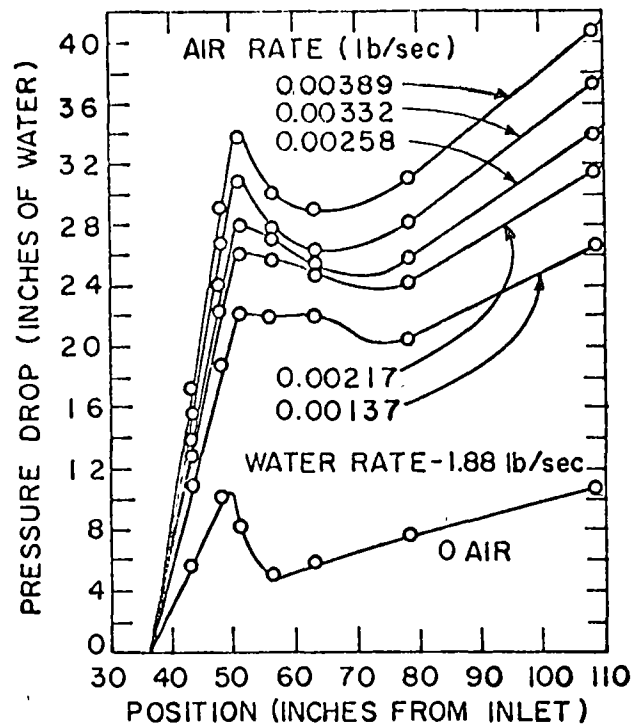


Figure I-5 Pressure drop as a function of length for air and water flowing in an expansion from .25x2" to .5x2" horizontal. The long developing length for the two phase flow is the significant feature. Reference 11.

and zero quality. This is the basis of the correlation scheme proposed below.

If we turn now to the pressure drop occurring in a fitting with two phases flowing it is possible to use the homogeneous model to formulate a model for correlating two phase pressure drop data for fittings. For a single phase flowing through a fitting, the pressure drop is given by equation (I-17).

$$\Delta P_o = \frac{K_1 G^2 (v_f)}{2g_o} \quad (I-17)$$

For two phases, the comparable homogeneous model equation is

$$\Delta P_{tp} = \frac{K_2 G^2 (v_f + x v_{fg})}{2g_o} \quad (I-18)$$

The ratio of these can be formed and will have the ratio of coefficients out front. This is not a convenient location as the zero quality asymptote is incorrect. For this reason, coefficient is redefined and put inside the brackets. The result is equation (I-19).

$$\frac{\Delta P_{tp}}{\Delta P_o} = (1 + C x \frac{v_{fg}}{v_f}) \quad (I-19)$$

If the homogeneous model were perfect, C would be one. It turns out, experimentally, that it can be either greater or less than one. Table I gives a listing of C for various fittings. The quality range over which these coefficients apply is limited because at a quality of 100%, the form of equation (I-19) must be incorrect. In the absence of any data, it is suggested that a "C" of one be used or perhaps a simple interpolation and extrapolation scheme constructed. A comparison of this scheme with a sample of data is shown in Figure (I-6). Table I shows the recommendations.

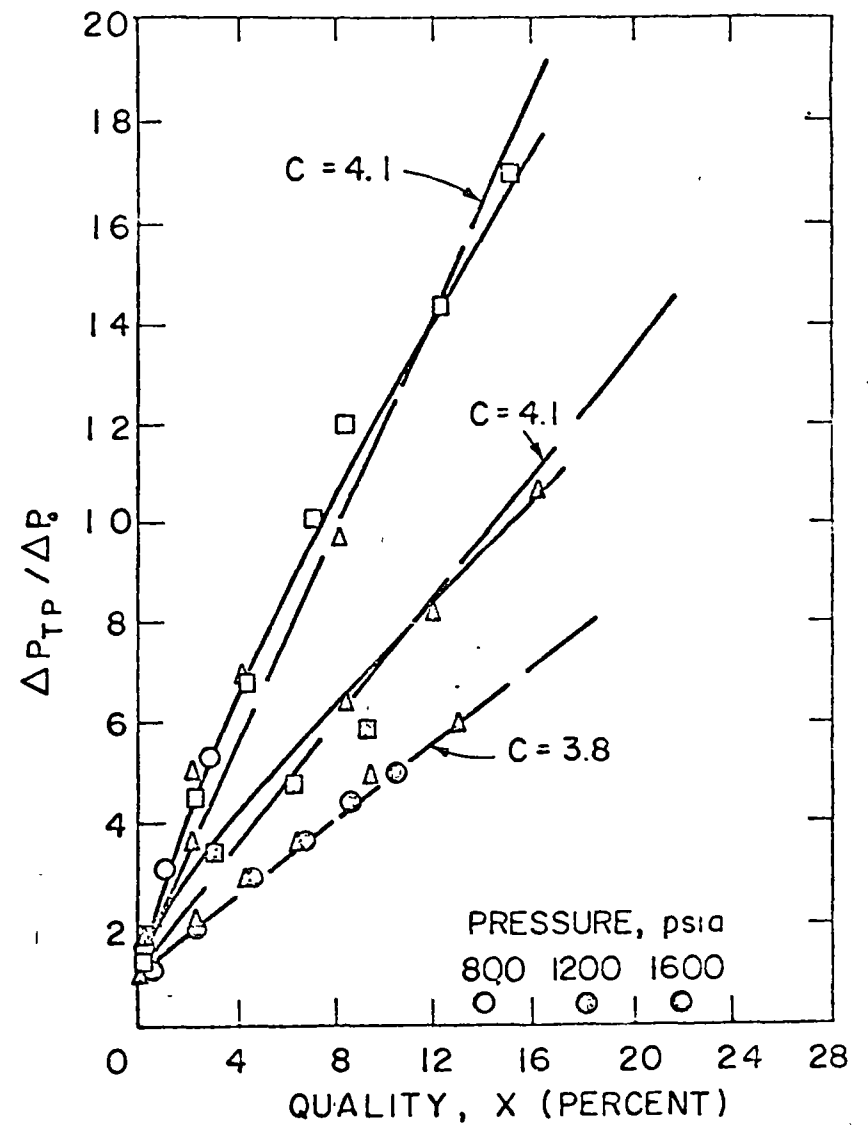
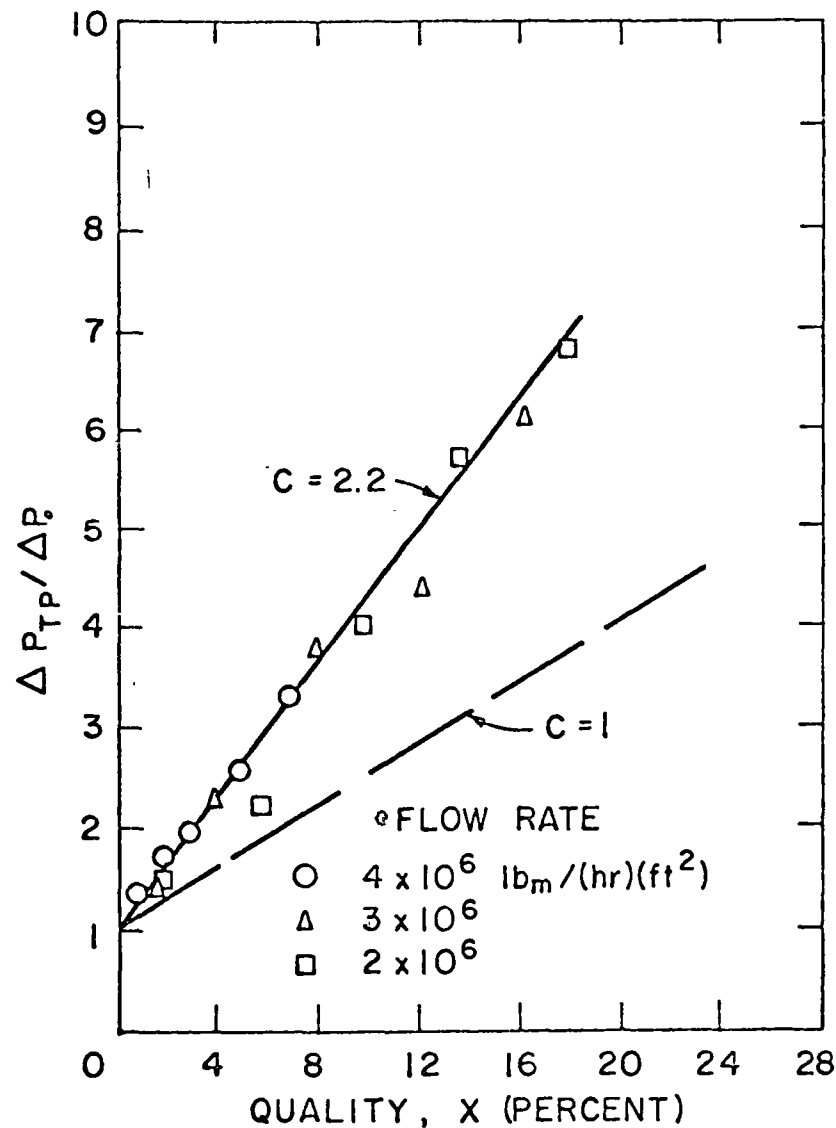


Figure 1-6 Pressure drop ratio for a bend for steam and water at 1,200 psia for a 2" diameter pipe and 10" bend radius. The first curve shows mass velocity as a parameter and the second shows pressure. Reference 10.

TABLE I

<u>Reference</u>	<u>Fitting</u>	<u>Pressure Range</u>	<u>Quality Range</u>	<u>C</u>
10,11	Bend-Short	P/Pc .15	0 to 10%	1.5
10	Bend-Short	P/Pc .1	0 to 50%	4.0
10	Long	P/Pc .1	0 to 50%	2.2
10	Tee (Serving as L)	P/Pc .1	0 to 50%	1.6
10	Gate Valve	P/Pc .1	0 to 50%	1.5
10,14,15	Contractions	.015 P/Pc 1	0 to 50%	1.0
10,11	Expansions	.15 P/Pc 1	0 to 50%	1.1

METHOD OF THOM

Thom's calculation method Reference [16] represents an improvement over the homogeneous method, in two respects. First the velocities of the two phases are allowed to differ, with the ratio of the velocities being a function of pressure alone. Second, the friction factor is allowed to become a function of quality. Both the velocity ratio variation and the friction factor variation are determined from data taken on heated and adiabatic system water systems in pipes between 0.2" and 2" in diameter, at pressure between 250 and 2300 psia and orientated both horizontally and orientated both horizontally and vertically. The mass velocity is above 0.5×10^6 lb/ft². In this section the method of computing the pressure

drop for unheated pipes will be given first. Then the method for computing uniformly heated pipes will be given. The basic equation for evaluating the pressure drop still is equation (I-8).

ADIABATIC PIPES

The friction pressure drop is evaluated by computing that which would occur if saturated liquid were flowing alone at the mixture mass velocity and multiplying by a dimensionless multiplier. This is shown in the equation (I-20) below. The multiplier r_f is given as a function of pressure and quality on Figure (I-7).

$$\Delta P_f = 4f \left(\frac{L}{D} \right) \left(\frac{G^2 v_f}{2g_o} \right) r_f \quad (I-20)$$

The friction factor in equation (I-20) is evaluated from the well known Moody curve for instance.

In order to evaluate the gravity pressure drop it is necessary to know the void fraction. The velocity ratio is given on Figure (I-8). Equation (I-4) relates the velocity ratio to the void. The void in turn is used in equation (I-21) to determine the gravity pressure drop. The equation is:

$$\Delta P_g = \left[\rho_f (1 - \alpha) + \rho_g (\alpha) \right] L \frac{g}{g_o} \sin \theta \quad (I-21)$$

Figure (I-9) shows the void fraction calculated from equation (I-4) and the curve of Figure (I-7) for several different pressures.

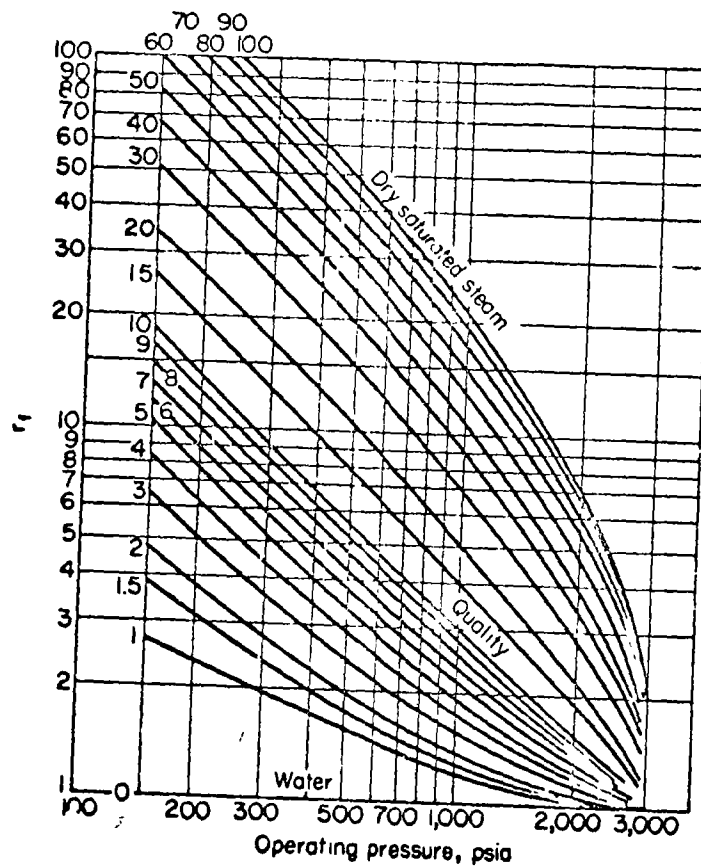


Figure I-7 Friction multiplier to use in Equation (I-20). Reference 16.

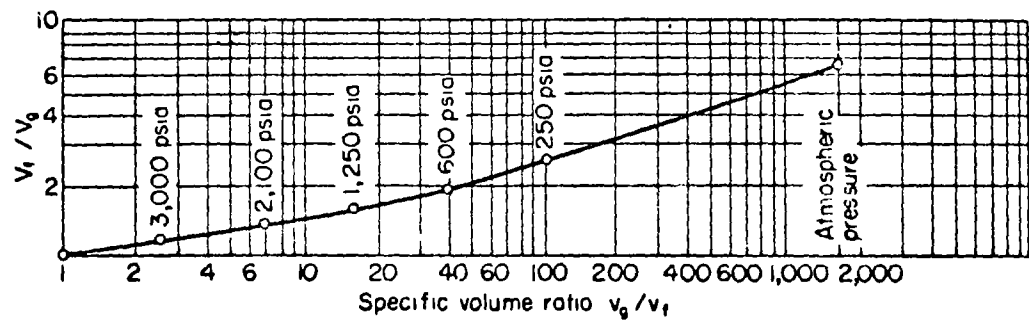


Figure I-8 Velocity ratio to use in Equation (I-4) to determine the void. Reference 16.

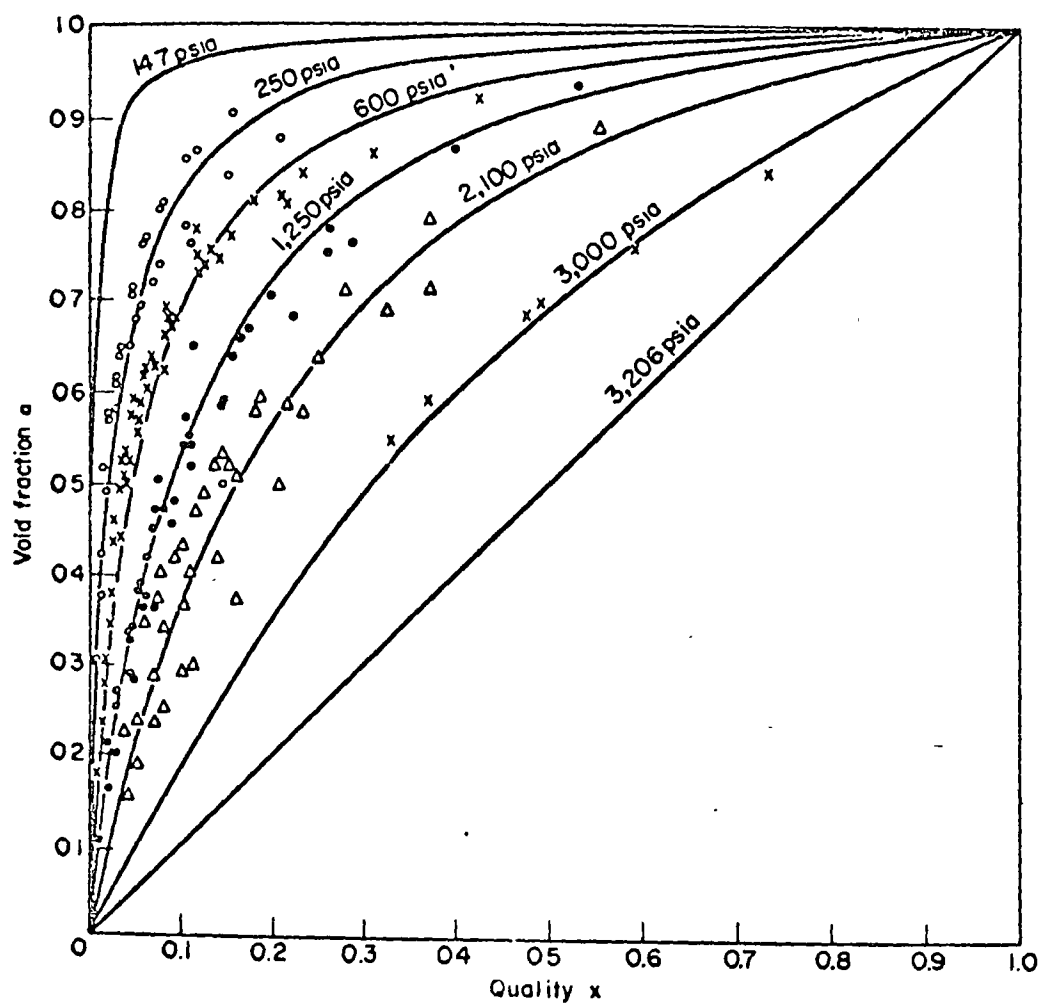


Figure I-9 Void fraction as a function of quality and pressure. Reference 16.

The acceleration pressure drop is still evaluated using equation (I-16). For unheated pipes it is normally small.

HEATED PIPES

For uniformly heated pipes, starting with saturated liquid at the entrance, the friction contribution to the pressure drop has been integrated and is presented in terms of a multiplier. The multiplier is a function of gravity and pressure for water. It is obtained from Figure (I-10) and is used in equation (I-22).

$$\Delta P_{fh} = 4f \left(\frac{L}{D}\right) \frac{G^2 v_f}{2g_o} r_{fh} \quad (I-22)$$

The friction factor is evaluated from the Moody Graph Section VII assuming saturated water is flowing at the mixture mass velocity.

The gravity contribution is evaluated assuming saturated water is flowing at the mixture mass velocity and using the multiplier obtained from Figure (I-11) in equation (I-23):

$$\Delta P_{gh} = \rho_f \left(\frac{g}{g_o}\right) L \sin \theta r_{gh} \quad (I-23)$$

The angle θ is measured from the horizontal. The gravity pressure drop is positive for up flow and negative for down flow. Figure (I-12) shows a comparison of the Thom calculation technique with some data.

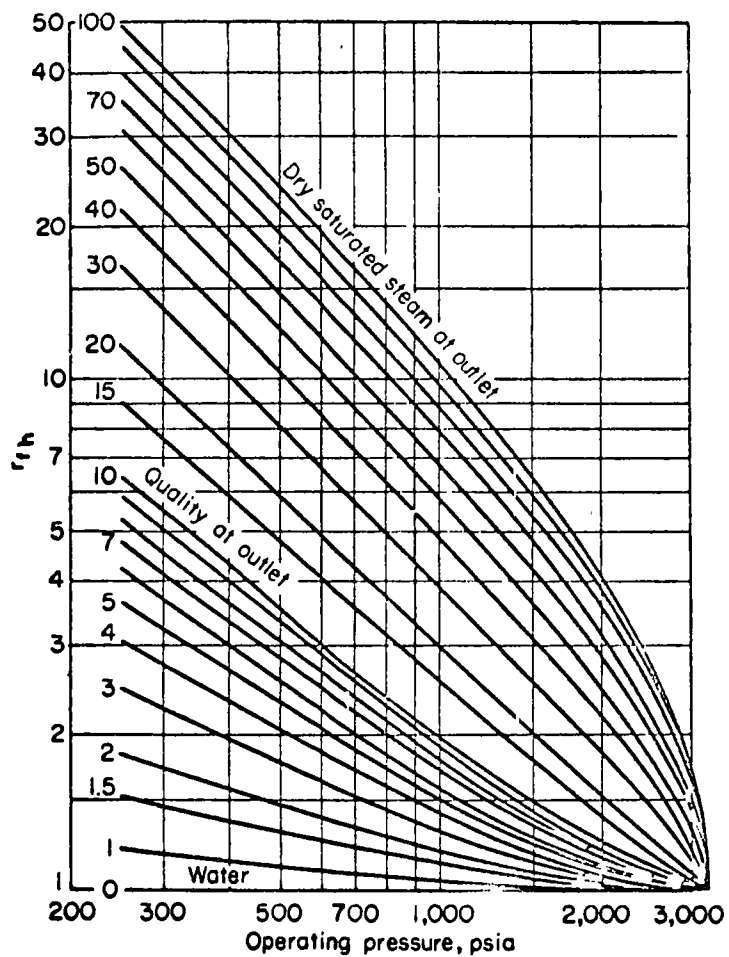


Figure I-10 Friction multiplier for heated pipes for substituting in Equation (I-22). Reference 16.

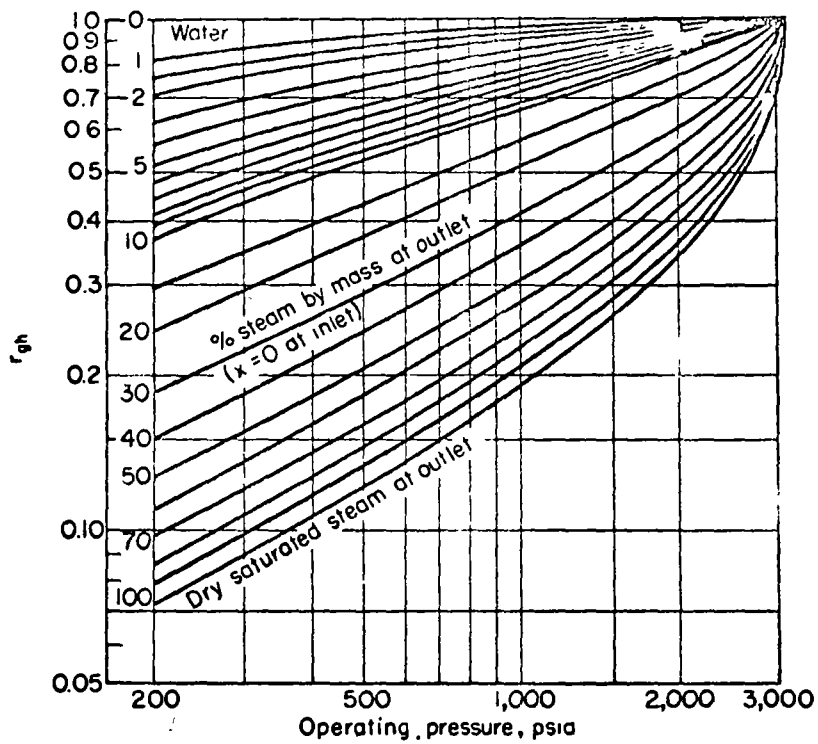


Figure I-11 Gravity pressure drop multiplier for substitution into Equation (I-23). Reference 16.

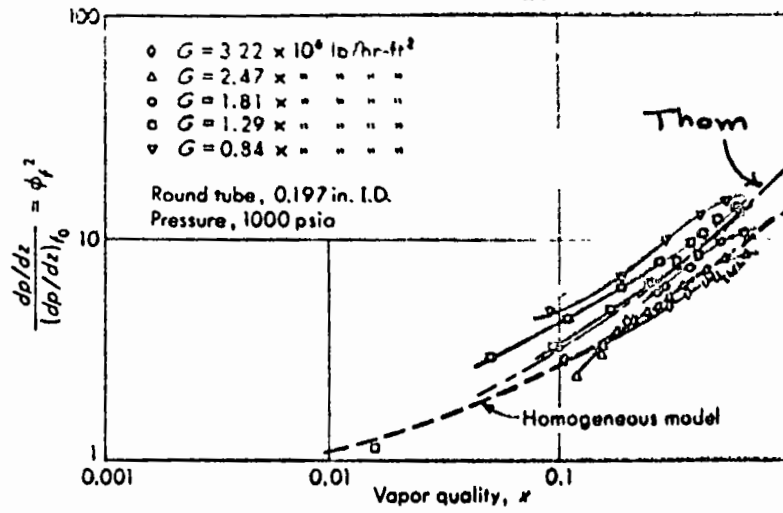


Figure I-12 Comparison of homogeneous and Thom model with data of Reference 17. $P = 1,000 \text{ psia}$ steam and water.

THE MARTINELLI CALCULATION TECHNIQUE FOR LOW PRESSURE ADIABATIC PIPES

The Martinelli two phase pressure drop calculation procedure has been supplanted at high pressure by that of Thom. At pressures close to one atmosphere pressure for a very wide variety of liquids, the Martinelli procedure [18] is still the best. The bulk of the data behind it were taken in horizontal pipes between 0.06 and 1.0 inches in diameter at about one atmosphere pressure using air and a wide variety of liquids. The basic pressure drop equation is still equation (I-8).

The pressure drop is determined by first calculating the pressure drop for either the liquid flowing alone, or the gas flowing alone in the pipe at its mass velocity. That is

$$\Delta P_{fa} = 4f \left(\frac{L}{D} \right) \frac{G_f^2 v_f}{2g_o} \quad (I-24)$$

$$\Delta P_{ga} = 4f \left(\frac{L}{D} \right) \frac{G_g^2 v_g}{2g_o} \quad (I-25)$$

The two phase friction pressure drop is then determined by multiplying either the liquid only pressure drop, or the gas only pressure drop, by the appropriate multiplier ϕ_f^2 or ϕ_g^2 . This multiplier is given in Figure (I-13) as a function of the variable X. The following procedure is used to determine the variable X.

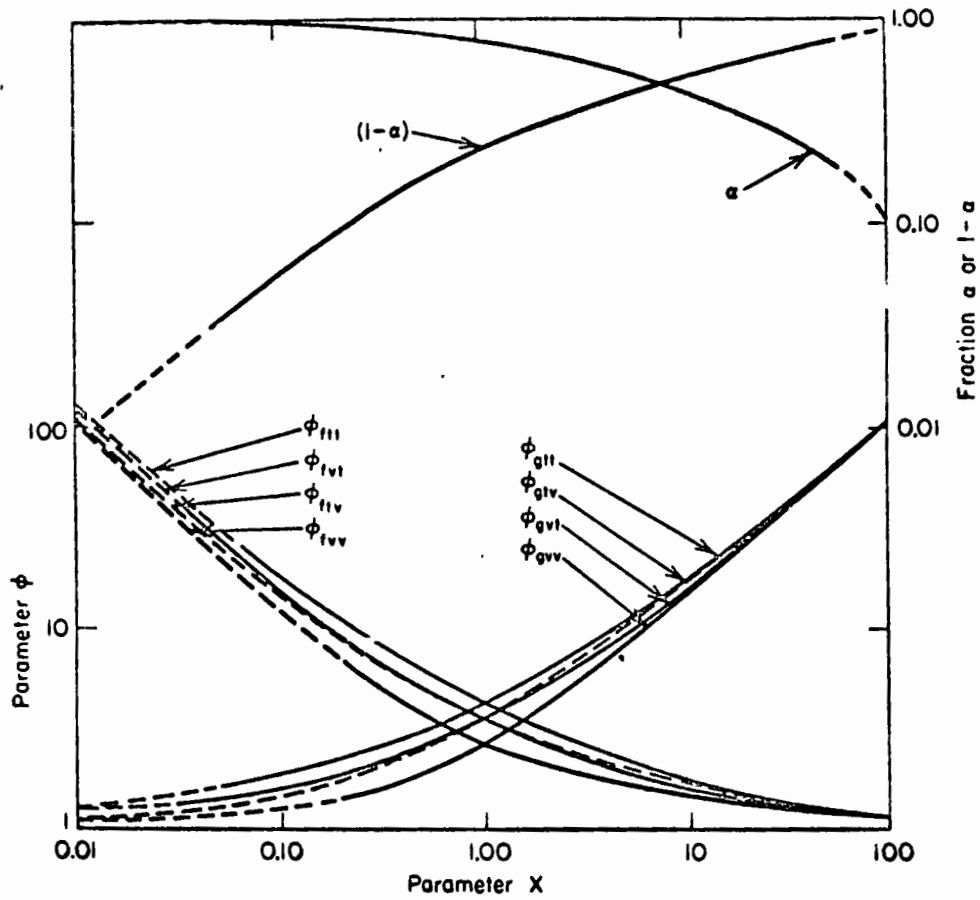


Figure I-13 Friction pressure drop multipliers for substitution into Equation (I-29) from Reference 18. This curve should be used for low pressure gas-liquid mixtures when the other fluid is not water.

Determine the Reynolds numbers for the gas flowing alone in the pipe, each at its own mass velocity. These Reynolds numbers are defined as

$$N_{Ref} = \frac{G_f D}{\mu_f} \quad (I-26)$$

$$N_{Reg} = \frac{G_g D}{\mu_g} \quad (I-27)$$

Go to Table II and determine which of the four combinations of viscous or turbulent flow for each phase exists for your problem. Go then to Table III and determine the coefficients for substituting into equation (I-28)

$$X^2 = \frac{N_{Reg}^m}{N_{Ref}^n} \frac{C_f}{C_g} \frac{w_f^2}{w_g} \frac{\rho_g}{\rho_f} \quad (I-28)$$

Finally, go to Figure (I-13) and determine the multiplier. For example, if one uses the liquid as the basis for the pressure drop calculation, and both N_{Ref} and N_{Reg} are turbulent, the friction pressure drop will be

$$\Delta P_f = \Delta P_{fa} \Phi_{ftt}^2 \quad (I-29)$$

For the other three possibilities of vt, tv or vv one simply uses the appropriate multiplier in an equation similar to equation (I-29). In

TABLE II

Flow Mechanism	Symbol	N_{Ref} is	N_{Reg} is
<u>Liquid</u> <u>Gas</u>			
Turbulent - Turbulent	t-t	> 2000	> 2000
Viscous - Turbulent	v-t	< 1000	> 2000
Turbulent - Viscous	t-v	> 2000	< 1000
Viscous - Viscous	v-v	< 1000	< 1000

TABLE III

	t-t	v-t	t-v	v-v
n	0.2	1	0.2	1.0
m	0.2	0.2	1.0	1.0
C_f	0.046	16	0.046	16
C_g	0.046	0.046	16	16

general, it is best to base the pressure drop calculation in phase which has the largest pressure drop when flowing alone in the pipe.

The gravity contribution is evaluated from equation (I-30) below and Figure (I-13).

$$P_g = \left[(\alpha) \rho_g + (1 - \alpha) \rho_f \right] \frac{g}{g_o} L \sin \quad (I-30)$$

For determining the momentum pressure drop it is recommended again that equation (I-16) be used.

When one is using the Martinelli method for heated pipes it is necessary to integrate the local values of pressure gradient over the length of the test section in order to get pressure difference across it. The pipes should be broken up into short sections and the pressure drop components summed up for each.

REFERENCES

1. S.W. Gouse, Jr. "An Index to the Two-Phase Gas Liquid Flow Literature," M.I.T., Vol. 1 - DSR Report 8734-1, May 1963; Vol. 2 - DSR Report 8734-4 September 1964; Vol. 3 - DSR Report 8734-6, January 1966. Also published as a book by the M.I.T. Press in 1967.
2. R.R. Kepple, T.V. Tung. "Two-Phase (Gas-Liquid) System: Heat Transfer and Hydraulics. An Annotated Bibliography," ANL-6734, July 1963.
3. E.H. Burr, V.A. Sternberg, F.J. Belusar, S.J. Green. "A Bibliography on Heat Transfer and Fluid Mechanics Aspects of Nuclear Reactors," WAPD - 315, Vols. I & II, August 1967.
4. L.S. Tong, "Boiling Heat Transfer and Two-Phase Flow," John Wiley and Sons, Inc., 1965.
5. S.S. Kutatedadze, M.A. Styrikovich. "Hydraulics of Gas-Liquid Systems," September 1950 in the USSR, Translated by the Technical Information Center of Wright Patterson AFB as ADN. 257824.
6. G.B. Wallis. "One Dimensional Two-Phase Flow," McGraw-Hill, 1969.
7. G.P. Gaspari, C. Lombardi, G. Peterlongo. "Pressure Drops in Steam-Water Mixtures," Report R-83 CISE, Milan, Italy, January 1964.
8. W.L. Owens, Jr. "Two-Phase Pressure Gradient," International Developments in Heat Transfer, Am. Soc. of Mech. Engrg. Paper 41, Vol. 2, pp. 363-368, 1961.
9. G.B. Andeen, P. Griffith. "Momentum Flux in Two Phase Flow," Trans. ASME, Vo. 90, Series C, No. 2, May 1968.
10. D.E. Fitzsimmons. "Two Phase Pressure Drop in Piping and Components," HW-08970 Rev. 1, March 20, 1964.
11. B.L. Richardson. "Some Problems in Horizontal Two-Phase Two-Component Flow," ANL 5949, December 1958.
12. Lorenz G. Straub, E. Siberman. "Air-Water Mixture Flow Through Orifices, Bends and Other Fittings in Horizontal Pipes," St. Anthony Falls Hydraulic Laboratory, University of Minnesota, Project Report No. 63, September 1960.
13. T.L. Deobald. "An Experimental Investigation of Two Phase Flow Pressure Losses in Pipe Elbows," M.S. Thesis in Chem. Engrg., University of Idaho, 1962.

14. J.K. Farrell, J.W. McGee. "Two-Phase Flow Through Abrupt Expansions and Contractions," Final Report Vol. III, USAEC Contract No. AT - (40-1) - 2950, June 1966, pp. 126-137 (North Carolina State University).
15. C.E. Geiger, W.H. Rohrer. "Sudden Contraction Losses in Two Phase Flow," ASME Paper 65-HT-41, August 1965.
16. J.R.S. Thom. "Prediction of Pressure Drop During Forced Circulation Boiling of Water," Int. J. Heat Mass Transfer, Vol. 7, pp. 709-724, 1964.
17. M. Muscettola. "Two-Phase Pressure Drop, UKAEA Report AEEW-R284, 1963.
18. Lockhart, R.W., R.C. Martinelli. "Proposed Correlation of Data for Isothermal Two Phase Two Component Flow in Pipes," Chem. Eng. Prog. 45, pp. 39-48, 1947.
19. G.F. Hewitt, N.S. Hall-Taylor. "Annular Two Phase Flow," Pergamon Press, 1970.
20. A.E. Dukler, et al. "Gas-Liquid Flow in Pipelines; I. Research Results II. Design Manual." Published by the American Gas Association and the American Petroleum Institute, 1970.

EL MATERIAL ADJUNTO ES PROPIEDAD EXCLUSIVA DEL
INSTITUTO TECNOLÓGICO DE MASSACHUSETTS.
REPRODUCCIÓN PARCIAL O TOTAL DEL MISMO ESTA
ABSOLUTAMENTE PROHIBIDA, SIN LA AUTORIZACIÓN DE:

W. M. ROHSENOW
DEPT. OF MECHANICAL ENGINEERING
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
CAMBRIDGE, MASS.

Final (rev. 8.1.10)
2/26/77

WMR-A-1

BOILING

by

Warren M. Rohsenow

The process of evaporation associated with vapor bubbles in a liquid is called boiling. Here attention will be focussed on boiling at solid heated surfaces of interest in engineering applications.

When a pool of liquid at saturation temperature is heated by an electrically heated wire or flat plate, data for q/A vs. $(T_w - T_{sat})$ usually appear as shown in Fig. 1, lower solid curve. As ΔT increases, the initial natural circulation regime, AB, changes to the nucleate boiling regime. The appearance of the first bubble at B requires a significant finite superheat.

As ΔT or heat flux is increased, more and more nucleation sites become active producers of bubbles. A peak nucleate boiling heat flux (called critical heat flux, CHF) is reached when bubbles stream forth from so many nucleation sites that the liquid is unable to flow to the heat surface. With electric heating (q/A the independent variable) the operating condition at point C of Fig. 1, changes rapidly to point E' which is at a very much higher temperature. If point E' is above the melting point of the solid surface, a "burnout" occurs. If it is below, then operation can be maintained along the curve ED which is called the "film boiling" regime, because a layer of vapor continually blankets the surface, and liquid does not contact the surface.

If the heat flux is reduced below point D at the minimum point on the curve, the operating condition reverts to nucleate boiling at point F. Then power again must be increased to point C before the process can revert to film boiling.

To operate in the transition region, C-D, between nucleate and film boiling, the surface temperature should be the independent variable, accomplished by a condensing vapor, etc. In this region the process may be alternating between nucleate and film boiling, with the liquid touching the surface intermittently.

The dashed line shown for pool boiling of a subcooled liquid lies near the curve for saturated liquids; it may lie either to the left or right depending on different natural convection effects resulting from different surface geometries.

Experimental data for forced convection - inside of ducts or across wires and plates - when plotted on the same coordinates, appears as shown in Fig. 1. If the liquid temperature T_b is subcooled, the heat flux is represented by

$$(q/A)_{\text{before boiling}} = h [(T_w - T_s) + (T_s - T_b)] \quad (1)$$

At high heat flux, the forced convection curves for a particular pressure and at various subcoolings and velocities appear to merge on log-log plots into a single curve called the "fully-developed boiling curve". For some systems this fully-developed region lies approximately on an extension of the pool boiling line for the same surface.

The literature on boiling heat transfer is now quite voluminous. For references in addition to those discussed here the reader is referred to the quite complete collection of annotated bibliography assembled by Gouse⁽¹⁾.

Nucleation

In carefully cleaned test systems, heating of pure liquids can produce high superheat at temperatures up to a limit of stability where homogeneous nucleation, vapor formation, takes place. For the case of heat transfer associated with boiling at solid heating surfaces, observed wall temperatures are very much lower than these. Further observations of nucleate boiling show bubble streams emerging at single spots on the surface. Microscopic observation of these spots (1) revealed a scratch or a cavity where a bubble formed. Earlier, Corty and Foust⁽²⁾, Bankoff⁽³⁾ and others developed the postulate that bubbles at a heating surface emerge from cavities in which a gas or vapor phase pre-exists, Fig. 2. As heat is added, more vapor forms in the cavity, Fig. 2a, and a bubble emerges and departs, Fig. 2b, after which liquid closes in over the cavity trapping vapor which is the

source of the next bubble. Surveys of nucleation phenomena are found in Refs. 4 and 5.

To put these notions on a quantitative basis, consider the equilibrium and growth of a spherical bubble in a large body of liquid. The following equilibrium conditions apply; equal and uniform temperatures in both phases equal chemical potential ($\mu = h - TS$) in both phases, and

$$p_v - p_l = \frac{2\sigma}{r} \quad (2)$$

With these the equilibrium conditions may be shown graphically in Fig. 3 where states b' , b'' and c' , c'' are corresponding vapor and liquid states for two different radii. The production of a Fig. 3 is rather tedious. A simplified procedure evolves from the observation that the locus of vapor states is very close to the saturation line. If it is assumed that the vapor temperature is the plane-surface saturation temperature corresponding to the pressure of the vapors, p_v , a much simpler analysis follows.

Calculate the ΔT corresponding to $p_v - p_l$ given by Eq. (1), integrating the Clausius-Clapeyron relation

$$\frac{dT}{dp} = \frac{T v_{fg}}{h_{fg}} \quad (3)$$

along the p - T saturation curve.

The following are results of such an integration for various assumptions:

a) If h_{fg}/v_{fg} T is constant and $T \approx T_{sat}$

$$T_v = T_{sat} \left(1 + \frac{2\sigma}{r} \frac{v_{fg}}{h_{fg}} \right) \quad (4a)$$

b) If h_{fg}/v_{fg} is constant

$$T_v = T_{sat} \exp\left(\frac{2\sigma}{r} \frac{v_{fg}}{h_{fg}}\right) \quad (4b)$$

c) If $v_{fg} \approx v_g = R_v T/p$

$$T_v = \frac{T_{sat}}{1 - \frac{T_{sat} R_v}{h_{fg}} \ln \left(1 + \frac{2\sigma}{r p_{sat}} \right)} \quad (4c)$$

d) Same as (c) with $2\sigma/p_{sat} r \ll 1$

$$T_v = \frac{T_{sat}}{1 - \frac{T_{sat} R_v}{h_{fg}} \frac{2\sigma}{r p_{sat}}} \quad (4d)$$

Other alternatives are as follows:

e) With Eq. (1) use an empirical fit of data for the saturation curve

$$\text{as } \log_{10} P(\text{psia}) = A + B/T(^{\circ}R)$$

With Eq. (1) this may be solved by trial and error, or with assumption

(c) above

$$\Delta T = \frac{T_{sat}^2}{+B} \log_{10} \left(1 + \frac{2\sigma}{r p_{sat}} \right) \left[1 + \frac{T_{sat}}{+B} \log_{10} \left(1 + \frac{2\sigma}{r p_{sat}} \right) \right] \quad (5)$$

for steam $A \approx 6.96$, $B \approx -3880.81$ at 14.7 psia; $A \approx 6.5728$, $B \approx -3608.17$ at 1000 psia.

f) With Eq. (1) use tabular values from properties tables for the saturation curve.

A comparison of results of the above calculation methods is given in the following table for $r = 0.0005$ inches and water at 14.7 and 1000 psia:

Method	p = 14.7 psia		p = 1000 psia	
	$T_{sat} = 212 \text{ F}$		$T_{sat} = 544.61 \text{ F}$	
Eq. 4a	$T_v = 216.6$	.60	$T_v = 544.66$	F
Eq. 4b	216.6	.63	544.66	
Eq. 4c	216.5	.50	544.68	
Eq. 4d	216.7	.71	544.63	
Eq. 5	216.5	.39	544.65	
(f) tables	216.5	.46	544.66	

The above equations represent an approximate expression for the superheat required for equilibrium of a bubble of radius r . Nuclei of radius greater than r from Eq. 5 should become bubbles and grow; those of smaller radius should collapse.

Some experiments have shown that at a heated surface in water at atmospheric pressure, boiling begins at around 30°F above saturation temperature. For this condition Eq. 5 predicts an equilibrium bubble radius of 10^{-4} in. This is about 10,000 times larger than the maximum cavity size expected from molecular fluctuations (14). Volmer (2) estimated a cavity formation rate of size 10^{-4} in. to be approximately one per cubic inch per hour. From this, we readily conclude that free vapor nuclei arising from molecular fluctuations are not important as nucleation cavities. On the other hand, gas nuclei will be very significant as nucleation cavities.

If inert gas molecules are present in the liquid and, hence, in the bubble, Eqs. 1 and 5 are modified as follows:

$$p_v - p = \frac{2\sigma}{r} - p_g \quad (6)$$

$$T - T_{\text{sat}} = \frac{R_v T^2}{p h_{fg}} \left(\frac{2\sigma}{r} - p_g \right) \quad (7)$$

which shows that the superheat required for a given size bubble to grow is decreased by the presence of the gas partial pressure, p_g .

The preceding calculation procedure relates to nucleation at a cavity in the heating surface in the following way. Griffith⁽⁵⁾,⁽⁷⁾ showed that the radius of curvature of a vapor interface for positions inside and outside of the cavity, Fig.2a, may be plotted as in Fig.4 as $1/r$ vs. vapor volume. In Eqs. (4) and (6) (7) the superheat required to cause a bubble to grow is determined by $1/r$. The maximum point in the $1/r$ curve, Fig.4, then determines the minimum superheat required for a bubble to grow at this cavity. The significant feature of Fig.4 is that while the shape of the curve depends on cavity geometry and contact angle β , the magnitude of r at the maximum point of the curve is equal to the radius of the cavity for any $\beta \leq 90^\circ$. Hence the cavity radius should determine the required superheat for nucleation for liquids which wet the surface.

Equation (8) has been verified experimentally by Griffith and Wallis⁽⁷⁾ for single cavities in a uniform temperature system.

It is possible for a cavity to continue to contain vapor even when the surface is highly subcooled. If the contact angle is such that the

curvature is as shown in Fig.2d, the vapor pressure may be very much less than the liquid pressure as the radius decreases down into the cavity. Here Eq.(2) has a minus sign, and vapor will be present at temperatures very much less than the normal saturation temperature of the liquid. Hence this cavity is immediately ready to produce bubbles on its next heating. In a similar way, re-entrant cavities, such as the one shown in Fig.2e, may also withstand high subcooling without collapsing the cavity. The cavity in Fig.2c would immediately fill with liquid upon cooling.

Incipient Boiling in Forced Convection

Consider a liquid flowing in a tube. As the heat flux or wall temperature is raised, boiling or nucleation begins at a particular value of the wall temperature. We wish to investigate this condition.

The following procedure was developed by Bergles⁽⁸⁾ based on a suggestion of Hsu and Graham⁽⁹⁾. In this flow inside the tube the expression for heat flux is

$$\frac{q}{A} = h(T_w - T_{liq}) = -k_l \left(\frac{\partial T}{\partial y} \right)_{y=0} \quad (8)$$

Here h is the heat transfer coefficient which is a function of the geometry, the fluid properties and the flow rate; k_l is the liquid thermal conductivity. For a given liquid temperature, both the temperature gradient $(\partial T / \partial y)_{y=0}$ and the wall temperature T_w increase as the heat flux increases. A series of curves representing the temperature distribution very near the heated wall is shown for increasing heat flux in Fig.5. Also shown in Fig. 5 is a curve labeled T_g^* , which is a plot of Eq. (4) with the radius of the cavity plotted as the distance from the heated surface. A possible theory: Nucleation takes place when the temperature curve in the liquid is tangent to the curve representing Eq. (4). The implication is that the surface contains cavities of various sizes (Fig.10) and when the temperature at the outer surface of the bubble reaches the critical value given by Eq.(4), the bubble grows at the cavity whose radius is represented by the distance between the wall and the point of intersection.

At the point of tangency, the radius of the first cavity to nucleate, solve Eqs. (4a) and (8) simultaneously, is

$$r_{nuc1} = \sqrt{\frac{2\sigma T v_{fg} k}{h_{fg} (q/A)}}$$

and the heat flux at incipient boiling is predicted by this postulate to be

$$\left(\frac{q}{A}\right)_{incip} = \frac{h_{fg} k}{8 \sigma T v_{fg}} (T_w - T_{sat})^2$$

For heat fluxes greater than this incipient condition, the radii which are active are on either side of the magnitude given by the above equation, intersections of the upper dashed line and the T_g^* curve of Fig. 5.

In commercially prepared surfaces a wide range of cavity sizes can be expected. Incipient boiling, then, should be independent of surface condition

for most commercially finished surfaces. The present analysis should, therefore, be valid in most practical applications. It is noted, however, that the position and slope of the remainder of the boiling curve should be dependent on the size range of active nuclei.

Equations (4) and (8) could be differentiated to determine the conditions at the point of tangency. However, this procedure does not lead to an explicit formulation for the incipient heat flux, $(q/A)_i$. Consequently, a graphical solution was found much more convenient. The results for $(q/A)_i$ were calculated for water over a pressure range of 15 to 2000 lb_F/in² abs and can be expressed quite accurately as follows:

$$(q/A)_i = 15.60p^{1.156}(T_w - T_s)^{2.30}/p^{0.0234} \quad (9)$$

where q/A is in Btu/hr ft², p is in lb_F/in² abs, and the temperatures are in deg F. Equation (9) is shown plotted for water in Fig. 6. Similar expressions may be calculated for other fluids.

It should be noted that although velocity does not appear in Eq. (9) the equation is valid for all fluid velocities.

As heat flux is increased above the condition for incipience, the upper dotted line, a range of cavity sizes as shown, become nucleated. A word of caution is in order here. The preceding calculation procedure presumes that cavities exist at sizes out to the point of tangency. At low heat fluxes, as in natural convection, the slope of the temperature distribution, Eq. (8), may be so small that the point of tangency occurs at cavity sizes larger than those present. Then the heat flux and wall superheat must be raised until the two curves intersect at the largest available active cavity. Hence for low heat flux cases as in natural convection, the procedure usually predicts superheats and (q/A) lower than the observed magnitude.

The intersection of the two curves of Fig. 5, Eqs. (4a) and (8), at the maximum available cavity, $r_{\max}$, gives

$$\left(\frac{q}{A}\right)_{\text{incip}} = \frac{k_l}{r_{\max}} (T_v - T_{\text{sat}}) - \frac{2\sigma T_v v_{fg} k}{h_{fg} r_{\max}^2}$$

With liquid metals, which have very high thermal conductivities, this same situation exists even in forced convection. For comparison, curves for both water and sodium are shown in Fig. 7 for incipient boiling conditions. Very much greater superheats are required in the liquid metals (10)

A number of other suggestions have been made for predicting the incipient boiling condition. Most of them reduce to the scheme shown in Fig. 5- except the T_l is plotted vs. y/n with T_g^* plotted vs. r . In the above n is taken as unity. In these other proposals the suggested magnitudes of n range from 0.67 to 2. Brown and Bergles⁽²⁶⁾ tried these suggestions with experimental data for various fluids and found n to range between 1 and 3.

The above procedure then appears to provide an estimate of the lower ΔT and $(q/A)_i$ at incipience. Actual ΔT and $(q/A)_i$ may be greater than these predicted magnitudes. These higher wall superheats may be required to provide sufficient bubble growth acceleration to remove it from the wall. Also thermocapillarity as described later may provide convection currents not accounted for in the above model.

Another study of nucleation (10a) attempted to account for flow pattern around the bubble surface protruding from the cavity, in forced convection. From comparison with only a small amount of data this study led to the following prediction for wall superheat in forced convection.

$$(T_w - T_{sat})_{incip} = 1.33 \left(\frac{q/A}{k} \right)^{0.4} \left(\frac{\tau_w}{\rho_l v^2} \right)^{0.1} \left(\frac{2\sigma T_s v_{fg}}{h_{fg}} \right)^{0.6}$$

More study of this with comparison with a wider variety of data is needed.

The thermal conductivity of liquid metals is so large that the temperature gradient in the liquid in the vicinity of the bubble at a cavity is essentially zero. Then the first cavity to nucleate should be at $r_{\max}$ if it is not inactive (filled with liquid). Then incipient boiling superheat should be predicted by Eq. (4-a) with $r = r_{\max}$.

Chen (10b) investigated the postulate that cavities could be deactivated to a predictable size by preconditioning by raising the pressure and temperature to specified levels of subcooling. This would establish a "negative" radius of the cavity from Eq. (4a) on the assumption the oxidation in the cavity would prevent further collapse to smaller radii. On subsequent reduction of pressure and increasing the degree of superheat, the radius, now taken as positive in Eq. (4a), should predict the superheat required for nucleation at the surface. The same formalism was re-explained (10c) in terms of re-entrant cavities, Fig. 2e, which eliminates the necessity of requiring oxidation to aid the process. Most test results fall below the predictions of this theory; however Singer and Holtz (10d) showed that this discrepancy was probably due to the presence of non-condensable gas. Their data showed that after extended periods of boiling to drive gas off the surface the measured superheats agreed well with the theory. Also surface finish had no effect on the results.

Possible Mechanisms of Heat Transfer in Boiling

As heat flux to a saturated liquid is increased beyond the incipient boiling point, the number of spots on the surface from which bubble columns rise increases. Photographs of Hsu and Graham suggest that between bubbles in pool boiling a thermal layer builds up and each bubble carries away with it the liquid in a region $2D_b$, thus pumping away the liquid superheat. In the region between the bubbles at the lower heat fluxes the heat transfer may be of the order of that associated with natural convection. Calculations based on this hypothesis were verified at low heat fluxes by Han and Griffith⁽¹¹⁾.

As heat flux is increased the nucleation site density increases and the columns with discrete bubbles change to columns of continuous vapor [Moissis and Berenson⁽¹²⁾]. The zones between these vapor columns become "return passages" bringing liquid to the heating surface where it is converted to vapor in the columns. As heat flux is increased further this situation becomes unstable and the heating surface becomes starved for liquid causing the transition to film boiling, point C of Fig. 1.

A typical history of the temperature at a cavity under a bubble is shown in Fig. 7a. At point A the bubble begins to grow on the surface. The major portion of the bubble grows in an essentially non-viscous way, except near the wall where viscosity predominates. Measurements of Cooper and Lloyd⁽¹³⁾ on toluene suggest that a thin liquid layer is left on the surface under the bubble, Fig. 7b. It is presumed that viscous force retards the motion of the liquid next to the wall as the bubble grows, thus leaving behind a liquid layer. At point B this layer has evaporated and the dry surface rises in temperature. At point C the bubble has departed and colder liquid covers the surface reducing its temperature to D. Then the surface continues to rise in temperature to point A when a new bubble forms. Calculations of heat flux from the surface suggest that major contributions to the heat flux occur while the thin liquid film evaporates under the bubble, AB, and when the colder liquid rushes into contact with the surface, CDA.

Cooper (13a) showed that this microlayer is significant only at low pressure. At pressures above 1 atm for fluids such as water, hydrocarbons and cryogenics the bubble growth is thermally controlled (not dynamically controlled); then the microlayer does not greatly influence the growth and the heat transfer.

Measurements (14) of temperature at a cavity in pool boiling of sodium suggest that for liquid metals the B-C portion of the curve is essentially non-existent and CD nearly coincides with AB. Here the primary mechanism of heat transfer is to the liquid which flows in behind a departing bubble. Because of the high thermal conductivity of the liquid metal the rate of rise of surface temperature is much slower and bubble frequencies are an order of magnitude less than in non-metals.

Boiling from a heated surface in highly subcooled liquids may be accompanied by a different effect which predominates. Cumo⁽¹⁵⁾ has observed photographically long "jets" of hot liquid streaming forth in front of bubbles forming at surfaces in subcooled liquid. From heat transfer measurements and photographs in highly subcooled liquids Brown⁽¹⁰⁾ suggests that because of evaporation at the hot side and condensation at the cold side of the bubble, a small temperature difference exists around the bubble. This results in a slightly lower surface tension at the outer edge of the bubble producing significant flow of the interface which induces large flows of the surrounding liquid to form the observed jets. Here the bubble appears to induce large convection currents which are heated by the hot surface. This process has been called thermocapillarity (10). It should have its most influence in boiling of subcooled liquids in forced convection, but is no doubt present in some degree at all times.

Thermocapillarity may tend to remove rapidly the thin layer of liquid

under a bubble in Fig.8b, reducing the time between points A and B, Fig.8a. Also the convection induced by thermocapillarity may require wall superheats higher than those predicted by Fig.6, as observed in some test data.

Probably all of the preceding mechanisms occur simultaneously and participate to varying degrees of importance in the heat transfer process under the various conditions. The boundaries where one or the other of the mechanisms predominate are not now well established.

Bubble Parameters

For liquids which wet the heating surface the size of bubbles at departure from the heating surface has been studied by a number of people. Fritz⁽¹⁶⁾ and Wark⁽¹⁷⁾ equated boundary and surface tension to determine the following expression for departure diameter:

$$D_b = C_d \beta \left[\frac{2 g_o \sigma}{g (\rho_l - \rho_v)} \right]^{1/2} \quad (10)$$

where C_d was found experimentally to be 0.0148 for H_2O and H_2 bubbles.

Mikic et al (19c) and Griffith and Lien (19d) studied bubble growth analytically and experimentally. Their results show clearly that all bubbles start out their growth dynamically controlled and in later stages become thermally controlled, vaporization being governed by heat conduction. Mikic (19c) shows that for a bubble growing at a wall at $T_w > T_{sat}$ into a liquid at $T_b < T_{sat}$

$$\frac{dR^+}{d\tau^+} = \left[\tau^+ + 1 - \theta \left(\frac{\tau^+}{\tau^+ + \tau_w^+} \right)^{1/2} \right]^{1/2} - (\tau^+)^{1/2} \quad (11)$$

where

$$R^+ \equiv R A/B^2; \tau^+ = \tau A^2/B^2$$

$$A \equiv \left[b \frac{(T_w - T_{sat}) h_{fg} \rho_v}{T_{sat} \rho_l} \right]^{1/2}; \quad B \equiv \left[\frac{12}{\pi} \frac{Ja^2 k_l}{\rho_l c_l} \right]^{1/2}$$

$$Ja \equiv \frac{(T_w - T_{sat}) \rho_l c_l}{h_{fg} \rho_v}; \quad \theta \equiv \frac{T_w - T_b}{T_w - T_{sat}}$$

Here for the bubble growing at the wall $b = \pi/7$. Mikic (19c) shows that the waiting time between bubbles may be approximated by

$$\tau_w = \frac{\rho_l c_l}{\pi k_l} \left\{ \frac{(T_w - T_b) R_c}{T_w - T_{sat} [1 + (2\sigma \sqrt{g}/R_c) h_{fg}]} \right\}^2$$

where R_c is the radius of the cavity. Mikic (19c) integrated Eq. (11) for various magnitudes of the parameters neglecting R_c , which is the bubble radius at $t = 0$, compared with R throughout the growth; the results are shown graphically in Fig. 8a.

For the dynamically controlled growth $t^+ \ll 1$ and $t_w^+ \gg t^+$,

Eq. (11) integrates to the following result

$$R^+ = t^+ \quad (11a)$$

For the thermally controlled region $t^+ \gg 1$, Eq. (11) and the curves of Fig. 8a are approximated by

$$R^+ = t^{+1/2} - \theta \left[(t^+ + t_w^+)^{1/2} - t_w^{+1/2} \right]$$

For a bubble growing in an initially uniformly superheated liquid ($t_w^+ \rightarrow \infty$) from an initial radius greater than the critical radius given by Eq. (4a), Eq. (11) integrates to

$$R^+ = \frac{2}{3} [(t^+ + 1)^{3/2} - t^{+3/2} - 1] \quad (12)$$

which is valid for both regimes of growth. Here $b = 2/3$ in A.

For the dynamically controlled region $t^+ \ll 1$, Eq. (12) becomes

$$R^+ = t^+ \quad (12a)$$

which is the Rayleigh solution, and is identical to Eq. (11a).

For the thermally controlled region $t^+ \gg 1$, Eq. (12) becomes

$$R^+ = \sqrt{t^+} \quad (12b)$$

which is the result obtained by Plesset and Zwick (19a) and Seriven (18).

Griffith and Lien (19d) performed definitive experiments for bubble growth in superheated water over a pressure range of 0.18 to 5.6 psia, superheats in the range 15 to 28 F, and $58 < Ja < 2690$. For pressures less than around 0.4 psia the duration of the bubble growth period remained in the dynamically controlled region, $R^+ = t^+$ and for pressures above around 5 psia the dynamically controlled region existed for only a short time and practically all of the growth was governed by heat conduction. Between these pressures the early stage of growth was dynamically controlled and the later stage thermally controlled. The mid-range of this region exists when $t^+ \approx 1$. The results agreed with the prediction of Eq. (12).

Applying Eq. (12) to bubble growth in sodium (19d) suggests that for superheats around 30 F the thermally controlled region governs for pressures above around 1.7 atm and the dynamically controlled region governs the pressures less than around 0.3 atm.

Experimental data for growth in non-uniform temperature fields for which waiting time t_w is recorded is limited but does agree (19c) with the prediction of Eq. (11a). Data for bubble growth on heated surfaces at reduced pressures do exist (20) but without observed magnitudes of t_w . These data fall below the prediction of Eq. (12), suggesting that Eq. (11) and Fig. 8a should apply and that the effect of waiting time on the data is significant.

The preceding solutions and results are for lower pressures where $\rho_v \ll \rho_l$. At higher pressures where bubble growth is thermally controlled Scriven showed the effect of going ^{to} higher pressures (significant magnitudes of ρ_v/ρ_l) is given by the curves of Fig. 8b where

$$R^+ = C t^+ \quad (12c)$$

Bubble departure diameters as given by Eq. (10) do not agree well with data. Cole and Rohsenow (20a) correlated the departure diameters D_b for various fluids as shown in Fig. 8c. The straight line portion of the two curves where $p/p_{crit} < 0.2$ is given by the following equations

$$\begin{aligned} \text{water:} \quad E_o^{1/2} &= (1.5 \times 10^{-4}) (Ja^*)^{5/4} \\ \text{other fluids:} \quad E_o^{1/2} &= (4.65 \times 10^{-4}) (Ja^*)^{5/4} \end{aligned} \quad (13)$$

$$\begin{aligned} \text{where} \quad E_o &\equiv \frac{g(\rho_l - \rho_v) D_b^2}{\rho_v \sigma} \\ Ja^* &\equiv \frac{\rho_l c_l T_{sat}}{\rho_v h_{fg}} \end{aligned}$$

The prediction of bubble frequency for growth at a heating surface is not well established. Early Jakob and Linke suggest $f D_b = 920$ ft/hr for hydrogen and water bubbles. Later Zuber and Prabhu and Garber (21) observed

$$f D_b = 1.18 \left(\frac{t_c}{t_c + t_w} \right) \left[\frac{\sigma g_c g(\rho_l - \rho_v)}{\rho_l^2} \right]^{1/4} \quad (14)$$

Cole (20b) showed from data that $1.18 (t_c/(t_c + t_w))$ ranged between 0.15 - 1.4 raising serious doubt regarding the validity of the above equation.

Later Ivey (20c) showed that the frequency-diameter relation depended on the regime of bubble growth.

Dynamically controlled $D_b f^2 = \text{constant}$

Thermally controlled $D_b f^{1/2} = \text{constant}$

In the intermediate region the exponent on f changes from 2 to $1/2$.

Mikic's analysis (19c) for the thermally controlled regime leads to

$$\frac{D_b f^{1/2}}{2 Ja \sqrt{\pi \alpha}} = \left(\frac{t_c}{t_c + t_w} \right)^{1/2} + \left(\frac{t_w}{t_c + t_w} \right)^{1/2} - 1$$

where t_c and t_w are bubble contact time and waiting time between bubbles, and $(t_c + t_w) = 1/f$. Over the range $0.2 < (t_c / (t_c + t_w)) < 0.8$. This becomes

$$D_b f^{1/2} = \frac{3}{4} \sqrt{\pi \alpha_L} Ja \pm 10\% \quad (14a)$$

In the dynamically controlled regime, Cole (20d) suggested

$$D_b f^2 = \frac{4}{3} \frac{g(\rho_L - \rho_v)}{C_{\text{Drag}} \rho_L} \quad (14b)$$

which for steam at 1 atm and $C_{\text{Drag}} \approx 1$ and $\rho_v \ll \rho_L$ ^{reduces to}

$$D_b f^2 = 1.32 g \quad (14c)$$

Information on f vs D_b is not well documented. The above relations should be considered to be approximate.

Effects of forced convection on D_b and f are not well established. Kowalczyk et al (20e) showed D_b to decrease nearly linearly with increase in velocity. No general prediction method is available.

As q/A increases the number of active nucleating cavities increase.
 Staniszeewski⁽²²⁾ observed

$$\frac{q}{A} \sim n^m \quad (15)$$

where $m = 1$ at low q/A and decreases to around $1/2$ at high q/A .

Factor Affecting Pool Boiling Data

In the following there is presented the results of experimental observations showing the effect on pool boiling data, Fig. 1, of changing the magnitude of various properties and conditions. Except as noted, all data discussed are for liquids which wet the heating surface.

Pressure and temperature difference have a marked effect on all regimes of boiling, Fig. 9 shows representative data. Imagine a cavity size distribution, number of cavities of a particular size range versus diameter of cavity, represented by the curve sketched in Fig. 10. The first cavity,

to be activated has an approximate diameter as calculated for the point of tangency in Fig. 5. Then, in accordance with Eq.(15), as q/A is increased, n must increase and more cavities must become activated at diameters spread on either side of D_{crit} (Fig.10). As shown in Fig.5, this requires greater wall superheat.

The slope of the q/A vs. ΔT curve expresses the change in ΔT necessary to increase the number of activated points sufficiently to accommodate the new heat flux. Although the slope is predominantly in the neighborhood of 3, observations are available with resulting slopes as low as unity for contaminated surfaces and as high as approximately 25 for clean, polished surfaces. The actual slope depends upon the uniformity or distribution of the size and shape of the nucleating cavities, Fig. 10.

Griffith and Wallis⁽⁷⁾ obtained data for a number of active nucleation sites vs. wall superheat and calculated the radius corresponding to this wall superheat from Eq.(5). For boiling water, methanol and ethanol on the same surface a single curve resulted for number of active nucleation sites vs. this radius, suggesting the existence of a characteristic cavity size distribution, Fig.10, for that surface.

Effect of Pressure

Pool boiling heat flux data plotted vs. wall superheat result in curves as shown in Fig. 9. At higher pressures the data lies at lower superheat magnitudes. Of significance in explaining this shift to lower wall superheat is that the fact that for all fluids, Eq.(4) predicts lower ΔT_{sh} required to activate given size cavities as pressure is increased. Figure 11 presents results for pool boiling of sodium showing the shift to lower ΔT_g with pressure.

Figure 9 also shows typical results for the effect of pressure on the critical (or peak) heat flux in pool boiling. This $(q/A)_{crit}$ typically goes through a maximum at a pressure well below the critical pressure.

Surface Coatings or Deposits

In addition to the effect described above in connection with aging there obviously will exist a temperature drop associated with conduction across the coating or deposit. Since coatings may be very thin and their properties not known, this additional temperature difference is usually included in the ΔT when plotting boiling data. Including this constant, conduction resistance causes q/A vs. ΔT curves to be farther to the right and at lower slopes than those for clean surfaces. Curves for surfaces with various coatings are spread apart even farther than those of Fig. 12.

Various investigators^(25,27,28,29) have reported data for the effect of surface deposits on critical heat flux. Though there is some disagreement, the following conclusions are suggested. Coated surfaces - mechanically or by oxidation - may have higher $(q/A)_{crit}$ for small diameter cylinders, by as much as a factor of 2 to 3; this increase seems to decrease as diameter increases, disappearing at diameter larger than 1/2 inch. No appreciable effect is noticed for flat plates.

For clean surfaces not prone to severe oxidation, such as chromel, silver, stainless steel, nickel etc., data for $(q/A)_{crit}$ scatter with a band of about $\pm 20\%$ for a given wire as well as for different materials⁽²⁸⁾.

Effect of Gases

Noncondensable gases dissolved in the water tend to come out of solution at the hot surface. In general they tend to move the q/A vs. ΔT curve to the left and to reduce the magnitude of $(q/A)_{crit}$.

Figure 15 shows the results when fresh liquid containing the gas is continually brought to the heating surface by forced convection. Gas bubbles may appear at temperatures well below the normal boiling point and additional convection, probably induced by thermocapillarity, may produce the higher heat fluxes as shown.

Figure 16 shows results for pool boiling. It is assumed that sand-blasting the surface caused air to be present in the nucleation cavities. As time goes on after heating is started, the boiling curve moves to the right to higher ΔT as the gas is pumped out of the cavities by the bubbles.

Another phenomenon which appears to be associated with the presence of gas in the nucleating cavities is a kind of unstable boiling or bumping, thus far observed only in boiling of liquid metals^(32,33). At moderate heat fluxes in nucleate boiling of sodium a thermocouple in the heating surface near the liquid-solid interface is observed to behave as shown in Fig. 17. The process seems to change back and forth between nucleate boiling and natural convection as active cavities become deactivated and reactivated. After a time during which gas was being removed from the cavity and the liquid, this oscillation ceases and steady natural convection takes place. In the absence of gas the performance follows the natural convection curve until the cavity is nucleated [Eq.(4)] and nucleate boiling occurs above this heat flux and the average ΔT is much lower, Fig. 18, though the surface temperature varies considerably under each bubble in liquid metals.

This minimum heat flux for nucleate boiling in the absence of gas is also the maximum heat flux for which the unstable boiling will occur when gas is present. This limit is predicted by Shah⁽³³⁾.

The effect of non-condensable gases on the critical heat flux in pool boiling appears to be most significant at high subcooling, Fig.21, and not very significant at temperatures near the normal boiling point.

Hysteresis

The overshooting of the wall temperature on increase of q/A in natural convection and significant drop in ΔT when boiling begins is known as a hysteresis effect. This overshooting is more pronounced with liquid metals, particularly alkali metals which wet practically all solid surfaces extremely well. This tends to deactivate cavities as discussed in connection with Fig.2c.

With liquid non-metals this overshooting occurs less frequently since cavities such as those of Fig. 2d are usually present and remain active cavities even after a system is cooled down. In non-metals if some of the cavities become deactivated, the hysteresis overshoot of Fig. 18 can occur

[Corty⁽²⁾]; however, on reducing the heat flux in the nucleate boiling region the data follows smoothly down the boiling curve to natural convection.

Size and Orientation of Heating Surface

There appears to be very little effect of size of wire or of orientation of the (q/A) vs. ΔT curve in the nucleate boiling region. This geometry effect, however, is pronounced in the non-boiling natural convection region which is the left asymptote of the curve and is predicted by the normal single-phase natural convection data. Also in the film boiling regime geometry influences the data, curves for smaller wire diameter lying to the left.

The critical heat flux (CHF) appears to be influenced more pronouncedly by geometry. There may be as much as a 25% variation in CHF in flat plate data - CHF for vertical heaters less than for horizontal^(34,35). The difference may be more for larger heater sizes. Critical heat flux appears to increase with wire size (28, 35, 36, 36a) from 0.01" up to 0.05 to 0.1 inches and level off beyond that size. For sizes below 0.01" down to 0.003" the magnitude of CHF levels off again (36a). In the range where CHF changes with D , the bubble sizes^{are} of the order of the wire diameter. Houchin and Lienhard^(36b) measured reduced critical heat flux magnitudes for thin ribbon heaters, thickness down to as low as 0.00035". Data for various thicknesses of aluminum, nickel, silver, tantalum and stainless steel was correlated with $pc\delta$ product where δ is the ribbon thickness with one side insulated. Data for water with D.C. heating decreased from $(q/A)_{crit} = 330,000$ down to 200,000 Btu/hr ft², when $pc\delta$ decreased from 0.06 to 0.0015 Btu/ft²F. They also found a more pronounced decrease with A.C. heating. For $pc\delta > 0.06$, both A.C. and D.C. gave the same $(q/A)_{crit} = 330,000$, independent of $pc\delta$.

Agitation

The effect of agitation such as with a propeller is quite similar to the velocity effect shown in Fig. 1. Curves for greater rpm are higher up on the graph. This effect was studied early⁽³⁷⁾ and more recently⁽³⁸⁾. Also the critical heat flux increases with increasing rpm. No attempt to

correlate this information is made, since the effect is greatly influenced by shape, size and position of the agitation.

Subcooling

The effect of subcooling on the position of the boiling curve appears to depend on the geometry for convection. Figure 19 shows data for water (39) boiling on an 0.0645 in-diam stainless steel tube. Here the boiling curves lie farther to the right as subcooling is increased. The left asymptotes are for natural convection and should have higher q/A for greater subcool at equal $(T_w - T_s)$.

In contradiction with this are results (40) for a horizontal flat nichrome heater, 4.25 in square 0.0165 in thick, shown in Fig. 20. Here the boiling curves lie farther to the left as subcooling increases.

The differences between the results of Figs. 19 and 20 may be due to different convection geometries or they may be due to different nucleating conditions of the surface.

Critical heat flux increases markedly as subcooling increases as shown in Fig. 21. Additional data are also available (42, 43, 44).

Non-wet Surfaces

When the liquid does not wet the heating surface, very large vapor bubbles form and cover much larger portions of the heating surface (45,46). The net result is very greatly increased magnitudes of $(T_w - T_s)$ required to transfer a given q/A . The critical heat is also very much reduced by factors of 10 to 20.

Gravitational Field

At the NASA laboratories, a ribbon heater immersed in a beaker of water was photographed during free-fall conditions, and the results were reported by Siegel and Usiskin⁽⁴⁸⁾. In each case the heat fluxes were in the nucleate boiling range under a normal gravity field. In the free-fall condition with the lower heat fluxes bubbles grew while remaining attached to the ribbon. At the higher heat fluxes a very large vapor volume formed around the ribbon. This suggests that nucleate boiling is essentially nonexistent under zero-g

Effect of Surface Finish

As expected surface finish can shift the position of the boiling curve markedly, probably because changing surface finish changes the cavity-size distribution curve, Fig. 10. The initial controlled experiments studying this effect were performed by Corty⁽²⁾. Figures 12 and 13 are typical of results for non-metals and metals. The data for the rougher surfaces lie to the left at lower wall superheat, presumably because active cavity sizes are smaller on the smoother surfaces.

The data in Fig. 14 show an interesting effect of direction of surface finishing (or scratch) orientation on boiling. Here axial and circumferential finishing of a horizontal tube produce quite different results at low heat fluxes. At low heat flux the liquid-vapor interface of bubbles moving circumferentially on the surface tend to fill the scratch or cavities with liquid, deactivating them. With axial scratches vapor tends to be trapped in the cavities, thus keeping them active. For this reason the horizontal scratches require lower superheat to produce the same heat flux. At higher heat fluxes this effect disappears because the bubbles tend to move perpendicularly away from the surface. Then the roughness effect of Figs. 12 and 13 predominates.

Of significance is the observation from Fig. 12 that surface finish does not influence the position of the data in film boiling, nor does it affect the position of the minimum heat flux in film boiling.

The influence of surface finish on the critical heat flux is shown in Fig. 12 to be rather small - about 20% reduction in $(q/A)_{crit}$ for very smooth surfaces.

Aging

Experience teaches that aged surfaces have higher required ΔT for a given q/A . On metallic surfaces a scale or deposit may form from the boiling liquid or a film may form from oxidation or other chemical reaction. In either case the vapor-trapping cavity may shrink, necessitating higher superheat for activation.

conditions.

Subsequently, Siegel and Usiskin added a small amount of friction to the free-fall system raising g to approximately 0.09. They reported verbally that under these conditions nucleate boiling appeared to continue throughout the fall. This indicates that only a small g field is needed to maintain nucleate boiling.

Merte and Clark⁽⁴⁹⁾ report the results of tests on a heated surface at the bottom of a pool. The system was placed in a centrifuge and rotated so that the resultant acceleration field was normal to the surface. Their boiling tests covered a range of 1 to 21 g with the results replotted as shown in Fig. 22. There seems to be very little effect on the position of the q/A vs. $T_w - T_{sat}$ curves at the higher heat fluxes. The displacement of the curves at the lower heat fluxes is probably due to the effect of superimposed natural convection effects.

Composite of Boiling Data

Temperature differences ($T_w - T_{sat}$) associated with nucleate boiling are small (1 to 100 F). In many applications this resistance is not of primary importance; hence an order of magnitude for this temperature difference is of interest. Figure 23 shows a composite of nucleate boiling data for various organic fluids and surfaces (50). Required superheat for boiling liquid metals is somewhat higher, Figs. 17, 18. The conditions for the 276 data points of Fig. 23 are combinations of the following:

1. Liquids Acetone, benzene, i-butanol, n-butanol, carbon disulfide, carbon tetrachloride, diethyl ether, diphenyl, ethanol, ethyl acetate, Freon-12_R, Freon-113_R, heptane, n-hexane, methanol, methyl chloroform, n-pentane, i-propanol, styrene, m-terphenyl, o-terphenyl, toluene.
2. Surfaces Brass, chromium plate, copper, gold, Inconel⁷¹, nickel, nickel plate, platinum, stainless steel, vitreous enamel, zinc (crystal)
3. Surface Treatment None; No. 36, 60, 120(lapped), 150, 200 and 320 grit polishes; mirror finish; acid and steel-wool cleaned; annealed and unannealed; fresh and aged.

4. Geometry 0.005 - 1.5" diameter cylinders (horizontal and vertical);
 3/4 - 3 3/4" diameter disks (vertical, horizontal, facing up);
 3/4 x 3/4" and 3/4 x 4" plates (vertical, horizontal, facing up and facing down).

Correlation of Pool Boiling Heat Transfer Data

Attempts have been made by various people to correlate pool boiling heat transfer data. The logic leading to the various forms of the correlations will be omitted here and the suggested equations presented along with comments of their applicability.

Referring to Figs. 12 and 13, one must inescapably conclude that any correlation equation which embodies only properties of the fluid (liquid or vapor) cannot be a "universal" correlation for all fluids or, for that matter, for any particular fluid. As a minimum the coefficient (and possibly even the exponents) must change in magnitude as the character of the solid surface changes, since the data for, say, Fig. 12 is all for the same fluid conditions only the solid surface condition has changed. This point cannot be over emphasized. This has lead to a great deal of confusion among boiling heat transfer researchers and users. At the present moment there is no satisfactory way to include quantitatively the effect of the solid surface in any correlation equation; in spite of this, some researchers present correlation equations with a fixed magnitude for the coefficient. The best that can be accomplished is to correlate the effect of pressure for a given fluid and solid surface.

Many of the proposed correlations were developed by analyzing a simplified model of boiling leading to some dimensionless groups of quantities. Various forms of Bubble Nusselt Numbers, Bubble Reynolds Numbers and Prandtl Number appear in these equations. One of the early correlations (Rohsenow⁵¹) employed such groups with the characteristic dimension D given by Eq. (10).

The equation proposed is

$$\frac{c_l (T_w - T_{sat})}{h_{fg}} = C_{sf} \left\{ \frac{q/A}{\mu_l h_{fg}} \left[\frac{g_o \sigma}{g(\rho_l - \rho_v)} \right]^{1/2} \right\}^r \left(\frac{c_l \mu_l}{k_l} \right)^s \quad (16)$$

where C_{sf} should be a function of the particular fluid heating surface combination. From Figs. 24a or 24b the exponent $r = 0.33$. A cross plot of $c_l \Delta T / h_{fg}$ vs. N_{Pr} for constant values of the ordinate shows $s = 1.0$ for water, but $s = 1.7$ for all other fluids. The final correlation is shown in Fig. 24c which

results in $C_{sf} = 0.013$ with a spread of approximately $\pm 20\%$. This process was repeated for other data with the results as shown in Table I.

Table I. Correlation Eq. (16) with $r = 0.33$

Surface-Fluid Combination*	C_{sf}	s
Water-nickel [52]	0.006	1.0
Water-platinum [53]	0.013	1.0
Water-copper [53]	0.013	1.0
Water-brass [54]	0.006	1.0
CCl_4 -copper [55]	0.013	1.7
Benzene-chromium [56]	0.010	1.7
n-pentane-chromium [56]	0.015	1.7
Ethyl alcohol-chromium [56]	0.0027	1.7
Isopropyl alcohol-copper [55]	0.0025	1.7
55% K_2CO_3 -copper [55]	0.0054	1.7
50% K_2CO_3 -copper [55]	0.0027	1.7
n-butyl alcohol-copper [55]	0.0030	1.7

*Numbers in brackets refer to source of data.

It should be emphasized that accurate values of fluid properties are essential to obtain a correlation or in using Eq. (16). Also the heating surface should be clean. The presence of a contamination or deposit of the heating surface can shift the relative position and slope of the curve thus changing r and s in addition to C_{sf} . It should also be noted that Eq. (16) includes a $g^{1/6}$ term which came from the expression for D in Eq. (10). As noted in Fig. 22, there appears to be no effect of g on the position of the boiling curve; therefore Eq. (16) should be used with $|s| = |g_0|$.

Later Forster-Zuber⁽⁵⁷⁾ used similar dimensionless groups, but with RR of Eq. (13) as the velocity times characteristic dimension in the Reynolds number and $2\sigma/\Delta\rho$ as the characteristic dimension in the Nusselt number.

$$Nu_* = A Pr_*^{n_1} Pe_*^{n_2} K_p^{n_3} K_t^{n_4} Ar_*^{n_5} \quad (19)$$

Authors (63a)	A	n_1	n_2	n_3	n_4	n_5
M.A. Kichigan and N. Yu. Tobilevich	$1.04 \cdot 10^{-4}$	0	0.7	0.7	0	0.125
S.S. Kutateladze	$7.0 \cdot 10^{-4}$	-0.35	0.7	0.7	0	0
V.M. Borishanskiy & F.P. Minchenko	$8.7 \cdot 10^{-4}$	0	0.7	0.7	0	0
G.N. Kruzhilin and Ye.A. Averin	0.082	-0.5	0.7	0	0.377	0
D.A. Zubuntsov	0.125	-0.32	0.65	0	0.35	0

$$Nu_* = \frac{h}{k} \left[\frac{g_o \sigma}{g(\rho_l - \rho_v)} \right]^{1/2}$$

$$Pe_* = \frac{q/A}{\alpha_t \rho_v h_{fg}}$$

$$K_p = \frac{p}{(g\sigma(\rho_l - \rho_v)/g_o)^{1/2}} \quad (20)$$

$$K_t = \frac{(\rho_v h_{fg})^2}{Jc_l T_s \rho_l [g\sigma(\rho_l - \rho_v)/g_o]^{1/2}}$$

$$Ar_* = \frac{g}{v^2} \left(\frac{g_o \sigma}{g(\rho_l - \rho_v)} \right)^{3/2} \left(1 - \frac{\rho_v}{\rho_l} \right)$$

Forster-Greif⁽⁵⁸⁾ modified the Forster-Zuber equation by not linearizing the Δp vs. ΔT relationship and suggested the following equation:

$$\frac{q}{A} \frac{1}{h_{fg} \rho_v} \left(\frac{2\sigma}{a_t \Delta p} \right)^{\frac{1}{2}} \left(\frac{\rho_L}{\Delta p} \right)^{\frac{1}{4}} = C_2 \left\{ \frac{\rho_L}{\mu} \left[\frac{c \rho_L (\pi a_t)^2 T_{sat}}{(h_{fg} \rho_v)^2 788} \Delta p \right]^2 \right\}^{\frac{5}{8}} Pr^{\frac{1}{3}} \quad (17)$$

where C_2 was suggested to be 0.0012 from data for water at 1 and 50 atm, n-butyl alcohol at 50 psia, analine at 35 psia, and mercury for 1 and 3 atm, Eq. (17) in the following form approximates the data:

$$\frac{q}{A} = K_{sf} \left(\frac{a_t c \rho_L T_S}{h_{fg} \rho_v \sigma^{\frac{1}{2}}} \right) \left[\frac{c T_S (a_t)^2}{(h_{fg} L \rho_v)^2} \right]^{\frac{1}{4}} \left(\frac{\rho_L}{\mu} \right)^{\frac{5}{8}} Pr^{\frac{1}{3}} (\Delta p)^2 \quad (18)$$

For this limited amount of data K_{sf} was suggested to be 43×10^{-6} . Recently this equation was compared with other data. For ethanol-chromium data of Cichelli-Bonilla⁽⁵⁶⁾ it was found that K_{sf} should be 82×10^{-6} and for water-platinum data of Addoms⁽⁵³⁾, $K_{sf} = 142 \times 10^{-6}$. Obviously the coefficient does indeed change in magnitude for various surface-fluid combinations. Equation (18) needs more extensive comparison with data.

Equation (18) is identical to the earlier Forster-Zuber equation, except that Δp and ΔT are related by Eq. (6). Also Eq. (18) yields a varying exponent of q/A vs. ΔT , increasing with ΔT in the range from 2 to 4. In many cases the curve cuts across natural boiling data at a lower slope.

Other correlations have been suggested by Gilmour⁽⁵⁹⁾ McNeilly⁽⁶⁰⁾ and Levy⁽⁶¹⁾. The Gilmour correlation contains a size effect not verified by experiment. The Levy procedure is dimensional and employs an empirical curve around which data scatter by a factor of five or more, which is about the same variation observed in C_{sf} of Table I. (Readers of the Levy paper as originally published should note the ordinate of its Fig. 4 should read multiplied by 10^{-5} instead of 10^{-6}).

Various Russian workers have suggested correlation equations in the following form (61a):

where

$$B = \frac{r_0^3}{2} \frac{2}{\sqrt{\pi}} \frac{g_0^{11/8}}{c^{9/8}} c_2^{5/3} c_3^{1/2} (c_1)^{-1} \quad (20c)$$

$$\phi^{m+1} = \frac{k_L^{1/2} \rho_L^{17/8} c_L^{19/8} h_{fg}^{(m-23/8)} \rho_v^{(m-15/8)}}{\mu_L [\rho_L - \rho_v]^{9/8} \sigma^{(m-11/8)} T_c^{(m-15/8)}} \quad (20d)$$

Here $C_2 = 0.00015$ for water and 0.000015 for other fluids (Eq. (13)) and $C_3 \approx 0.5$ (Eq. (14)); C_1 , r_0 and m are to be determined from the cavity size distribution Eq. (20a). Note that B is solely a function of cavity size distribution and ϕ is a function of fluid properties, except for the exponent m . Further if the cavity size distribution has a slope of n then the q/A vs ΔT curve should have a slope of $n+1$.

For most fluids $(q/A)_{sat} \ll (q/A)_b$ in Eq. (20b) and may be neglected.

Cavity size distribution for most surfaces for which data is available has not been measured. In these cases the boiling data may be used to determine m and B . Mikic (69b) shows that Eq. (20c) used in this way does correlate existing boiling data.

It should be emphasized again that the coefficient A in Eq. (19) is not a constant, but varies with the surface fluid combination.

Equation (16) has correlated the pressure effect for a variety of pool boiling data. The state of our knowledge is such that the coefficient C_{sf} must be determined from limited data for each fluid-surface combination. This, of course, is true for any of the other pool-boiling correlations.

It should also be emphasized that the actual metal of the surface is perhaps less important than the surface character as represented by the cavity-size distribution of Fig. 10, which is unknown for practically all of the surfaces tested.

Mikic (63b) has attempted to show how the cavity-size distribution for any surface influences the position of the q/A vs ΔT boiling curve in pool boiling. Starting with the description of the boiling process as outlined on pg. 7 and assuming that the number of cavities of radius greater than r is expressible by

$$n = C_1 \left(\frac{r_s}{r} \right)^m \quad (20a)$$

where r_s is the radius of the largest cavity present and C_1 , r_s and m are determined from cavity-size distribution measurements, Brown and Bergles (63c). The following expressions are obtained:

$$\frac{q}{A} = \frac{A_{nc}}{A} \left(\frac{q}{A} \right)_{nc} + \left(\frac{q}{A} \right)_b \quad (20b)$$

where A_{nc}/A is the fraction of the area where bubbles are not being formed, $(q/A)_{nc}$ is the natural convection heat transfer at this area and $(q/A)_b$ is given by

$$\frac{(q/A)_b}{\mu_L h_{sc}} \sqrt{\frac{\rho_L \sigma}{g(\rho_L - \rho_V)}} = B (\phi \Delta T)^{m+1} \quad (20c)$$

Correlation of Critical Heat Flux, Nucleate Boiling

As heat flux is increased in nucleate boiling, more nucleation sites become active. Equations for correlating the critical flux data emerge from models which suggest (a) as nucleation sites increase, a critical "bubble packing" is reached to produce vapor blanketing (65), (b) as nucleation sites increase, the liquid flow to the heating surface is restricted sufficiently to produce vapor blanketing (62) and (c) as heat flux and number of nucleation sites increase, bubbles in a column overtake each other producing a vapor column in which there are liquid droplets which fall back to the surface if horizontal. When the vapor velocity is sufficient to carry these liquid droplets away from the surface, vapor blanketing occurs (61a). Regardless of the model used, all correlations thus far proposed contain the relation between $(q/\Lambda)_{\text{crit}}/\rho_v h_{fg}$, an average vapor velocity normal to the surface and some function of, or measure of, pressure level. The following are some correlations which have been proposed for liquid non-metals.

Zuber (62) (modified):

$$\frac{(q/\Lambda)_{\text{crit}}}{\rho_v h_{fg}} = 0.13 \left[\frac{\sigma(\rho_l - \rho_v)g g_o}{\rho_v^2} \right]^{1/4} \left[\frac{\rho_l}{\rho_l + \rho_v} \right]^{1/2} \quad (21)$$

Here 0.18 agrees better with the data than 0.13 as originally suggested by Zuber.

Kutateladze (63):

$$\frac{(q/A)_{\text{crit}}}{\rho_v h_{fg}} = 0.16 \left[\frac{\sigma(\rho_l - \rho_v) g g_o}{\rho_v^2} \right]^{1/4} \quad (22)$$

Chang and Snyder (64):

$$\frac{(q/A)_{\text{crit}}}{\rho_v h_{fg}} = 0.145 \left[\frac{\sigma(\rho_l - \rho_v) g g_o}{\rho_v^2} \right]^{1/4} \left[\frac{\rho_l + \rho_v}{\rho_l} \right]^{1/2} \quad (23)$$

Rohsenow and Griffith (65):

$$\frac{(q/A)_{\text{crit}}}{\rho_v h_{fg}} = 143 \left(\frac{g}{g_o} \right)^{1/4} \left[\frac{\rho_l - \rho_v}{\rho_v} \right]^{0.6}, \text{ ft/hr} \quad (24)$$

which is dimensional but agrees well with most data.

The quantities in the right hand side of Eqs. (21, 22, 23) are essentially the same except very close to the critical pressure. The Zuber equation with 0.18 instead of 0.13 is recommended.

Lienhard and Shrock⁽⁶⁶⁾ and Lienhard and Watanabe⁽⁶⁷⁾ extended the Zuber type equation by applying the Law of Corresponding States, relating various parameters to conditions at the critical pressure and temperature. The suggested correlation is of the following form:

$$\frac{(q/A)_{\text{crit}}}{(g\sigma)^{1/4} \frac{p_c}{\rho_l - \rho_v} \left(\frac{8 p_c}{3RT_c} \right)^{3/4}} = f_1(R') \cdot f_2 \left(\frac{p}{p_c} \right) \quad (25)$$

where f_1 is a function of geometry alone and f_2 is a function of reduced pressure alone. Here

$$R' = L \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} \quad (26)$$

where L is a characteristic length like diameter for wires.

For liquid metals Noyes and Lurie⁽⁶⁸⁾ suggest

$$(q/A)_{\text{crit}} = 0.16 \rho_v h_{fg} \left[\frac{\sigma(\rho_l - \rho_v) g g_o}{\rho_v^2} \right]^{1/4} + K_{NL} \quad (27)$$

where $K_{NL} = 400,000$ for sodium (0.2 to 22 psia) data. Here the first group of quantities is the Kutateladze Eq.(22) and the K_{NL} is a constant which perhaps has different magnitudes for each fluid. Potassium (0.15 to 22 psia) data of Balzheiser and Colver⁽⁶⁹⁾ is correlated by Eq.(27) with $K_{NL} = 300,000$. In terms of pressure in psia, these equations become:

$$\text{Sodium: } (q/A)_{\text{crit}} = (4.0 + 1.75 p^{0.457}) \times 10^5 \quad (28)$$

$$\text{Potassium: } (q/A)_{\text{crit}} = (3.0 + 1.00 p^{0.444}) \times 10^5$$

The above equations apply to saturated liquid. For subcooled liquids Zuber⁽⁷⁰⁾ modified Eq. (21) as follows:

$$\frac{(q/A)_{\text{crit sub}}}{(q/A)_{\text{crit sat}}} = 1 + \frac{5.3}{\rho_v h_{fg}} \sqrt{k_l \rho_l c_l} \left[\frac{\sigma(\rho_l - \rho_v) g g_o}{\rho_v^2} \right]^{-1/8} \left[\frac{g(\rho_l - \rho_v)}{g_o \sigma} \right]^{1/4} (T_{\text{sat}} - T_{\text{sub}}) \quad (29)$$

which was tested and agreed with data for water and ethyl alcohol at pressures below 142 psia. Ivey and Morris⁽⁴²⁾ suggest a simpler relation

$$\frac{(q/A)_{\text{crit sub}}}{(q/A)_{\text{crit sat}}} = 1 + \frac{0.1}{\rho_v h_{fg}} \left(\frac{\rho_v}{\rho_l} \right)^{1/4} c_l \rho_l (T_{\text{sat}} - T_{\text{sub}}) \quad (30)$$

which was tested against and agreed with data for water, ethyl alcohol, ammonia, C Cl_4 , and iso-octane over a pressure range of 4.5 to 500 psia.

In eqs. (21) through (27) the $(q/A)_{\text{crit}}$ is predicted to vary with $g^{1/4}$. Test data for wires in water in a drop tower (48) fall on or above a curve of $(q/A)_{\text{crit}} \sim g^{1/4}$ in the range $0.02 < (g/g_o) < 1$. The data points

above the curve may be due to a transient condition resulting from the short drop time.. Adams⁽⁷¹⁾ using cylindrical graphite heaters in a centrifugal field found $(q/A)_{crit} \sim g^{0.15}$ for $1 < g < 10$ and $(q/A)_{crit} \sim g^{0.25}$ for $10 < g < 100$. Bessant and Jones⁽⁷²⁾ found the exponent to drop from 0.27 to 0.14 as pressure increased from 14.7 to 300 psia.

In the equations $(q/A)_{crit} \sim \sigma^{1/4}$ limited data of Adams⁽⁷¹⁾ shows for vertical heating surfaces (2 to 4 ins. high) $(q/A)_{crit} \sim \sigma$.

None of the correlation equations include an effect of the condition of the solid surface. Berenson⁽²⁵⁾ and Ivey and Morris⁽²⁸⁾ obtain less than a 20% variation in $(q/A)_{crit}$ for a wide variety of clean surface materials and surface finishes. Oxidized or coated surfaces which tend to increase wettability, tend to have higher magnitudes of $(q/A)_{crit}$, (28), (73).

Cryogenic Fluids, Pool Boiling

Cryogenic fluids - liquid oxygen, nitrogen, hydrogen, helium, etc. - in pool boiling behave like non-cryogenic fluids. The position of the q'' vs. $(T_w - T_s)$ curve depends strongly on the nature of the solid surface. A survey of such data is presented by Zuber and Fried⁽⁷⁴⁾, Richards et al⁽⁷⁵⁾ and Clark⁽⁷⁶⁾. Figure 25 shows a composite plot of a wide variety of pool boiling and also film boiling data for oxygen. Brentani and Smith⁽⁷⁷⁾ attempted to correlate all of this data with a modified equation of Kucateladze

$$q'' = 4.87 \times 10^{-11} \left[\frac{c_{pl}}{h_{fg} \rho_v} \right]^{1.5} \left[\frac{k_l \rho_l^{1.282} p^{1.75}}{\sigma^{0.906} \mu_l^{0.626}} \right] (T_w - T_s)^{2.5} \quad (31)$$

which represents the solid line through the central portion of the data. Quite obviously the coefficient of the equation must vary by a factor of 10 or so to accommodate all of the data. Similar composite data for nitrogen, hydrogen and helium (77).

Data for critical heat flux of the various cryogens are shown to be correlated (77), (78), (79)^(79a), rather well by Eqs. (21, 22 and 24) except at pressures approaching the critical pressure where the data falls significantly below the predictions.

Eq. (70) shows that a modified form of Eq. (16) also correlates the cryogenic boiling data

$$\frac{q/A}{h_{fg}} \left[\frac{g_0 \sigma}{\rho_l (\rho_l - \rho_v)} \right]^{-1/2} = 3.25 (10^5) \left[\frac{C_{pf} \Delta T}{h_{fg}} \left(\frac{T/T_c}{Pr_l} \right)^{1.8} \right]^{2.89} \quad (31a)$$

Here Eq. (16) was modified by including an additional pressure effect empirically in the form of T/T_c , the exponent on the Prandtl number was changed, and single magnitude of C_{pf} was selected.

Forced Convection

Forced convection here refers to flow inside of tubes, annuli, parallel plates, outside of rod bundles, etc. The flow is a two-phase flow of liquid and its vapor. Adiabatic flow occurs in a variety of flow regimes, but two-phase flow with heat addition usually occurs as shown in Fig. 26 which represents liquid entering subcooled where boiling takes place at the wall, called local boiling. At some place along the tube the liquid reaches the saturation point. Beyond this point the process is called bulk boiling. At low quality (below 5-8%) and in the subcooled region, the flow is bubbly flow. At higher qualities the flow becomes annular with a thin liquid layer on the wall and a vapor core, which may or may not contain significant amounts of liquid droplets, in the center. In the annular flow regime there may or may not be nucleate boiling. If the liquid film on the wall is thin enough, heat is transferred by conduction through the liquid and vaporization occurs at the liquid vapor interface. Under these conditions the wall temperature is below that required for nucleation or activation of the cavities. At higher heat fluxes nucleation will take place under these thin liquid layers at the wall. A more detailed study of flow patterns is given by Tippets⁽⁸⁰⁾. The simplified description above is adequate for classifying heat transfer data.

At some point down the channel the heat^{ed} wall may become dry. Beyond this point wall temperature usually rises significantly for a particular heat flux. This is the critical condition. The cause of the dryout may be intensive nucleation if in the nucleation regime, or tearing off of the liquid film by high velocity of the vapor which increases progressively down the channel.

We separate our discussion of forced convection boiling into the low quality and subcooled regions (essentially bubbly flow) and the high quality regions (essentially annular flow). The region beyond the critical condition is a mist-flow, dry-wall regime and is handled separately. This may be called film-boiling in forced convection.

Low Quality and Sub-Cooled Forced Convection

Data in this regime typically follow a pattern as shown in the upper left corner of Fig. 1. No boiling occurs to the left of the incipient boiling line, determined as discussed in convection with Figs. 5 and 6. This curve is applicable for any velocity and subcooling. Beyond this line nucleate boiling occurs and the curves, regardless of velocity or subcooling, appear to merge into a fully-developed boiling curve, the location of which is quite insensitive to the magnitude of either velocity or subcooling. Evidence of this is seen in Fig. 27 where data for various velocities and subcooling converge at high heat flux to a single fully developed boiling curve.

Shown on the same graph, Fig. 27, is data for pool boiling taken on the same surface obtained by cutting open one of the tubes axially and boiling on the inner surface (half of the tube), the outer surface being insulated. Note that the lower end of the pool boiling data appears to lie on an extension downward of the fully-developed boiling curve drawn with a 3 to 1 slope. In this case for pool boiling on the inner curved half tube it is suspected that induced convection effects caused the pool boiling data at the higher heat flux to have a slope greater than 3 to 1 and lie above the dashed curve. There is insufficient evidence to conclude that the pool boiling data will always lie on an extension of the fully-developed boiling curve.

The above observations lead to the following two suggestions for drawing the forced convection boiling curve for a particular velocity and subcooling. On a graph of q'' vs. $(T_w - T_s)$, Fig. 28, draw the forced convection non-boiling curve, q''_{FC} for a particular velocity and subcooling, Eq.(1). Then from procedure of Fig. 5, or for water, Fig. 6 or Eq.(9), locate the point of incipient boiling. Then draw the fully developed boiling curve, q''_B , if known, at a slope of 3. This may be approximated by the pool boiling curve, if known, for the same surface. Locate the magnitude q''_{Bi} immediately below the incipient boiling point. Revising a suggestion of Kutateladze⁽⁸¹⁾, Bergles and Rohsenow⁽³⁹⁾ suggest the following interpolation equation:

$$\frac{q''}{q''_{FC}} = \left[1 + \left\{ \frac{q''_B}{q''_{FC}} \left(1 - \frac{q''_{Bi}}{q''_B} \right)^2 \right\}^{1/2} \right] \quad (32)$$

which is a curve asymptotic to the q''_{FC} curve at the incipient boiling point and asymptotic to the q''_B at high heat flux. An alternative suggestion (39) is a simple additive or superposition equation having the same asymptotes

$$q'' = q''_{FC} + q''_B - q''_{Bi} \quad (33)$$

Either equation follows the data reasonably well.

The data in Fig. 14 show the strong effect of roughness and scratch orientation of when filling the system on the position of the boiling curve in pool boiling. Tests were performed on these same surfaces for forced convection in an annulus around the tube. In this case, Fig. 29, (Brown⁽²⁶⁾) the data show very little effect of surface finish in forced convection. This may result from the fact that the position of the incipient boiling point as predicted by the procedure of Fig. 5 is independent of surface finish so long as cavities exist of the size represented by the point of tangency in Fig. 5. This fact may be the explanation for why surface finish has less influence in forced convection boiling. Nevertheless the actual shape of the cavity-size distribution curve, Fig. 10, for a particular surface probably does influence the position of the fully-developed boiling curve in forced convection.

Based on the assumption that surface characteristics do not influence significantly the position of the fully-developed boiling curve in forced convection, the following two equations have been proposed for subcooled and low-quality forced convection of water:

$$\text{Jens-Lottes}^{(82)}: T_w - T_s = \frac{60}{e p / 900} \left(\frac{q''}{10^6} \right)^{1/4} \quad (34)$$

$$\text{Weatherhead}^{(83)}: T_w - T_s = (90 - 0.127 T_s) \left(\frac{q''}{10^6} \right)^{1/4} \quad (34a)$$

where T_s is in °F and p in psia. Both equations are shown in Fig. 30 along with some data (84). More recently Them et al (83a) suggested

$$\text{Them (83a)}: T_w - T_s = \frac{72}{e p / 1260} \left(\frac{q''}{10^6} \right)^{1/2} \quad (35)$$

which appears to be in better agreement with data, Fig. 29, even at low pressures.

High-Quality, Forced Convection

Farther along a flow passage, Fig. 26, at qualities above 5 to 10% the flow regime becomes annular. As more vapor is generated along the tube the vapor velocity progressively increases down the tube, thus increasing the heat transfer coefficient. This effect is shown in Fig. 31 for R-22 refrigerant. The effect is more pronounced at the higher mass flow rates. At high flow rates, high qualities and lower heat flux, the liquid layer on the wall may be thin enough to keep the wall temperature below the nucleation temperature. Then no actual boiling takes place. Heat is transferred by conduction across the liquid layer and evaporation takes place at the liquid vapor interface (86). Evidence of suppressed nucleation is shown in Fig. 32 by Bennett et al (87). The data to the left are at low quality and are in the nucleation regime. The right hand portion of the data for higher quality are in the no-nucleation regime and appear to be correlatable by the Lockhart-Martinelli two-phase flow parameter, X_{tt} .

Various investigators have suggested a variety of correlation equations which are gathered together in Table 1. Some equations refer the actual h in the two-phase region to h_{LO} or h_{LP} with correction factors F to account for nucleation. The Chen equation is recommended for the lower quality region and Dugler and Addoms for the higher quality regions.

As in the case of pool boiling a kind of hysteresis has been observed in forced convection boiling. Figure 35 shows some data obtained at CISE (10) for the mist-annular flow regime. On increasing the heat flux, the wall temperature follows the curve a-b interpreted as a non-boiling curve. At point b the wall temperature dropped to point c. Further heating went to point d and subsequent successive reduction in heat flux caused the data to follow down curve d-c-e, interpreted as a curve representing the presence of nucleation or boiling. A possible interpretation is that nucleation is delayed along the path a-b because cavities have become deactivated.

Author	Guerrieri and Talty(42)	Collier, Lacey and Pulling(35)	Wright(34)			
Fluid	methane, cyclohexane, benzene, pentane, heptane,	water	water			
Direction of flow	up (nat. circ.)	up	down			
Channel geometry	circular	annular	circular			
Diameter, D (in)	0.75, 1.0	0.625/0.866, 0.375/0.551	0.719, 0.472			
Length, L (ft)	6.0, 6.5	4.0, 2.42	5.67, 4.69			
Pressure level, p (psia)	atmospheric	15.5 - 80	15.8 - 68.2			
Mass flux, G/10 ⁵ (lb/hr ft ²)	around 3.5	0.987 - 7.94	3.96 - 25.2			
Heat flux, q" (Btu/hr ft ²)	2170 - 17,400	31700 - 252,000	13,800 - 88,000			
Quality range, x	0 - 0.116	0 - 0.659	0 - 0.19			
Recommended correlation	$\frac{h}{h_{LP}} = A \left(\frac{1}{X_{tt}} \right)^n F$ The correlations apply to "convective " regime only, in general. The correction coefficient F is intended to take into account the nucleation effects. <table><tr><td>$A = 3.4 \quad n = 0.45$ for $X_{tt} < 0.3$</td><td>$A = 2.065 - 3.772$ $n = 0.491 - 0.749$ Recommend : $A = 2.167$ $n = 0.699$</td><td>$A = 2.721 \quad n = 0.581$</td></tr></table> No recommendation for F $F = 0.187 \left(\frac{r_b}{\delta} \right)^{-5/9}$ r_b from $\Delta T = \frac{R}{M} \frac{T_w T_{sat}}{h_{fg}} \ln \left(1 + \frac{2\sigma}{r_b p} \right)$, $(dp/d\ell)_{TPF}$ calculated from Lockhart-Martinelli (friction only) $\delta = 10 \frac{\mu_\ell}{\rho_\ell} \sqrt{\frac{4 \rho_\ell}{(dp/d\ell)_{TPF} D_{gc}}}$ thickness of lam. film adj. the wall [lb_f/ft³]			$A = 3.4 \quad n = 0.45$ for $X_{tt} < 0.3$	$A = 2.065 - 3.772$ $n = 0.491 - 0.749$ Recommend : $A = 2.167$ $n = 0.699$	$A = 2.721 \quad n = 0.581$
$A = 3.4 \quad n = 0.45$ for $X_{tt} < 0.3$	$A = 2.065 - 3.772$ $n = 0.491 - 0.749$ Recommend : $A = 2.167$ $n = 0.699$	$A = 2.721 \quad n = 0.581$				
Properties evaluated at:	T_{sat}	$T_f = T_{sat} + 0.33 \Delta T_{sat}$	T_{sat}			
No of data points/runs	192 / 96	1542 / 153	1272 / 106			
Error in correlating data	88 % within ± 20 %	26.9 % (std deviation)	15.7 % (average)			
Eq. No.	36a	36b	36c			

Author	Dengler and Addoms (88)	Chen (97)
Fluid	water	water, methanol, cyclohexane, pentane, heptane, benzene
Direction of flow	up	up and down
Channel geometry	circular	circular and annular
Diameter, D (in)	1.0	large range
Length, L (ft)	20	large range
Pressure level, p (psia)	7.5 - 40	8 - 505
Mass flux, G/10 ⁵ (lb/hr ft ²)	0.44 - 10	large range
Heat flux, q" (Btu/hr ft ²)	2100 - 21,800	2000 - 760,000
Quality range, x	0 - 0.70	0 - 0.71
Recommended correlation	$\frac{h}{h_{LO}} = 3.5 \left(\frac{1}{X_{tt}} \right)^{0.5}$ for convective region only: $0.25 < X_{tt} < 70$ To take into account nucleation multiply by F : $F = 0.67 \left[(\Delta T - \Delta T_1) \left(\frac{\delta p}{\delta T} \right)_{sat} \frac{D}{\sigma} \right]_{T_w}^{0.1}$ if $F > 1$, = 1 otherwise $\Delta T_1 = 10 (\bar{V})^{0.3} \quad [\bar{V}] = \text{ft/sec}$ $\bar{V} = G\bar{v}/3600 \approx Gv_g/3600(1-\alpha)$ α from measurements or Lock-Martinelli	$h = h_{mic} + h_{mac} =$ $= 0.00122 \frac{k_l^{0.79} c_l^{0.45} \rho_l^{0.49} g_c^{0.25}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_v^{0.24}}$ $. (\Delta T)^{0.24} (\Delta p)^{0.75} S + h_{LP} F$ with : $\Delta p = \frac{\Delta T h_{fg}}{T_{sat} v_{fg}} \left(\frac{\text{diff. in sat. vap. press. corr. to}}{\Delta T = T_w - T_{sat}} \right)$ $g_c = 32.2(3600)^2$, S and F given in Fig.
Properties evaluated at :	T_{sat}	T_{sat}
No of data points/ runs	150	data of many investigators
Error in correlating data	85 % within ± 20 % for convective region	85 % within ± 15.1 %
Eq. No.	37	38

Author	Schrock and Grossman(71)	Wright (89)
Fluid	water	water
Direction of flow	up	down
Channel geometry	circular	circular
Diameter, D (in)	0.1162 - 0.4317	0.719, 0.472
Length, L (ft)	1.3 - 3.3	5.67, 4.69
Pressure level, p (psia)	42 - 505	15.8 - 68.2
Mass flux, G/10 ⁵ (lb/hr ft ²)	1.76 - 32.8	3.96 - 25.2
Heat flux, q" (Btu/hr ft ²)	60,000 - 1,450,000	13,800 - 88,000
Quality range, x	0 - 0.57	0 - 0.19
Recommended correlation	$\frac{h}{h_{LO}} = C_1 Bo + C_2 \left(\frac{1}{x_{tt}}\right)^{2/3}$ $C_1 = 7400$ $C_2 = 1.11$ for $0 < x < 0.50$ Correlation presented also in graphical form	$h = 4.192(Re_{LP})^{0.455} q''^{0.289} x^{0.379} Pr_L^{0.4}$ Wright presents a total of 12 different correlations which all fit his data. (See also page 1)
Properties evaluated at :	T _{sat}	T _{sat}
No of data points / runs	~900/~180	12/2 / 106
Error in correlating data	35 %	14.9 % (average) 10.8 % (average)
Eq. No.	39	40 41

Author	Pierre (99)	Altman, Norris and Staub (100)	Sachs and Long
Fluid	F-12, F-22 (F-11, CH ₃ Cl)*	F-22	F-11
Direction of flow	horizontal	horizontal	up
Channel geometry	circular	circular	annular
Diameter, D (in)	0.471 - 0.708	0.343	0.5/0.97
Length, L (ft)	11.7 - 27	8	1.0
Pressure level, p (psia)	atmospheric	atmospheric	atmospheric
Mass flux, G/10 ⁵ (lb/hr ft ²)		0.585 - 4.63	0.16 - 0.8
Heat flux, q" (Btu/hr ft ²)		0.198 - 25 ^o F sup.	5000 - 23,000
Quality range, x	0.9 or 11 ^o F sup.		-
Recommended correlation	$Nu = C \left((Re_{LO})^2 \left(\frac{J \Delta x h_{fg}}{L} \right)^{0.5} \right)^v$ for $x_{in} = 0 - 0.56$ $x_{out} = 0.08 - 1.0$ $Re_{LO}^2 K_f = 10^9 - 0.7 \times 10^{12}$ $C = 0.0009, v = 0.5$ for $x_{in} < 0.5$ $x_{out} = 1.0 - 11^o F \text{ sup.}$ $Re_{LO}^2 K_f = 10^9 - 0.7 \times 10^{12}$ $C = 0.0082, v = 0.4$ Valid for average h along the tube		$Nu = 0.020 Re_v^{0.8} Pr^{1/3} \left(\frac{D_2}{D_1} \right)^{0.5}$ where : the Nu is calculated using the hydr. diam. $D_2 > D_1$ $Re_v = \frac{V_v d_v \rho_v}{\mu_v}$ The actual velocity of the vapor phase, V_v, and the hydr. diam. of the vapor core, d_v, are obtained from void fraction measurements.
Properties evaluated at :	T_{sat}	T_{sat}	T_{sat}
No of data points/runs			27
Error in correlating data	-10% to +20% (-20%+35%)*		
Eq. No.	42	43	44

* using data of other investigators

General Definitions and Symbols For TABLE ISubscripts

ℓ	liquid
v	vapor
LO	all liquid flowing in tube
LP	only liquid phase flowing in tube
b	bulk
sat	saturation
i	at point of suppression of nucleation (or at incipience of nucleation)

$$\Delta T = T_w - T_b, \text{ where } T_b = T_{\text{sat}}$$

$$\Delta T_i = (T_w - T_b)_i$$

α Void fraction

$$Bo = \frac{q''}{h_{\ell} G} \quad \text{Boiling number}$$

$$Pr_{\ell} = \frac{c_{\ell} \mu_{\ell}}{k_{\ell}} \quad \text{Liquid Prandtl number}$$

$$Re_{LO} = \frac{G D}{\mu_{\ell}} \quad Re_{LP} = \frac{G (1-x) D}{\mu_{\ell}} \quad \text{Reynolds numbers}$$

$$Nu = \frac{h D}{k_{\ell}} \quad \text{Nusselt number}$$

$$K_{\ell} = \frac{J \Delta x h_{\ell}}{\Delta L}$$

$$h_{LO} = 0.023 \frac{k_{\ell}}{D} \left(\frac{G D}{\mu_{\ell}} \right)^{0.8} Pr_{\ell}^{0.4} \quad \text{Heat transfer coefficient}$$

$$h_{LP} = 0.023 \frac{k_{\ell}}{D} \left(\frac{G(1-x)D}{\mu_{\ell}} \right)^{0.8} Pr_{\ell}^{0.4} \quad \text{Heat transfer coefficient}$$

$$\frac{i}{X_{tt}} = \left(\frac{x}{1-x} \right)^{0.9} \left(\frac{v}{v_{\ell}} \right)^{0.5} \left(\frac{\mu}{\mu_{\ell}} \right)^{0.1} \quad \text{Martinelli parameter}$$

Critical Heat Flux in Forced Convection

Our ability to predict and describe the conditions at the critical heat flux is far inferior for boiling with forced convection than for pool boiling. Most test data are taken with uniformly heated test sections for flow inside round tubes, rectangular channels and annuli; and some data are taken for flow along rod bundles. Some data are available for non-uniform heating - cosine distribution and discontinuous distributions.

Typical of test results in the vicinity of burnout is the curve shown in Fig. 36 with four regions identified (102). Beyond a critical heat flux, the surface temperature begins to rise sharply and to oscillate significantly (Region III) until the wall is completely dry and film boiling is established (Region IV). The detailed description of what occurs along this curve is intimately associated with the two-phase flow regime that exists, the vapor quality and the total flow rate in a particular geometry. Regardless of the detailed description, the critical condition is associated with a sharp reduction in ability to transfer heat from the surface, for example, a reduced heat flux at a given temperature difference or an increased temperature difference at the same heat flux.

Imagine the curve of Fig. 36 to represent the test results taken at constant inlet enthalpy, constant flow rate and uniform heat flux in a channel of uniform cross section. Region I will usually be a region of no boiling, that is, ordinary single-phase forced convection. We focus attention on the exit condition. The description of Regions II and III depends largely on the flow rate.

At low flow rate it is possible for vapor to form at stray nucleation sites and, because the heat flux would be low, to establish annular flow where evaporation can continue at the liquid-vapor interface by pure conduction through the annular liquid film without nucleate boiling. In any case, Region II is characterized by an increased heat-transfer coefficient (decrease in slope of the curve) because of the increased velocity caused by the presence of the vapor. As the quality increases down the channel, the vapor velocity may become large enough to "tear" the liquid off the wall, resulting in the temperature oscillation associated with Region III. At sufficiently high

heat flux, the wall would be dry, and a fog flow would exist in the core, resulting in Region IV that might be called film boiling.

At higher flow rates, the condition in Region II is identified as nucleate boiling. If the flow rate is sufficiently high, the critical heat flux can occur at very low exit-vapor quality, in which case the mechanism in Region III is similar to that associated with pool-boiling burnout. Here, in Region IV, the core may be primarily liquid in bubbly flow with a vapor film separating the liquid from the wall.

These two cases perhaps represent extremes of all possible flowing conditions at the critical heat flux that might occur in nuclear reactors. In the low-flow case, it is possible that no nucleate boiling ever really existed, and the critical heat flux was in fact not a departure from nucleate boiling. Also, the heat fluxes under these flow conditions would be low enough to result in a wall temperature in Region IV that would be well below the melting point of the wall; hence, no physical burnout would result. At the other extreme, high flow rate, the critical heat flux would represent a condition just below a departure from nucleate boiling. Also, the heat flux would be rather high, causing the wall temperature in Region IV to be well above the melting point of the wall; hence, physical burnout would result.

In the light of the above discussion, terms such as "burnout", "departure from nucleate boiling", "maximum heat flux" and so forth should be discarded in favor of the more noncommittal "critical heat flux" defined as the heat flux just below the point, on a curve such as that of Fig. 36, where the wall temperature begins to rise sharply.

In the preceding discussion we identify the point that we would like to call the "critical heat flux point". Detecting and measuring the corresponding critical heat flux $(q/A)_{crit}$ is quite another matter. Some detectors respond to amplitude of temperature oscillation of a temperature-measuring device usually placed on the outside surface of the test section and close to the exit; others respond to rate of rise of such a temperature indicator; still others respond to rate of rise of the ratio of the voltage drop over the last quarter of the electrically heated test section to the voltage drop over the entire test section. Clearly, each of these detectors, set to be

tripped at various magnitudes of temperature oscillation or various magnitudes of rate of rise of temperature or voltage ratio, can indicate widely differing magnitudes of $(q/A)_{crit}$ for a particular test condition. The possible variation is not known, but a $\pm 10\%$ variation is not unreasonable to expect.

The magnitude of $(q/A)_{crit}$ seems to be seriously affected by the system dynamics of the entire test loop. It was demonstrated early in the testing programs (103) that even with subcooled liquids introduced at the inlet it is necessary to place a well-throttled valve just ahead of the test section in order to obtain reproducible results for $(q/A)_{crit}$. More recently (107), the quantitative reduction of $(q/A)_{crit}$ due to insufficient inlet throttling was measured in a particular system. In the late 1950's, data appeared for test sections with inlet conditions in the quality regime (104,105,106,107) by mixing liquid and vapor streams. Also, compressible volumes (108) were intentionally introduced at the test-section inlet. In both these latter cases, system dynamics or instabilities were found to influence significantly the magnitude of measured $(q/A)_{crit}$.

The heat transfer mechanism associated with the $(q/A)_{crit}$ is intimately related to the kind of two-phase flow regime existing at its occurrence.

All of these complicating effects have made it virtually impossible to describe adequately the conditions existing when $(q/A)_{crit}$ occurs and to obtain adequate correlations of the data for $(q/A)_{crit}$. Some correlation equations have been suggested for particular fluids (primarily water), specific geometries and for limited ranges of operating conditions.

Influence of System Instabilities on $(q/A)_{crit}$

The data shown in Fig. 37 were taken (108) on a once-through flow system with an expander vessel teed to the flow path just ahead of the test section. The upper curve is obtained when the expander vessel is filled with well-subcooled water, or is not in the system. In this case, the flow is quite stable. The lower curve is obtained when the expander vessel is either with a non-condensable gas, saturated vapor, or the test liquid at saturation temperature. In these cases, flow oscillates rather violently, presumably due to the compressible volume in the expander.

Another operating condition that produces similar results and instabilities is with the inlet to the test section in the vapor quality regime. The research groups at Harwell (England) and CISE (Italy) have operated test systems with inlet quality by mixing liquid and vapor streams ahead of the test section.

In this test system, the data for $(q/A)_{crit}$ are influenced greatly by the system oscillations and are not representative of the stable forced-convection-boiling processes. To show this, the CISE group ran tests at similar operating conditions but attempted to reduce or eliminate the system oscillations by placing orifices just ahead of the test section. Figure 38 shows these results. Curve A is obtained when there were no orifices between the test-section inlet and the point of mixing the liquid and vapor streams. Curve B is obtained when the two orifices are placed in series upstream of the heated section. The $(q/A)_{crit}$ in the dip region was increased probably because of the damping of the system oscillations. When five orifices were placed just ahead of the test section, the damping of the system oscillations, particularly as they influenced the flow in the test section, must have been nearly complete, because the resulting curve A is characteristic of data obtained when a system is in stable operation.

We conclude here that data for $(q/A)_{crit}$ taken in a system that is experiencing unstable flow is significantly lower than that which would be observed if a stable flow condition existed. The amount of the reduction in $(q/A)_{crit}$ will vary from system to system. The only data representative of the test-section boiling process that can be correlated or interpreted are data taken in stable operation.

Variables affecting Critical Heat Flux

An energy balance for a particular fluid (properties) at a pressure (p) and mass velocity (G) in a duct (L, D_e) with an inlet enthalpy (h_i) is

$$h_e - h_i = \frac{4}{GD_e} \int_0^L q'' dz \quad (45)$$

For the case of uniform heat flux this becomes

$$h_e - h_i = \frac{4L}{G D_e} q'' \quad (46)$$

Experiments to determine q''_{crit} for a particular fluid may be performed by setting the magnitudes of the following independent variables: L, D_e, G, h_i, p . Then the power to test section is raised until the critical condition is reached. At this condition the exit enthalpy may be considered to be $h_{e,crit}$. Then in functional form

$$h_{e,crit} = f_1(G, h_i, L, D_e, p) \quad (47)$$

or for uniform heat flux

$$q''_{crit} = f_2(G, h_i, L, D_e, p) \quad (48)$$

These two formulations assume the existence of the energy balance, Eq.(46) which may be used to replace any one of the variables in Eqs. (47) or (48). For example, h_i may be replaced by h_e using Eq.(46); then

$$q''_{crit} = f_3(G, h_e, L, D_e, p) \quad (49)$$

The tests for q''_{crit} could have been run taking h_e instead of h_i as an independent variable; then there would have been an $h_{i,crit}$ corresponding to q''_{crit} .

Other quantities, related to those above, may be useful in correlating data:

$\Delta h_i = h_{sat} - h_i$, inlet subcooling enthalpy

L_s , saturation length measured downstream from the position where the heat added would bring the fluid to the saturated liquid state at equilibrium

X_e , exit quality (related to h_e)

X_i , inlet quality (related to h_i); note X_i and X_e may both be negative if subcooled

W , total power added to heated section,

W_s , power added over length L_s

W_{crit} , power added to location where critical condition occurs

$W_{s,crit}$, power added from point where liquid is saturated to point where critical condition occurs.

Critical Heat Flux Data

Data obtained when flow oscillations exist is greatly influenced by the nature of the entire flow system. In order to exclude that data, Figs. 37, 38, only the data taken with subcooled inlet or very high quality inlet should be considered as providing stable, reproducible data independent of a particular flow circuit. This puts severe limitations on any test system in attempting to cover the entire range of variables desired.

A common way to present critical heat flux data for uniform q''_{crit} is shown in Fig. 39 as q''_{crit} vs. h_e (or X_e) with lines of constant G for L , D_e and p constant.

Tong⁽¹⁰⁹⁾ pointed out, that because two relations - the critical condition relations and the energy balance - must be satisfied, the magnitude of q''_{crit} cannot vary with one variable alone. In Fig. 39 h_i varies along each of the curves. Alternatively, then it is possible to plot the same data against h_i as in Fig. 40. Note the maximum X_e in Fig. 39 corresponding to saturated liquid at inlet. In order to obtain data beyond these points longer test sections must be used.

Tong^(109a) further shows the difficulty of showing trends in q''_{crit} because at least two variables always change simultaneously. Figures 41 and 42 show lines of constant tube diameter as parameters. One plot shows larger q''_{crit} for larger D , the other shows smaller q''_{crit} for larger D . The dashed line for $L = 60$ " in Fig. 42 is the same data on Fig. 41 at $G = 10^6$ lb/hr ft². Similarly Tong shows the different way q''_{crit} varies with pressure when h_e is held constant, Fig. 43, and when h_i is held constant, Fig. 44.

The preceding data was for water. Figure 45 shows data for R-12 refrigerant plotted as in Fig. 39 for water.

In the subcooled boiling (115) region - Figs. 46 and 47 show the effect of diameter and length on q''_{crit} in small diameter tubes. In Fig. 46 the data fall asymptotically to a constant value for $D > 0.25$ in.

Critical Heat Flux Correlations for Uniform q''

No forced-convection critical heat flux correlation has been evolved that applies universally to data for all fluids and all geometries. This discussion will first direct its attention toward correlations for a particular fluid and a particular flow cross-section shape and uniform heat

flux distribution.

The correlations suggested prior to 1958 are summarized by De Bertoli et al⁽¹¹⁶⁾ and Tong^(109b). Some are purely empirical, others attempt to include some fluid properties by resorting to a description of a postulated mechanism and a limited dimensional analysis. Most of these earlier correlations are limited to rather small ranges of variables, and some even show the wrong influence of flow rate at higher qualities. Here the discussion is limited to the more recently proposed correlations.

It should again be emphasized that much of the available data may have been taken with flow oscillations in the system, particularly that taken with inlet quality greater than zero, as shown in Fig. 37 and Fig. 38. These data should not be included in an attempt to correlate successfully the critical heat flux data.

Tong^(109c) reviews the various idealized physical models employed by various people; in attempting to arrive at a logical correlation equation. In the subcooled region these models focus on the forces acting on a bubble at the wall and on a superheat layer near the wall. In the mist-annular flow regime (quality greater than 5 to 10%) the critical condition is visualized to occur when the liquid film dries out or disappears from the wall. This occurs when the rate of evaporation plus the entrainment exceeds the rate of deposition of liquid droplets from the core. Obviously these processes are greatly influenced by the vapor velocity and take place either with or without actual nucleation at the wall. If nucleation (nucleate boiling) is taking place in the liquid layer, the resulting disruption of the surface of the layer will cause the wall to become dry of liquid at a vapor velocity lower than that required when nucleation is suppressed.

Tong^(109d) presents and reviews many of the proposed correlations, which have been developed primarily for water as the fluid. Only a few will be presented here.

Correlation of q''_{crit} , Subcooled Forced Convection

For subcooled liquid Gambill⁽¹¹⁷⁾ developed an additive correlation

$$q''_{crit} = q''_{b,crit} + q''_{c,crit} \quad (50)$$

where

$$q''_{b,crit} = 0.145 h_{fg} \rho_v \left[\frac{\sigma g_o g (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4} \left[1 + \left(\frac{\rho_l}{\rho_v} \right)^{0.923} \left(\frac{c_l \Delta T_{sub}}{25 h_{fg}} \right) \right] \quad (51)$$

$$q''_{c,crit} = h_c (T_{w,crit} - T_{liq}) \quad (52)$$

and $T_{w,crit}$ is evaluated from Bernath's⁽¹¹⁸⁾ relation

$$T_{w,crit} = 102 \ln p - 97 \left(\frac{p}{p+15} \right) - \frac{V}{2.22} + 32 \quad (53)$$

where p is absolute pressure in psia, and V is flow velocity in ft/sec assuming all liquids flow. The magnitude of h_c in Eq.(52) is the heat transfer coefficient assuming all liquids flow in the channel.

When compared with low pressure, small tube data for water (Dergles⁽¹¹⁵⁾) Eq. (50) predicted results too low.

Another empirical correlation for the subcooled region for water only was developed at Westinghouse APD, Tong⁽¹²¹⁾

$$q''_{crit} = (0.23 \times 10^6 + 0.094 G) (3 + 0.01 \Delta T_{sub}) \times (0.435 + 1.23 e^{-0.0093 L/D_e}) (1.7 - 1.4 e^{-a}) \quad (54)$$

$$\text{where } a \equiv 0.532 \left(\frac{\rho_l}{\rho_v} \right)^{1/3} \left[\frac{h_{sat} - h_i}{h_{fg}} \right]^{3/4}$$

Eq.(54) correlates 95% of the water data to $\pm 20\%$ for the following ranges of variables:

tube, rectangular channels, rod bundles (not annulus)

uniform heat flux

$G = 0.2 \times 10^6$ to 8×10^6 lb/hr ft²

$p = 800$ to 2750 psia

$L/D_e = 21$ to 365

$h_i > 300$ Btu/lb

$\Delta T_{sub} = 0.$ to $228F$

$D_e = 0.1$ to 0.54 in.

Correlation of q''_{crit} , Quality Regions

Macbeth Correlation

An attempt has been made to correlate the data for one particular fluid, water, using only those independent and dependent variables of Eqs. (46) to (49) for specific geometries (Macbeth⁽¹¹⁹⁾).

In order to correlate conveniently the data with Eqs. (46) to (49), it is necessary to seek simplification of the functional relation. By careful examination of the world's data, excluding all data points taken with inlet conditions in the quality range that are suspected of being associated with flow instabilities, Macbeth found the $(q/A)_{crit}$ to be related linearly with Δh_1 . The equation was further simplified by employing the local-condition hypothesis that $(q/A)_{crit}$ depends only on the conditions at the exit in a uniformly heated test section as it approaches the critical condition from the nucleate boiling or the wetted-wall condition. Then Eq. (49) should not include L ; therefore

$$(q/A)_{crit} = f_2(G, h_c, D, p) \quad (55)$$

Using data from carefully planned experiments at Winfrith, Barnett⁽¹²⁰⁾ shows these equations may be written in the following form:

$$(q/A)_{crit} = \frac{A + 1/4 CD(G \times 10^{-6})\Delta h_1}{1 + CL}$$

or

(56)

$$(q/A)_{crit} = A - 1/4 CD(G \times 10^{-6})h_{fg}x_e$$

where A and C are functions of D , G and p .

Macbeth⁽¹¹⁹⁾ applied these equations to practically all available data for uniformly heated round tubes and rectangular channels, using only data for which the inlet condition was subcooled, and no flow oscillations were reported. The coefficients A and C were determined empirically. Macbeth found he could present the results more simply by dividing the representative equations into two zones - high and low flow regions, as defined by Fig. 48.

In the low-flow regime for uniformly heated round tubes, the critical heat flux is given by

$$(q/A)_{crit} \times 10^{-6} = 0.00633 h_{fg} D^{-0.1} (C \times 10^{-6})^{0.51} (1 - x_e) \quad (57)$$

In the low-flow regime for uniformly heated narrow channels of width 1 in., the critical heat flux is given by

$$(q/A)_{crit} \times 10^{-6} = 0.264 h_{fg} S^{1.73} (G \times 10^{-6})^{-0.1} (1 - x_e) \quad (58)$$

where S is the internal spacing between the flat heating surfaces in a rectangular channel.

In the high-flow regime, $(q/A)_{crit}$ is given by Eq. (56) with A and C given by

$$A = y_0 D^{y_1} (G \times 10^{-6})^{y_2} \quad (59)$$

$$C = y_3 D^{y_4} (G \times 10^{-6})^{y_5}$$

where optimized y values are given in Table 3 for various pressures for water.

For rectangular channels, replace 1/4 by 0.555 and D by S in Eq. (55). The optimized values are given in Table 4.

In the range of reactor design, these equations appear to predict the $(q/A)_{crit}$ to well within 10% rms error; however, they should not be used beyond the range of the magnitudes of D, G, L and p for which data exist. Also, Δh_1 must be greater than zero. These equations are not applicable to tube bundles and annuli.

Table 3. Optimized y Values and Rms Errors for High-Velocity Regime Burnout Data for Round Tubes (Eq.(59))

Pressure* (psia)	y ₀	y ₁	y ₂	y ₃	y ₄	y ₅	Rms Error(%)	No. of Test Data Points
15	1.12	-0.211	-0.324	0.0010	-1.4	-1.05	13.8	88
250(nom.)	1.77	-0.553	-0.260	0.0166	-1.4	-0.937	4.7	237
530(nom.)	1.57	-0.566	-0.329	0.0127	-1.4	-0.737	5.7	170
1000	1.06	-0.487	-0.179	0.0085	-1.4	-0.555	7.4	405
1570(nom.)	0.720	-0.527	0.024	0.0121	-1.4	-0.096	3.4	133
2000	0.627	-0.268	0.192	0.0093	-1.4	-0.343	9.0	362
2700(nom.)	0.0124	-1.45	0.489	0.0097	-1.4	-0.529	4.7	37

Table 4. Optimized y Values and Rms Errors for High-Velocity Regime
Burnout Data for Rectangular Channels (Eq.(59)).

Pressures* (psia)	y_0	y_1	y_2	y_3	y_4	y_5	Rms Error(%)	No. of Test Data Points
600	23.4	-0.472	-3.29	0.123	-1.4	-3.93	6.1	22
800	0.445	-1.01	-0.384	0.0096	-1.4	-0.0067	12.9	28
1200	1.88	-0.081	-0.526	0.0035	-1.4	-1.29	4.9	42
2000	0.546	-0.315	-0.056	0.0027	-1.4	-0.725	9.4	359

*1 psia = 0.07031 kg/cm² = 0.06805 atm.

For uniform heat flux the group at CISE⁽¹⁰⁷⁾ suggest the following form of equation

$$q''_{crit} = (a - x_e) \frac{G h_{fg}}{4\beta} \quad (60)$$

where

$$a = \frac{1 - p/p_{cr}}{1.11(G/10^6)^{1/2}} ; a \leq 1 \quad (61)$$

$$b = 62 \left(\frac{p_{cr}}{p} - 1 \right)^{0.4} D^{0.4} (G/10^6)$$

Ranges of variables for which 84% of the data is predicted to $\pm 15\%$ are:

$$D > 0.28''$$

$$p = 640 \text{ to } 2130 \text{ psia}$$

$$G/10^6 = 0.74 (1 - p/p_a)^3 \text{ to } 2.95$$

$$x_1 = - (a/\beta)(L/D) \text{ to } 0.5a.$$

Note that Eqs. (60) and (56) have the same form but the coefficients are slightly different.

Tong⁽¹²¹⁾ also presents an empirical correlation for the region when exit quality is greater than zero.

$$h_{crit} - h_i = 0.529 \Delta h_i + (0.825 + 2.3 e^{-17D_e}) h_{fg} e^{-\frac{1.5G}{10^6}} - 0.41 h_{fg} e^{-0.0048L/D_e} - 1.12 h_{fg} \frac{\rho_v}{\rho_l} + 0.548 h_{fg} \quad (62)$$

where D_e is in inches, h in Btu/lb and G in lb/hr ft².

For 95% of the data a $\pm 25\%$, the following are the limits of variables considered:

Tubes, rectangular channels, rod bundles (not annuli)

$$G = 0.4 \times 10^6 \text{ to } 2.5 \times 10^6 \text{ lb/hr ft}^2$$

$$p = 800 \text{ to } 2250 \text{ psia}$$

$$L = 9 \text{ to } 76 \text{ inches}$$

$$D_e = 0.1 \text{ to } 0.54 \text{ inches}$$

$$h_f = 400 \text{ Btu/lb to sat.}$$

$$q''_{\text{crit}} = 0.1 \times 10^6 \text{ to } 1.8 \times 10^6 \text{ Btu/hr ft}^2$$

$$x_e = 0 \text{ to } 90 \%$$

Quite a number of other correlation equations have been proposed. A survey of many of these is presented by Tong^(109d). The ones selected for presentation here appear to agree with data over a wider range of variables. Nevertheless all of these correlations have shortcomings. Where possible it is best to use actual data for a system operating close to the desired conditions. Collections of such data are found in references (109d) and (121) through (124).

A Limiting Vapor Velocity

In the higher quality regions the simultaneous effects of evaporation, droplet deposition and entrainment suggest a critical magnitude of vapor flow. Some of the British data (125) suggest a critical $G_{v,\text{crit}} \approx 0.45 \times 10^6$ lb/hr ft² to $\pm 30\%$ for $G = 1 \text{ to } 3 \times 10^6$ and $p = 200 \text{ to } 1400$ psia. Mozharov⁽¹²⁶⁾ suggests the following dimensionless relation for this critical flow velocity:

$$\left(\frac{G_{v,\text{crit}}^2 D}{g_o \sigma \rho_v} \right)^{1/2} = 115 \left(\frac{x}{1-x} \right)^{1/4} \quad (63)$$

The actual process of liquid film dryout is undoubtedly much more complicated than these two simple relations suggest and is surely significantly influenced by whether or not nucleation is taking place at the wall.

Effect of Non-Uniform Heat Flux

The effect of non-uniform axial heat flux distribution on the critical condition is of interest in nuclear reactor and combustion chamber cooling jacket design. Tong^(109d) surveys the various test data. More recently Todreas⁽¹²⁷⁾ obtained data for a variety of flux distributions including discontinuous spikes for water at lower pressures in aluminum tubes. Figure 49 shows the location of the critical condition (actual tube rupture) for various flux shapes. Here M is the ratio of maximum-to-minimum heat flux along the tube.

With uniform heat flux the critical condition occurs at the outlet end of the tube, but with non-uniform flux it sometimes occurs upstream from the outlet.

For uniform heat flux since the critical condition occurs at the tube exit and since the energy balance Eq.(46) is a simple relation, an adequate correlation can be obtained in any of the forms of equations (47), (48) or (49), or for that matter in terms of total power, W , added instead of q'' . These formulations are essentially all equivalent for uniform flux distribution. This is not so for non-uniform flux distribution.

Earlier Tong^(121,109d) suggested that non-uniform flux critical condition in the quality regime could also be predicted by Eq. (62) where $h_{crit} - h_i$ is a measure of the integrated power added.

The group at CISE (107) showed that both uniform and non-uniform heat flux critical data may be correlated by use of boiling length, L_s , from where the liquid is calculated to be saturated to the dryout location and the total or integrated power W_s added over the boiling length L_s . The correlation equation is written as follows:

$$\frac{W_s}{w h_{fg}} = \frac{a}{1 + b D/L_s} \quad (64)$$

where a and b are given by Eq.(61) for water, w is flow rate, lb/hr and $W_s = w \times_{crit} h_{fg}$. For Refrigerant-12, at a reduced pressure of 0.27, Eq.(64) correlated the data with the 62 in Eq.(61) for b replaced by 118.

Following the completely empirical approach of Macbeth, Kirby⁽¹²⁸⁾

correlated available data for non-uniform heat flux as follows:

$$q''_{crit} = A - B X_{crit} \quad (65)$$

where q'' varies with length and q''_{crit} and X_{crit} are the appropriate magnitudes at the critical location. The data determined magnitudes of the constants expressed as

$$A = Y_1 G^{Y_2} D^{Y_3}$$

$$B = Y_4 G^{Y_5} D^{Y_6} \quad (66)$$

where the magnitudes of Y_n are given in Table 5.

Table 5 Optimized Constants

	560	1000	1250	1500	1600	1800	2000
Y_1	2.411	0.7468	0.988	0.7423	0.776	0.616	6.18
Y_2	-0.271	-0.1285	-0.191	-0.1057	-0.118	-0.155	0.231
Y_3	0.0*	-0.544	0.0*	-0.2335	-0.452	-0.293	2.907
Y_4	3.511	1.344	2.918	2.384	3.130	2.086	50.8
Y_5	0.345	0.526	0.287	0.6910	0.739	0.743	0.827
Y_6	0.0*	-0.2278	0.0*	0.0903	-0.296	-0.695	4.228
* No diameter variation							
No. of Expts.	34	269	22	79	98	80	69
Rms % Error	3.66	5.73	3.37	4.82	3.42	5.95	6.60

Overall Rms Deviation = 5.30%

The agreement with data is rather good. The following ranges of variables were covered:

$D = 0.37$ to 1.11 inches

$L = 39$ to 192 inches

$G = 0.25$ to 7.01×10^6 lb/hr ft²

WIR-A-47

Figure 50 shows a plot of heat flux variation along a tube and also a plot of incipient heat flux q''_i as calculated from, say, Eq.(8) and Fig. 5. Of course, this incipient heat flux would occur at a wall temperature different from that actually existing at any position, except where $q'' = q''_i$ at any point. The lower curve is a plot of the ratio of q''/q''_i vs. position along the tube. This can be placed anywhere on the $q'' = q''_i$ vs. h plot merely by changing the level of heat flux and the magnitude of the inlet enthalpy. This lower curve of Fig. 50 is an operating curve whose shape and position depends upon the actual flux distribution.

Todreas⁽¹²⁷⁾ found that the data at the actual burnout or tube rupture location for various flux shapes, including those with sharp spikes would all fall within a narrow band on a q''_{crit}/q''_i vs. h_{crit} at burnout as shown in Fig. 51. For q''_{crit}/q''_i below a magnitude around 1, no nucleation occurs and the burnout data appeared to fall on a nearly vertical line which for a given G is a particular value of G_v or vapor quality, lending support to the suggestion that there may be a critical G_v at which the liquid layer on the wall dries out. At higher magnitudes of q''_{crit}/q''_i , nucleation occurs and the burnout takes place at a lower h or x or G_v , suggesting that the presence of nucleation disrupts the liquid film and the wall dries out at a lower magnitude of G_v .

This same type of plot tends to show how burnout sometimes occurs at the tube exit and sometimes upstream. Since the operating curve can be placed anywhere on this plot there is shown in Fig. 51 two operating curves, a and b, for $G = 1 \times 10^6$. For curve a the critical local G_v (or h) is reached at an upstream position, nucleation being present. For curve b the critical G_v (or h) is reached at the end of the tube; here nucleation was suppressed. This same reasoning shows why tubes with heat flux spikes sometimes burnout upstream and sometimes at the exit.

The same phenomena was shown for flux distributions with spikes (129) for water data at high pressures.

Effect of Unheated Walls of h_{crit} .

For flow in an annulus with one wall unheated, or flow along clusters of rod bundles in an unheated container wall, the curve of q''_{crit} vs. h_e as in

Fig. 39 is shifted to lower h_e or quality. This is understandable, since the unheated wall contributes no vapor generation, hence, for a particular local critical condition at the heated wall the averaged vapor generation (or h_e) is less by an amount which depends on the ratio of the heated perimeter P_H to the total flow perimeter P . In fact Becker et al⁽¹³⁰⁾ found that critical heat flux data for various numbers of rod bundles and annulus sizes followed a definite linear pattern, Fig. 52, when the exit critical enthalpy is plotted vs. P_H/P . In Fig. 52 the ordinate is the ratio of the quality at the critical condition in the particular geometry to that in a round tube with the entire perimeter heated.

Liquid Metals

Critical heat flux data for liquid metals is very limited in quantity. Noyes and Lurie⁽⁶⁸⁾ present data for sodium in an annulus covering the range from around 70 F subcooled to around 15% exit quality. The resulting curves look very much like the lower curve of Fig. 37 suggesting the presence of severe instability. The lower curve in Fig. 37 resulted from a compressible volume at the inlet of the test section. The Noyes and Lurie system had a short heated length with a compressible volume in the condenser located very close to the exit. Non-condensable gases were allowed to collect in this condenser. It is quite likely that this compressible volume caused premature burnout as in Fig. 37.

Miscellaneous Effects

For flow of subcooled water in vertical annuli, Durant and Mirshak⁽¹³¹⁾ report increases in q''_{crit} by as much as 100%. For water in annuli in the high-quality, annular flow regime, Jansen⁽¹³²⁾ reports decreases in q''_{crit} of as much as 50% for sand blasted (300 microinch) surfaces. On the other hand, Moussez^(109e) reports a 20% increase in q''_{crit} when roughness was increased from 20 to 200 microinches.

In contrast to results for pool boiling, the effect of dissolved gas in flow boiling q''_{crit} has been found to be insignificant (109f).

As a means of delaying the film dryout twisted ribbons in tubes and vortex flow at the inlet have produced increases in q''_{crit} , (133 and 134).

Film Boiling

Film boiling as a cooling process has not had wide commercial applications because of the accompanying high surface temperatures; it may find use in the future as better materials become available. It is, however, often encountered in chemical process equipment and in cryogenic systems.

In film boiling, vapor is generated at the liquid-vapor interface by conduction and radiation from the heating surface through the vapor film. Using the hydrodynamic instability of the liquid-vapor boundary, Zuber⁽¹³⁵⁾ arrived at the following equation representing the minimum heat flux for film boiling:

$$(q/A)_{\min} = 0.09 \rho_{vf} h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\rho_l + \rho_v} \right]^{1/2} \left[\frac{g_o \sigma}{g(\rho_l - \rho_v)} \right]^{1/4} \quad (67)$$

where ρ_{vf} is the vapor density at average film temperature and the other properties are evaluated at saturation temperature. The coefficient 0.09 was determined empirically by Berenson⁽¹³⁶⁾ from data such as those shown in Fig. 12. It is significant that the data for any surface finish converge on this point, Fig. 12, and in the film boiling region are independent of surface finish. In the transition boiling region the data are influenced by the same factors that influence nucleate boiling data, suggesting that the film does occasionally touch the heating surface.

Extending Zuber's analysis, Berenson arrives at the following expression for ΔT at the minimum heat flux point:

$$\Delta T_{\min} = 0.127 \frac{\rho_{vf} h_{fg}}{k_{vf}} \left[\frac{g(\rho_l - \rho_v)}{\rho_l + \rho_v} \right]^{2/3} \left[\frac{g_o \sigma}{g(\rho_l - \rho_v)} \right]^{1/2} \left[\frac{\mu_f}{g_o (\rho_l - \rho_v)} \right]^{1/3} \quad (68)$$

The stable film regime was studied experimentally and analytically by Bromley^(137,138) for horizontal tubes and vertical plates. By balancing the buoyant and frictional forces on the vapor flowing in the film on the outside of a horizontal tube, Bromley arrived at the following equation representing the h_c associated with conduction alone:

$$h_c = 0.62 \left[\frac{k_v^3 \rho_v (\rho_l - \rho_v) (h_{fg} + 0.4 c_{pv} \Delta T)}{D_o \mu_v (T_w - T_{sat})} \right]^{1/4} \quad (69)$$

Radiation contributes to the heat transfer and increases the vapor film thickness, reducing the effective contribution of the conduction. The total heat transfer coefficient is given by

$$h = h_c \left(\frac{h}{h_c} \right)^{1/3} + h_r \quad (70)$$

where h_r is calculated for radiation between two parallel planes.

For forced convection flow of the liquid across the tube, Bromley⁽¹³⁸⁾ suggests the following equation when $V_\infty \geq 2\sqrt{gD_o}$:

$$h_c = 2.7 \sqrt{\frac{V_\infty k_v \rho_v (h_{fg} + 0.4 c_{pv} \Delta T)}{D_o (\Delta T)}} \quad (71)$$

and

$$h = h_c + \frac{7}{8} h_r \quad (72)$$

Film boiling on a horizontal surface was analyzed by Berenson⁽¹³⁶⁾ and compared with data for pentane (Fig.12), CCl_4 , benzene, and ethyl alcohol. The following equation results:

$$h = 0.425 \left[\frac{k_v^3 \rho_v (\rho_l - \rho_v) g (h_{fg} + 0.4 c_{pv} \Delta T)}{\mu_f (\Delta T) \sqrt{\frac{g_o \sigma}{g (\rho_l - \rho_v)}}} \right] \quad (73)$$

Note this is identical in form with Eq.(69) with D_o replaced by $\sqrt{g_o \sigma / g (\rho_l - \rho_v)}$ which is proportional to bubble diameter, Eq.(10).

WMR-A-51

Data for film boiling of cryogenic horizontal cylinders was correlated (77) by the Breen & Westwater⁽¹³⁹⁾ equation:

$$(q/A)_{\text{cond}} = \left\{ 4.94 \frac{(\rho_l - \rho_v)}{\sigma^{1/8}} + 0.115 \left(\frac{\sigma^{3/8}}{D(\rho_l - \rho_v)^{1/8}} \right) \right\} \left(\frac{k_v^3 h'_{fg} \rho_v}{\mu_v} \right)^{1/4} (T_w - T_s)^{3/4}$$

$$h'_{fg} = \frac{[h_{fg} + 0.340 c_{pv} (T_w - T_s)]^2}{h_{fg}} \quad (74)$$

The minimum heat flux in film boiling on horizontal cylinders is correlated (77) by the Lienhard and Wong⁽¹⁴⁰⁾ equation:

$$(q/A)_{\text{min cond}} = 0.114 \left(\frac{h_{fg} \rho_v}{D} \right) \left[\frac{2g(\rho_l - \rho_v)}{\rho_l - \rho_v} + \frac{4\sigma}{D^2(\rho_l - \rho_v)} \right]^{1/2}$$

$$\left[\frac{g(\rho_l - \rho_v)}{g_o \sigma} + \frac{2.0}{D^2} \right]^{-3/4} \quad (75)$$

Radiation for $T < 425^\circ\text{R}$ is usually $< 5\%$ of q''_{total} .

For quenching of spheres Frederking and Clark⁽¹⁴¹⁾:

$$\frac{hD}{k} = 0.15 \text{ Ra}^{1/3} \quad (76)$$

For $\text{Ra} > 5 \times 10^7$ where

$$\text{Ra} \equiv \left[\frac{D^3 g(\rho_l - \rho_v)}{\nu_v^2 \rho_v} \text{Pr}_v \left(\frac{h_{fg}}{c_{p,v}(T_w - T_{\text{sat}})} + 0.4 \right) \frac{a}{g} \right]$$

where a is local acceleration, equal to g for a stationary sphere on earth.

In flowing forced convection systems heat transfer beyond the point where the liquid film dries out is essentially a film boiling process. Here the vapor flows down the tube carrying along liquid droplets. The process is described by Quinn⁽¹⁴²⁾ in Fig. 53. In the region of the dry-out the liquid film oscillates back-and-forth with accompanying temperature fluctuations - the transition region. In the next region the heat flux and wall temperatures are steady and the ratio of $q''/(T_w - T_{sat})$ may go through a minimum depending on the flow rate.

The process of film boiling was studied in more detail by Lavery⁽¹⁴³⁾ and Forslund⁽¹⁴⁴⁾ for nitrogen. Typical measurements of wall temperature are shown in Fig. 54 for various heat fluxes. The curves merge asymptotically to the wall temperatures predicted for 100% vapor flow far downstream well beyond the point where enough heat has been added to evaporate all of the vapor. Clearly the vapor must have been superheated with liquid droplets entrained. The solid line asymptotes are the predicted wall temperatures for 100% vapor flow and the arrows show the location where the quality (X_E') would have been 100% had there been no superheating of the vapor. These curves plus visual observations show that liquid droplets exist far downstream of the $X_E' = 1.00$ point; hence significant vapor superheat was present.

Since vapor is generated along the tube the vapor flow accelerates and drags the liquid droplets along. Forslund⁽¹⁴⁴⁾ predicts wall temperatures and heat fluxes along the tube by calculating the heat transfer from the wall to the vapor and then from the vapor to the droplets. In the accelerating vapor stream, the droplets are taken to break up at a critical Weber Number, $(V_v - V_l)^2 \rho_v D / \sigma = 7.5$, which determines local droplet diameters with drag coefficients on spheres. The prediction process involves a stepwise calculation of all conditions down the tube. Calculated wall temperatures, droplet diameters and vapor superheat agree well with observations. Since the calculation procedure is somewhat complicated it is not reproduced here.

Knowledge of heat transfer in this beyond-dryout region is limited. Data over limited ranges of variables have been correlated by simple dimensionless equations, but no universally applicable prediction scheme has

evolved other than the calculation procedure by Forslund(144).

Additional data and empirical correlations applicable over limited ranges of data are summarized by Tong(109) and Silvestri(145).

Other Surveys of Boiling Heat Transfer

The reader is referred to the following other surveys of Boiling Heat Transfer by Gouse(1), Rohsenow(147), Tong(109), Silvestri(145), Leppert and Pitts(146), Westwater(148), Dwyer(149).

A	area of heating surface
A_b	surface area of a bubble
α_t	thermal diffusivity, $k/\rho c_p$
C_{sf}	coefficient of Eq.(16) which depends on the nature of the heating-surface-fluid combination
c_l	specific heat of saturated liquid, $\text{Btu}/\text{lb}_m \text{ } ^\circ\text{F}$
D	tube diameter
D_b	diameter of the bubble as it leaves the heating surface, ft
D_e	equivalent diameter = $4(\text{flow area})/\text{perimeter}$
f	frequency of bubble formation, 1/hr
G	mass velocity in a channel or tube, $\text{lb}_m/\text{hr ft}^2$
G_b	mass velocity of bubbles at their departure from the heating surface, $\text{lb}_m/\text{hr ft}^2$
g	acceleration of gravity
G_o	conversion factor, $4.17 \times 10^8 \text{ lb}_m \text{ ft}/\text{hr}^2 \text{ lb}_f$
h, h	enthalpy, Btu/lb
h_{fg}	latent heat of evaporation, Btu/lb_m
h	$(q/A)/\Delta T$, film coefficient of heat transfer, $\text{Btu}/\text{hr ft}^2 \text{ } ^\circ\text{F}$
J	778 ft lb/Btu
J_a	Jakob Number, $\rho_l c_l (T_h - T_{sat})/\rho_v h_{fg}$
k_l	thermal conductivity of saturated liquid, $\text{Btu}/\text{hr ft } ^\circ\text{F}$
L	length
M	molecular weight
Nu	Nusselt number = hd/k
Nu_b	bubble Nusselt number, defined by Eqs.
n	number of points of origin of bubble columns per ft^2 of heating surface
Pr_l	Prandtl number = $C_l \mu_l / k_l$
P_c	critical pressure
P_L	pressure on liquid side of interface of radius r
P_v	pressure on vapor side of interface of radius r
$q/A, q''$	heat transfer rate per unit heating surface area, $\text{Btu}/\text{hr ft}^2$
$(q/A)_b$	heat transfer rate to bubble per unit heating surface while bubble remains attached to the surface, $\text{Btu}/\text{hr ft}^2$
R	gas content; also bubble radius
Re_b	bubble Reynolds number, defined by Eqs.

Nomenclature

r	radial distance
T_b, T_{liq}, T_l	temperature of liquid, °F
T_w	temperature of heating surface, °F
T_{sat}, T_s	saturation temperature, °F
t	time
v	volume ft ³ /lb
W, w	flow rate, lb/hr
X_{tt}	two phase flow parameter, defined in Eq. (36)
z	depth
β	bubble contact angle, defined in Fig.2
$\Delta T_{subcool}$	$T_{sat} - T_{liq}$
α	thermal diffusivity, ft ² /hr
μ_l	viscosity of saturated liquid, lb _m /ft hr
ρ	density of saturated liquid, lb _m /ft ³
ρ_v	density of saturated vapor, lb _m /ft ³
σ, σ_{lv}	surface tension of liquid-vapor interface, lb _f /ft
σ_{sl}	surface tension of solid-liquid interface, lb _f /ft
σ_{vs}	surface tension of vapor-solid interface, lb _f /ft
ν	kinematic viscosity, μ/ρ

35. Bernath, L., "A Theory of Local-Boiling Burnout and its Application to Existing Data," Chem.Eng. Prog. Symp.Series. 56, No.30, 95-116 (1960).
36. Costello, C., Bock, C.O. & Nichols, C.C., "A Study of Induced Convective Effects on Saturated Pool Boiling Burnout" Nat. Ht/Trans. Conf. Cleveland, AIChE preprint no. 7. (1964).
- 36a. Pitts, C.C. & Leppert, G., Int. Jrl. of Heat & Mass Trans., Vol 9, 365-377, April 1966.
- 36b. Honchin, W.R. & Lienhard, J.H., "Boiling Burnout in Low Thermal Capacity Heaters", ASME Paper 66-WA/HT-40, Nov. 1966.
37. Austin, L, Mitt. Forsch, 7, 75 (1903).
38. Pramuk, F.S. & Westwater, J.W. Chem.Eng. Prog. Symp.Ser. 52, No.18, 79 (1956).
39. Berjes, A.E. & Rohsenow, W.M., ASME Trans. J. of Heat Trans., Aug.1964.
40. Duke, E.E. & Shrock, V.E., Fluid Mech & Ht. Trans. Inst., 130-145, June '61.
41. Ellion, M.E., "A Study of the Mechanism of Boiling Heat Transfer" Jet Propulsion Lab. Memo 20-88, C.I.T. (1954).
42. Ivey, H.J. & Morris, D.J., "On the Relevance of the Vapor-Liquid Exchange Mechanism for Subcooled Boiling Heat Transfer at High Pressure," UK Report AEEW-R-137, Winfrith (1962).
43. Kutateladze, S.S., Otdelenie Tekhnicheskikh Nauk, No. 4, pp 529 - 536 (1951).
44. Kutateladze, S.S. & Schneiderman, L.L. "Experimental Study of Influence of Temperature of Liquid on Change in Rate," pp 95-100 "Problems of Heat Transfer during a Change of State", USAEC Rep. AEC-tr-3405(1953).
45. Jakob, M., & Fritz, W., Forsch, Giebete, Ing. 2, 434 (1931).
46. Averin, E.K., AERE Lib/Trans. 562, From Izvest. Akad.Nank. SSSR(OTN) No. 3, pp 116-122, 1954.
47. Jakob, M. & Linke, W., Forsch. Giebete.Ing. 4, 75 (1933).
48. Siegel, R., & Usiskin, C., "Photographic Study of Boiling in Absence of Gravity" Trans. ASME, J. Heat Trans 81, 3(Aug.1959).
49. Merte, J.Jr., & J.A. Clark, "Study of Pool Boiling in an Accelerating System" U. Mich., Rep. No. 2646-21-T, Tech. Rep. No. 3(Nov.1959)
50. Armstrong, R.J., "Temperature Difference in Nucleate Boiling", Int. J. of Heat & Mass Trans., 9, 1148-9, Oct. 1966.

51. Rohsenow, W.M., "A Method of Correlating Heat Transfer Data for Surface Boiling of Liquids" ASME Trans. No. 48 (July 1952).
52. Rohsenow, W.M. & Clark J.A., "Heat Transfer and Pressure Drop Data for High Heat Flux Densities to Water at High Sub-Critical Pressures", Heat Transfer Fluid Mech. Inst., Stanford, Calif. (1951).
53. Addoms, J.N., "Heat Transfer at High Rates to Water Boiling Outside Cylinders," D.Sc. Thesis, Chem. Eng. Dept. MIT (June 1948).
54. Cryder, D.S., & Finalborgo, A.C., "Heat Transmission from Metal Surfaces to Boiling Liquids: Effect of Temperature of the Liquid on Film Coefficient" Am. Inst. Chem. Eng. 33, 346 (1937).
55. Piret, E.L. & Isbin, H.S., "Two-phase Heat Transfer in Natural Circulation Evaporators," AIChE Heat Trans. Symp., St. Louis (Dec.1953).
56. Cichelli, M.T. & Bonilla, C.F., Trans. AIChE, 41, 755-87 (1945).
57. Forster, K. & Zuber, N., AIChE Jrl. 531-535 (1955).
58. Forster, K. & Greif, R., "Heat Transfer to a Boiling Liquid: Mechanism and Correlations," Prog. Rep. No. 7, Dept. of Eng., UCLA, L.A. (1958).
59. Gilmour, C.H., "Nucleate Boiling - A Correlation," Chem.Eng.Prog.54, No. 10, 77-79 (1958).
60. McNeilly, M.J., "A Correlation of the Rates of Heat Transfer to Nucleate Boiling Liquids", J. Imp.Coll.Chem. Eng. Soc. 7, 18-34 (1953).
61. Levy, S., "Generalized Correlation of Boiling Heat Transfer," Trans/ASME, Ser.C., J. Heat Trans. 81, 37-42 (1959).
- 61a. Wallis, G.W., Int. Heat Trans. Conf., Boulder, Colo. Aug. 1961, Vol.II.
62. Zuber, N., "Hydrodynamic Aspects of Boiling Heat Transfer," USAEC Rep. AECU-4439 (1959). (Ph.D. thesis, U. of Calif. Los Angeles, 1959).
63. Kutateladze, S.S., "Heat Transfer in Condensation and Boiling," USAEC Rep. AEC-tr-3770 (1952).
- 63a. Kutateladze, S.S., "Fundamentals of Heat Transfer." Academic Press, 1963 p. 362.
64. Chang, Y.P. & Snyder, N.W., "Heat Transfer in Saturated Boiling," Chem. Eng. Prog. Symp.Ser. 56, No.30, 25-38 (1960).
65. Rohsenow, W.M. & Griffith, P., "Correlation of Maximum Heat Transfer Data for Boiling of Saturated Liquids," Chem. Eng. Prog. Symp.Ser. 52, No. 18, 47 (1956).
66. Lienhard, J.H. & Schrock, V.E., Trans. ASME, Jrl. Heat Trans. 85c, 265-272, August 1963.

67. Lienhard, J.H. & Watanabe, K., Trans. ASME, Jrl. Ht. Trans. 88c, 94-100, February 1966.
68. Noyes, R.C. & Lurie, H., "Boiling Sodium Heat Transfer," 3rd Int. Heat Trans. Conf., Chicago, August 1966, Vol.V, pp 92-100.
69. Balzheiser, R.E. et al, "Investigation of Liquid Metal Boiling Heat Transfer," Wright-Patterson Air Force Base Rep. RTD-TDR-63-4130 (1963).
- 69a. Sciance, C.T., Colver, C.P. & Sliepcevich, C.M., Paper No. C-4, 1966 Cryogenics Engrg. Conf. Boulder.
70. Zuber, N., Tribus, M. & Westwater, J.W., "The Hydrodynamic Crisis in Pool Boiling of Saturated and Subcooled Liquids," pp 230-236 Int. Dev. in Heat Transfer, Pt.II, ASME (1961).
71. Adams, J.M., "A Study of the Critical Heat Flux in an Accelerating Pool Boiling System," NSF G-19697, U. of Washington (1962).
72. Beasant, W.R. & Jones, H.W., "The Critical Heat Flux in Pool Boiling under Combined Effect of High Acceleration and Pressure," UK Rep. AEEW-R 275, Winfrith (1963).
73. Costello, C.P. & Frea, W.J., "A Salient Non-hydrodynamic Effect on Pool Boiling Burnout of Small Semi-cylindrical Heaters," AIChE preprint 15, 6th Nat. Ht. Trans. Conf., Boston (1963).
74. Zuber, N. & Fried, E., "Two Phase Flow and Boiling Heat Transfer to Cryogenic Liquids," Amer. Rocket Soc. Propellants, Combustion & Liquid Rocket Conf. April 1961.
75. Richards, R.J., Steward, W.G. & Jacobs, R.B., "Survey of Literature on Heat Transfer to Cryogenic Fluids" NBS TN 122, Boulder Labs., Oct.1961.
76. Clark, J.A., "Heat Transfer", Chap. 5, Cryogenic Technology, Ed. R.W. Vance, John Wiley 1963.
77. Brentari, E.G. & Smith, R.V., "Nucleate and Film Pool Boiling Design Correlations for O_2 , N_2 , H_2 and He." Int. Adv. In Cryogenic Engrg. Vol.10, Timmerhaus, Ed., Aug. 1964 Conf. Phila., Plenum Press pp 325-341.
78. Lyon, D.N., Int. Adv. Cryo. Engrg. Vol. 10 pp 371-379.
79. Lyon, D.N., Kosky, P.G. & Harman, B.N., Int. Adv. In Cryo. Engrg., Vol.9, August 1963, Conf. Boulder, Plenum Press, pp 77-87.
80. Tippets, F.E., "Critical Heat Fluxes and Flow Patterns in High Pressure Boiling Water Flows," ASME Paper 62-WA-162 (1962).
81. Kutateladze, S.S., "Boiling Heat Transfer," Int. J. Ht Mass Trans. 4, 31-45 (1961).
82. Jens, W.H., & Lottes, P.A., "Analysis of Heat Transfer, Burnout, Pressure Drop and Density Data for High Pressure Water," USAEC Rep. ANL-4627(1951).

83. Weatherhead, R.J., "Nucleate Boiling Characteristics and the Critical Heat Flux Occurrence in Subcooled Axial-Flow Water System," USAEC Rep. ANL-6675 (1962).
84. Buchberg, H., Romie, F.E., Lipkis, R. & Greenfield, M., "Heat Transfer, Pressure Drop, and Burnout Studies with and without Surface Boiling for Deaerated and Gassed Water at Elevated Pressures in a Forced Flow System" pp 171-191 in Proc. 1951 Heat Trans. and Fluid Mech. Inst. Stanford, Calif., University Press (1951).
85. Anderson, S.W., Trans. ASME, J. Heat Trans. 196-197, August 1960.
86. Hewitt, G.F., Kearsley, H.A., Lacey, P.M.C. & Pulling, D.J., "Burnout and Nucleation in Climbing Film Flow" UK Rep. AERE-R 4374 (1963).
87. Bennett, J.A.R., Collier, J.G., Pratt, H.R.C. & Thornton, J.D., "Heat Transfer to Two-Phase Gas-Liquid Systems, Part I: Steam-Water Mixtures in the Liquid-Dispersed Region in an Annulus." Trans. Inst. Chem. Engrs. (London) 39, 113-126 (1961).
88. Dengler, C.E. & Addoms, J.N., Chem.Eng. Prog. Ser. Vol.52, No.18, 95-103 (1956).
89. Wright, R.M., "Downflow Forced Convection Boiling of Water in Uniformly Heated Tubes," USAEC Rep. UCRL-9744 (1961).
90. Collier, J.G., Lacey, P.M.C. & Pulling, D.J., Trans. Inst. of Chem/Engrs. 42, 1964.
91. Shrock, V.E. & Grossman, L.M., "Forced Convection Boiling Studies Forced Convection Vaporization Project," USAEC Rep. TID-14632 (1959).
92. Guerrieri, S.A. & Talty, R.D., Chem. Eng. Prog. Symp.Ser., Vol.52, No.18, 69-77, 1956.
93. Shrock, V.E. & Grossman, L.M., "Local Pressure Gradients in Forced Convection Vaporization," Nucl. Sci. Eng. 3, 245-250 (1959).
94. Boiling Symposium, ASHRAE Meeting, Houston (1966) Publ. by ASHRAE.
95. Handbook of Fundamentals, ASHRAE 1967, Chap. 3, Heat Transfer by J.B. Chaddock, pp 52-53.
96. Michaud, E.R. & Chaddock, J.B., "Forced Convection Boiling of Refrigerants" phase 2, Report HL-106, Herrick Lab., Purdue U., Aug. 1966.
97. Chen, J.C., "A correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow," ASME Paper 63-HT-34 (1963).
98. Chen, J.C., "Existence of Non-Equilibrium, Inverse Temperature Profiles in Boiling Two-Phase Flow," BNL 9296 (1965).
99. Pierre, B., Kylteknisk Tidskrift, No.3, May 1957, p 129; also Svenska Flaktfabriken Review, Vol. 2, No. 1, 1955.

100. Altman, M., Norris, R.H., & Staub, F.W., ASME Jrl. of Heat Transfer, 189-198, Aug. 1960.
101. Silvestri, M., Advances in Heat Transfer, Vol.I., Irvine & Hartnett.
102. Levy, S. & Bray, A.P., "Reliability of Burnout Calculations in Nuclear Reactors," Nuclear News (Feb. 1963) 3.
103. Rohsenow, W.M. & Clark, J.A., "Heat Transfer and Pressure Drop Data for High Heat Flux Densities to Water at High Sub-Critical Pressures," Heat Transfer Fluid Mech. Inst. Stanford, Calif. (1951).
104. Bennett, A.W., Collier, J.G. & Lacey, P.M.C., AERE-R3804 (Aug.1961).
105. Silvestri, M., Paper No. 39, 1961 Int. Heat Trans. Conf., Boulder.
106. Bennett, A.W., Collier, J.G. & Lacey, P.M.C., AERE-R3934, Part III (1963).
107. Bertoletti, S., Gaspari, G.P., Lombardi, C., Peterlongo, G., Silvestri, M., Tacconi, F.A., "Heat Transfer Crisis in Steam-Water Mixtures," Energia Nucleare, Vol. 12, No.3, March 1965.
108. Aladyev, I.T., Z.I. Miropolsky & Boroshchuk, V.E. & Styrikovich, M.A., Int. Dev. in Heat Transfer, Vol. II, Boulder (Aug. 1961).
109. Tong, L.S., "Boiling and Two-Phase Flow," Wiley 1965: (a) p.183, (b) p.172, (c) pp 137-155, (d) pp 155-178, (e) p.188, (f) pp 194-5, (g) pp 130-134.
110. Weatherhead, R.J., "Nucleate Boiling Characteristics and the Critical Heat Flux Occurrence in Subcooled Axial-Flow Water System," USAEC Rep. ANL 6675 (1962).
111. Macbeth, R.V., "Forced Convection Burnout in Simple, Uniformly Heated Channels: A Detailed Analysis of World Data," European Atomic Energy Society Symp. on Two-Phase Flow, Steady State Burnout and Hydrodynamic Instability, Sweden (1963).
112. Aladyev, I.T., Miropolsky, S.L., Doroschuk, V.E., & Styrikovich, M.A., "Boiling Crisis in Tubes," pp 237-243 of Int. Dev. in Heat Trans. Part II, ASME (1961).
113. Macbeth, R.V., "Burnout Analysis, Pt.2, The Basic Burnout Curve," U.K. Rep. AEEW-R 167, Winfrith (1963). Also Pt. 3, "The Low Velocity Burnout Regime," AEEW-R 222 (1963): Pt.4, "Application of Local Conditions Hypothesis to World Data for Uniformly Heated Round Tubes and Rectangular Channels," AEEW-R 267 (1963).
114. Kirby, G.J., "Burnout in Climbing Film Two-Phase Flow - Theories", AEEW-R 470, 1960.

115. Bergles, A.E., "Forced Convection Surface Boiling Heat Transfer and Burnout in Tubes of Small Diameter," Ph.D. Thesis, Mech. Eng. Dept. MIT (June 1962).
116. De Bertoli, Green, S.J., LeTourneau, B.W., Troy, M. & Weiss, A., "Forced Convection Heat Transfer Burnout Studies for Water in Rectangular Channels and Round Tubes at Pressures above 500 psi," Rep. WAPD-188, Bettis Atomic Power Lab. (1958).
117. Gambill, W.R., "Generalized Prediction of Burnout Heat Flux for Flowing Subcooled, Wetting Liquids," AIChE Preprint 17, 5th Nat. Heat Trans. Conf. Houston (1962).
118. Bernath, L., "A Theory of Local Boiling Burnout and its Application to Existing Data," Chem. Eng. Prog. Symp. Ser. 56, No.30, 95-116 (1960).
119. Macbeth R.V., "Burnout Analysis. Part 4: World Data for Uniformly Heated Round Tubes and Rectangular Channels," AEEW-R 267, Winfrith (1963); also Part 3, AEEW-R222 (1963) and Part 2: AEEW-R167 (1963).
120. Barnett, P.G., "An Investigation into the Validity of Certain Hypotheses Implied by Various Burnout Correlations," AEEW-R214 (1963).
- 121a. Tong, L.S., Currin, H.B. & Engel, F.C. "DNB (BUrnout) Studies in an Open Lattice Core," USAEC Rep. WCAP-3736 (1964).
- 121b. Tong, L.S., Currin, H.B. & Thorp, A.G., "New Correlations Predict DNB Conditions," Nucleonics 21, No. 5, 43-47 (1963).
122. Stein, R.P. & Lottes, P.A., "Status of Boiling Burnout for Reactor Design," in Selected Topics in Reactor Technique, Leonard Link (Ed.) USAEC Rep. TID-8540 (1964).
123. See No. 119.
124. Firstenberg, H. et al, "Compilation of Experimental Forced Convection Quality Burnout Data with Calculated Reynolds Numbers," NDA-2131-16 (1960).
125. Bennett, A.W., Collier, J.G. & Lacey, P.M.C. AERE-R3934, Part III.
126. Mozharov, N.A., "An Investigation into the Critical Velocity at which a Moisture Film Breaks away from the Wall of a Steam Pipe," Teploenerg 6, (2), 50-53(1959) DSIR-trans-RTS-1581.
127. Todreas, N.E. & Rohsenow, W.M. "Effect of Non-Uniform Axial Heat Flux Distribution," MIT Rep. 9843-37, Sept. 1965; also 3rd Int. Ht. Trans. Conf. ASME-AIChE Chicago, Aug. 1966, Paper 89, Vol.III, pp 78-85.
128. Kirby G.J., "New Correlation of Non-Uniformly Heated Tubes Burnout Data," AEEW-R500 (1966).
129. Todreas, N.E., "Effect of Local Flux Peaks on Critical Heat Flux in Annular two-phase Flow", Amer. Nucl. Soc. Mtg. Nov.1966, Pittsburgh.

130. Becker, K.M. & Persson, P., "Analysis of Burnout Conditions for Flow of Boiling Water in Vertical Round Ducts," AE-113, Aktiebolaget Atomenergi, Stockholm, Sweden (1963) See also: AE-114 (1963).
131. Durant, W.S. & Mirshak, S., "Roughening of Surfaces to Increase Burnout Heat Flux," USAEC Rept. DP-380 (1959); and DPST-60-284 (1960).
132. Janssen, E., Levy, S. & Kervinen, J.A., "Investigations of Burnout - Internally Heated Annulus Cooled by Water at 600 to 1450 psia," ASME Paper 63-WA-149 (1963).
133. Gambill, W.R. & Greene, N.D. "Boiling Burnout with Water in Vortex Flow," Chem. Eng. Prog. 54, No. 10, 68-76 (1958).
134. Viskanta, R., "Critical Heat Flux for Water in Swirling Flow," Nucl. Sci. Eng. 10, 202-203 (1961).
135. Zuber, N. & Tribus, M., "Further Remarks on the Stability of Boiling Heat Transfer," UCLA, Rep. No. 58-5, (1958).
136. Berenson, P., "Transition Boiling Heat Transfer from a Horizontal Surface" AIChE Paper No. 18, ASME-AIChE Heat Transf. Conf. Buffalo, August 1960.
137. Bromley, L.A. Chem. Eng. Prog. 46:5, 221-27 (1950).
138. Bromley, L.A. et al, Ind. Eng. Chem 45, 2639-46 (1953).
139. Breen, B.P. & Westwater, J.W., Chem. Eng. Prog., 58, No. 7, 67 (1962): also preprint AIChE No. 19, 5th Nat. Mt. Transf. Conf. Houston 1962.
140. Lienhard, J.H. & Wong, P.T.Y., ASME Paper 63-HT-3, 1963.
141. Frederking, T.H.K. & Clark, J.A., "Natural Convection Film Boiling on a Sphere," Adv. Cryogenic Engrg. No. 8, 1962, Ed. Timmerhaus.
142. Quinn, E.P., "Forced Flow Heat Transfer Beyond the Critical Heat Flux," ASME Paper 66-Wa/Ht-36, Nov. 1966.
143. Laverty, W.F. & Rohsenow, W.M., "Film Boiling of Saturated Nitrogen Flowing in a Vertical Tube," ASME Paper No. 65-WA/HT-26
144. Forslund, R. & Rohsenow, W.M., "Dispersed Flow Film Boiling." MIT Report No. 75312-44, Jan. 1967.
145. Silvestri, M., "Two Phase Annular Dispersed Flow," Adv. in Heat Transf. Vol I, Ed. T.F. Irvine, Jr. & J.P. Hartnett, Acad. Press, 1964 pp 355-446.
146. Leppert, G. & Pitts C.C., "Boiling" Adv. in Heat Transf. Vol. I Ed. T.F. Irvine, Jr. & J.P. Hartnett, Acad. Press. 1964 pp 185-265.
147. Rohsenow, W.M., "Boiling Heat Transfer" Dev. in Heat Transf., Ed. W.M. Rohsenow, MIT Press 1964, pp 169-260.
148. Westwater, J.W. "Boiling of Liquids", Adv. in Chem Engrg. Ed. T.B. Drew and J.W. Hoopes, Jr., 1956, 1958, Acad. Press.

149. Dwyer, O.E., "Recent Developments in Liquid Metal Heat Transfer," Atomic Energy Review, Vol. IV, No. 1, Int. Atomic Energy Agency, Vienna, 1966.

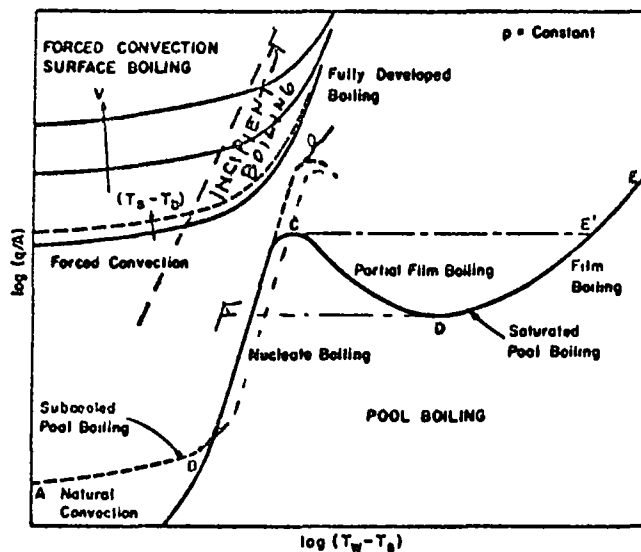


Fig. 1. Regimes in boiling heat transfer.

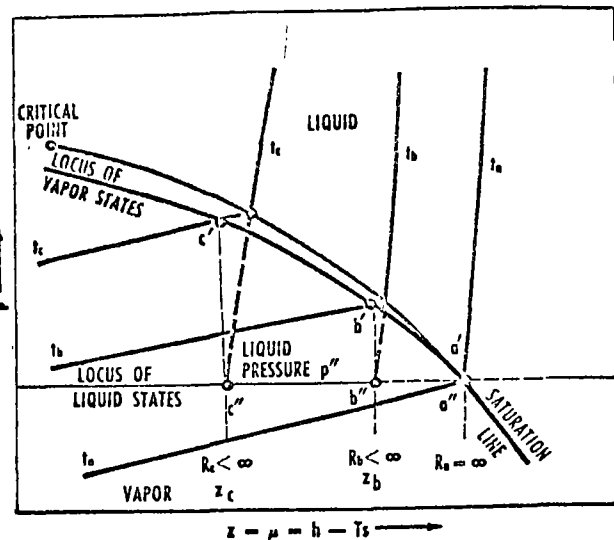


Figure 3. Approximation of vapor-liquid equilibrium at a curved interface of a vapor bubble in a one-component system

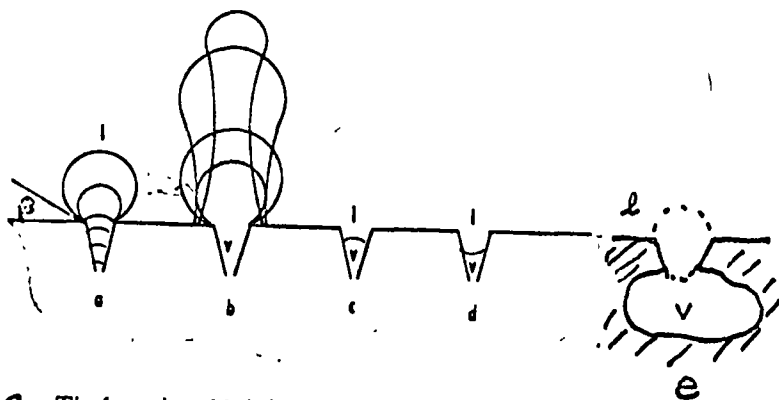


Figure 2. The formation of bubbles of vapor over cavities in a heated surface

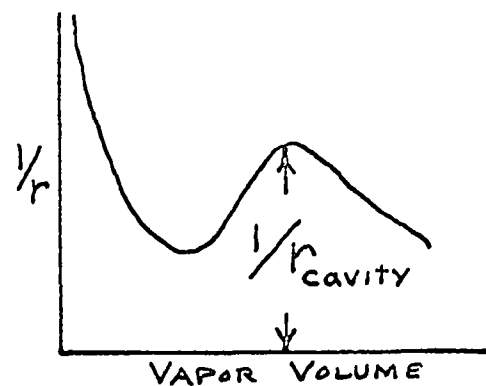


Fig 4. Interface Curvature vs Bubble Volume for Cavity shown in Fig. 2a.

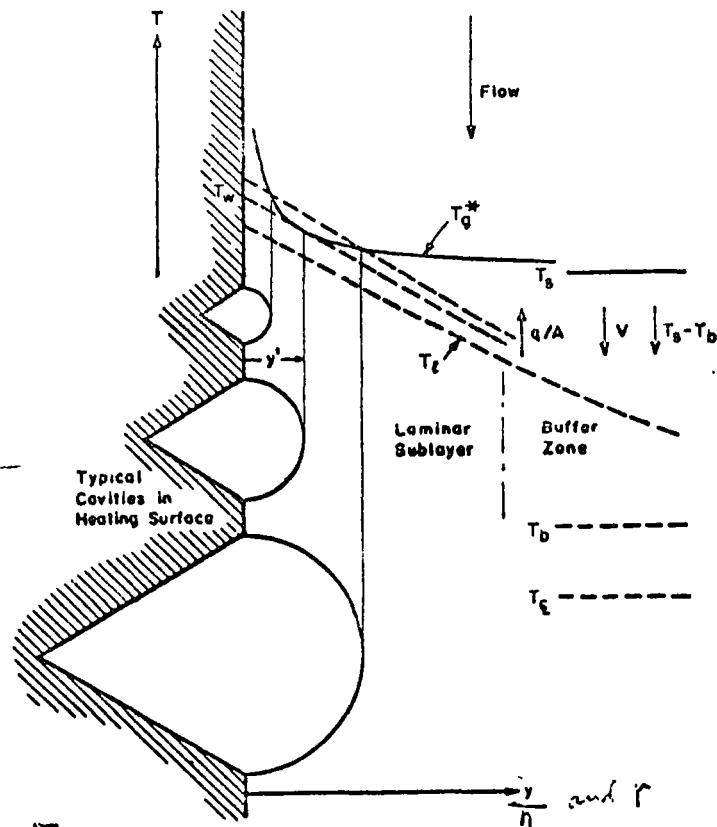


Fig. 43. Initiation of bubble growth in forced-convection surface boiling [5].

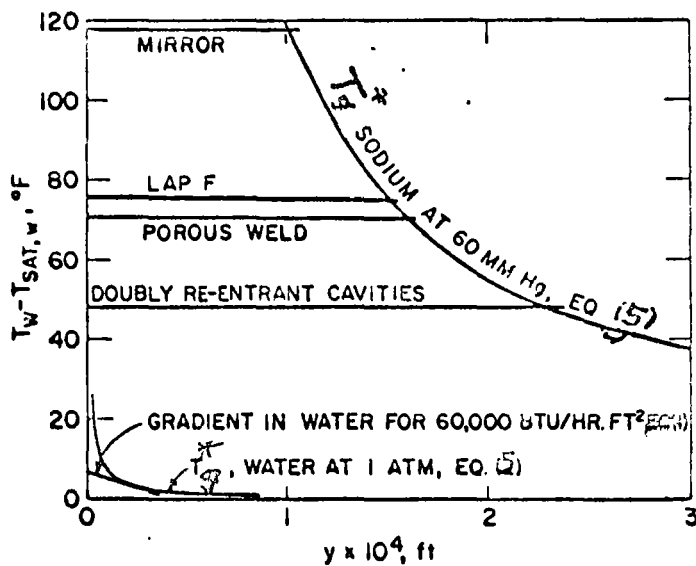


Fig 7 Incipient boiling for sodium and water [10]

MAXIMUM CAVITY RADIUS (FT) REQUIRED FOR VARIOUS PRESSURES

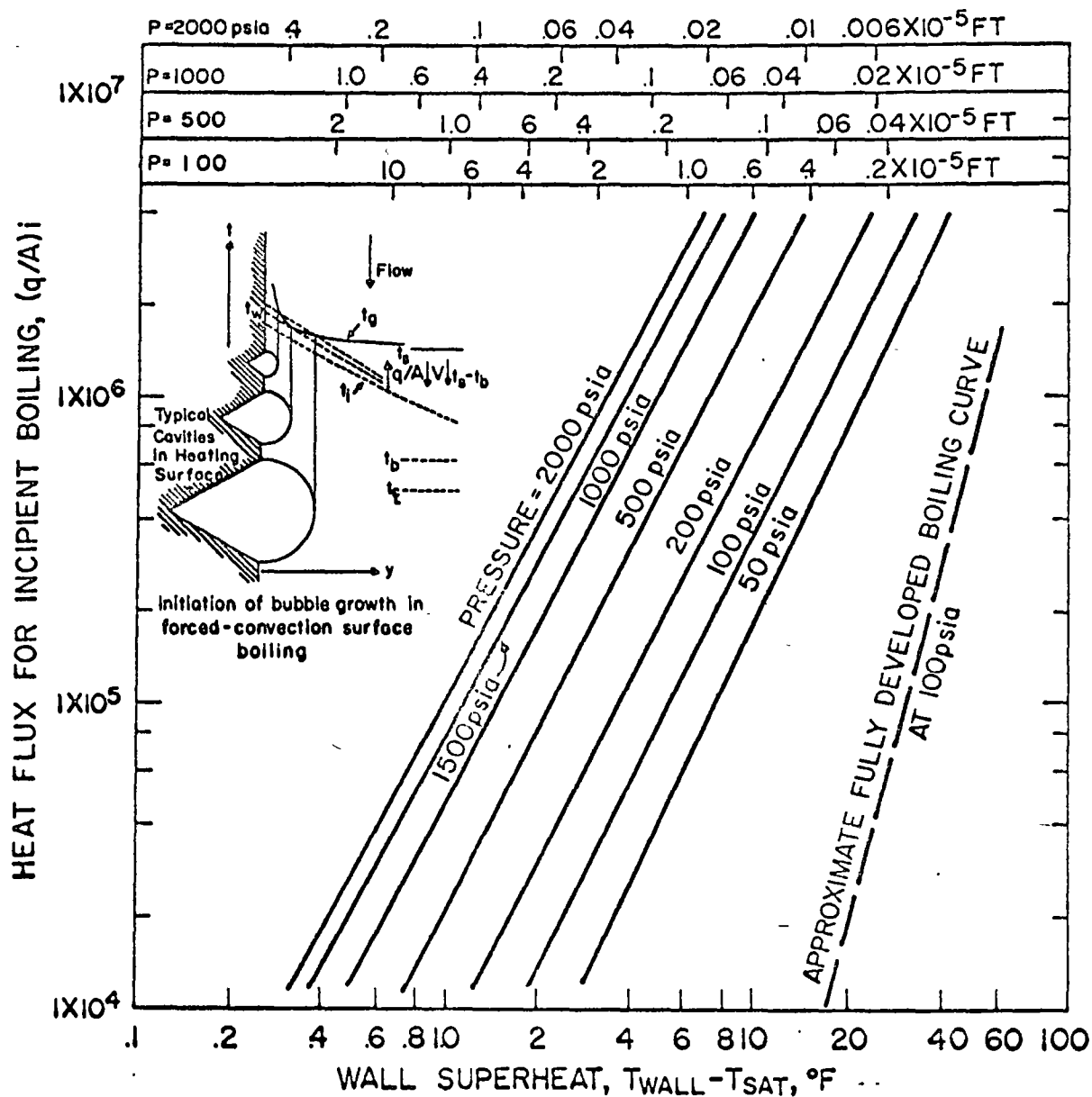


FIGURE 6 BERGLES-ROHSENOW CRITERIA (REF 39) FOR
 INCIDENT BOILING $(q/A)_i = 15.60 p^{1.156}$
 $(T_{\text{wall}} - T_{\text{sat}})^{2.30} / p^{0.0234}$

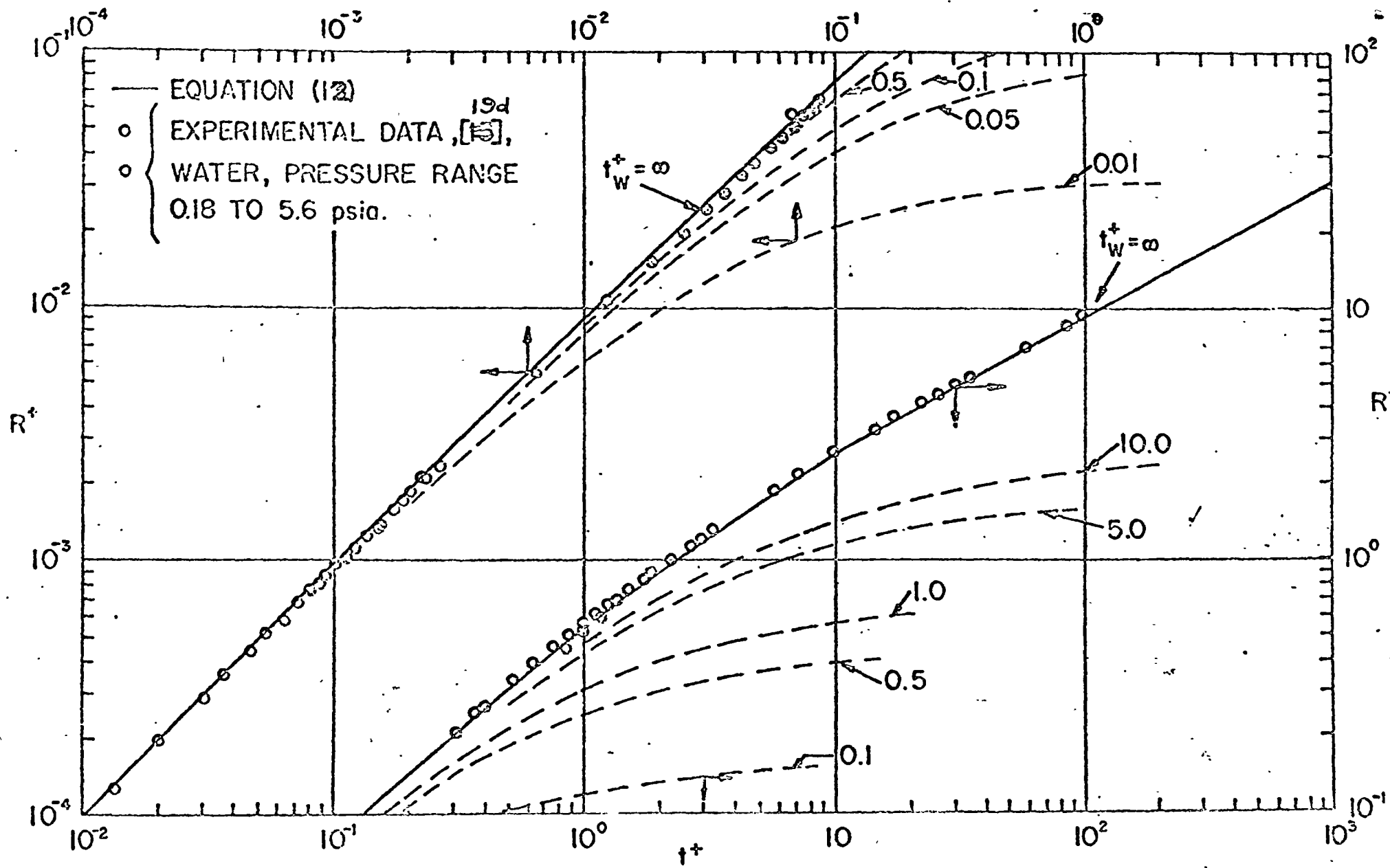


FIG. 8a BUBBLE GROWTH CURVES, COMPARISON WITH EXPERIMENTAL DATA [19c]

EUGENE DIETZGEN CO.
MADE IN U. S. A.

LOGARITHMIC
3 CYCLES X 3 CYCLES

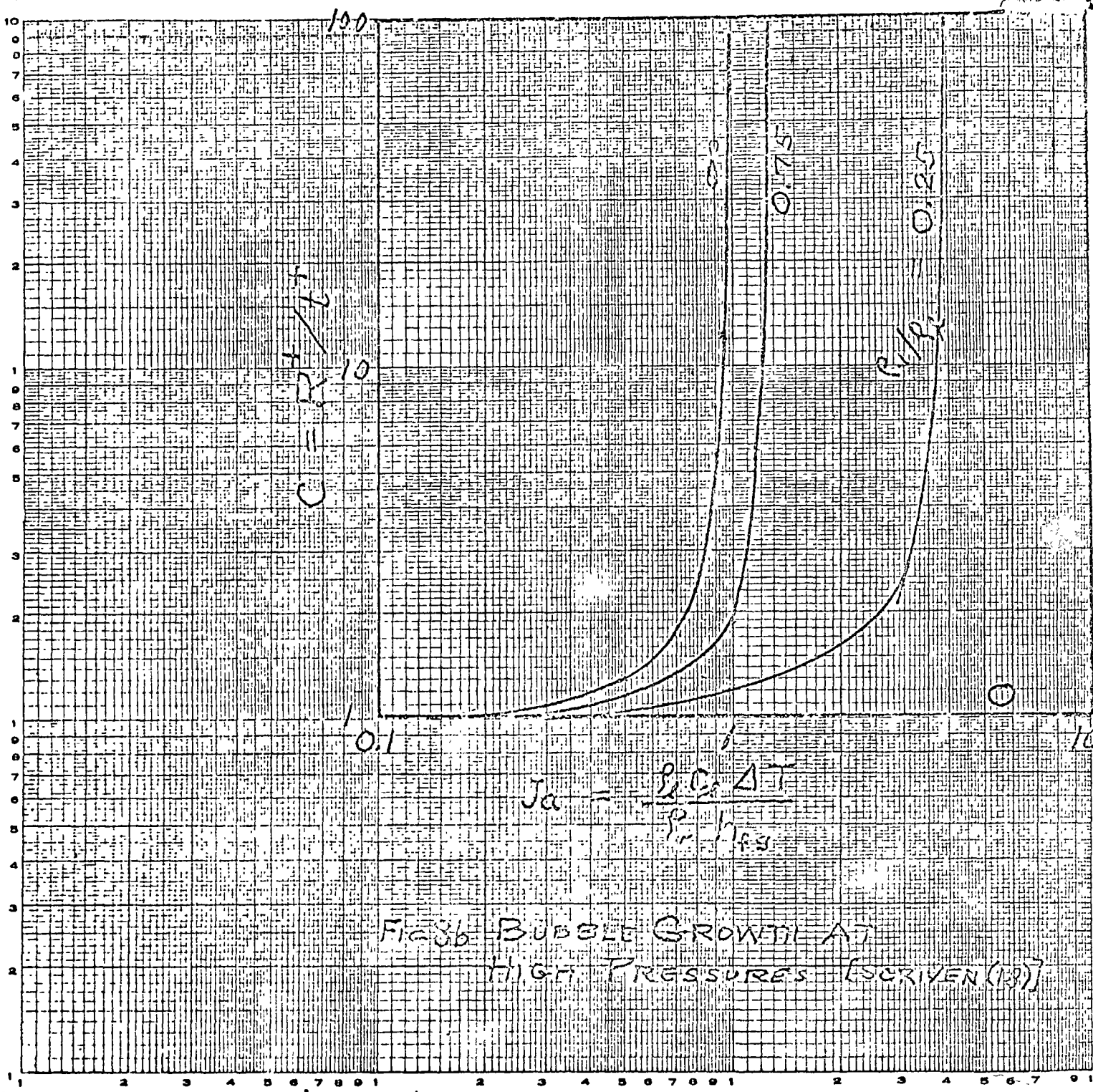
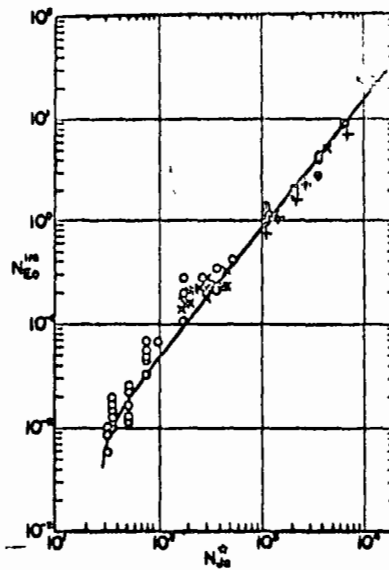
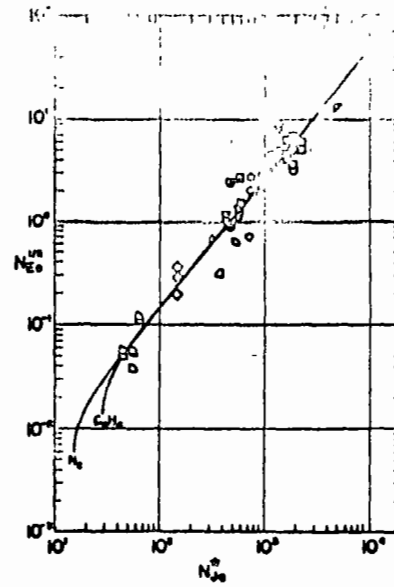


Fig. 86 BUBBLE GROWTH AT
HIGH PRESSURES (SCRIVEN (19))



+ Water (4)
 o . . . (8)
 o . . . (2)
 o . . . (10)
 x . . . (9)
 o Benzene (9)
 o Freon 12 (9)

Departure diameter correlation for water.



o Acetone (2)
 o Carbon Tetrachloride (2)
 o (14)
 o (6)
 o Methanol (2)
 o . . . (13)
 o . . . (9)

Departure diameter correlation for other liquids.

FIG 8c

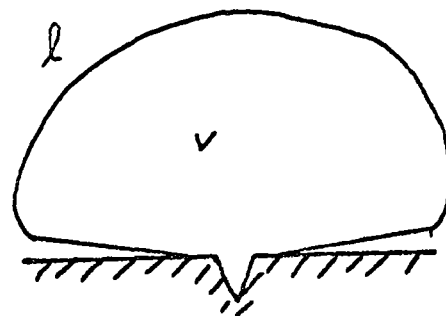
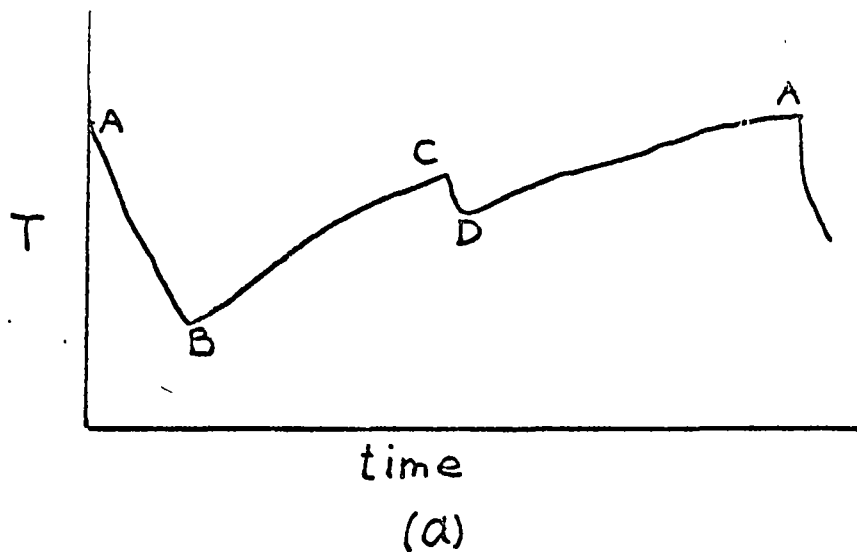


FIG 8 TEMPERATURE OF WALL UNDER A BUBBLE

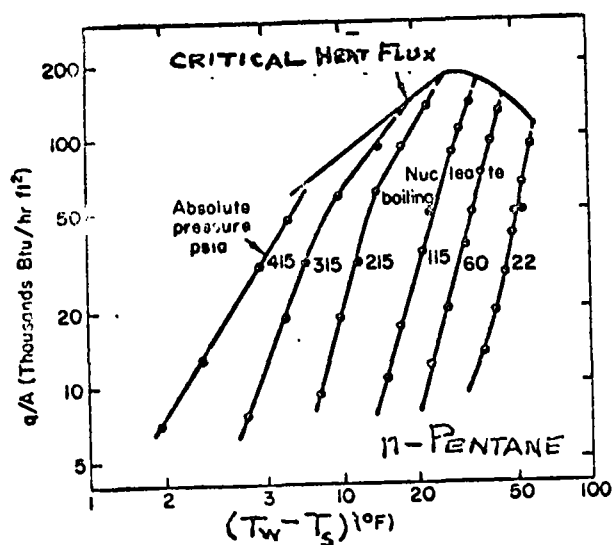


FIG 9 Effect of pressure on pool-boiling curve [23]

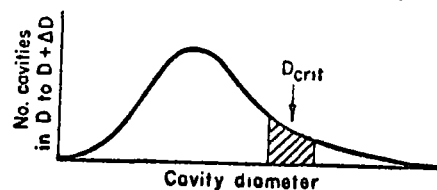


FIG 10 Cavity size distribution.

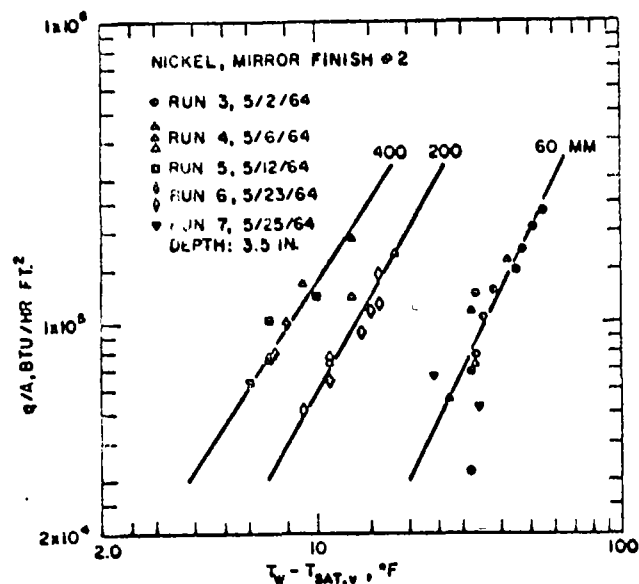


FIG 11 Effect of pressure on nucleate pool boiling

WMR-A-F4

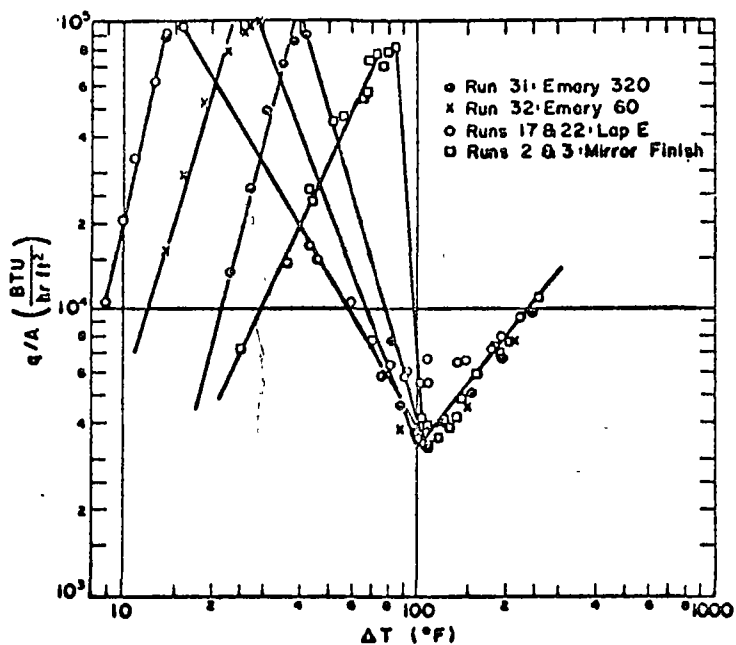


Fig 12 Copper pentane test results; effect of roughness [25]

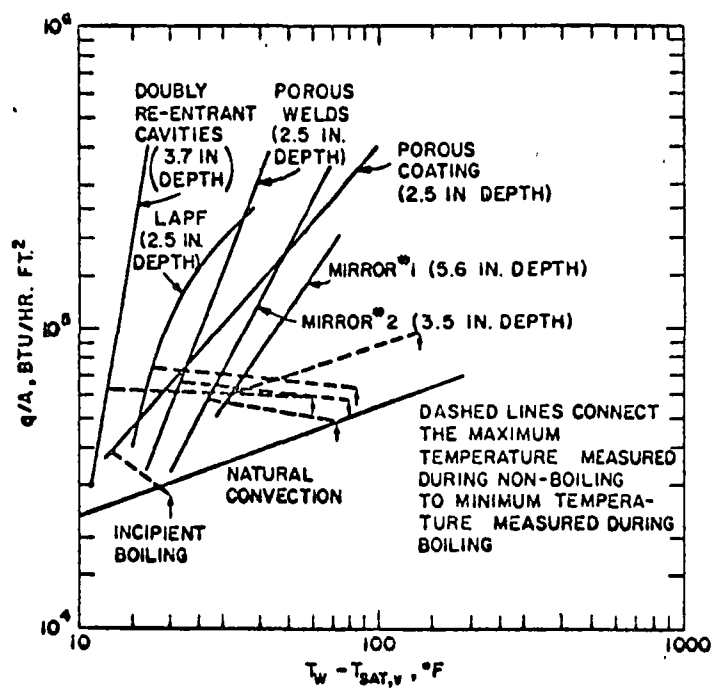


Fig 13 Comprehensive plot of sodium heat-transfer data at an average pressure of 65 mm Hg, MARTO [24].

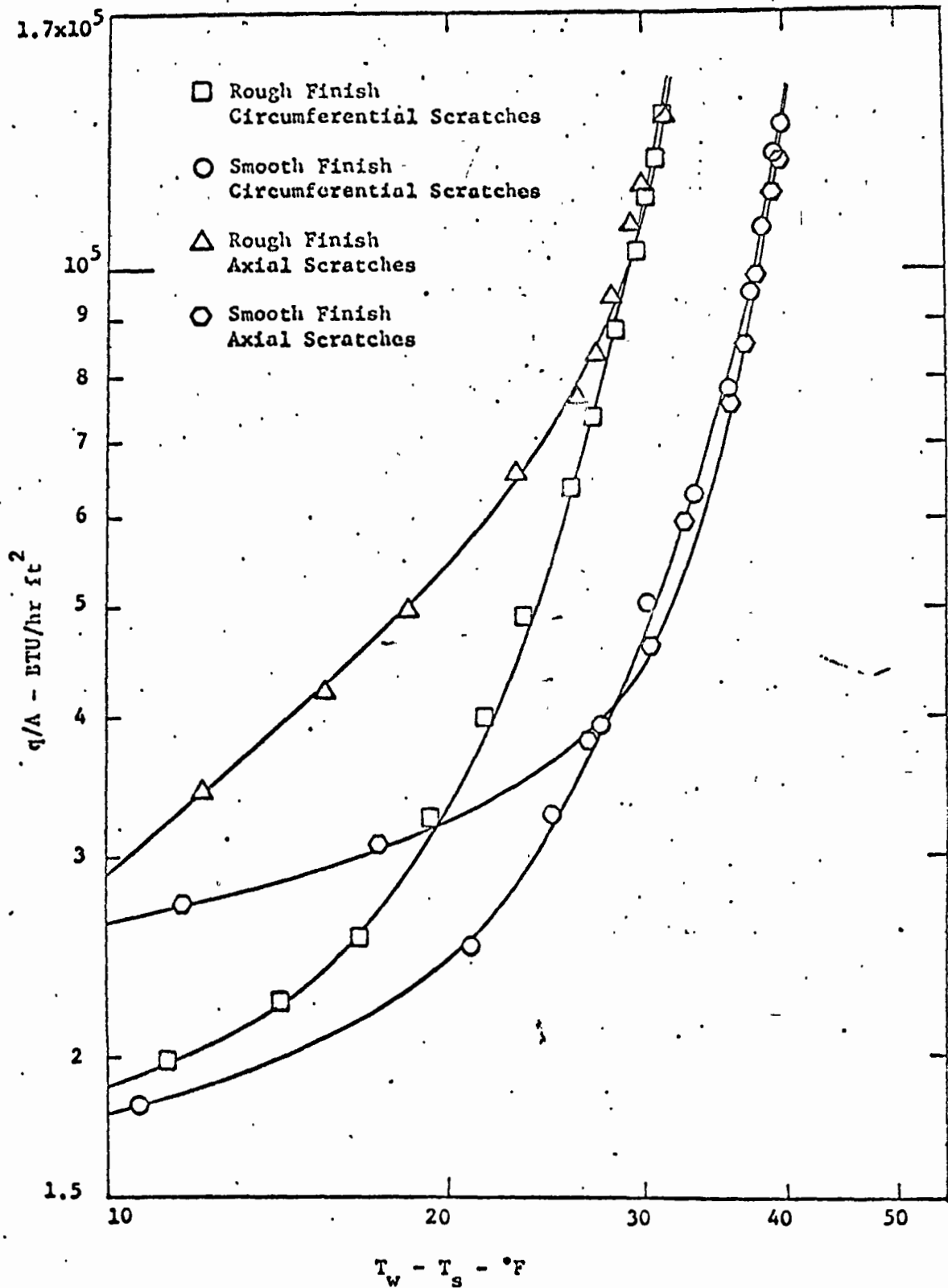


FIG. 24- INFLUENCE OF SURFACE FINISH WHEN POOL BOILING WATER FROM THE OUTSIDE OF A HORIZONTAL TUBE. BROWN [26]

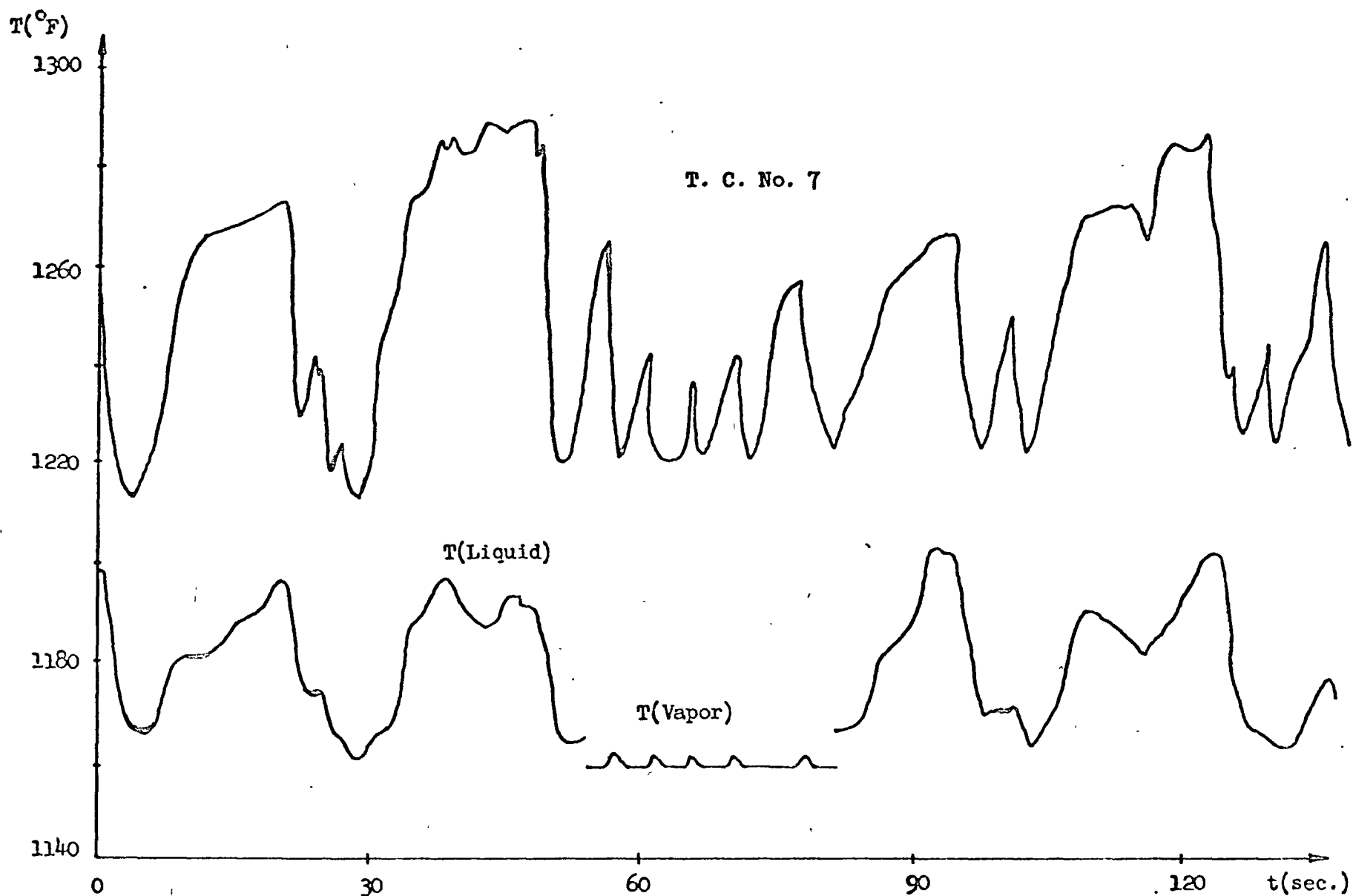


FIG. #1 TEMPERATURE VARIATIONS DURING BUMPING
SODIUM POOL BOILING [33]

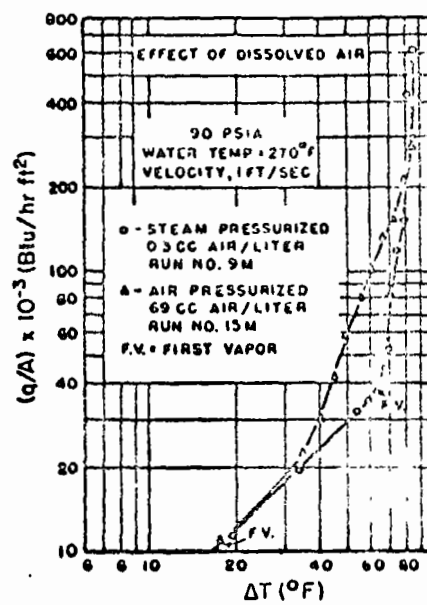


Fig. 15. Effect of dissolved gas on boiling curve [30].

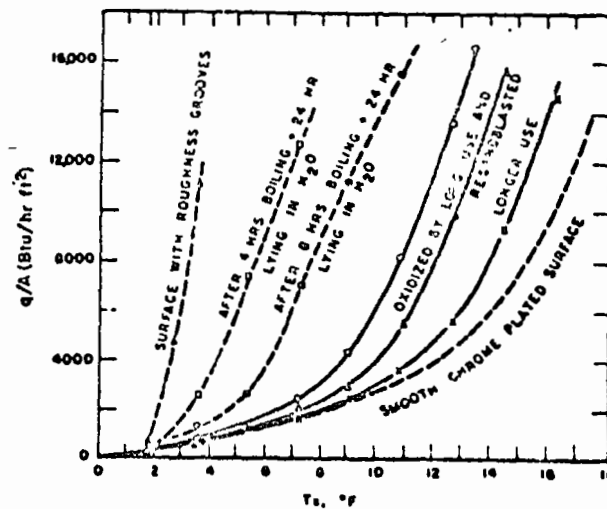


Fig. 16. Effect of surface sandblasting on pool boiling [31]

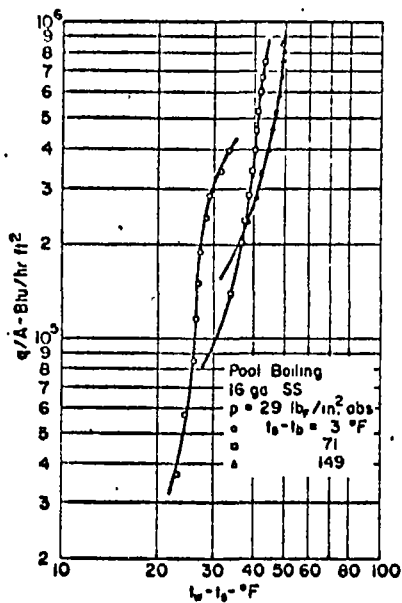


Fig. 19 Influence of subcooling on pool boiling
BERGLES [39]

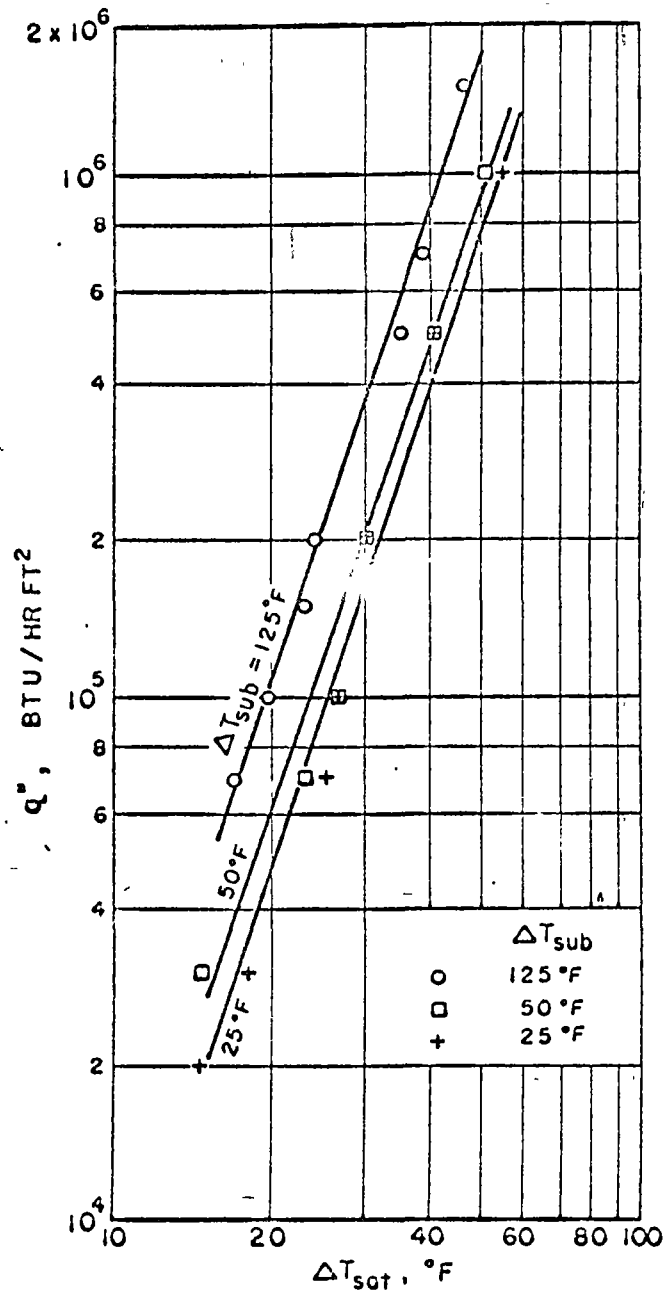


FIG. 20 HEAT TRANSFER DATA AT 115 psia
(DUKE + SHROCK) [40]

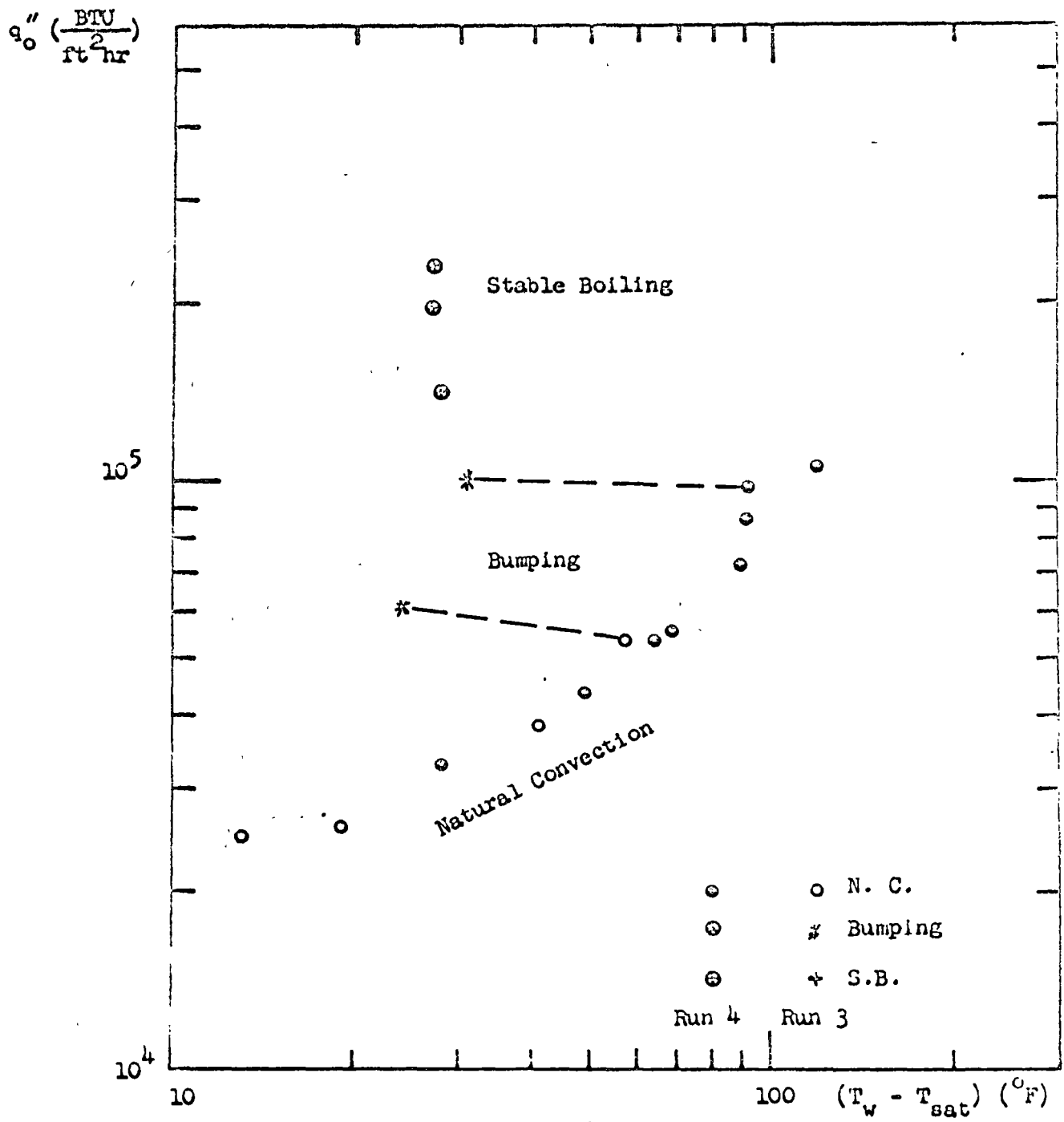


FIG 18 SODIUM POOL BOILING [33]

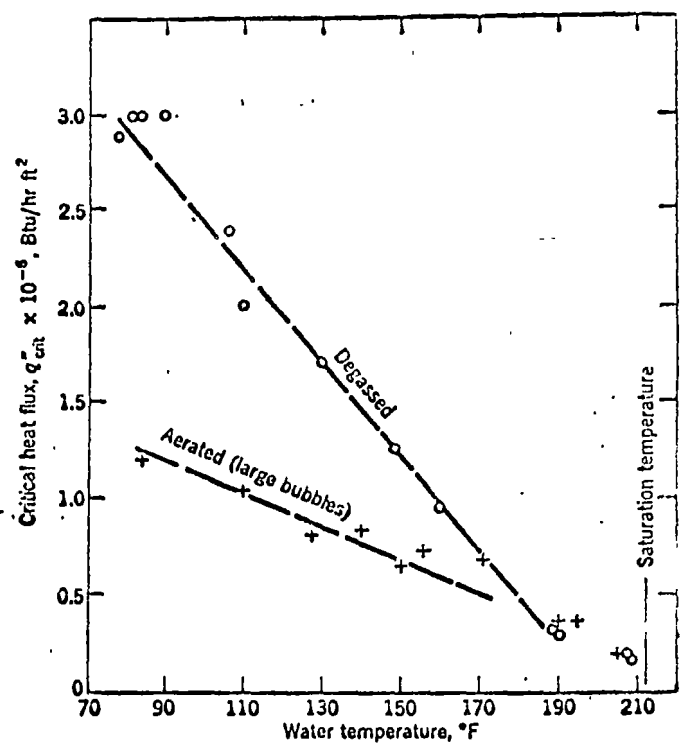


Figure 215 Effect of dissolved gas and subcooling on critical heat flux under atmospheric pressure. From Ellison (27).

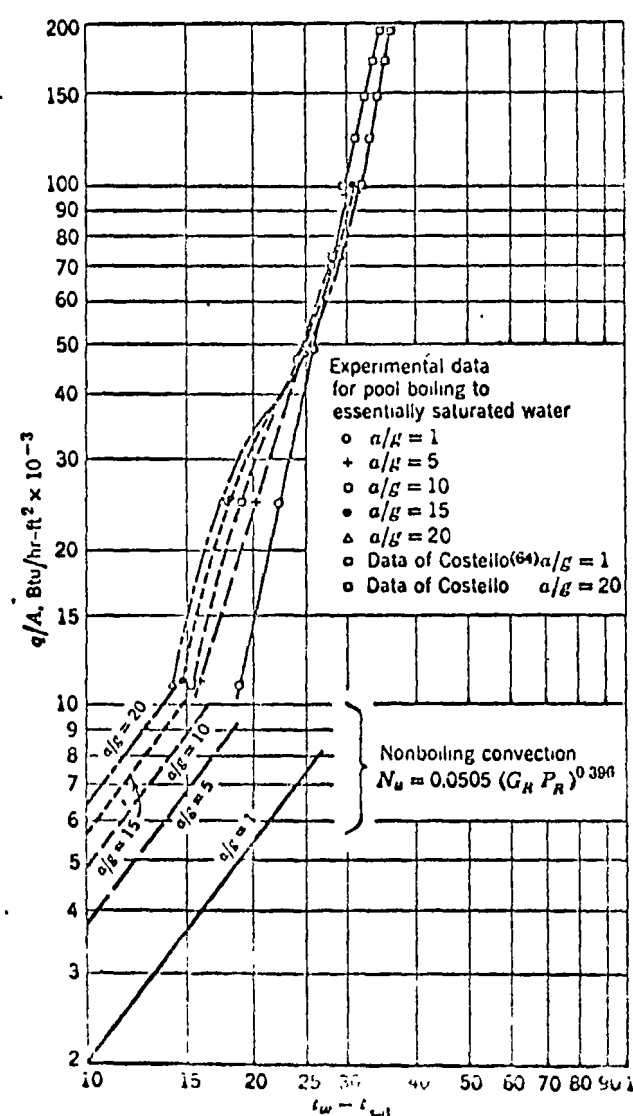


Figure 22 Influence of acceleration normal to heating surface on convection

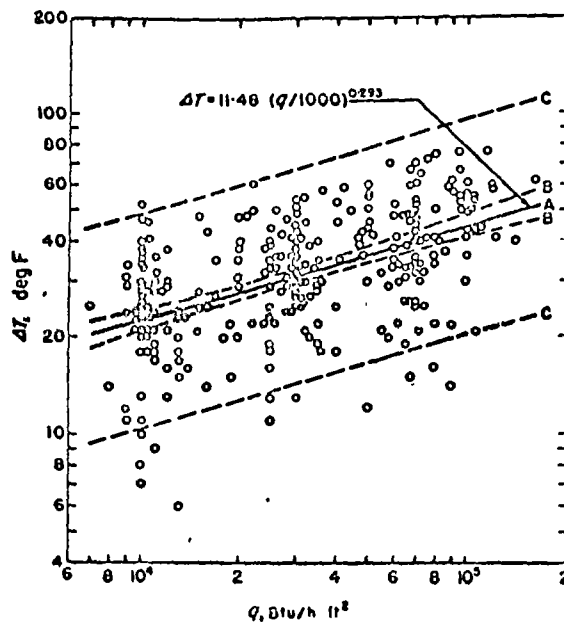


FIG. 23 Nucleate pool boiling of organic liquids at 1 atm:
A, regression line; B, 95 per cent confidence limits on regression line; C, 95 per cent probability limits on ΔT predicted by regression. *ARMSTRONG(50)*

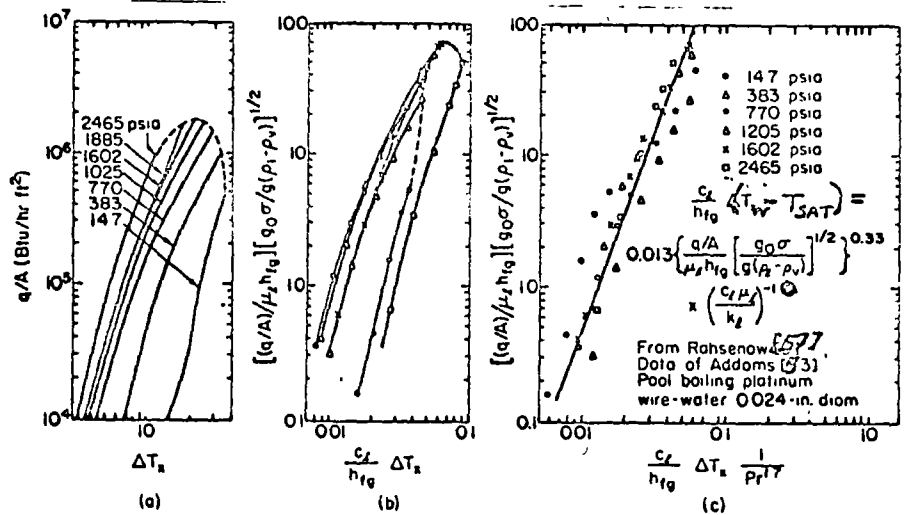


Fig. 24 Correlation of nucleate-boiling data for water [Rohsenow, 57].

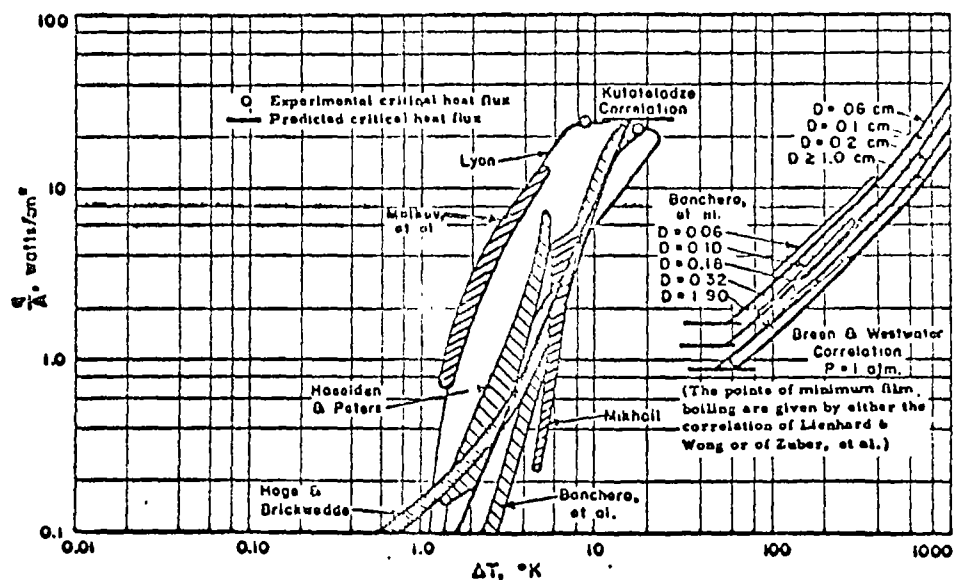


Fig. 25 Experimental nucleate and film pool boiling of oxygen at 1 atm compared with the predictive correlations of Kutateladze and Green and Westwater. 777

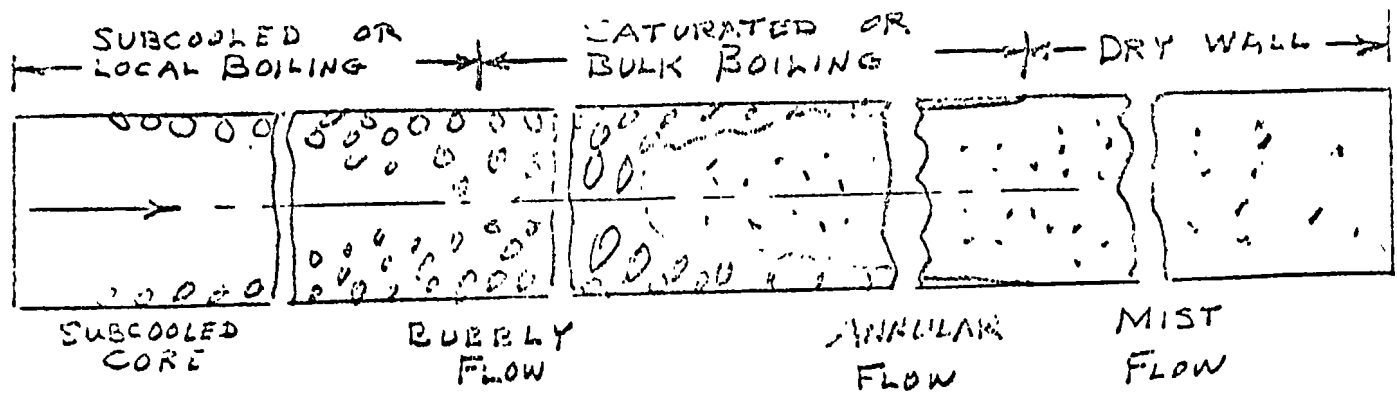
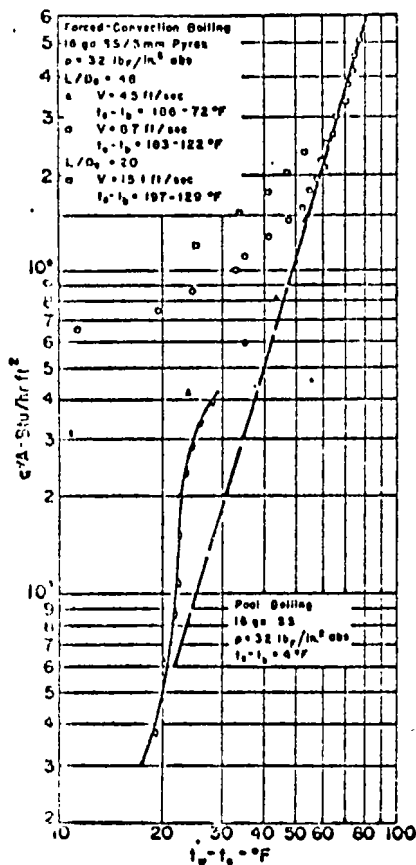


Fig 26. Flow Regimes in a Vertical Heated Channel



27
Fig. 27 Forced-convection surface-boiling data and pool-boiling data for stainless-steel (39)

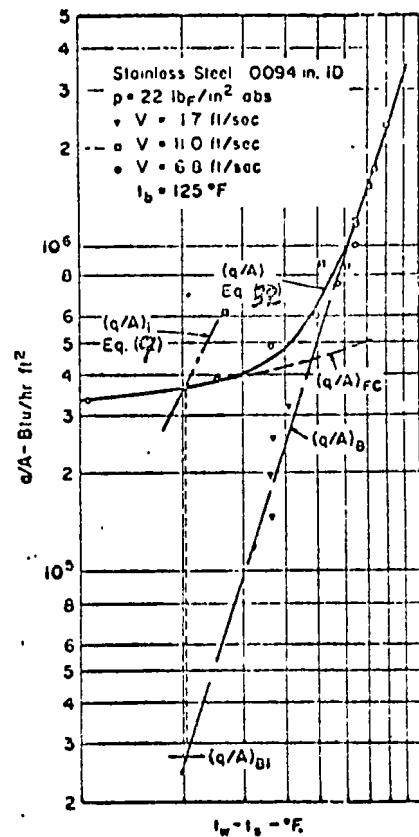


Fig. 28 Recommended procedure for construction of curve for forced-convection surface boiling

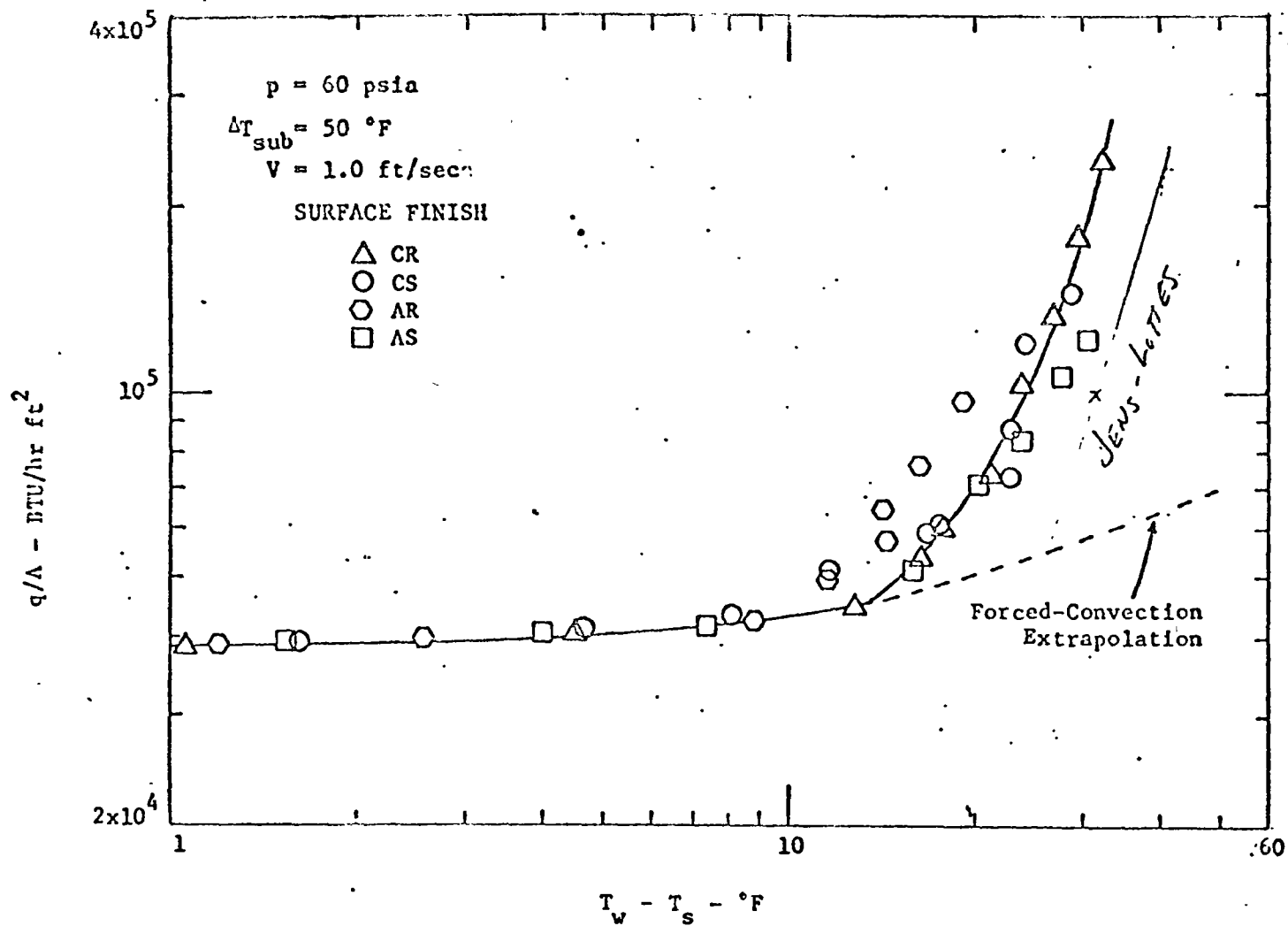


FIG. 29 FLOW SURFACE BOILING DATA - LOW HEAT FLUX (BROWN 2C)

MMR-A-P14

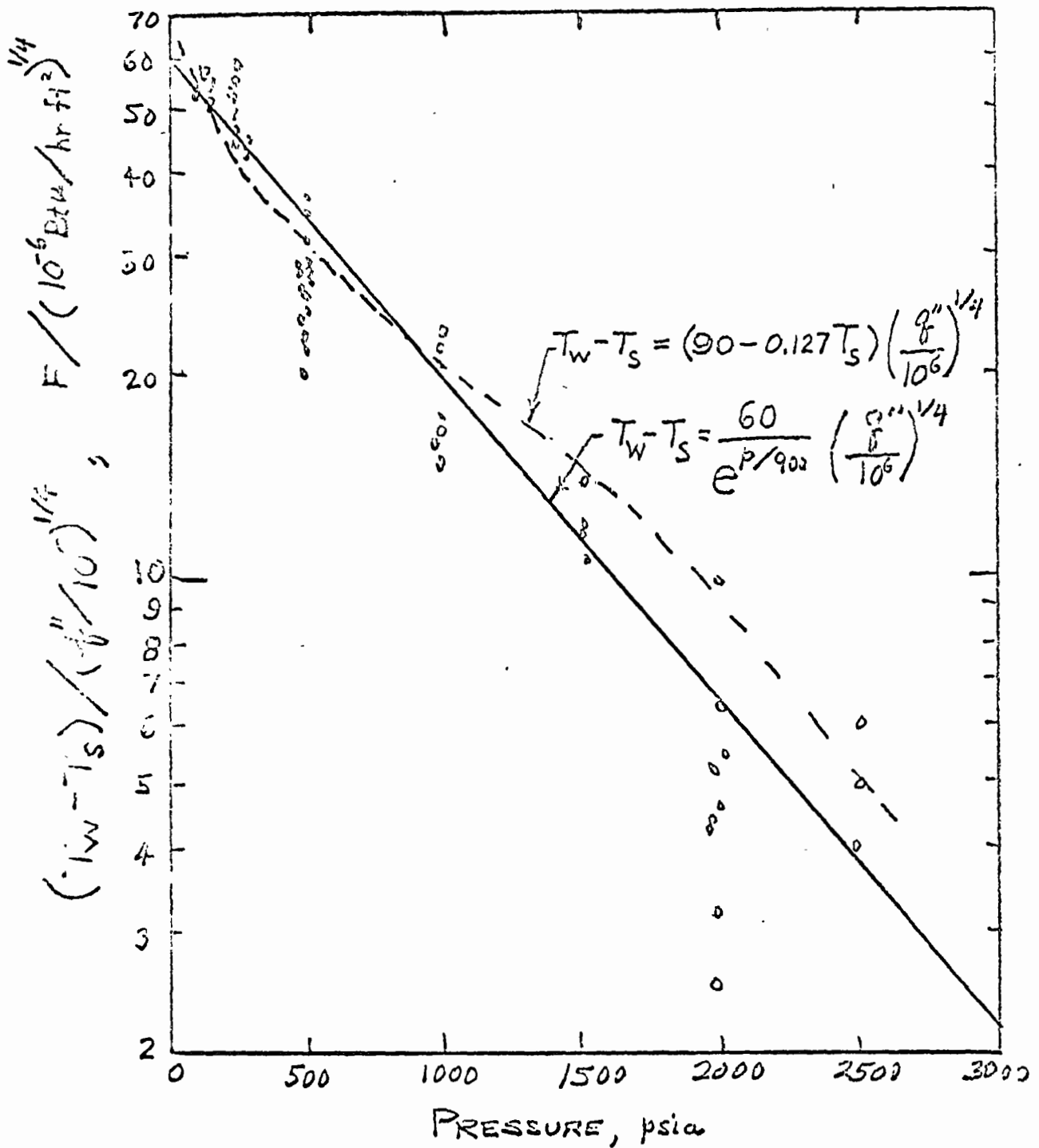


Fig 30. Jens-Lottes and Weatherhead Equations for Fully- Developed Boiling Curve for Low-Quality or Subcooled Forced Convection.

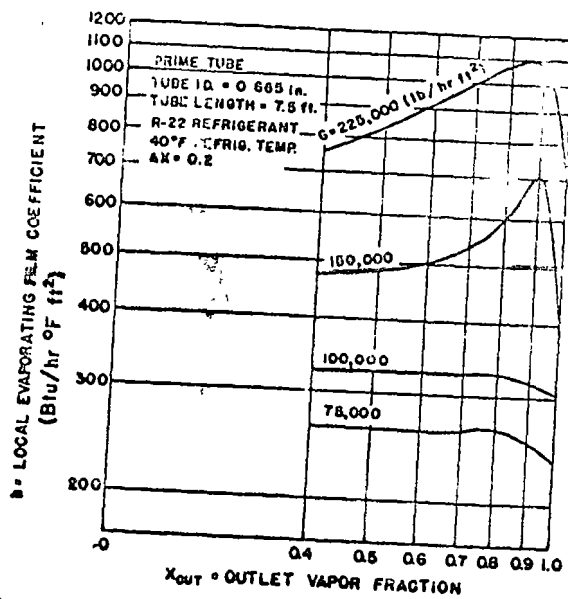


Fig. 31. Variation of h with vapor fraction for forced-convection boiling. [85].

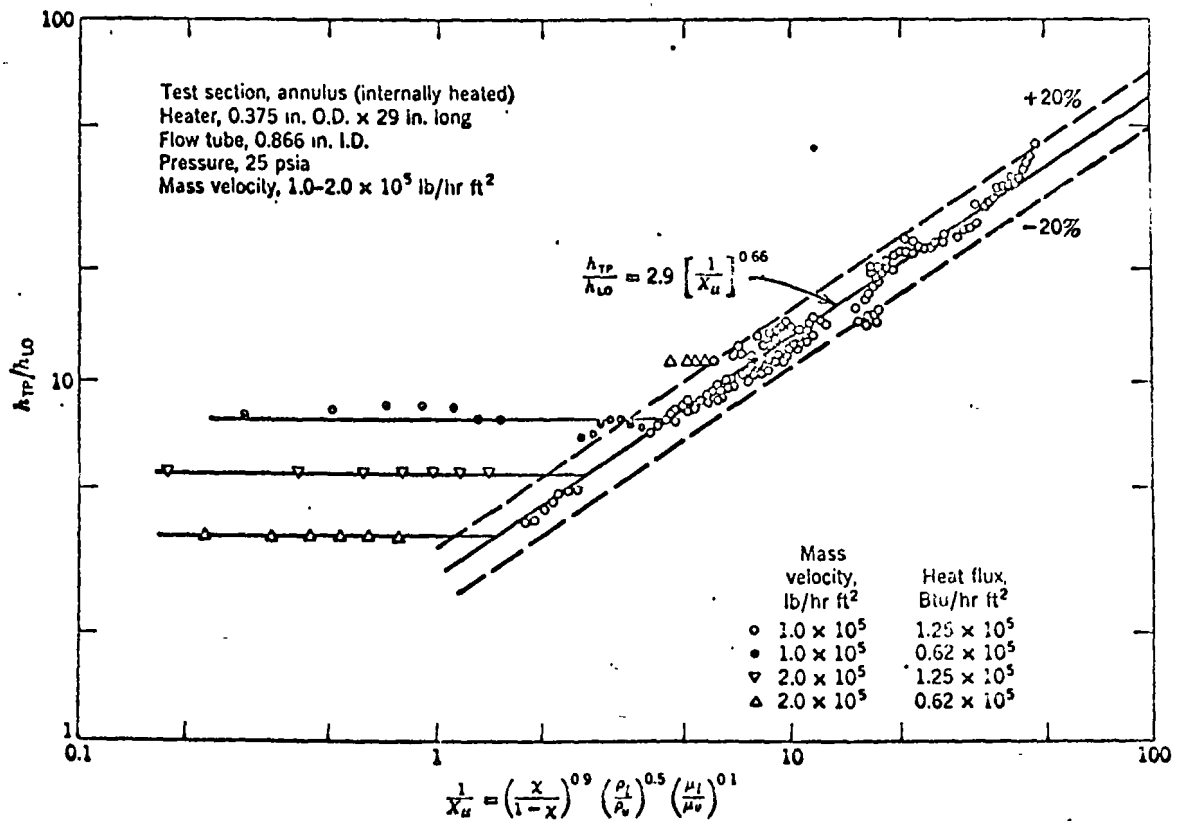
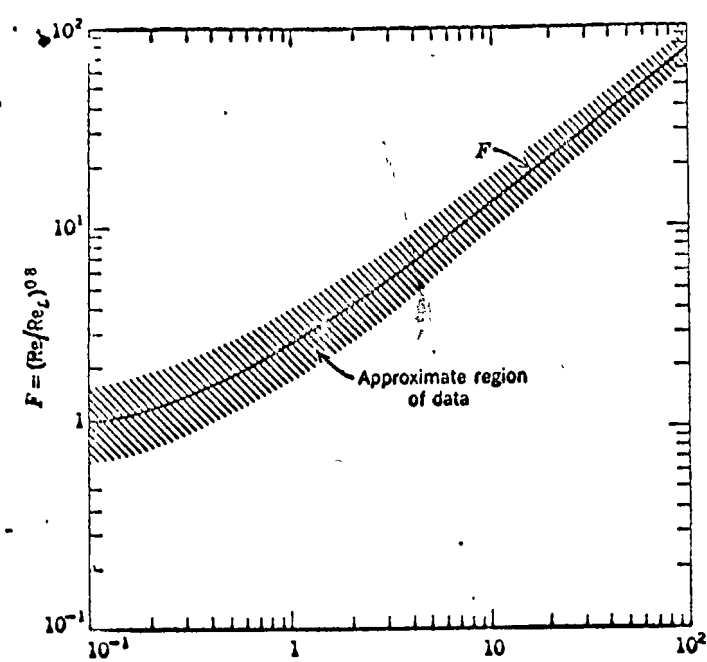


Figure 32. Variation of heat transfer coefficient ratio with Martinelli parameter, series G. From Bennett et al. [87].



$$\frac{1}{X_u} = \left(1 - \frac{x}{L}\right)^{0.9} \left(\frac{\rho_l}{\rho_v}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.1}$$

Figure 33 Reynolds number factor, F .

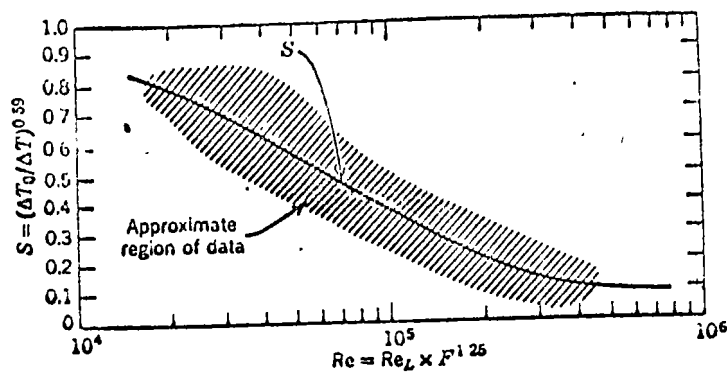


Figure 34 Suppression factor, S .

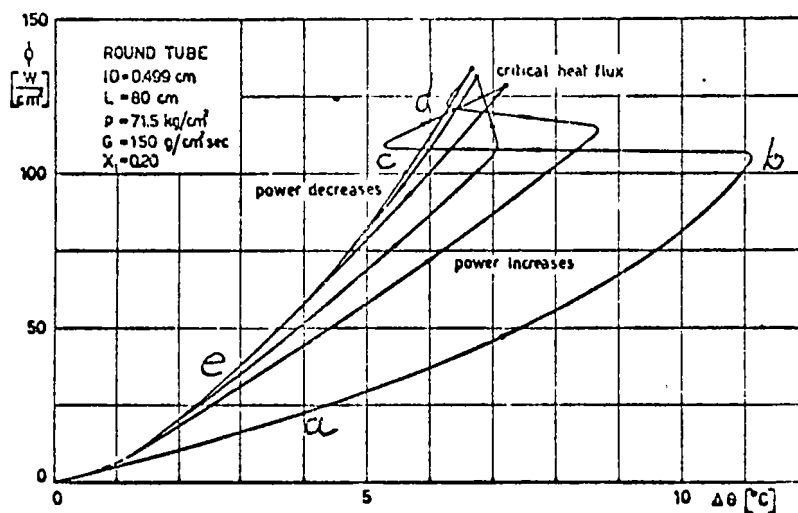


FIG. 35. Example of the wall to coolant temperature difference below CHF, as a multivalued function of the heat flux. CISE (101)

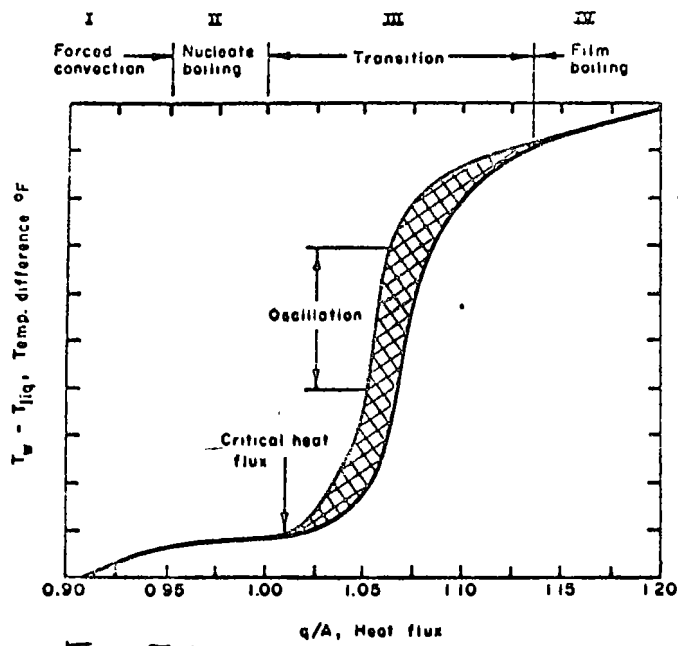


Fig. 36 Typical temperature history during burnout [102].

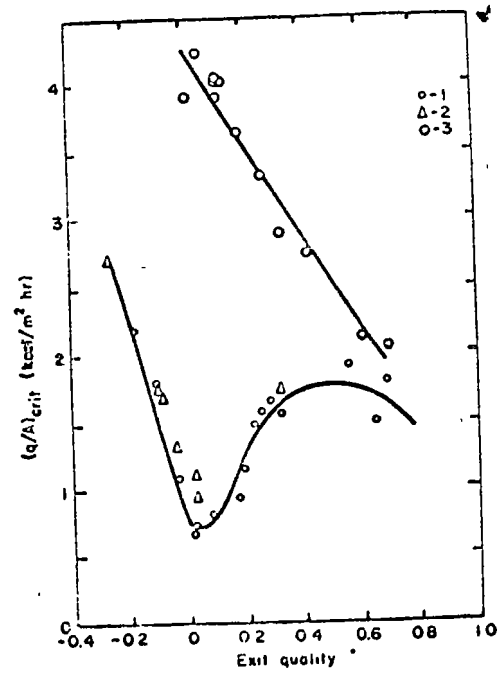


Fig. 37. Effect of a compressible volume on $(q/A)_{crit}$ [108].

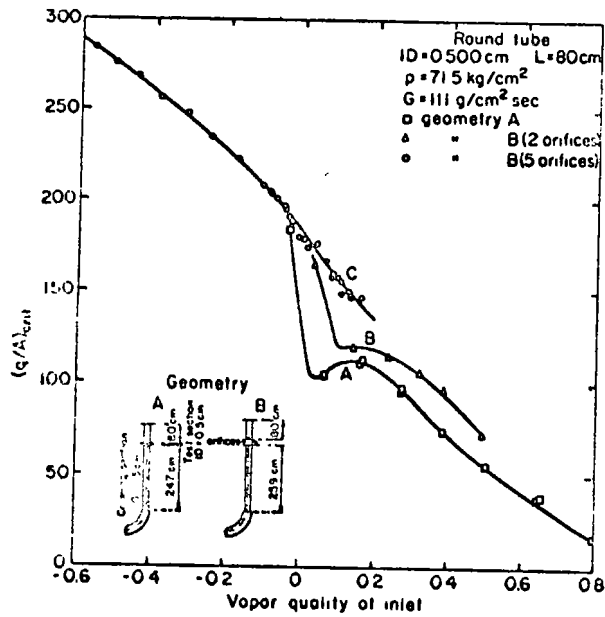
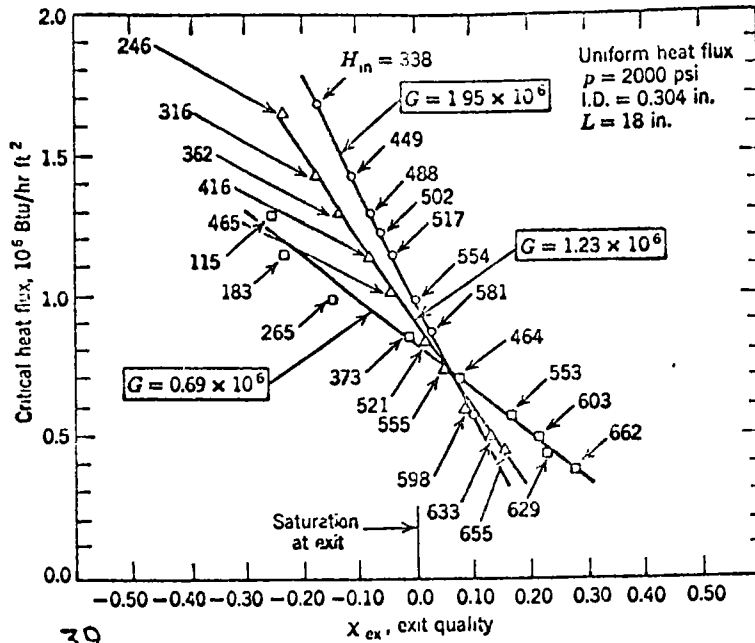
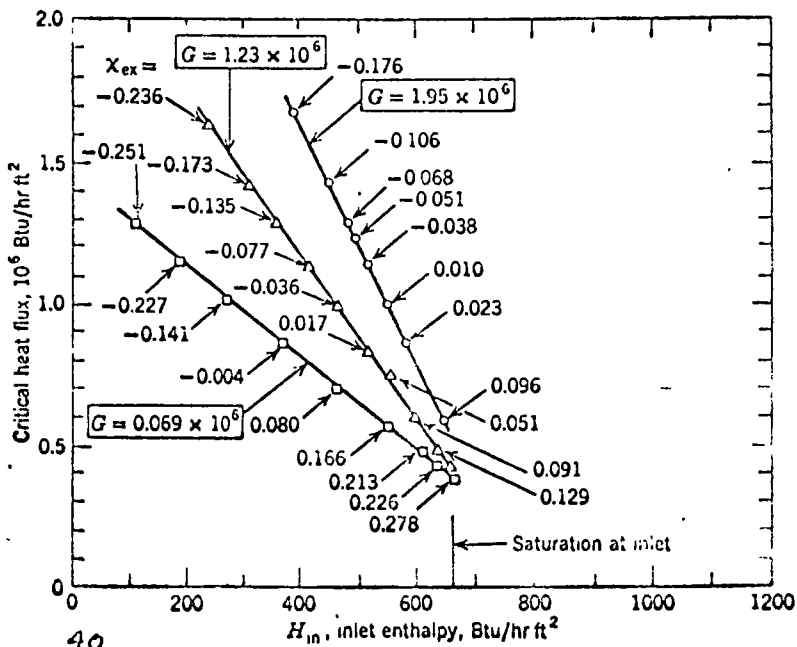


Fig. 38. Effect of throttling at inlet [107].



39
Figure 39 Mass velocity effect on critical heat flux (local condition concept). From (33).



40
Figure 40 Mass velocity effect on critical heat flux (system parameter concept) (same data as Figure 39). From (33).

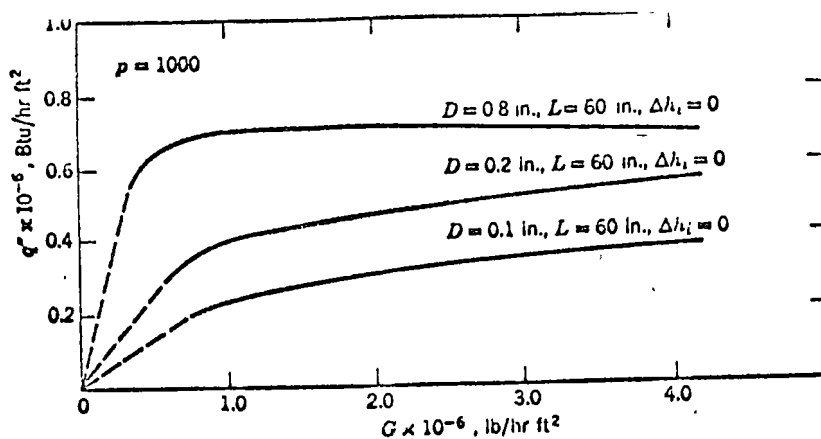


Fig 41

Parametric effects on burnout.

[111]

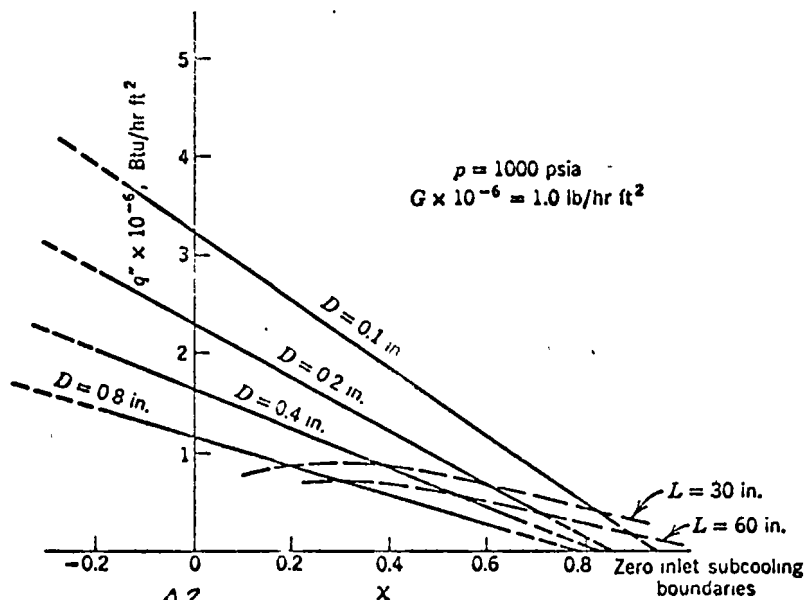


Figure 42

Parametric effects on burnout. From [111]

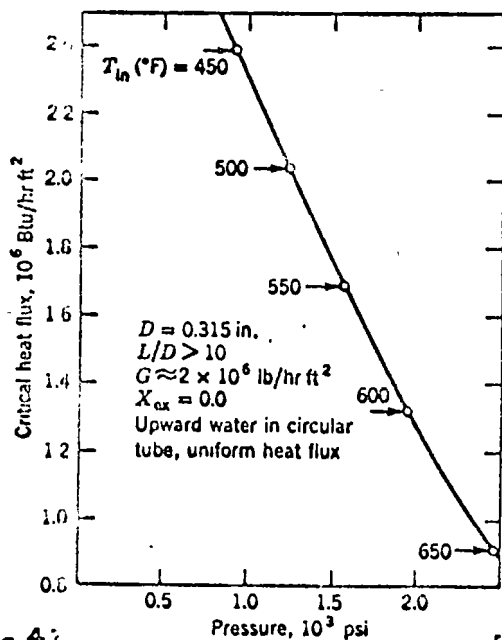


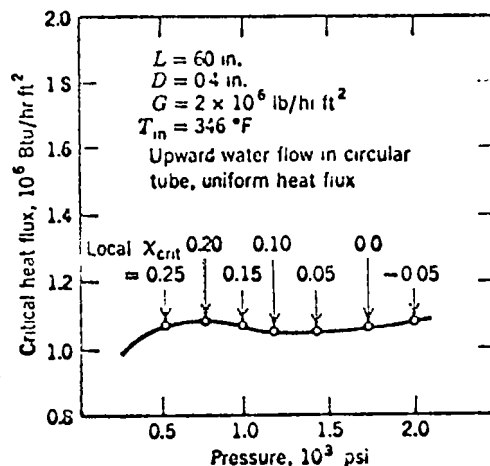
Fig 43

Effect of pressure and inlet subcooling on critical heat flux.

[112]

Figure 44

Pressure and local enthalpy effects on critical heat flux



From [111]

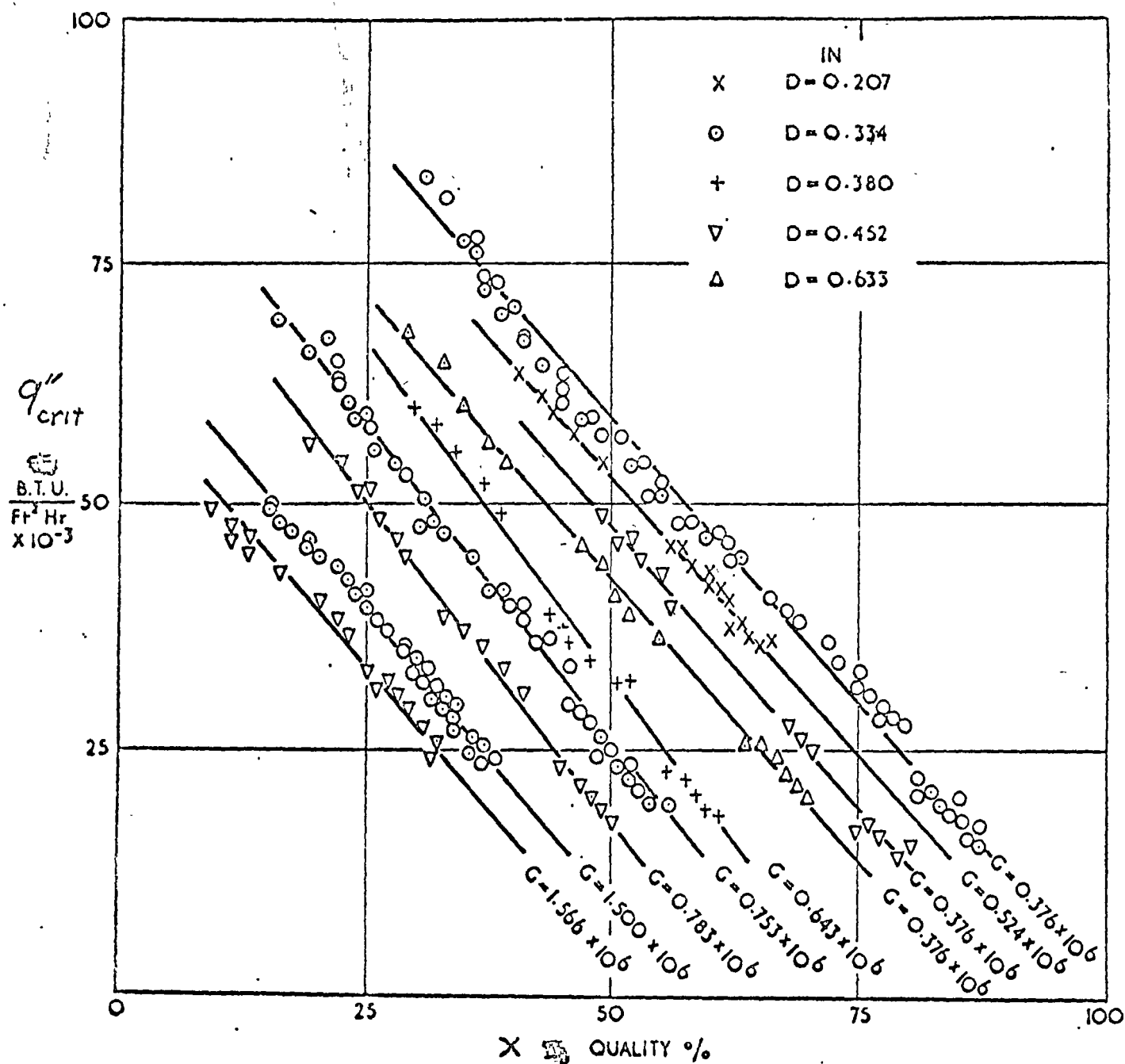


FIGURE 4-5 CRITICAL HEAT FLUX FOR FREON-12 AT 155 psia. KIRBY [114]

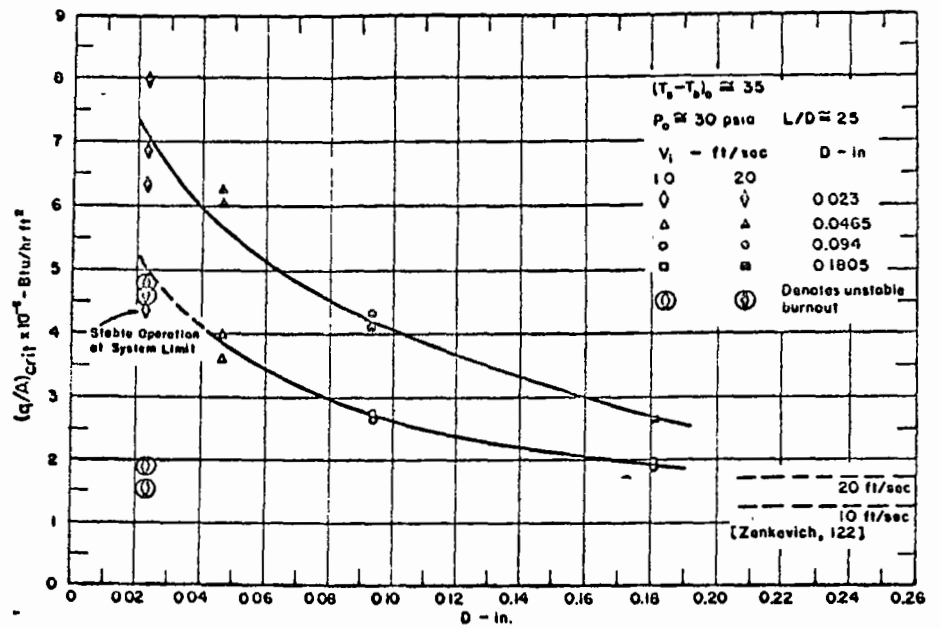


Fig. 46 Dependence of burnout heat flux on tube diameter and velocity for surface boiling [5]. 115

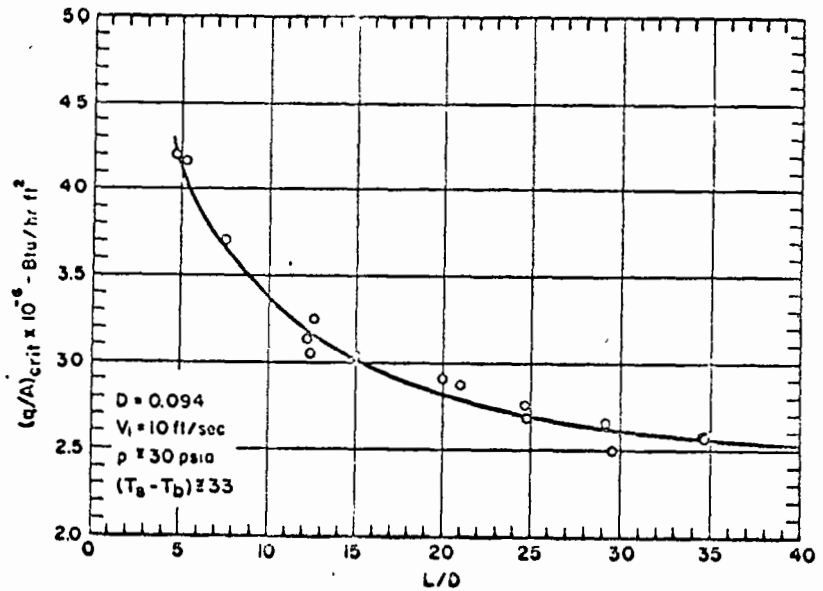


Fig. 47. Dependence of burnout heat flux on length-diameter ratio [5]. 115

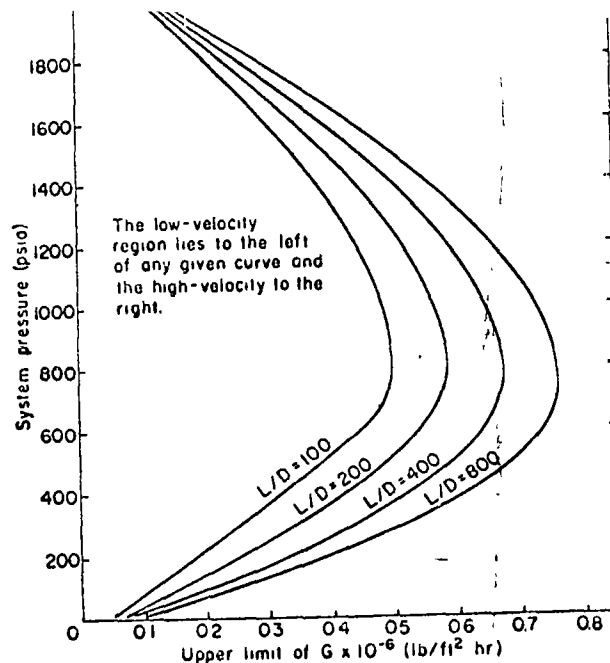


Fig. 48 Approximate boundary limits of the low-velocity and high-velocity burnout regimes for round tubes [Macbeth, 129].

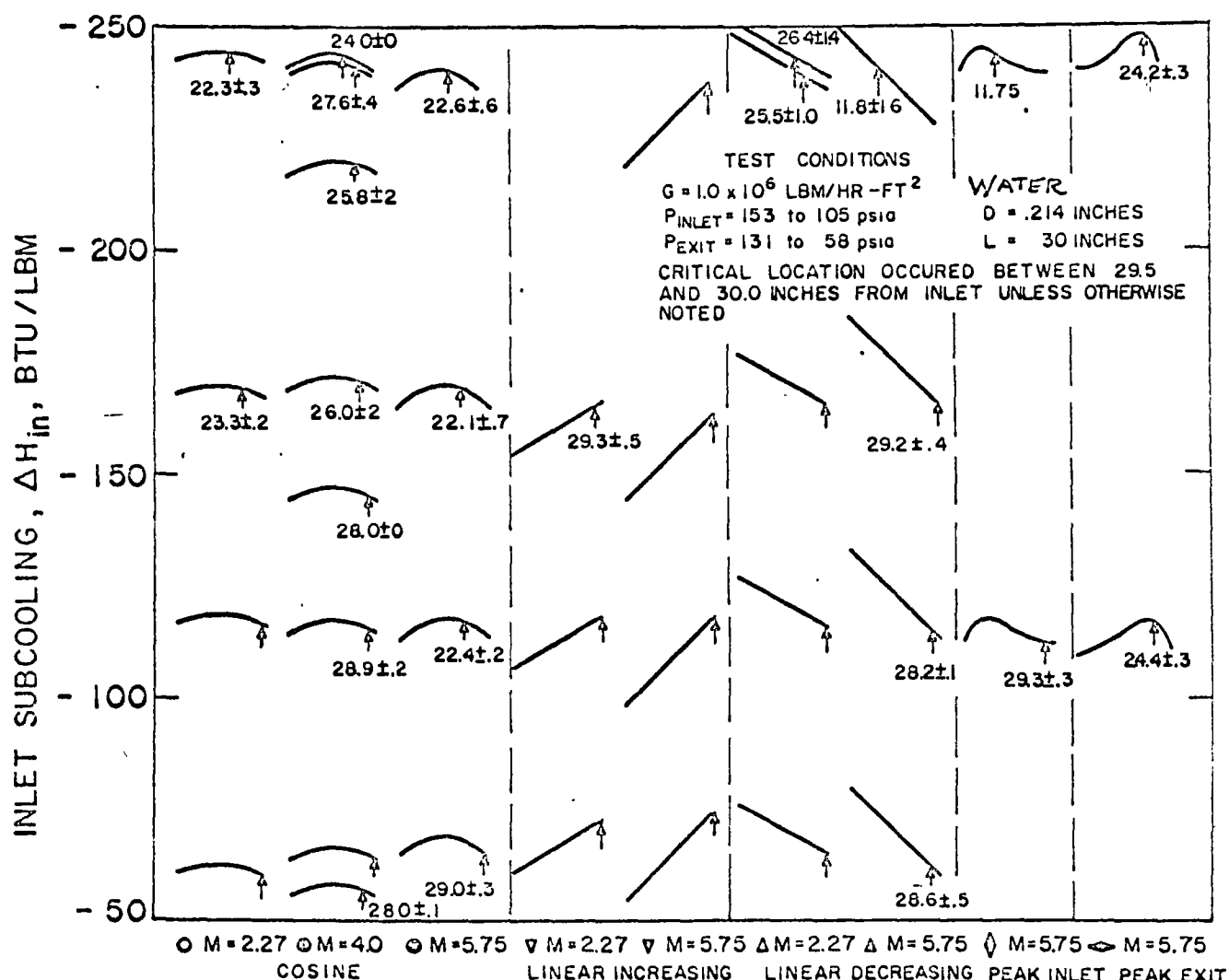


FIGURE 4-9. CRITICAL LOCATIONS (INCHES FROM INLET) FOR THE NONUNIFORM AXIAL FLUX DISTRIBUTIONS INVESTIGATED AT $G = 1.0 \times 10^6$ LBM/HR-FT² [127]

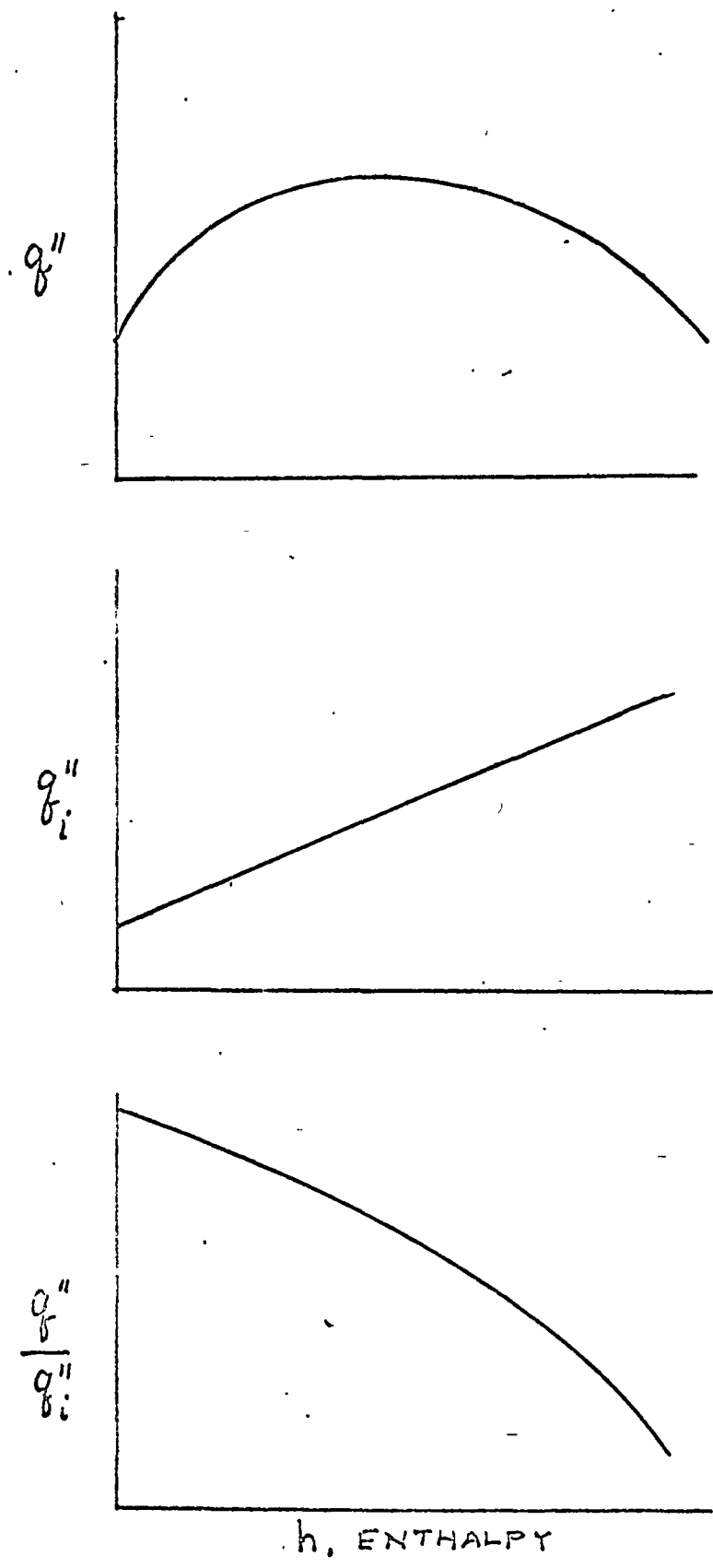


FIG. 50 HEAT FLUX DISTRIBUTION

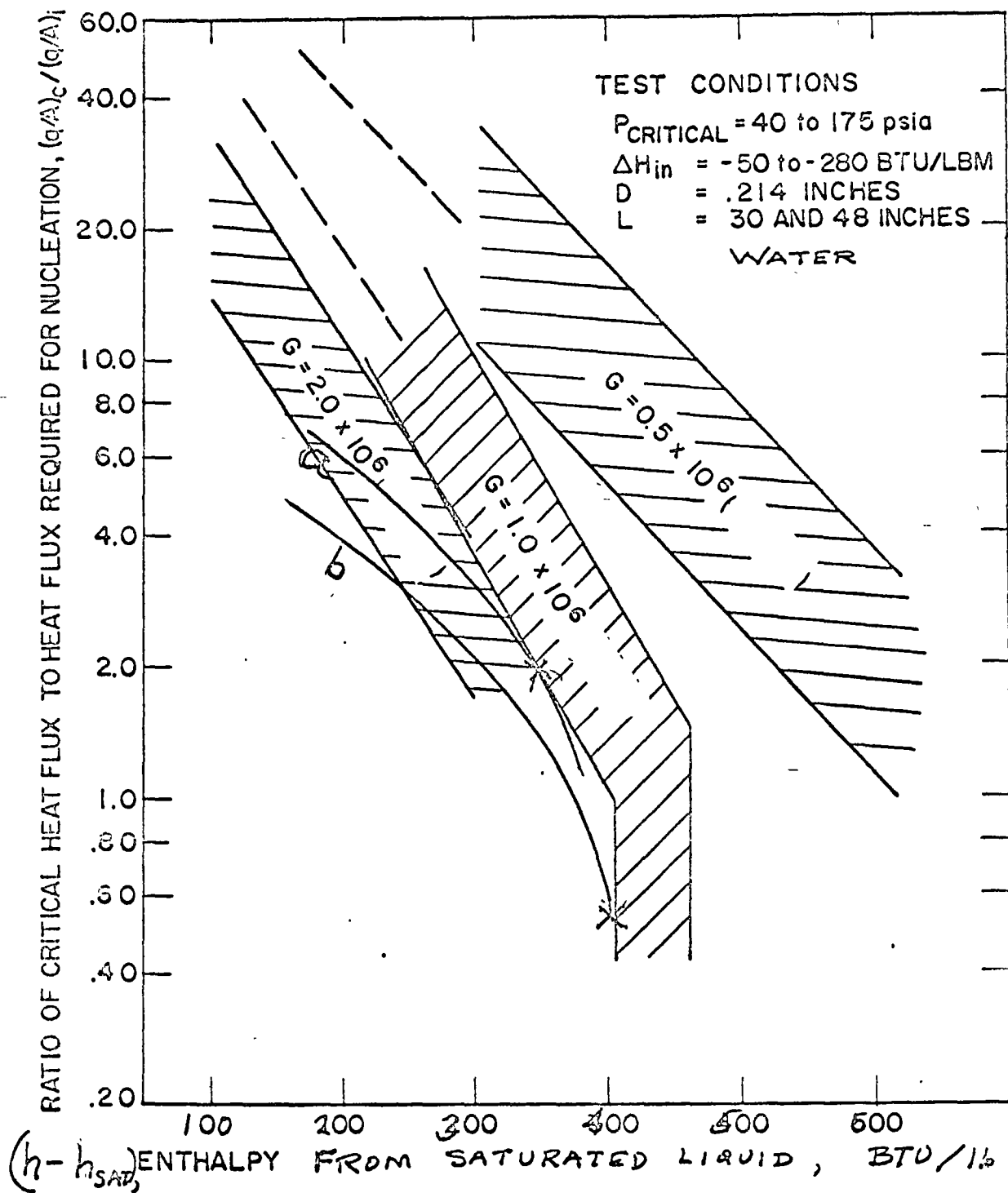


FIGURE 5/ COMPARISON OF CRITICAL FLUX RESULTS FOR THE RANGE OF MASS VELOCITIES INVESTIGATED, TOPREAS(127)

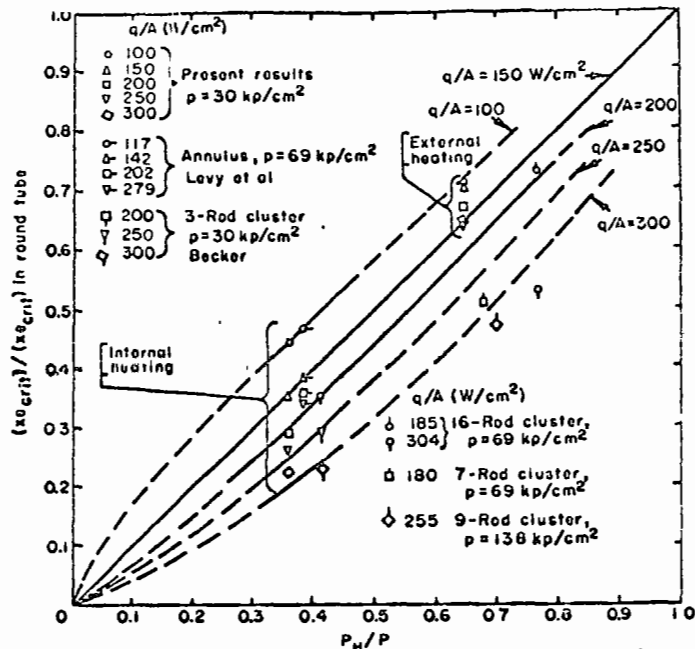


Fig. 52. Effect of perimeter ratio on critical conditions [130].

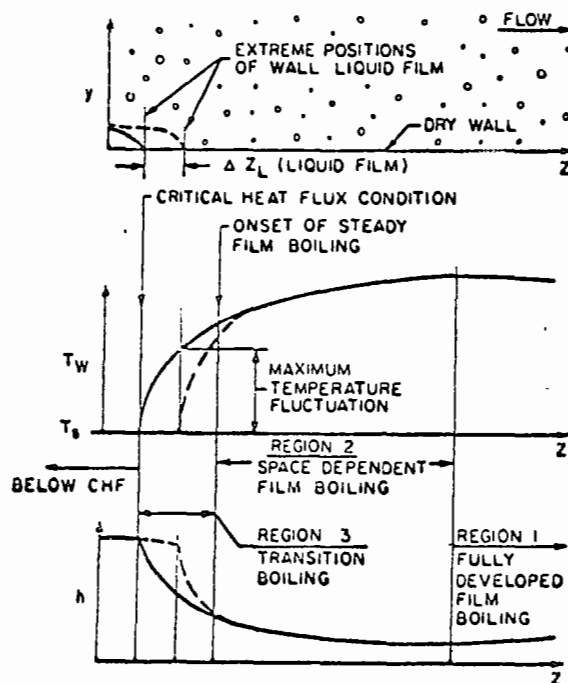
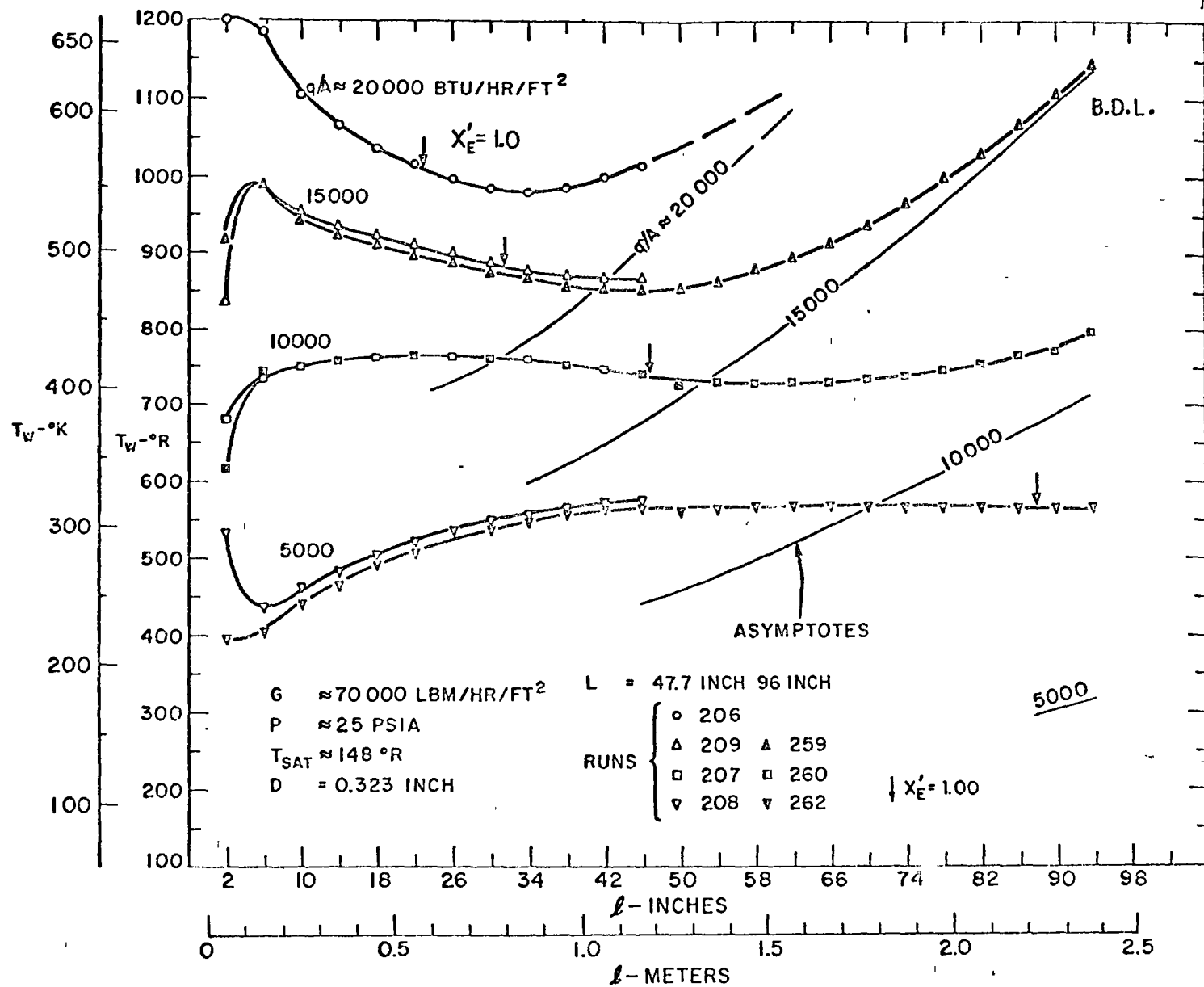


Fig. 53 Heat-transfer behavior beyond the CHF (uniform heat flux) [142]

Fig. 54. Wall Temperatures: film boiling, forced convection.
Lipid Nitrogen, 0.323 " ID tube. Forslund(144)



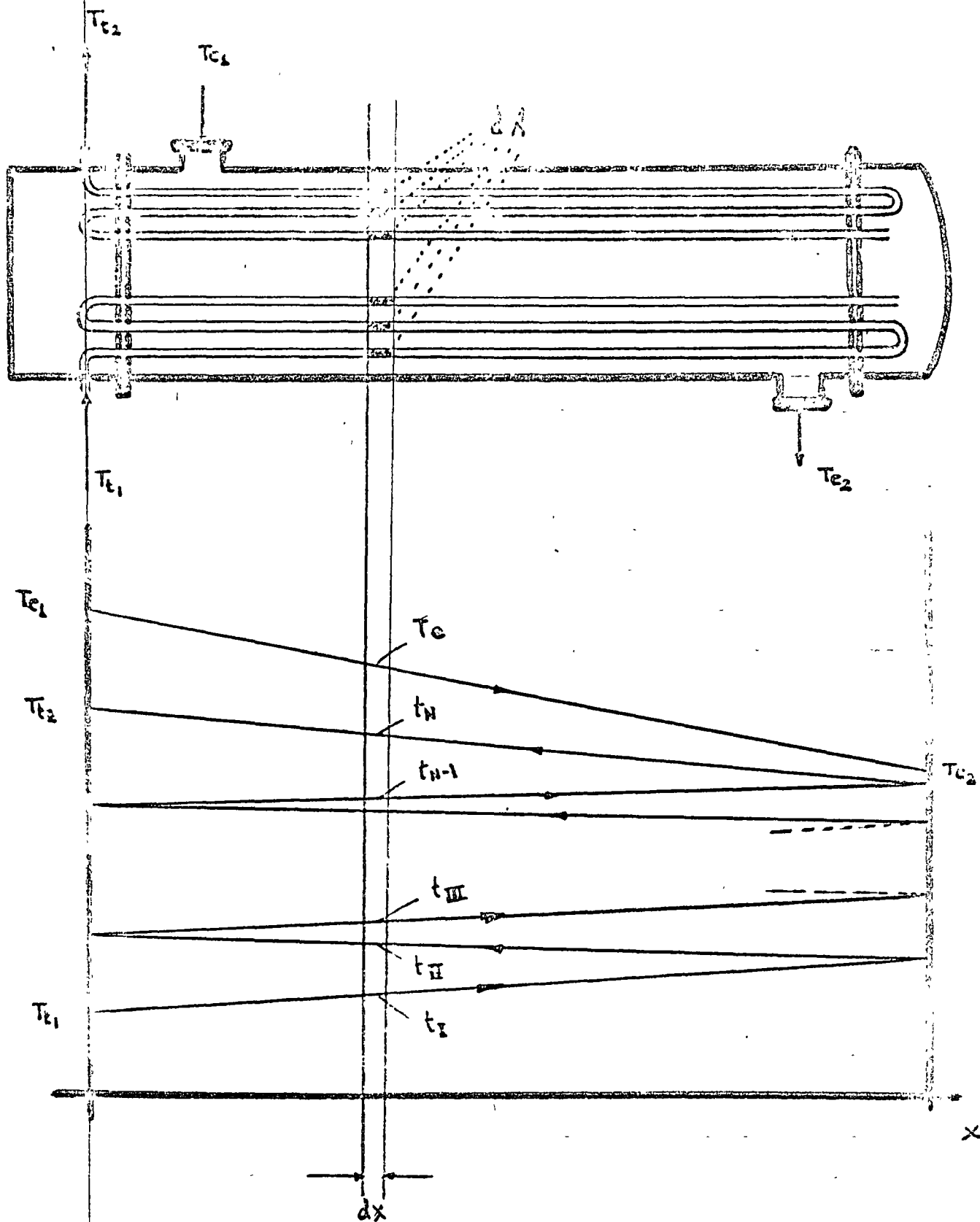
NOTAS PARA EL CURSO DE TRANSFERENCIA DE CALOR

" APLICACIONES EN CAMBIADORES DE CALOR "

Julián Castellanos Fernández
(INSTITUTO MEXICANO DEL PETROLEO)

CONTENIDO:

	Pag
1 Introducción	1
2 Modelo Matemático	1
3 Coeficientes Individuales de Transfe- rencia de Calor.	10
4 Caídas de Presión	18
5 Secuencia General de Cálculo para el Diseño de Cambiadores de Calor x	22
6 Simulación de Trenes de Intercambio Calorífico.	26
7 Optimización	30
8 Bibliografía	42
9 Apéndice	43



Transferencia de calor por paso en el elemento diferencial:

$$\text{Paso 1} \quad Wt \, C_{ptI} \, dt_I = U_{IA}(T_e - t_I)dx \text{-----}(1)$$

$$\text{Paso 2} \quad -Wt \, C_{ptII} \, dt_{II} = U_{II}A(T_e - t_{II})dx \text{-----}(2)$$

$$\text{Paso 3} \quad Wt \, C_{ptIII} \, dt_{III} = U_{III}A(T_e - t_{III})dx \text{-----}(3)$$

.....

$$\text{Paso N-I} \quad Wt \, C_{pt_{N-I}} \, dt_{N-I} = U_{N-I}A(T_e - t_{N-I})dx \text{-----}(4)$$

$$\text{Paso N} \quad -Wt \, C_{ptN} \, dt_N = U_N A(T_e - t_N)dx \text{-----}(5)$$

La capacidad calorífica puede considerarse una función lineal de la temperatura:

$$C_{pt_i} = a + b \, t_i \text{-----}(6)$$

El coeficiente total de transferencia de calor U_i representa la inversa de la suma de las resistencias individuales a la transferencia que presentan los fluidos por dentro de los tubos y por la envólvente y puede expresarse en términos de los coeficientes individuales de película:

$$U_i = \frac{h_i \, h_o}{h_i + h_o} \text{-----}(7)$$

Substituyendo ecuaciones (6) y (7) en ecuaciones (1) a (5) :

$$\frac{dt_I}{dx} = \frac{A}{Wt} \frac{1}{a + b \, t_I} \frac{h_{iI} \, h_{oI}}{h_{iI} + h_{oI}} (T_e - t_I) \text{-----}(8)$$

$$-\frac{dt_{II}}{dx} = \frac{A}{Wt} \frac{1}{a + b \, t_{II}} \frac{h_{iII} \, h_{oII}}{h_{iII} + h_{oII}} (T_e - t_{II}) \text{-----}(9)$$

.....

$$\frac{dt_{N-I}}{dx} = \frac{A}{Wt} \frac{1}{a + b \, t_{N-I}} \frac{h_{i_{N-I}} \, h_{o_{N-I}}}{h_{i_{N-I}} + h_{o_{N-I}}} (T_e - t_{N-I}) \text{-----}(10)$$

$$-\frac{dt_N}{dx} = \frac{A}{Wt} \frac{1}{a + b \, t_N} \frac{h_{iN} \, h_{oN}}{h_{iN} + h_{oN}} (T_e - t_N) \text{-----}(11)$$

Balance de Calor en el Elemento Diferencial del Cambiador:

Calor cedido por el fluido por la envolvente = Calor ganado por el fluido por los tubos.

$$Q_t = -Q_c \quad \text{-----} (12)$$

$$Q_t = W_t \left[\int_{t_I}^{t_I+dt_I} (a+bt_I) dt_I - \int_{t_{II}}^{t_{II}+dt_{II}} (a+bt_{II}) dt_{II} + \int_{t_{III}}^{t_{III}+dt_{III}} (a+bt_{III}) dt_{III} + \dots + \int_{t_{N-I}}^{t_{N-I}+dt_{N-I}} (a+bt_{N-I}) dt_{N-I} - \int_{t_N}^{t_N+dt_N} (a+bt_N) dt_N \right] \quad \text{-----} (13)$$

$$Q_e = W_e \int_{T_e}^{T_e+dT_e} (A+BT_e) dT_e \quad \text{-----} (14)$$

El conjunto de ecuaciones diferenciales (8) a (11), (13) y (14) constituyen el modelo matemático representativo de un elemento diferencial de un cambiador de calor, y el modelo correspondiente al cambiador integral quedará determinado por la integración de las ecuaciones diferenciales mencionadas.

Para la integración de estas ecuaciones se introducen una serie de simplificaciones al modelo que permiten su resolución mediante procedimientos analíticos y las ecuaciones integradas tendrán aplicación en aquellos equipos donde las simplificaciones efectuadas en el modelo matemático tengan justificación en el modelo físico correspondiente.

A continuación se presentan algunas de las ecuaciones integradas, derivadas del modelo diferencial presentado, indicando las simplificaciones que se introdujeron para su derivación:

a) Un cambiador de calor con un sólo paso por los tubos, considerando que las capacidades caloríficas de los dos fluidos y el coeficiente total de transferencia (U) son constantes independientes de la temperatura:

$$Q = U A \frac{(T_{e1} - T_{t2}) - (T_{e2} - T_{t1})}{\ln \frac{(T_{e1} - T_{t2})}{(T_{e2} - T_{t1})}} \quad \text{-----} (15)$$

b) Un cambiador de calor con un solo paso por los tubos, considerando que las capacidades caloríficas de ambos fluidos son constantes independientes de la temperatura y que el coeficiente total de transferencia es una función lineal de la temperatura del fluido por dentro de la envolvente:

$$Q = A \frac{U_1 \Delta T_2 - U_2 \Delta T_1}{\ln \frac{(U_1 \Delta T_2)}{(U_2 \Delta T_1)}} \quad \text{-----} (16)$$

c) Un cambiador con dos pasos por los tubos, considerando que las capacidades caloríficas de ambos fluidos y el coeficiente total de transferencia de calor (U) son constantes independientes de la temperatura:

$$Q = U A \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} F_t \text{ ----- (17)}$$

$$F_t = \frac{\sqrt{R^2 + 1} \ln \frac{1-S}{1-R S}}{(R-1) \ln \frac{2 - S(R+1 - \sqrt{R^2 + 1})}{2 - S(R+1 + \sqrt{R^2 + 1})}} \text{ ----- (18)}$$

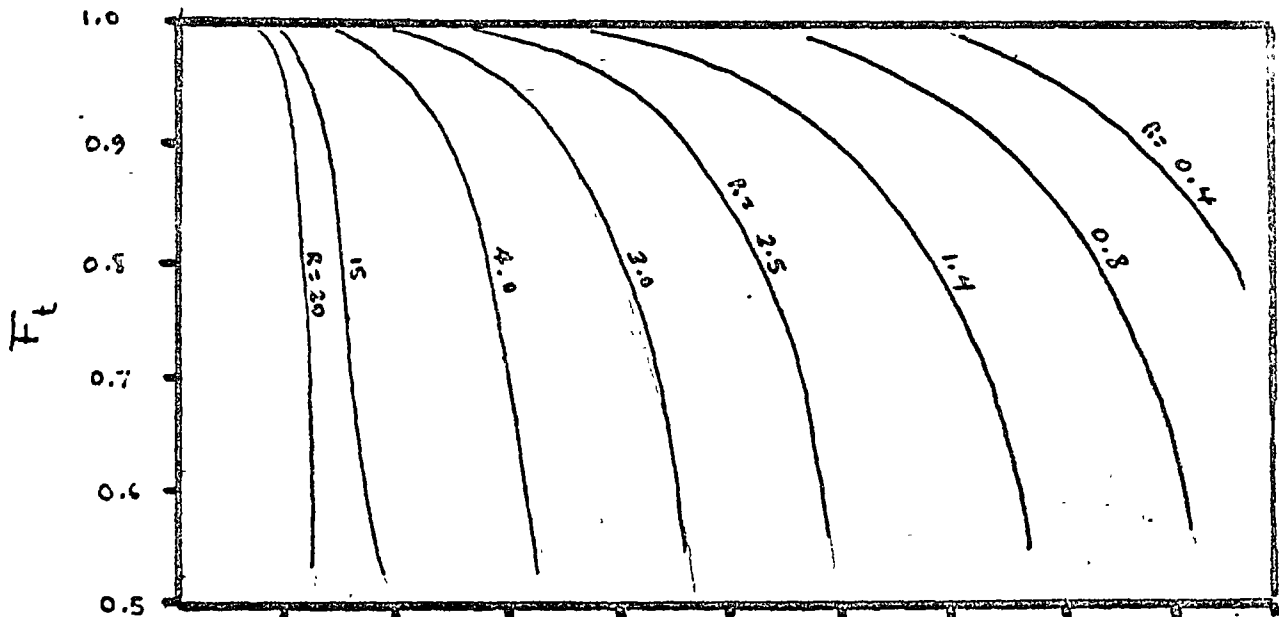
d) N cambiadores en serie con dos pasos cada uno por los tubos, considerando que las capacidades caloríficas de ambos fluidos y el coeficiente total de transferencia (U) son constantes independientes de la temperatura:

$$Q = U A \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} F_t \text{ ----- (19)}$$

El factor F_t se presenta para su cálculo como una serie de gráficas de F_t contra S a diferentes parámetros de R

$$R = \frac{T_{e1} - T_{e2}}{T_{t2} - T_{t1}} \text{ ----- (20)}$$

$$S = \frac{T_{t2} - T_{t1}}{T_{e1} - T_{t1}} \text{ ----- (21)}$$



La estimación del factor F_t mediante el uso de estas gráficas presenta interés práctico para cálculos de escritorio, sin embargo para un cálculo estricto con computadoras resulta preferible utilizar las expresiones analíticas correspondientes:

Siendo n el número de cambios en serie y R y S funciones de las temperaturas de entrada y salida de los fluidos por los tubos y la envolvente, definidas por las ecuaciones (20) y (21), se tiene:

Para $R=1$

$$X = \frac{S}{n - nS + S} \text{ ----- (22)}$$

$$F_t = \frac{1.414 \left(\frac{X}{1-X} \right)}{\text{Ln} \left(\frac{\frac{2}{X} - 1 - R + \sqrt{R^2 + 1}}{\frac{2}{X} - 1 - R - \sqrt{R^2 + 1}} \right)} \text{ ----- (23)}$$

Para $R \neq 1$

$$X = \frac{1 - \left(\frac{1-RS}{1-S} \right)^{\frac{1}{n}}}{R - \left(\frac{1-RS}{1-S} \right)^{\frac{1}{n}}} \text{ ----- (24)}$$

$$F_t = \frac{\frac{\sqrt{R^2 + 1}}{R - 1} \text{Ln} \left(\frac{1-X}{1-RX} \right)}{\text{Ln} \left(\frac{\frac{2}{X} - 1 - R + \sqrt{R^2 + 1}}{\frac{2}{X} - 1 - R - \sqrt{R^2 + 1}} \right)} \text{ ----- (25)}$$

e) Resolución estricta del modelo matemático sin introducir ninguna simplificación:

Como se dijo anteriormente, el conjunto de ecuaciones diferenciales (8) a (11) representan el comportamiento de un elemento diferencial de un cambiador de calor. Cuando se tengan M cambiadores con N pasos por los tubos, se tendría un conjunto de NxM ecuaciones diferenciales, esto es, una ecuación por cada paso de cada cambiador, que expresa la variación de la temperatura correspondiente con respecto a la longitud del cambiador.

Para la resolución de un sistema como éste, resulta necesario recurrir a un método numérico de integración, de los cuales puede elegir el método de Runge-Kutta:

Para n ecuaciones diferenciales:

$$\frac{dy_i}{dx} = F_i(x, y_i), \quad i=1, n \quad \text{-----} (26)$$

Dado un incremento finito (Δx) a la variable independiente, (x), los incrementos correspondientes (Δy_i) de las variables dependientes (y_i), involucra el cálculo de cuatro grupos de constantes:

Primer Grupo:

$$K(i, 1) = \Delta x F_i(x, y_i), \quad i=1, n \quad \text{-----} (27)$$

Segundo Grupo:

$$K(i, 2) = \Delta x F_i(x + \Delta x/2, y_i + K(i, 1)/2), \quad i=1, n \quad \text{-----} (28)$$

Tercer Grupo:

$$K(i, 3) = \Delta x F_i(x + \Delta x/2, y_i + K(i, 2)/2), \quad i=1, n \quad \text{-----} (29)$$

Cuarto Grupo:

$$K(i, 4) = \Delta x F_i(x + \Delta x, y_i + K(i, 3)), \quad i=1, n \quad \text{-----} (30)$$

Una vez calculados los cuatro grupos de constantes, los incrementos correspondientes Δy_i pueden obtenerse:

$$y_i = 1/6 (K(i, 1) + 2K(i, 2) + 2K(i, 3) + K(i, 4)), \quad i=1, n \quad \text{-----} (31)$$

En las ecuaciones diferenciales (8) a (11) y en general en las NxM ecuaciones que se tendrían para M cambiadores, está involucrada la temperatura de la envolvente T_e , pudiéndose establecer una relación entre ésta y las temperaturas $t_I, t_{II}, \dots, t_N$ mediante un balance de calor desde el extremo izquierdo del cambiador (Fig 1) y cualquier punto intermedio a una distancia x .

$$Q = Wt \left[\int_{t_{I_0}}^{t_I} (a + bt_I) dt_I - \int_{t_{II_0}}^{t_{II}} (a + bt_{II}) dt_{II} + \dots + \int_{t_{N-I_0}}^{t_{N-I}} (a + bt_{N-I}) dt_{N-I} - \int_{t_{N_0}}^{t_N} (a + bt_N) dt_N \right] \quad \text{-----} (32)$$

$$Q = W_t \left[(t_{T1} - t_{T0}) \left\{ a + b/2(t_{T1} - t_{T0}) \right\} - (t_{T11} - t_{T10}) \left\{ a + b/2(t_{T11} + t_{T10}) \right\} + \dots \right. \\ \left. + (t_{N-1} - t_{N-10}) \left\{ a + b/2(t_{N-1} + t_{N-10}) \right\} - (t_N - t_{N0}) \left\{ a + b/2(t_N + t_{N0}) \right\} \right] \text{-----(33)}$$

La expresión (33) representa el calor ganado por el fluido por dentro de tubos en el tramo de cambiador correspondiente a $x = 0$ al $x = x$ y debe ser igual al calor cedido por el fluido dentro de la envolvente en el mismo tramo:

$$Q = -W_e \int_{T_{e1}}^{T_e} (A + BT_e) dT_e \text{------(34)}$$

$$Q = -W_e \left[A(T_e - T_{e1}) + B/2(T_e^2 - T_{e1}^2) \right] \text{------(35)}$$

De donde se puede despejar:

$$T = -A/B \pm \sqrt{(A/B)^2 + 2/B(AT_{e1} + B/2T_{e1}^2 - Q/W_e)} \text{------(36)}$$

El signo del radical será positivo cuando la variación de la capacidad calorífica sea directa con respecto a la temperatura ($B > 0$) y negativo cuando la variación sea inversa ($B < 0$).

Para poder resolver el sistema de ecuaciones diferenciales por métodos numéricos, es necesario conocer los valores iniciales de las variables dependientes (a $x=0$) ya que a partir de éstos, se calculan los incrementos correspondientes a un incremento de x . Para el caso del sistema de ecuaciones analizado, es necesario partir de valores para: t_{T0} , t_{T10} , ..., t_{N0} de los cuales sólo t_{T0} es conocido ya que corresponde a la temperatura de entrada del fluido por los tubos.

La naturaleza del problema, implica pues la necesidad del empleo de un método iterativo para su resolución.

Analizando la Figura 1 tenemos que: Para $x=0$ $t_i = t_{i+1}$ donde i es número par, y para $x=L$ $t_i = t_{i+1}$ donde i es número non. El método iterativo puede consistir pues en los siguientes pasos:

- a) Suponer valores iniciales que cumplan con las condiciones para $x=0$
- b) Resolver las ecuaciones diferenciales hasta $x=L$
- c) Checar que se cumplan las condiciones para $x=L$
- d) Si las condiciones checan el cálculo se da por terminado y en caso contrario se vuelven a suponer valores iniciales y se repite el cálculo.

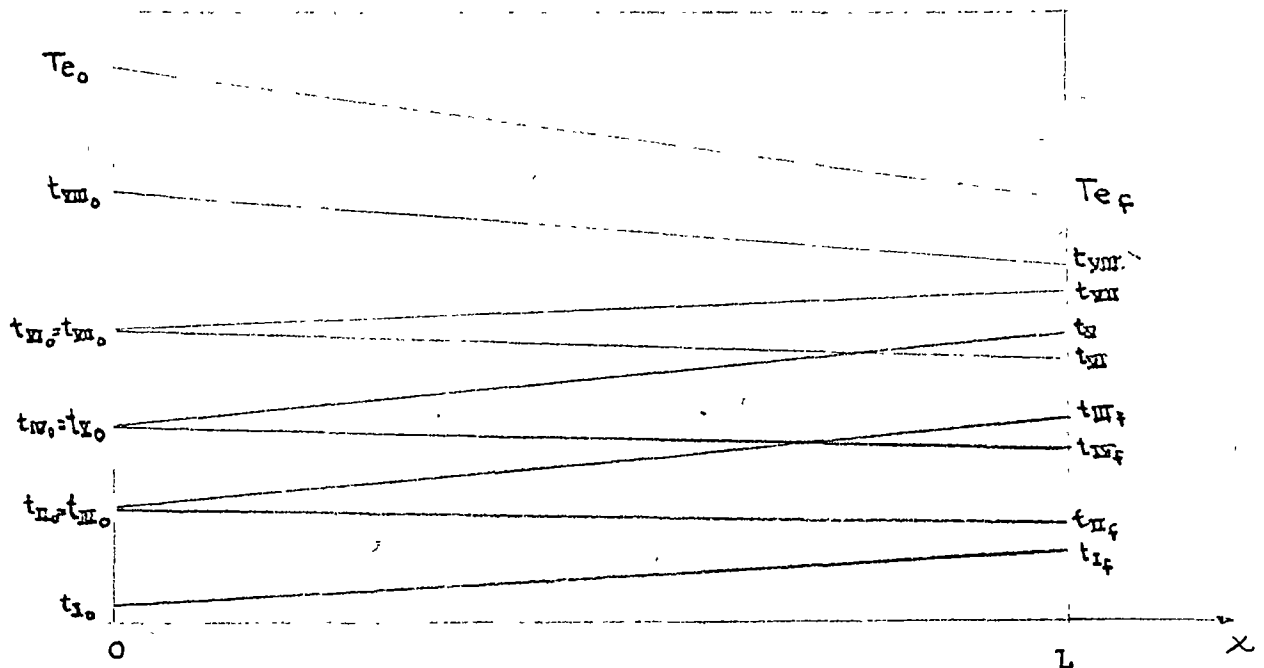
El procedimiento que se sigue para resuponer los valores iniciales de las variables independientes es el siguiente:

A) Se calcula la suma de calores transmitidos en cada par de pasos 1-2; 3-4; ... N-1-N y en general $i-i+1$

B) Se determinan los nuevos valores iniciales de temperaturas, haciendo un balance de calor de los puntos 1 a 2; 3a4; ... N-1 a N despejando las temperaturas de los puntos pares en función de la de los puntos nones y el calor calculado en (A) y se igualan las temperatu -

iones $(i+1)$ a las pares (i) de acuerdo a las condiciones requeridas a $x=0$.

Este procedimiento queda más claro mediante una figura que muestre los perfiles de temperatura obtenidos cuando las suposiciones iniciales no son las correctas:



- Después de haber calculado un perfil de temperaturas correspondiente al del diagrama, los nuevos valores iniciales de temperaturas para la iteración siguiente se calcularían siguiendo la siguiente secuencia:
- Calcular calor transferido en los pasos I y II por balance entre las temperaturas $(t_{I0}$ a $t_{IF})$ y $(t_{IIF}$ a $t_{II0})$
 - Calcular nueva temperatura t_{II0} por balance de calor entre las temperaturas $(t_{I0}$ a $t_{II0})$ usando el calor calculado en A)
 - Calcular calor transferido en los pasos III y IV por balance entre las temperaturas $(t_{III0}$ a $t_{IIIF})$ y $(t_{IV0}$ a $t_{IVF})$
 - Calcular nueva temperatura t_{IV0} por balance de calor entre las temperaturas $(t_{III0}$ a $t_{IV0})$ usando t_{III0} igual al nuevo valor de t_{II0} calculado en B) y con el calor calculado en C)
 - Calcular calor transferido en los pasos V y VI por balance entre las temperaturas $(t_{V0}$ a $t_{VF})$ y $(t_{VIF}$ a $t_{VI0})$
 - Calcular nueva temperatura t_{VI0} por balance de calor entre las temperaturas $(t_{V0}$ a $t_{VI0})$ usando t_{V0} igual al valor calculado para t_{IV0} en D) y el calor calculado en E)
 - Calcular calor transferido en los pasos VII y VIII por balance de calor entre las temperaturas $(t_{VII0}$ a $t_{VIIF})$ y $(t_{VIIIF}$ a $t_{VIII0})$
 - Calcular nueva temperatura t_{VIII0} por balance de calor entre las temperaturas $(t_{VII0}$ a $t_{VIII0})$ usando t_{VII0} igual al nuevo valor de t_{VI0} calculado en F) y el calor calculado en G)

- I) Igualar temperaturas ($t_{TII_0} = t_{TIO}$) ; ($t_{V_0} = t_{VIO}$) ; ($t_{VII_0} = t_{VIO}$)
 J) Resolver nuevamente el sistema de ecuaciones diferenciales con los nuevos valores iniciales calculados para las temperaturas de cada uno de los pasos y volver a obtener el perfil de temperaturas.

3. COEFICIENTES INDIVIDUALES DE TRANSFERENCIA DE CALOR

En un cambiador de calor ocurre una transferencia de calor de un fluido caliente a otro frío que se ponen en contacto a través de una superficie metálica y que en el caso de un cambiador de haz de tubos y envolvente corresponde a la superficie de los tubos. La transferencia de calor es provocada por la diferencia de temperaturas entre los dos fluidos presentándose una serie de resistencias a dicha transferencia constituidas por:

- A) Resistencia a la transferencia de calor por conducción en la pared metálica, la cual viene dada en función de las dimensiones del tubo y de la conductividad térmica del material del que está constituido:

$$R_m = \frac{D_2}{2k} \ln \frac{D_2}{D_1} \text{-----} (37)$$

- B) Resistencia a la transferencia de calor por convección presentada por el fluido por dentro de los tubos y que corresponde a la inversa del coeficiente individual de transferencia: h_i
 C) Resistencia a la transferencia de calor por convección presentada por el fluido por la envolvente y que corresponde a la inversa del coeficiente individual de película h_o
 D) Resistencia a la transferencia de calor presentada por las incrustaciones que se forman poco a poco sobre la superficie metálica tanto en la pared externa como la interna. Estas resistencias tienen el valor de cero cuando el cambiador está limpio existiendo valores recomendados para las mismas para diferentes tipos de fluidos, valores que corresponden a la resistencia estimada por incrustación después de un periodo de operación del cambiador de 6 meses: R_{di} y R_{do}

La resistencia total a la transferencia vendrá dada por la suma de las resistencias individuales y corresponde a la inversa del coeficiente total de transferencia de calor:

$$R_t = \frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{D_2}{2k} \ln \frac{D_2}{D_1} + R_{di} + R_{do} \text{-----} (38)$$

Para calcular los coeficientes individuales de película h_i y h_o se dispone de correlaciones semiempíricas determinadas experimentalmente

Estas correlaciones están dadas en su gran mayoría como ecuaciones que establecen la relación de diversos grupos adimensionales:

Número de Nusselt:

$$Nu = \frac{h d}{k}$$

Número de Reynolds:

$$Re = \frac{d v \rho}{\mu}$$

Número de Prandtl:

$$Pr = \frac{C_p \mu}{k}$$

y la forma más común de la ecuación que establece la relación entre estos grupos adimensionales es:

$$Nu = a Re^b Pr^c$$

donde las constantes a b y c son determinadas experimentalmente.

Coefficientes Individuales por dentro de Tubos:

Existen varias correlaciones para los coeficientes de película por dentro de tubos, reportadas en la literatura, las cuales se pueden dividir en tres grupos atendiendo al régimen de flujo:

A) Régimen Laminar ($Re < 2100$)

Ecuación de Sieder y Tate:

$$Nu = 2.02 Gr^{.3333} \quad (39)$$

Ecuación de Drew:

$$Nu = 1.75 Gr^{.3333} \quad (40)$$

Ecuación de Colburn:

$$Nu = 1.65 Gr^{.3333} (1 + 0.15 Gr)^{.3333} \quad (41)$$

Ecuación de Ruthod:

$$Nu = 1.06 (Re Gr Pr^2)^{.113} \quad (42)$$

Ecuación Eusank:

$$Nu = 1.75 [(Gr + 0.4)(D/L Gr Pr)^{.75}]^{.3333} \quad (43)$$

Ecuación de Kern-Othnor:

$$Nu = 2.02 Gr^{.3333} \psi \quad (44)$$

Ecuación de Oliver:

$$Nu = 1.75 [Gr + .00056(Gr Pr L/D)^{.7}]^{.3333} \quad (45)$$

Donde:

$$Gr = \text{No de Graetz} = \frac{U C_p}{k L} \quad (46)$$

$$\psi = \text{Factor de corrección por conducción natural} = \frac{2.25(1 + .01 Gr^{.3333})}{\ln(Re)} \quad (47)$$

U = Gasto en masa del fluido por los tubos lb/h

Cp = Calor específico del fluido por los tubos Btu/h °F

k = Conductividad térmica del fluido por dentro de los tubos Btu/h ft °F

L = Longitud de los tubos ft

$$Gr = \frac{D^3 \rho^2 g \beta \Delta t_a}{\mu^2}$$

g = aceleración de la gravedad $4.18 \cdot 10^8$ ft/h²

β = factor de expansión volumétrica del fluido 1/°F

Δt_a = diferencia de temperaturas entre el fluido y la pared del tubo °F

ρ = densidad del fluido lb/ft³

μ = viscosidad del fluido lb/ft h

D = diámetro interno de tubos ft

B) Régimen Turbulento ($Re \geq 10000$)

Ecuación de Sieder y Tate:

$$Nu = 0.027 Re^{0.8} Pr^{0.3333} \quad (48)$$

Ecuación de Van Glann:

$$Nu = \frac{6.8}{1 + Pr} + 0.023 \left(1 - \frac{1}{A}\right) Re^{0.8} Pr^m \quad (49)$$

donde:

$$A = 1 + 19 Pr + 100 Pr^3 \quad (50)$$

$$m = 0.41 Re^{0.04} \left(\frac{1}{1 + Pr}\right)^{0.1} \quad (51)$$

Ecuación de Weismann:

$$Nu = 0.023 Re^{0.8} Pr^m \quad (52)$$

donde:

$$m = 1 - 1.22 Re^{-0.07} \quad (53)$$

C) Régimen Transicional ($2100 \leq Re \leq 10000$)

Ecuación de Norris:

$$Nu = 0.0067 Re Pr^{0.2} \quad (54)$$

Ecuación de Kuznetson y Leonenke:

$$Nu = 0.021 Re^{0.8} Pr^{0.4} f(Re) F(Re) Gr \quad (55)$$

Donde:

$$f(Re) = 1.14 \frac{Re - 1800}{Re - 710} \quad (56)$$

$$F(Re) = -0.167 \ln(f(Re)/2.3) \quad (57)$$

Ecuación de Hausen:

$$Nu = 0.116 + (Re^{0.6666} - 125) Pr^{0.3333} \cdot 1 + (D/L)^{0.6666} \quad (58)$$

Ecuación de Metais:

$$Nu = 4.69 Re^{0.27} Pr^{0.21} Gr^{0.07} (D/L)^{0.36} \quad (59)$$

Correlación Gráfica de Kern:

Esta correlación corresponde a una gráfica de un factor J_h dado a diferentes valores de Re y con diferentes parámetros de (L/D) . Determinado este valor de las gráficas, se tiene la siguiente relación:

$$Nu = J_h Pr^{0.3333} \left(\frac{L}{D}\right)^{0.14} \quad (60)$$

Coefficientes Individuales de Película por la Envolvente.

A) Método de Kern

Kern presenta una correlación gráfica de un factor Jh en función del número de Reynolds:

$$Re = \frac{De G}{\mu} \text{-----} (61)$$

Donde:

$$De = \text{Diámetro equivalente} = \frac{4 (Pt^2 - \pi Do^2/4)}{12 \pi Do} \text{-----} (62)$$

Pt = Distancia entre centros de tubos in

Do = Diámetro externo de tubos in

G = Masa velocidad = Ws/As lb/h ft²

Ws = Flujo del fluido por la envolvente lb/h

As = Área de flujo = $Ds C B/144$ Pt ft²

Ds = Diámetro interno de la envolvente in

C = Claro entre tubos = Pt - Do in

B = Distancia entre mamparas in

Una vez calculado el factor Jh el coeficiente de película se calcula de acuerdo a la ecuación:

$$Nu = Jh Pr^{0.3333} \left(\frac{\mu}{\mu_w} \right)^{0.14} \text{-----} (63)$$

B) Método de Donohue

Donohue propone una correlación entre los números adimensionales de Nusselt, Reynolds y Brandt:

$$Nu = 0.22 Re^{0.6} Pr^{0.3333} \left(\frac{\mu}{\mu_w} \right)^{0.14} \text{-----} (64)$$

Donde el número de Reynolds está definido en base a una masa velocidad media geométrica entre las masas velocidades transversal y longitudinal:

$$Re = \frac{G_m Do}{\mu} \text{-----} (65)$$

$$G_m = \sqrt{G_t G_l}$$

Gt = Masa velocidad transversal = Ws/At lb/h ft²

At = Área de flujo transversal ft² = $(Ds - E)B/144$

Ds = Diámetro interno de la envolvente in

E = Distancia entre mamparas in

B = Espacio ocupado por los tubos en un diámetro de la envolvente in = $Nt Do$

Do = Diámetro externo de tubos in

Nt = Número de tubos en un diámetro de la envolvente =

$$= \frac{Do - C}{nt} + 1$$

P_t = Distancia entre centros de tubos in

C = Claro entre tubos in

G_l = Masa velocidad longitudinal lb/h ft² = W_s/A_l

A_l = Area de flujo en la ventana de las mamparas ft²

$$= \frac{\% \text{Corte } 4(D_s^2 - N_{tt}D_o^2)}{144}$$

N_{tt} = Número de tubos totales en el haz.

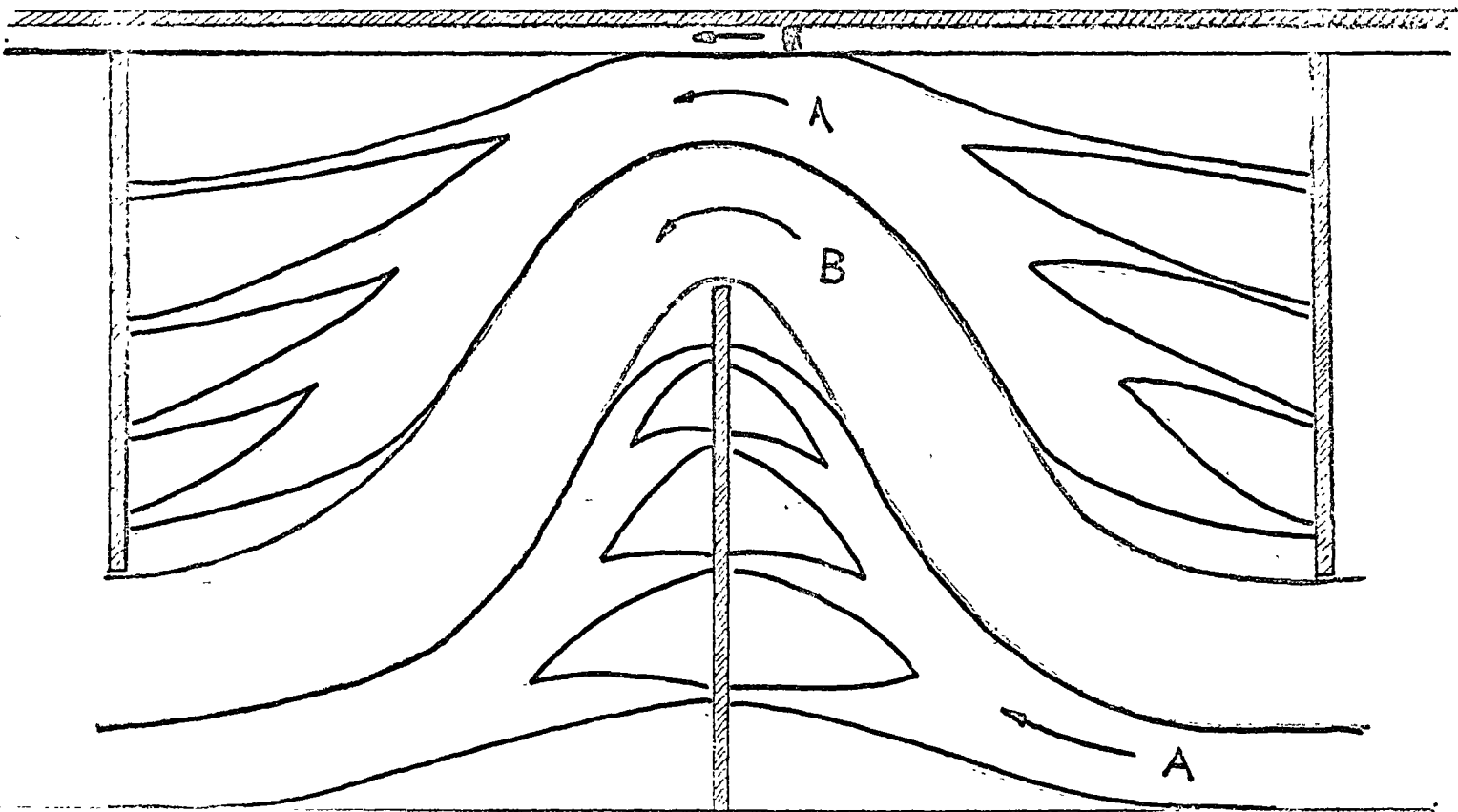
$\% \text{ Corte}$ = Relación del área de la ventana al área total de la mampara.

C) Método de Tinker

El método de Tinker es uno de los métodos más rigurosos reportados en la literatura, ya que analiza los diferentes tipos de corrientes que se generan en la envolvente de un cambiador de calor y toma en consideración que la transferencia de calor depende básicamente del flujo efectivo a través del haz de tubos y no del flujo total por la envolvente.

En el análisis presentado por Tinker, se proponen la existencia de seis corrientes fundamentales en la envolvente:

- 1 Una corriente transversal B a través del haz de tubos.
- 2 Un flujo A por el espacio libre entre los tubos y los barrenos en las mamparas.
- 3 Un flujo longitudinal a través de las ventanas de las mamparas.
- 4 Un efecto combinado de los tres mecanismos anteriores en los bordes de las mamparas.
- 5 Una corriente C que rodea el haz de tubos sin pasar a través de ellos.
- 6 Derrames E sobre los espacios libres entre las mamparas y la envolvente.



La corriente B se considera como la suma de las corrientes a través de la ventana de las mamparas y a través del espacio entre agujeros y tubos (A). La cantidad de esta corriente a través del haz de tubos (B) - de la que depende fundamentalmente la transferencia de calor será pues una fracción del gasto total por la envolvente:

$$Q_B = F_h W_s \text{ ----- (66)}$$

Donde F_h será una fracción que depende de la geometría del cambiador que queda definido por la relación:

$$F_h = \frac{1}{1 + N_h \sqrt{D_s/P_t}} \text{ ----- (67)}$$

D_s = Diámetro de la envolvente (in)

P_t = Distancia entre centros de tubos (in)

N_h = Factor de capacidad definido como:

$$N_h = \psi_1 \frac{(C_{14} - 1)^{3/2}}{C_{14}^{1/2}} + \psi_2 \frac{C_{15}}{C_{14}^{1/2}} + \psi_3 C_{16} C_{14}^{1/2} \text{ ----- (68)}$$

Los coeficientes ψ_1, ψ_2 y ψ_3 son funciones del espaciamiento entre mamparas, diámetro externo de tubos y separación entre centros y para arreglo triangular están definidos por:

$$\psi_1 = 1.26 \left[\frac{P_t/D_o}{P_t/D_o - 1} \right]^{3/2} \text{ ----- (69)}$$

$$\psi_2 = \frac{0.82 C_3}{(P_t/D_o)^2} \frac{D_s}{B} \left[\frac{P_t/D_o}{P_t/D_o - 1} \right] \text{ ----- (70)}$$

$$\psi_3 = 1.48 \left(\frac{D_s}{B} \right) \left(1 - \frac{H}{D_s} \right) \left[\frac{P_t/D_o}{P_t/D_o - 1} \right] \text{ ----- (71)}$$

Para arreglos cuadrado y rotado, deben introducirse los siguientes multiplicadores a ψ_1, ψ_2 y ψ_3

	ψ_1	ψ_2	ψ_3
□	1.000	0.805	0.930
◇	0.707	0.687	0.800

D_o = Diámetro externo de tubos (in)

B = Separación entre mamparas (in)

H = Altura de la ventana (in)

$C_3 = 1 - C_2$

C_2 = Relación del área de la ventana al área total de la mampara

Las constantes C_{14}, C_{15} y C_{16} dependen de la construcción del cambiador y vienen dados por las siguientes expresiones:

$$C_{14} = D_s / D_h$$

$$C_{15} = (D_a - D_o) / D_a$$

$$C_{16} = (D_s - D_b) / D_s$$

D_h = Diámetro del haz de tubos (in)

D_a = Diámetro de los barrenos en las mamparas para introducción de tubos (in)

D_b = Diámetro de la mampara.

Algunos valores característicos de construcción de estas relaciones son los siguientes:

	C_{14}	C_{15}	C_{16}
Cabezal Fijo	1.040	0.045	0.008
Cabezal flotante	1.075	0.045	0.008

En algunos casos con el objeto de evitar las corrientes parásitas E y C el cambiador se construye con fajas de sello y para tales casos, - la constante C_{14} se ve modificada de acuerdo a la siguiente relación:

$$C_{14} = 1 + \frac{(C_{14} - 1)}{4} + 1.5 C_{16} \quad (72)$$

El área de flujo para determinar la masa velocidad con la que se calculará un número de Reynolds corresponde a la media geométrica entre el área de flujo transversal y el área de flujo por la ventana:

$$A_m = \sqrt{A_x A_w} = M A_x \quad (73)$$

A_m = Área de flujo media (ft²)

A_x = Área de flujo transversal (ft²)

A_w = Área de flujo de la ventana (ft²)

M = Factor para determinar el área de flujo media.

$$A_x = C_a B D_h \quad (74)$$

$$\begin{array}{ll} \text{Arreglo } \triangle & C_a = \frac{C_4 (Pt/Do) - 1}{(Pt/Do)} \quad (75) \\ \text{Arreglo } \square & \end{array}$$

$$\text{Arreglo } \diamond \quad C_a = \frac{1.414 C_4 (Pt/Do) - 1}{(Pt/Do)} \quad (76)$$

Donde C_4 depende de la construcción del cambiador y toma generalmente un valor de 0.97

$$M_w = m C_{14}^{1/2} \quad (77)$$

$$m = \left[\frac{0.785 C_2 \alpha D_s}{1.44 C_a B} \right]^{1/2} \quad (78)$$

α = Fracción del área transversal de la envolvente no ocupada por tubos.

$$M = \left[\frac{1}{1 + \frac{0.70 B}{D_s} \frac{1}{M_w^{0.6}} - 1} \right]^{1.67} \quad (79)$$

En base del área transversal A_x , el factor M , el factor F_h y el flujo total W_s se determina una masa velocidad:

$$G = \frac{F_h W_s}{M A_x} \quad (80)$$

Con la masa velocidad se determina el número de Reynolds:

$$Re = \frac{0.0344 D_o G}{\mu} \quad (81)$$

μ viscosidad del fluido por la envolvente (c. poises)

Tinker presenta varias gráficas donde se puede determinar un factor J_h en función del Reynold usando como parámetro la relación (P_t/D_o) .

Finalmente, el coeficiente de película puede calcularse de acuerdo a la siguiente relación:

$$Nu = J_h Pr^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} \quad (82)$$

Por otro lado, el método Tinker toma en consideración que las turbulencias ocasionadas en el flujo por la envolvente, no se presenta a lo largo de la longitud total de los tubos, ya que existen tramos en los extremos del cambiador, en los que se encuentran con un espaciamento diferente al de la totalidad de las rampas, estableciéndose por este concepto el concepto de eficiencia a la transferencia, la cual es función de la longitud total de los tubos, separación entre rampas y separación de las boquillas de entrada y salida de flujo.

$$E = \frac{L_b + (L - L_b) \frac{2 B^{0.5}}{L - L_b}}{L} \quad (83)$$

L = Longitud total de tubos (ft)

L_b = Longitud entre boquillas de entrada y salida (ft)

El coeficiente de película determinado mediante la ec (82) debe aún corregirse por concepto de esta eficiencia:

$$h_o = E h_o \quad (84)$$

Debe también hacerse notar que Tinker recomienda que los cortes de las rampas se hagan en función de la relación (D_s/B) con el objeto de mantener constancia en la relación del área de la ventana al área de flujo para cualquier diseño.

Los valores que se recomiendan para estas relaciones son:

$\frac{D_s}{B}$	$\frac{H}{D_s}$	C_p - Corte
1.0	0.46	0.45
1.5	0.34	0.30
2.0	0.25	0.20
3.0	0.20	0.15
4.0	0.16	0.10
5.0	0.16	0.10

En varios de los métodos mencionados para el cálculo de coeficientes de película tanto por dentro de tubos como por la envolvente; es necesario el determinar la relación (/).¹⁴ Lo que implica calcular la temperatura en la pared del tubo.

= Viscosidad a la temperatura del fluido

= Viscosidad a la temperatura de la pared del tubo.

Para hacer este cálculo se puede considerar que la temperatura de la pared es única tanto en la superficie interna como la externa, lo que corresponde a despreciar la resistencia presentada por el material metálico del tubo, consideración muy cerca de la realidad ya que esta resistencia es muy pequeña comparada por la presentada al flujo de calor por convección en cada uno de los fluidos.

Bajo esta consideración, la temperatura de pared vendrá dada por la siguiente expresión:

$$T_w = \frac{h_i T_i + h_o T_o}{h_i + h_o} \quad (35)$$

Según esto, tenemos que para calcular la temperatura de pared se necesitan conocer los coeficientes de película y para calcular los coeficientes de película se requiere como dato la temperatura de pared, lo que implica un método iterativo para la resolución.

T_w = Temperatura de la pared del tubo.

T_i = Temperatura del fluido dentro de tubos.

T_o = Temperatura del fluido por la envolvente.

4. CAIDAS DE PRESTON

El cálculo de caídas de presión es otro aspecto fundamental a considerar en el estudio de cambiadores de calor tanto para el problema de dimensionamiento como en el de análisis de equipos instalados.

El cálculo de la caída de presión por dentro de tubos no ofrece ninguna dificultad, sin embargo el cálculo de caídas de presión por la envolvente se hace complicado dado a la geometría del sistema.

Caida de Presión por Dentro de Tubos.

La caída de presión por dentro de tubos viene dada por la siguiente expresión:

$$\Delta P_t = \frac{f G_t^2 L N_p}{5.22 \times 10^{10} D_i s \left(\frac{\mu}{\mu_s} \right)^{0.14}} \quad (86)$$

ΔP_t = Caída de presión en los tramos rectos de tubos. (psi)

G_t = Masa velocidad (lb/h ft²)

f = Factor de fricción

L = Longitud total de tubos (ft)

D_i = Diámetro interno de tubos (ft)

N_p = Número de pasos por los tubos

s = Densidad relativa.

La masa velocidad se calcula en función del flujo total del fluido por los tubos y el área de flujo

$$G_t = \frac{W_t}{A_t} \quad (87)$$

El área de flujo corresponde a la suma de las áreas transversales de los tubos entre el número de pasos:

$$A_t = \frac{.785 N_t D_i^2}{N_p} \quad (88)$$

El factor de fricción se calcula en función del número de Reynolds ($D_i G_t / \mu$) usando la correlación gráfica presentada por Kern o bien mediante la siguiente expresión analítica obtenida de dicha gráfica:

$$(Re \leq 1000) \quad f = .4821037 Re^{-0.99355} \quad (89)$$

$$(Re > 1000) \quad f = .00311321 Re^{-0.2636089} \quad (90)$$

Cuando el cambiador de calor presenta dos o más pasos por los tubos, resulta necesario considerar la caída de presión que produce en los retornos. Esta caída de presión puede calcularse con la siguiente expresión:

$$\Delta P_R = 1.68152 N_p V^2 / \rho \quad (91)$$

ΔP_R = Caída de presión en los retornos. (psi)

V = Velocidad del fluido por dentro de tubos (ft/seg)

ρ = Densidad del fluido dentro de tubos (lb/ft³)

La caída de presión total será la suma de las caídas de presión en el tamo recto y en los retornos:

$$\Delta P_T = \Delta P_t + \Delta P_R \quad (92)$$

Caida de Presión por el lado de la Envolvente.

A) Método de Kern

Kern propone la siguiente expresión para calcular la caída de presión por el lado de la envolvente:

$$\Delta P_s = \frac{f G^2 D_s (N_b + 1)}{5.22 \cdot 10^{10} D_e s \left(\frac{\mu}{\mu_w}\right)^{1.4}} \quad (93)$$

f = Factor de fricción.

ΔP_s = Caída de presión (psi)

G = Masa velocidad por la envolvente (lb/h ft²)

N_b = Número de mamparas.

D_e = Diámetro equivalente (ft)

D_s = Diámetro interno de la envolvente (ft)

s = Densidad relativa.

μ = Viscosidad del fluido por la envolvente a la temperatura dentro de la envolvente

μ_w = Viscosidad del fluido por la envolvente a la temperatura de la pared externa de tubos

El factor de fricción se presenta como una correlación gráfica de f - contra número de Reynolds ($Re = D_e G / \mu$)

La masa velocidad se calcula en base al flujo total y a un área de flujo definida por la ecuación:

$$A_s = \frac{d_s C B}{144 P_t} \quad (94)$$

A_s = Área de flujo (ft²)

d_s = Diámetro de la envolvente (in)

C = Claro entre tubos (in)

B = Separación entre tubos (in)

P_t = Distancia entre centro de tubos (in)

Y el diámetro equivalente viene dado por la siguiente expresión:

$$D_e = \frac{4 (P_t^2 - \pi D_o^2 / 4)}{12 \pi D_o} \quad (95)$$

D_o = Diámetro externo de tubos (in)

B) Método de Tinker.

Para el cálculo de caídas de presión por la envolvente, Tinker hace consideraciones semejantes a las hechas para la estimación de coeficientes de película, con respecto a las diferentes corrientes que se presentan.

Se considera en primer término, que el flujo que contribuye fundamentalmente a la caída de presión es una fracción del flujo total por la en volvente viniendo dado por la expresión:

$$Q_{Bp} = F_p Ws \text{ ----- (96)}$$

Para la definición del factor F_p se considera que la corriente A no - existe, debido a un bloqueo de los espacios entre barrenos y tubos, y - que la corriente B queda constituida en un 75% por la corriente a través de la ventana y en un 25% por penetraciones de las corrientes C y E. En base de esto, el factor F_p queda expresado por:

$$F_p = \frac{1}{.80 + N_p \quad D_s/P_t} \text{ ----- (97)}$$

5. SECUENCIA GENERAL DE CALCULO PARA DISEÑO DE CAMBIADORES DE CALOR

El diseño termodinámico de un cambiador de calor corresponde al dimensionamiento de cada elemento del mismo orientado a satisfacer una transferencia de calor especificada sin sobrepasar caídas de presión máximas permitibles.

Como es de esperarse, para un mismo servicio requerido pueden encontrarse diversas combinaciones en las dimensiones de los elementos del cambiador que satisfagan los requerimientos, pudiéndose por lo mismo encontrar se diversas soluciones para un mismo problema.

En este capítulo no se considerarán los criterios para la selección de las diferentes posibles alternativas, ya que este tema será tratado en el capítulo de optimización y nos concretaremos a establecer una secuencia lógica para el dimensionamiento de algunos de los elementos del cambiador bajo la consideración de que se fijen ciertas variables de acuerdo a un criterio dictado por la experiencia.

Las variables que se consideraran fijas para el propósito de diseño serán:

- A) Diámetro y tipo de arreglo de los tubos (triangular, cuadrado o rotado.
- B) Separación entre centros de tubos.
- C) Longitud de los tubos.
- D) Régimen de flujo (contracorriente o paralelo)
- E) Tipo de Cabezal
- F) Opción por fajas de sello.
- G) Caída de presión máxima permisible por los tubos.
- H) Caída de presión máxima permisible por la envolvente.
- I) Velocidad máxima permisible por los tubos

Estas variables quedarán fijas antes de iniciar los cálculos, siguiéndose los siguientes criterios generales para su selección:

A) La selección del diámetro y del arreglo de tubos dependerá fundamentalmente del flujo y del tipo de fluido manejado por dentro de tubos, recomendándose tubos con diámetros entre $3/4$ a $1 1/4$ de pulgada eligiéndose diámetros pequeños para bajos flujos y fluidos poco viscosos, y diámetros mayores para altos flujos y flujos viscosos. El tipo de arreglo se selecciona fundamentalmente en forma triangular para fluidos poco viscosos y poco incrustantes, ya que es un arreglo que presenta ventajas desde el punto de vista de transferencia de calor, arreglo cuadrado para fluidos viscosos y sucios con el objeto de disminuir las caídas de presión y los problemas por incrustación.

B) La longitud de tubos se elige de acuerdo al espacio disponible y a una proporción entre diámetro de envolvente/longitud, en base a un área requerida estimada. En ocasiones cuando el cambiador va a formar parte de un tren de equipos de intercambio calorífico, los tubos se seleccionan de las mismas características de las del resto de los equipos con el objeto de reducir inventarios y problemas de almacenamiento de tubos de diferentes características.

C) Separación entre centros de tubos. La selección de la separación de tubos vendrá dictada por las características de viscosidad e incrustación de los fluidos manejados, siendo preferibles valores pequeños de separación para fluidos poco viscosos e incrustantes con el objeto de favorecer la transferencia de calor, y valores altos de separación para reducir problemas por caídas de presión en fluidos muy viscosos y de incrustación en fluidos sucios.

D) El régimen de flujo se seleccionará generalmente en contracorriente ya que presenta ventajas en la transferencia de calor por permitir mayores gradientes de temperatura a lo largo del cambiador y sólo se selecciona flujo en paralelo para servicios muy particulares donde se requiera favorecer una rápida disminución o elevación de temperatura en algún fluido por condiciones de proceso o bien para casos donde se quiere asegurar que la temperatura de salida de alguno de los fluidos no exceda un valor máximo o no baje de un valor mínimo.

E) El tipo de cabezal se selecciona fundamentalmente de acuerdo a las diferencias de temperaturas del fluido caliente al fluido frío, usándose cabezal fijo para diferencias pequeñas y cabezal flotante para diferencias grandes.

F) Opción por fajas de sello. Las fajas de sello se instalan con el propósito de evitar corrientes parásitas en la envolvente del cambiador, mejorando la transferencia de calor, pero por otro lado la introducción de las mismas trae como consecuencia mayores caídas de presión. Su instalación dependerá pues de las características del fluido y de los flujos por la envolvente.

G) Caída de presión máxima permisible. Las caídas de presión tanto por tubos como por envolvente, son variables fundamentales para la optimización de cambiadores ya que un aumento de las mismas trae como consecuencia mejores coeficientes de transferencia y por lo mismo equipos de menores dimensiones con la consecuente reducción de los costos fijos y la inversión por concepto del cambiador, pero por otro lado si el aumento de caída de presión trae como consecuencia el aumento del consumo de energía de las bombas que manejan los flujos, se tendrá un aumento correspondiente en los costos de operación debiéndose tener pues un valor óptimo donde los costos totales presenten un valor mínimo. En ocasiones sin embargo, se tienen cambiadores de calor instalados entre equipos que operan con una diferencia positiva de caídas de presión, pudiéndose manejar los flujos sin necesidad de la instalación de equipos de bombeo, en estos casos, las caídas de presión máximas permisibles vendrán dadas por las diferencias de presión disponibles y las caídas estimadas en las líneas y accesorios del sistema de tuberías de interconexión de equipos.

H) La velocidad máxima permisible depende fundamentalmente de las características de erosividad de los fluidos existiendo algunos valores recomendados para ciertos fluidos.

Antes de iniciar los cálculos correspondientes para el dimensionamiento del cambiador es necesario tener los siguientes datos:

- A) Temperaturas de entrada de los fluidos por los tubos y envolvente.
- B) Flujos por tubos y envolvente.
- C) Propiedades de los fluidos a las temperaturas de entrada y salida tanto de tubos como de envolvente, incluyendo: densidad, conductividad térmica, capacidad calorífica y viscosidad.
- D) Separación entre boquillas, para poder estimar la efectividad de transferencia por la envolvente. Para conocer este valor tendrá que dimensionarse las boquillas en base de velocidades recomendadas y posteriormente localizar su posición de acuerdo al diámetro determinado con una distancia de los espejos por recomendaciones mecánicas
- E) Factores de incrustación para los fluidos por dentro y fuera de tubos, en base a recomendaciones para el tipo de fluidos manejados.

Una vez fijas algunas de las variables y conocidos los datos necesarios, pueden iniciarse los cálculos de diseño siguiendo la siguiente secuencia:

- A) Suponer un coeficiente total de Transferencia optimista, es decir superior al valor esperado.
- B) Determinar la diferencia de temperaturas media logarítmica.
- C) Calcular el factor de corrección de diferencia de temperaturas, - determinándose simultáneamente el número de cambiadores en serie necesarios para obtener un valor superior a 0.8 para dicho factor
- D) Determinar el área total de transferencia necesaria en base al calor a transferir, la diferencia de temperaturas corregida y el - coeficiente total de transferencia supuesto.
- E) Determinar el área de transferencia por cambiador de calor, que - corresponderá al área total entre el número de cambiadores necesarios
- F) En base del área por cambiador, la longitud de los tubos y el diámetro externo de los mismos, determinar el número de tubos necesario para dar el área de transferencia requerida.
- G) Considerando que el cambiador es de un solo paso por los tubos de terminar a que diámetro estándar de envolvente le corresponde un número de tubos igual o mayor que el calculado en el punto (F).
- H) En el caso de que el número de tubos sea mayor que el que pueden - contenidos por una envolvente de diámetro máximo, aumentar un - cambiador de calor en paralelo y repetir el cálculo (F) y (G)
- I) Con el número de tubos estándar correspondiente al diámetro de envolvente calculado en el punto (G) determinar el área de transferencia real por cambiador y el área total disponible.
- J) En base del área total disponible calculada en (I), del calor total que se va a transferir y de la diferencia de temperaturas media logarítmica calculada en (P) corregida por el factor calculado en (C) determinar el coeficiente total de transferencia que satisface la ecuación general de transferencia $Q = U A \quad T_L$
- K) Calcular la velocidad por los tubos.
- L) Calcular la caída de presión por los tubos.
- M) Si alguna de las dos magnitudes calculadas, velocidad o caída de

presión por los tubos exceden el valor máximo permisible, aumentar un cambiador en serie y volver a repetir cálculos empezando en el punto- (E)

- N) Si ninguna de las dos magnitudes, velocidad y caída de presión exceden los valores máximos permisibles, considerar un cambiador con dos pasos por los tubos y repetir cálculos empezando en (E)
- O) Si para dos pasos no se han sobrepasado los límites máximos de velocidad y caída de presión, considerar cambiadores de calor con 4, 6, 8, ... etc pasos hasta sobrepasar los límites, y escoger un número de pasos que corresponda al cálculo inmediato anterior al que sobrepasó los límites.
- P) Una vez definido el número de pasos por los tubos, calcular el coeficiente individual de película H_i con las propiedades del fluido a la temperatura de entrada y a la temperatura de salida.
- Q) Calcular caída de presión por la envolvente considerando la separación máxima entre mamparas, esto es una separación igual al diámetro de la envolvente correspondiente a una relación D_s/B igual a 1, donde D_s es el diámetro de la envolvente y B la separación entre mamparas.
- R) Si la caída de presión calculada en (Q) excede el valor máximo permisible aumentar un cambiador en paralelo y repetir cálculos a partir del punto (E) volviendo a empezar con un cambiador de un solo paso por los tubos.
- S) Si la caída de presión por la envolvente para $D_s/B = 1$ no excede el valor máximo permisible, analizar envolventes con separación entre mamparas menores, correspondientes a relaciones $D_s/B = 1.5, 2, 3, 4, 5$ y 6, calculando las correspondientes caídas de presión.
- T) Si aún para el acercamiento de $D_s/B = 5$ no se excede la caída de presión máxima, se elige este valor para la separación entre mamparas.
- U) Si para alguna de las relaciones de D_s/B se excede el valor máximo de caída de presión, se elige la separación entre mamparas correspondiente a una relación D_s/B inmediata inferior a la que sobrepasó el límite.
- V) Una vez definida la separación entre mamparas, se determinan los coeficientes de película H_o a las condiciones del fluido por la envolvente a la temperatura de entrada y salida.
- W) Calcular los coeficientes globales limpios de transferencia de calor U_1 y U_2 en los dos extremos del cambiador.
- X) Calcular un coeficiente global limpio en base a la siguiente relación

$$U = \frac{\frac{\ln \frac{T_2}{T_1}}{T_2 - T_1}}{\frac{U_1 T_2 - U_2 T_1}{\ln \frac{U_1 T_2}{U_2 T_1}}}$$

- Y) Con el coeficiente global limpio y los factores de ensuciamiento Rd_t y Rd_e determinar el coeficiente total de transferencia:

$$U_s = \frac{1}{\frac{1}{U} + Rd_t + Rd_e}$$

- Z) Si el coeficiente calculado en (Y) es mayor o igual al calculado en el punto (J) dar por terminados los cálculos, y si el valor resulta menor, usar un valor supuesto de coeficiente total de transferencia igual al calculado en (Y) e iniciar cálculos en el punto (D)

6. SIMULACION DE TRENES DE INTERCAMBIO CALORIFICO

El objetivo fundamental que tienen las técnicas de simulación es la predicción del comportamiento a régimen permanente de un conjunto de equipos interconectados, haciendo uso de un modelo matemático adecuado. Para el caso particular de simulación de trenes de cambiadores de calor, la simulación involucra la predicción de las temperaturas de salida de cada equipo, tanto por tubos como por envolvente.

Para poder llevar a cabo una simulación es necesario disponer de una descripción matemática precisa de la siguiente información:

- (a) La naturaleza de las corrientes del proceso, esto es, los nombres y definiciones de las propiedades de interés en cada corriente del proceso.
- (b) La naturaleza del material que constituye las corrientes, esto es, un modelo que describa las propiedades termodinámicas y de transporte con valores numéricos de los parámetros específicos necesarios para la evaluación de las propiedades.
- (c) La naturaleza del equipo de procesamiento, es decir, las relaciones matemáticas entre las propiedades de entrada y salida de las corrientes como una función de las características físicas específicas de los equipos y de los parámetros de operación de cada equipo.
- (d) La configuración específica del proceso, es decir, la topología del proceso describiendo como están interconectadas las diferentes unidades del proceso y especificando los valores específicos del diseño y parámetros de operación de cada equipo.
- (e) Los nombres de las especies químicas, composiciones y flujos en cada corriente de entrada al proceso.

El método de organización de toda esta información dependerá de la técnica particular de simulación, pero cualquiera que ésta sea debe de incluir de algún modo o de otro esta información.

Analizando el problema de simulación de cambiadores de calor, la información que sería necesaria sería la siguiente:

- a) En la naturaleza de las corrientes de proceso habría que indicar básicamente si las corrientes son líquidos, vapores o mezclas de éstos así como el flujo y la composición de las especies químicas que constituyen la corriente, debiendo también estar incluida la información de cuales son las propiedades físicas que deben calcularse para dicha corriente (viscosidad, conductividad térmica, capacidad calorífica, densidad, entalpia).

- b) En la naturaleza del material que constituye las corrientes, sería necesario disponer de una lista de las propiedades físicas de los componentes individuales que constituyen las corrientes, a condiciones estandarizadas de presión y temperatura, así como reglas de mezclas y correcciones por presión y temperatura actual de las corrientes, que permitan calcular las propiedades físicas en función de la composición de las corrientes y de las condiciones de presión y temperatura de las mismas. Por otro lado sería también conveniente el incluir en este punto los parámetros de una ecuación de estado que permitiera la estimación de densidades, desviaciones de entalpía y de capacidad calorífica de las corrientes.
- c) En la naturaleza del equipo de procesamiento quedaría incluido el modelo matemático representativo de un cambiador de calor que ya fué analizado en el punto (2).
- d) En la configuración específica del proceso, quedaría incluida la información de como están interconectadas las corrientes en el tren de intercambio calorífico, debiendo indicarse la proveniencia de las corrientes que entran a tubos y a envolventes de cada cambiador de calor. En este punto, quedaría también incluida la información de las características de diseño de cada cambiador de calor.

La forma de organizar esta información y combinarla con una secuencia de cálculo puede efectuarse de muy diversos modos. A continuación se expondrá una técnica sencilla que permite la resolución de cualquier arreglo de cambiadores de calor.

- A) Establecer el número de cambiadores que constituyen el tren de intercambio calorífico.
- B) Definir el área total de intercambio calorífico para cada cambiador de calor.
- C) Definir los flujos por tubos y envolvente por cada cambiador de calor.
- D) Definir los valores de capacidad calorífica, coeficiente de película por dentro de tubos y por la envolvente, a dos temperaturas para cada cambiador de calor.
- E) Definir las temperaturas de entrada a cada cambiador de calor. En esta definición, las temperaturas de corrientes externas quedarán fijas pero las provenientes de algún otro cambiador de calor serán solo supuestas.
- F) Suponer todas las temperaturas de salida de cada cambiador de calor por tubos y por envolvente.
- G) Definir la topología del sistema. La forma para esta definición puede efectuarse por la asignación de cuatro índices de arreglo a cada cambiador de calor:
 - I₁ Representa el número del cambiador de calor del que proviene la corriente que entra a los tubos. (En el caso de provenir de una corriente externa, toma el valor de 0)
 - I₂ Define si dicha corriente proviene de tubos o de la envolvente del cambiador I₁ (Toma el valor de 1 cuando proviene de tubos, de 2 cuando proviene de la envolvente y de 0 cuando es corriente externa)

T_3 Representa el número del cambiador de calor del que proviene la corriente que entra a la envolvente. (En el caso de provenir de una corriente externa toma el valor de 0)

T_4 Define si dicha corriente proviene de tubos o de envolvente del cambiador T_3 (Toma el valor de 1 cuando proviene de tubos, de 2 cuando proviene de envolvente y de 0 cuando es una corriente externa)

Estos índices serían los siguientes para el tren de intercambio representado en la figura anexa.

Cambiador	T_1	T_2	T_3	T_4
1	0	0	2	2
2	0	0	3	1
3	0	0	4	2
4	2	1	5	2
5	1	1	0	0

H) En base a las capacidades caloríficas del fluido por dentro de tubos y por envolvente y de los coeficientes de película interno y externo definidos a dos temperaturas determinar las constantes de las siguientes relaciones lineales:

$$C_{pei} = A_i + B_i T_e$$

$$C_{pti} = a_i + b_i T_t$$

$$h_{ii} = c_i + d_i T_t$$

$$h_{oi} = C_i + D_i T_e$$

que permitan una interpolación lineal en función de las temperaturas.

I) En base a las temperaturas supuestas de entrada y salida del fluido por dentro de tubos, calcular el calor total por balance de calor, en cada cambiador de calor.

$$Q_i = W_{ti} \left[\left\{ a_i + b_i / 2 (T_{t2i} + T_{t1i}) \right\} (T_{t2i} - T_{t1i}) \right]$$

J) En base del calor calculado en (I) y de la temperatura de entrada su-
puesta por la envolvente, determinar las temperaturas de salida de en-
volvente de cada cambiador de calor que satisfaga el balance de calor.

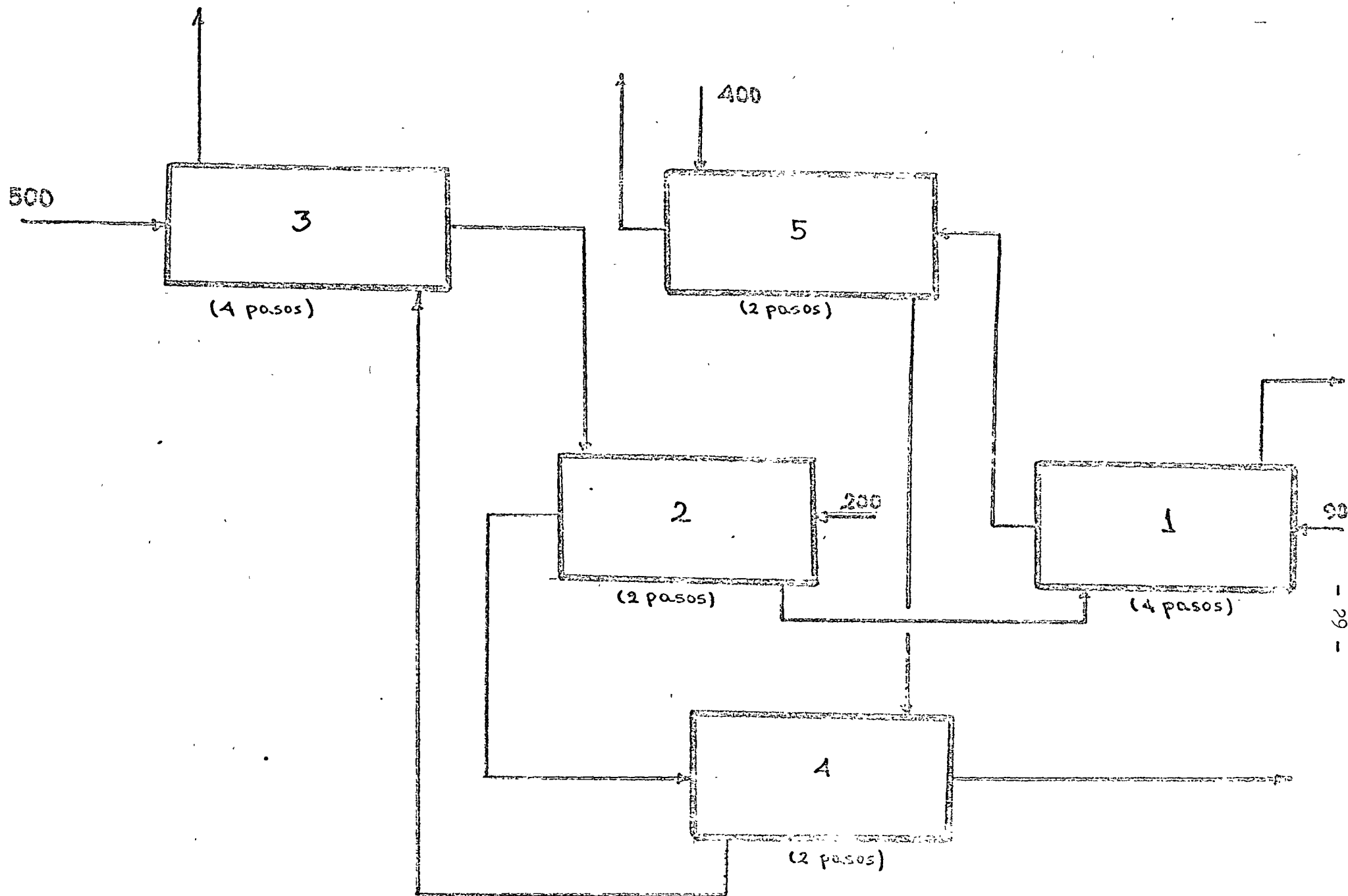
$$T_{e2i} = -A_i/B_i + \sqrt{(A_i/B_i)^2 + 2/B_i(A_i T_{e1i} + B_i/2 T_{e1i}^2 - Q_i/W_{ei})}$$

K) Con las temperaturas de entrada y de salida definidas, calcular los coe-
ficientes de película interno y externo, haciendo uso de las constantes
calculadas en el punto (H)

L) Calcular el coeficiente total de transferencia en cada extremo del cam-
biador usando los coeficientes individuales de película, y los factores
de incrustación.

M) Calcular el calor total haciendo uso de la ecuación general de transfe-
rencia:

$$Q_{Ti} = A_i \frac{U_{1i} \Delta T_{2i} - U_{2i} \Delta T_{1i}}{\ln \frac{U_{1i} \Delta T_{2i}}{U_{2i} \Delta T_{1i}}} F_{Ti}$$



- N) Calcular las temperaturas de salida de tubos en base a las temperaturas de entrada, los flujos por dentro de tubos, y el calor calculado en (M), para cada cambiador de calor.

$$T_{t2i} = -a_i/b_i + \sqrt{(a_i/b_i)^2 + 2/b_i(a_i T_{t1i} + b_i/2 T_{t1i}^2 - Q_{Ti}/W_{t_i})}$$

- O) Comparar los valores de las temperaturas de salida supuestas para tubos y envolvente con los valores calculados para dichas temperaturas en los puntos (J) y (N). Cuando los valores estén próximos, dar por terminado el cálculo y cuando estén desviados, iniciar los cálculos en el punto (T) con valores supuestos de temperaturas de salida igual a los calculados.

En toda esta secuencia de cálculo, debe efectuarse la igualación de las temperaturas de entrada a tubos y envolvente de cada cambiador de calor a los valores calculados de salida de temperaturas de tubos o envolvente, de acuerdo a la topología del sistema que se resuelve, haciéndose uso para esto de los índices de identificación definidos en el punto (G)

El orden de resolución de los cambiadores de calor debe hacerse de acuerdo a la topología del tren, siguiendo el siguiente criterio:

Cuando se están calculando las temperaturas de salida de envolvente, (Punto J), deben resolverse primeramente aquellos cambiadores que tengan alimentaciones provenientes de corrientes externas o de los tubos de otro cambiador de calor y hacer el cálculo de los cambiadores con corrientes provenientes de alguna envolvente de otro cambiador de calor, sólo cuando este cambiador ya se haya resuelto.

Cuando se están calculando las temperaturas de salida de los tubos, (Punto N), deben resolverse primeramente aquellos cambiadores de calor cuyas corrientes de alimentación a tubos provenga de la envolvente de otro cambiador de calor o bien de una corriente externa, dejando la resolución de los equipos con corrientes de alimentación a tubos provenientes de los tubos de otro cambiador de calor hasta después de haber resuelto estos cambiadores.

7. OPTIMIZACION

La optimización consiste en el proceso de selección de los valores de las variables independientes que determinan el mejor valor posible de la medida de comportamiento de un sistema.

Los pasos básicos en la optimización de un sistema son:

- a) Descripción del sistema, que se puede conseguir mediante la proposición de un modelo matemático que lo represente o mediante la ejecución de los experimentos necesarios que determinen el comportamiento del sistema. Para el caso del estudio de cambiadores de calor, esto correspondería al modelo matemático desarrollado en la sección 2 o bien a una serie de experimentos para determinar el comportamiento de uno o varios cambiadores de calor ya instalados bajo diferentes condiciones de operación.

- b) Establecimiento de una medida única del comportamiento del sistema. En otras palabras es necesario dejar perfectamente establecido que es lo que interesa optimizar, y si el valor óptimo representa un valor máximo o un valor mínimo de la medida del comportamiento. En el caso del estudio de cambiadores de calor, esta medida de comportamiento puede quedar expresada de varias maneras:

Como un área total de transferencia necesaria para un servicio dado que interesa minimizar.

Como una transferencia de calor total en un equipo instalado que interesa maximizar.

Como una inversión para la instalación de un equipo que cumpla con un servicio especificado que interesa minimizar.

Como costos de operación de un equipo que interesa minimizar.

Como costos totales que interesa minimizar.

- c) Establecer una relación unequivoca entre la medida de comportamiento y las variables independientes (variables bajo control)

Esto dependerá naturalmente del caso particular bajo estudio pudiendo presentar varias alternativas en el caso del estudio de cambiadores de calor:

Establecer la relación entre el área total de transferencia y las temperaturas intermedias en un tren de cambiadores de calor.

Establecer la relación entre un calor total transferido y las temperaturas intermedias de un tren de intercambio calorífico.

Establecer la relación entre la inversión y las características físicas (dimensiones) del cambiador de calor, etc...

- d) Aplicación de las técnicas necesarias para llegar a encontrar los valores de las variables independientes (controlables) que optimizan la función objetivo (medida de comportamiento del sistema).

De las técnicas disponibles para la optimización se pueden mencionar tres grandes grupos

Optimización de una función expresada en forma algebraica por el método indirecto.

Optimización de una función expresada mediante un modelo matemático o en forma experimental mediante métodos directos.

Optimización de una función algebraica lineal con restricciones representadas mediante expresiones también lineales (programación lineal)

Optimización de los resultados de una serie de etapas interconectadas, donde el resultado de cualquiera de las etapas depende de los resultados de las demás etapas (Programación Dinámica)

- 4) Optimización por el método indirecto.

Para poder aplicar el método indirecto para la optimización de una función, es necesario tener una expresión algebraica explícita de la función objetivo:

$$Y = F(X_1, X_2, X_3, \dots, X_n)$$

El método de optimización consiste en la resolución de un sistema simultáneo de ecuaciones algebraicas que resultan con la igualación a 0 de las derivadas parciales de la función con respecto a cada variable-

$$80(A_2)(400-T_2) = 100,000(T_2-T_1)$$

$$A_2 = \frac{1250(T_2-T_1)}{400 - T_2}$$

Balance de calor en el cambiador 1

$$300,000(T_1-100) = 100,000(300-T_5)$$

$$T_1-100 = 300-T_5 \quad 300-T_1 = T_5-100 = T_{m1}$$

$$Q_1 = U_1 A_1 T_{m1} = (120)(A_1)(300-T_1)$$

$$Q_1 = WC_p(T_1-100) = 100,000(T_1-100)$$

$$(120)(A_1)(300-T_1) = 100,000(T_1-100)$$

$$A_1 = \frac{833(T_1-100)}{300 - T_1}$$

Determinar las temperaturas intermedias del tren que minimizan el área necesaria.

La función a optimizar será:

$$A_1 + A_2 + A_3 = F(T_1, T_2) = 12,500 - 25T_2 + \frac{1250(T_2-T_1)}{400 - T_2} + \frac{833(T_1-100)}{300 - T_1}$$

Igualando las derivadas parciales con respecto a T_1 y T_2 a 0

$$\frac{F(T_1, T_2)}{T_1} = \frac{-1250}{400-T_2} + \frac{166,600}{(300-T_1)^2} = 0$$

$$\frac{F(T_1, T_2)}{T_2} = -25 + \frac{500,000-1250T_1}{(400-T_2)^2} = 0$$

$$1250 T_1^2 - 750,000 T_1 + 166,600 T_2 + 45,860,000 = 0$$

$$25 T_2^2 - 20,000 T_2 + 1,250 T_1 + 3,500,000 = 0$$

Resolviendo el sistema de ecuaciones simultaneas mediante el método iterativo de Newton Rapson:

$$T_1 = 180$$

$$T_2 = 295$$

Y la función objetivo:

$$A_1 + A_2 + A_3 = 7,050 \text{ ft}^2$$

B) Métodos de Optimización Directos.

Los métodos de optimización directos, por el contrario a los indirectos, no intentan encontrar el punto óptimo en un solo paso, sino por el contrario emplean alguna estrategia exploratoria que tiende a un movimiento de los valores de las variables independientes hacia el punto óptimo. De los varios métodos directos existentes, se explicará a continuación el Método Simplex de Nelder y Mead.

independiente:

$$\frac{\partial F}{\partial X_1} = 0$$

$$\frac{\partial F}{\partial X_2} = 0$$

$$\frac{\partial F}{\partial X_3} = 0$$

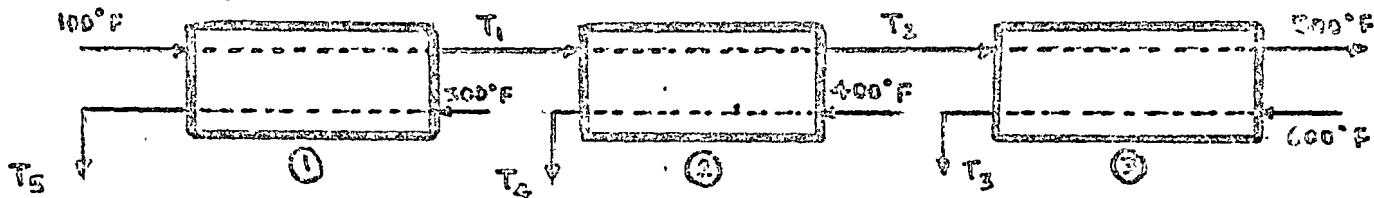
.....
.....
.....

$$\frac{\partial F}{\partial X_n} = 0$$

La resolución del sistema de n ecuaciones para $X_1, X_2, X_3, \dots, X_n$ da los valores de las variables independientes que optimizan la función y el valor óptimo de la función corresponderá a la substitución de las variables independientes calculadas en la función.

Ejemplo:

Sea el siguiente tren de cambiadores de calor:



Todos los valores correspondientes a los productos WC_p son iguales a 100,000 Btu/h °F

Los coeficientes totales de transferencia de calor son los siguientes: $U_1 = 120$ Btu/h ft² °F $U_2 = 80$ Btu/h ft² °F $U_3 = 40$ Btu/h ft² °F

Balance de calor en el cambiador 3

$$100,000(600 - T_3) = 100,000(500 - T_2)$$

$$T_3 - T_2 = 100$$

$$\Delta T_{m1} = 100$$

$$Q_3 = WC_p(500 - T_2) = 100,000(500 - T_2)$$

$$Q_3 = U_3 A_3 \Delta T_{m1} = (40)(A_3)(100)$$

$$100,000(500 - T_2) = (40)(A_3)(100)$$

$$A_3 = 12,500 - 25 T_2$$

Balance de calor en el cambiador 2

$$100,000(T_2 - T_1) = 100,000(400 - T_4)$$

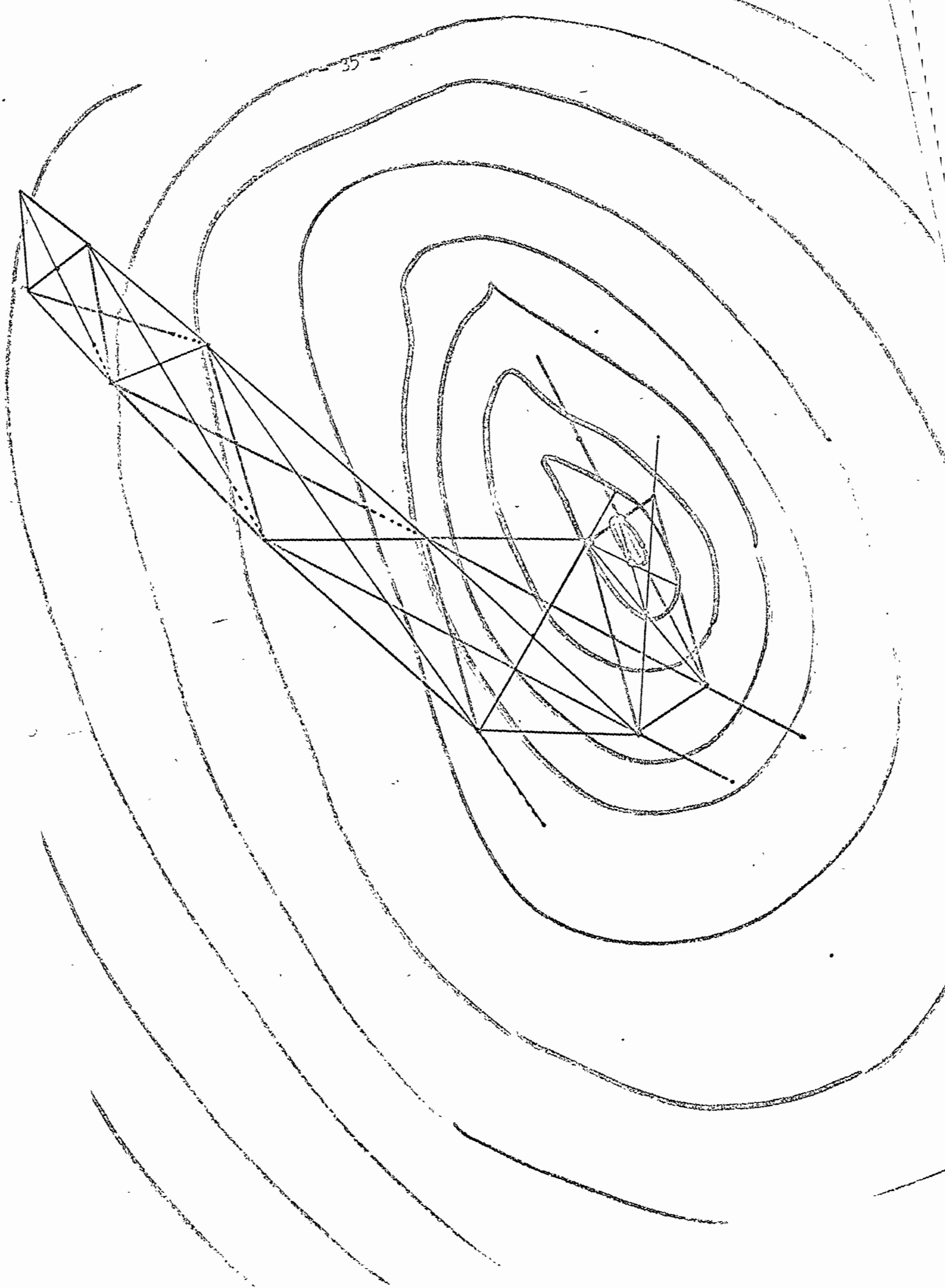
$$T_2 - T_1 = 400 - T_4 = 400 - T_2 = T_4 - T_1 = \Delta T_{m2}$$

$$Q_2 = U_2 A_2 \Delta T_{m1} = (80)(A_2)(400 - T_2)$$

$$Q_2 = WC_p(T_2 - T_1) = 100,000(T_2 - T_1)$$

Método Simplex de NIELDER Y MEAD

- a) Reflexión del peor punto en la dirección determinada por la línea que une este punto con el centroide de los otros N puntos
- b) Expansión en la misma dirección cuando la reflexión encuentra un punto mejor.
- c) Contracción en la misma dirección cuando la reflexión produce otro punto peor.
- d) Contracción del simplex hacia el mejor punto cuando una contracción previa produce otro punto peor.



Reflexión:

$$P_i^* = (1 + \alpha) \bar{P}_j - \alpha P_{w,j} \quad j=1, N$$

P_i^* es el punto reflejado

P_w es el peor punto

$\bar{P}$ es el centroide definido como:

$$\bar{P}_j = \frac{\sum_{i=1}^N P_{i,j} (x_i)}{N} \quad j=1, N$$

α es el coeficiente de reflexión
que representa la relación
de las distancias:

$$\frac{\|P^* - \bar{P}\|}{\|P_w - \bar{P}\|}$$

Expansión :

$$P_j^{**} = \gamma P_j^* + (1-\gamma) \bar{P}_j \quad j=1, N$$

donde :

P_j^{**} es el punto expandido

γ es el coeficiente de expansión ($\gamma > 1$) y corresponde a la relación de las distancias:

$$\gamma = \frac{\|P_j^{**} - \bar{P}\|}{\|P_j^* - \bar{P}\|}$$

Contracción del punto reflejado:

$$P_j^{**} = \beta P_j^* + (1-\beta) \overline{P}_j \quad j=1, n$$

donde:

P_j^{**} es el punto contraído

β es el coeficiente de contracción
($\beta < 1$) y corresponde a
la distancia:

$$\beta = \frac{\|P_j^{**} - \overline{P}_j\|}{\|P_j^* - \overline{P}_j\|}$$

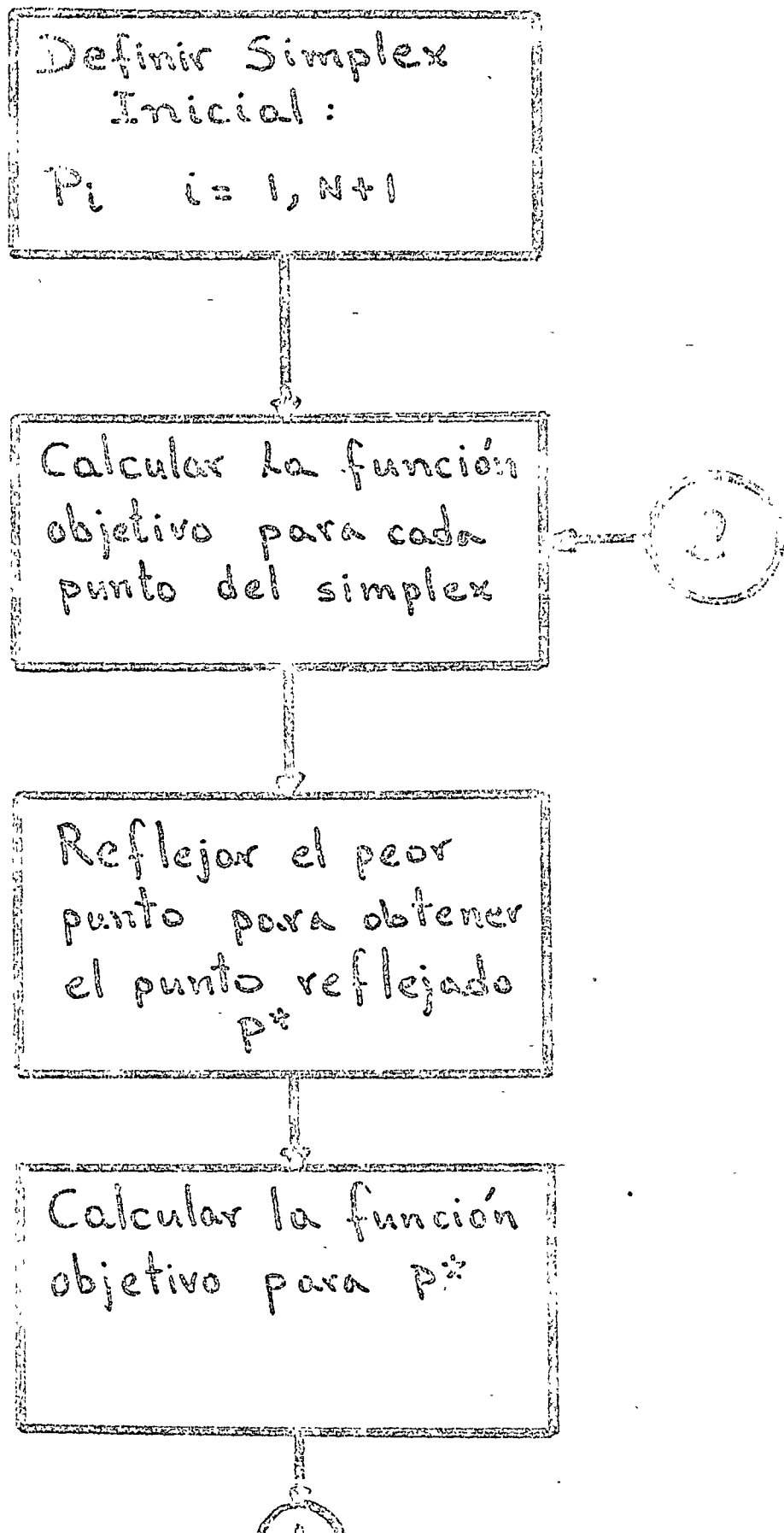
Contracción hacia el mejor punto

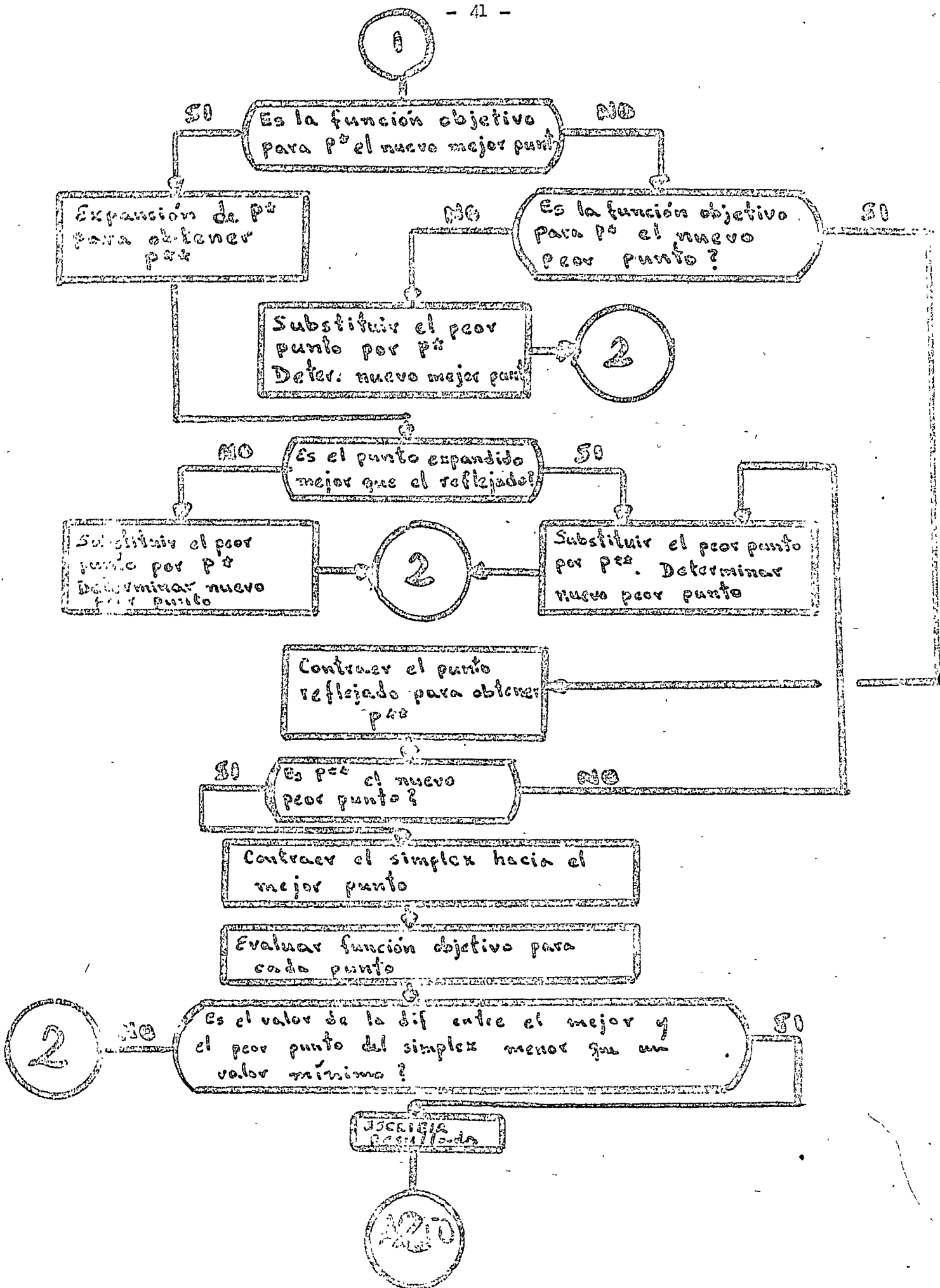
$$P_{i,j}^* = (P_{i,j} + P_{b,j})/2 \quad j=1,n), i=1,n+1$$

$P_{i,j}$ es el punto i del simplex actual.

$P_{b,j}$ es el punto mejor del simplex actual.

$P_{i,j}^*$ es el punto i del simplex contraído.





8 BIBLIOGRAFIA

Colburn A.P. Mean Temperature difference and heat transfer coefficient in liquid heat exchangers. Industrial & Engineering Chemistry Vol 45 No 8 August 1933 pag 873-76

Buthod A.P. How to estimate heat exchangers. The Oil and Gas Journal Vol 58 No 3 January 18 1960 pag 67-82

Devore Abe. Try this simplified method for rating baffled exchangers. Tinker's method. Petroleum Refiner. May 1961 Pag 221-223

Short E.E. Flow geometry and heat exchanger performance. Chemical Engineering Progress Vol 61 No 7 1965 pags 63-70

Rubin F.L. Practical heat exchangers design. Chemical Engineering Progress. Vol 64 No 12 December 1968 Pags 44-48

P.R. Jones et al. Computer usage for evaluation of design parameters and cost of heat exchangers with no change of phase and tube side pumping costs as prime parameters. January 1969 Clearinghouse pag 1-19

Jenssen S.K. Heat exchanger optimization. Chemical Engineering Progress Vol 65 No 7 July 1969 Pags 59-66

Fisher J.U. Optimization of a heat exchanger. Brit. Chem. Eng. Vol 9 1964 Pags 450-456

Bergwerk. Better heat exchanger design. Brit. Chem. Eng. Vol 10 1965 Pags 323-326

Tinker T. Shell side characteristics of shell and tube heat exchangers. Transaction ASME January 1958

Kern D.O. Heat Transfer Mc Graw Hill 1950

Carroll G.W. The created response surface technique for optimizing non linear restrained systems. Operations Research Vol 9 April 1960 Pags 169-184

Nelder J.A. y Mead R. A simplex method for function minimization. The Computer Journal Vol 7 (1965) pags 308-313

J. Landgrave Optimización de Combinadores de Calor sin Cambio De Fase. Tesis Profesional Facultad de Química U.N.A.M. (1972)

Fisher J. and Parker R. New Ideas on heat exchanger design. Hydrocarbon Processing. July 1969 Pags 147-154

Culley D.L. How to calculate weighted MTD's Hydrocarbon Processing Vol 45 No 6 June 1966 Pags 116-122

C U R S O

Aplicaciones de la transferencia de calor en proceso
Industriales.

C O N F E R E N C I A

Cambiadores de calor de Superficie extendida.

Centro de educación continúa Facultad de Ingeniería
UNAM

Ing. Julio Santamaria
Bufete Industrial.

PROLOGO

El presente trabajo tiene como objeto desarrollar un procedimiento adecuado para la optimización de un equipo de enfriamiento, usando como medio para tal función el aire.

En los primeros 2 capítulos se hace un estudio sobre los primeros usos del aire como medio de enfriamiento, sus aplicaciones a casos industriales, así como sus características, mencionando sus ventajas y desventajas.

En el 3er. capítulo se trata el diseño termodinámico de ambos lados del cambiador, desglosando cada paso que debe hacerse para el cálculo del mismo.

En el 4º capítulo se hacen las consideraciones de diseño y económicas necesarias para obtener el cambiador más adecuado para un buen funcionamiento y a un mínimo costo.

En el 5º capítulo se estudia y se resuelve un problema particular, para demostrar el procedimiento desarrollado en el capítulo anterior, obteniéndose los resultados esperados.

I. INTRODUCCION

1. Breve historia
2. Aplicaciones
3. Características

III. DESCRIPCION DE EQUIPO

1. Generalidades
2. Características

III. TEORIA

1. Lado del aire
2. Lado de los tubos

IV. DISEÑO

1. Consideraciones de diseño
2. Consideraciones económicas

V. OPTIMIZACION DE UN CASO ESPECIFICO

1. Desarrollo
2. Ejemplo

VI. CONCLUSIONES

VII. NOMENCLATURA

VIII. BIBLIOGRAFIA

INTRODUCCION

INTRODUCCION

1. Breve Historia

1.1 Primeras aplicaciones del aire como medio de enfriamiento.- El primer cambiador de calor comercial enfriado por aire del que se tiene noticia, fué de tipo vertical, con un ventilador en el plano vertical forzando aire a través del haz de --- tubos, Fig. 1, más tarde, las unidades cuadradas o rectangulares con haces de tubos verticales fueron construidas, Fig. 2 - pero este diseño muy pronto cayo en desuso debido a los efectos de la variable dirección del viento.

Durante 1935, las unidades horizontales con tubos aletados, de tiro inducido y forzado Fig. 3, entraron en operación. Posteriormente, en 1940, el tipo "V" fué diseñado con el fin de conseguir un ahorro en el terreno ocupado, Fig. 4, estos diseños presentan el problema inherente del cambio de dirección del viento, requiriendo pantallas para el viento y una adecuada atención a la recirculación del aire caliente.

1.2 El uso del aire como medio indirecto de enfriamiento ha sido practicado por muchos años en la Industria, prácticamente todo el exceso de calor ha sido disipado por este medio.

Este medio indirecto de enfriamiento ha acarreado infinidad de problemas, pues el agua que es enfriada por el aire en torres de enfriamiento, es usada posteriormente en cambiadores, y al terminar su servicio o es desechada trayendo consigo una

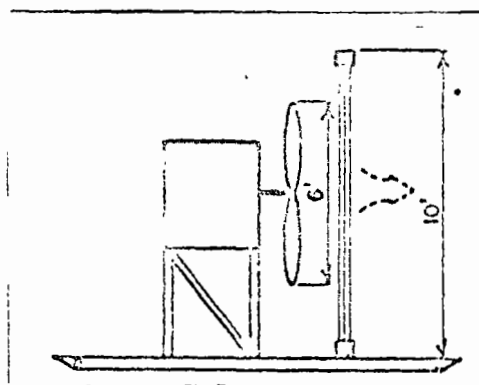


FIGURA 1

Enfriador vertical de aire, tiro forzado.

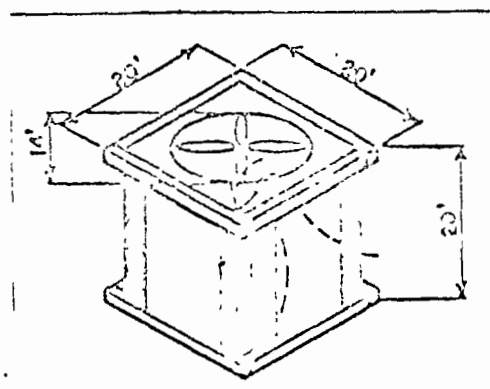


FIGURA 2

Enfriador vertical de aire, tiro inducido.

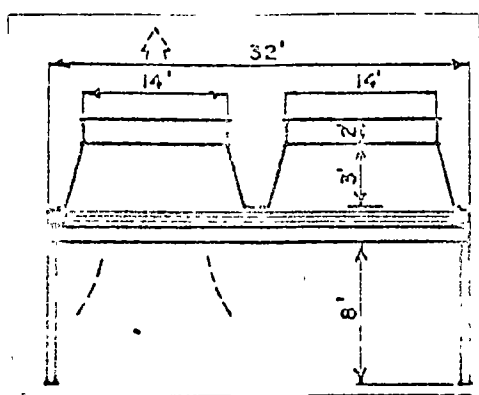


FIG. 3a.

Enfriador horizontal de
aire, tiro inducido.

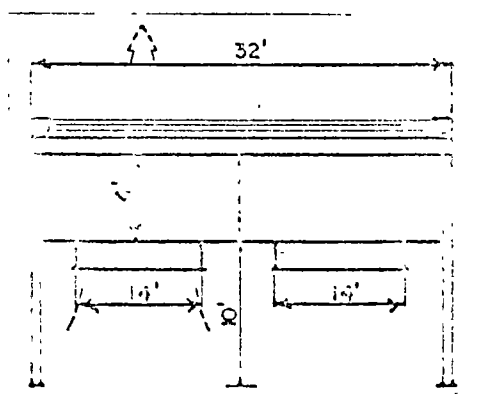


FIG. 3b

Enfriador horizontal de
aire, tiro forzado.

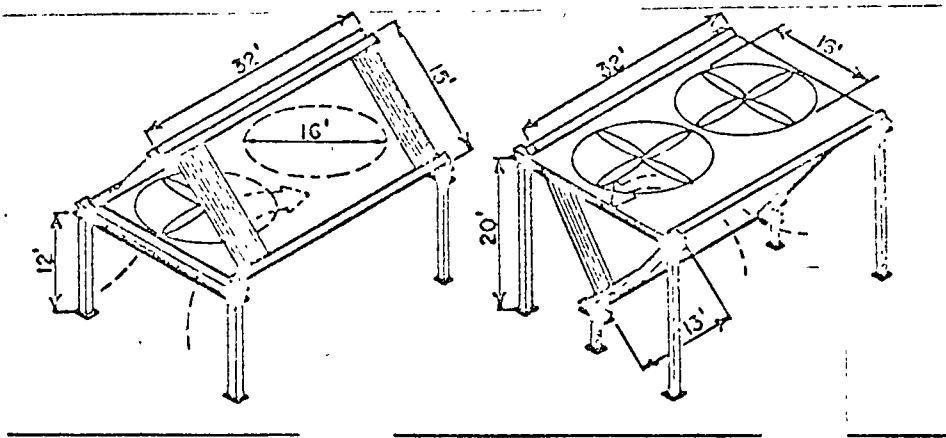


FIGURA 4a, b

Fig. 4a. Enfriador en "V", tiro inducido

Fig. 4b. Enfriador en "V", tiro forzado

posible contaminación térmica y en el caso en que existan -- fugas en los cambiadores, una contaminación con sustancias -- que posiblemente no sean biodegradables; o es recirculada -- provocando un costoso sistema de tratamiento y acondiciona-- miento de la misma. El uso del aire como medio directo de -- enfriamiento no crea estos problemas y hasta hace unos años, la industria del gas natural comenzó a utilizar el aire como medio de enfriamiento.

Actualmente un gran número de refinerías y plantas químicas han sido construídas o estan siendo diseñadas utilizando en lo mas posible el aire directamente como medio de enfriamien-- to.

2. Aplicaciones

2.1 Aplicaciones Generales.- Tal vez la razón mas importan-- te para el desarrollo de cambiadores de calor enfriados por -- aire, es la localización de las plantas en áreas donde el su-- ministro de agua es limitado, esto es, en zonas desérticas, -- como es el caso de las refinerías que se encuentran instala-- das en la Arabia Saudita, Irán, o lugares que se asemejan a -- tales condiciones.

El agua apropiada para enfriamiento en estas zonas es escasa, y las Industrias del Petróleo y de Procesos han cambiado por lo tanto a otro medio de enfriamiento, aire.

El aire es abundante, fácil de manejar, físicamente estable y

no corrosivo, excepto en áreas de contaminación de aire que son poco usuales. No existe además ningún límite superior -- a el cual el aire pueda ser calentado lo cual no es posible -- con el agua.

La óptima transferencia de calor se obtiene ya que el aumento de temperatura permisible puede ser hecho el máximo, para el cual es posible diseñar las superficies de transferencia de -- calor de un cambiador de calor enfriado por aire, otra razón para usar el aire como medio de enfriamiento es el marcado -- aumento en usos Industriales del agua, causando problemas en el suministro de esta, aún en áreas donde ha sido abundante o encarecida, debido a usos competitivos más provechosos.

A medida que nuevos procesos son perfeccionados, las Indus--- trias se expanden y las demandas de energía son aumentadas, -- la cuestión del uso más económico del agua, o su eliminación como medio de enfriamiento se convierte más importante.

2.2 Aplicaciones específicas.- Las unidades enfriadas por -- aire han sido usadas económicamente y con buen éxito para enfriamiento de fracciones del Petróleo, Aceites, Vapores Orgánicos de altos puntos de ebullición, condensación de vapor de agua, amoníaco, servicios, etc.

Desde un punto de vista general, la economía de aplicación -- favorece acercamientos de 30 a 40°F entre la temperatura de -- salida de proceso y la temperatura del aire ambiente, éstas --

unidades son usadas muy a menudo en conjunción con cambiadores de calor enfriados con agua, esto es, unidades que toman el fluido de proceso a la temperatura de salida del enfriador de aire y la llevan a la temperatura final deseada con agua. En algunas ocasiones, la unidad enfriada con aire puede ser llevada hasta 20-25°F de la temperatura de aire ambiente, si éste es el punto final requerido, más bien que añadir un enfriador de agua para este pequeño decremento de temperatura.

Como la temperatura se acerca a la del aire ambiente, el consumo de energía aumenta rápidamente y puesto que la superficie afecta los costos iniciales, un balance entre estos y los de operación es requerido para el diseño óptimo del cambiador, para una carga térmica fija.

3. Características.- Es imposible evitar comparaciones con los cambiadores enfriados con agua, puesto que estos comprenden el grupo mas grande de equipo para enfriamiento y son los estandares contra los cuales cualquier comparación debe ser hecha.

A continuación se enlistan una serie de ventajas y desventajas comparativas. Dependiendo del servicio, no todos los puntos se aplicarán, o en algunos casos pueden ser de menor importancia.

Cada aplicación debe ser evaluada sobre sus propios méritos,

y el enfriamiento con aire resulta ser la óptima selección en un sorprendente número de casos cuando un análisis total de todos los costos es hecho.

3.1 Ventajas.

3.1.1 El aire está siempre disponible, su uso ahorra agua para muchos servicios críticos. Grandes cantidades de agua son requeridas para servicios Industriales aún con recirculación en las torres de enfriamiento, hasta un 3% o más de la corriente de agua recirculada es requerida como repuesto.

3.1.2 El aire elimina el costo y el espacio requeridos para servicios de agua, bombas, equipo de tratamiento, torres de enfriamiento, filtros, áreas para evaluación de purgas, y líneas de evaluación y distribución.

3.1.3 La localización de la planta no está tan restringida como para enfriamiento con agua.

3.1.4 Los materiales de construcción del cambiador son más simples y baratos, porque, a diferencia del agua, el aire es raras veces corrosivo. Esto es, tubos encamisados no son requeridos.

3.1.5 Los enfriadores con agua que siguen a los enfriadores con aire estarán menos expuestos a la incrustación y corrosión que en el de enfriamiento total con agua, debido a las temperaturas más bajas.

3.1.6 La temperatura de la pared del tubo está limitada so-

lamente por los materiales de construcción, aunque uniones especiales entre aletas y tubos son requeridas a altas temperaturas. La temperatura de salida del agua generalmente tienen que ser sostenida hasta 120°F ó 130°F como máximo, para prevenir una Incrustación excesiva y esto restringe ambos, la temperatura del agua.

3.1.7 La resistencia del lado del aire es despreciable. Esto contrasta con la formación del lado del agua de una fuerte --- incrustación, corrosión y depósito de sales, etc. Lo cual es un problema con muchas aguas y es agravado por el método común de control de flujo de agua cuando su temperatura es baja, ya que regulando el flujo mediante válvulas, aumenta la formación y depósito de incrustaciones.

3.1.8 Se requieren menos paros, porque la incrustación del --- lado de la aleta es despreciable, muchas unidades pueden ser limpiadas con lanceados de aire o vapor durante la operación. Aún este simple paso es raramente necesario.

3.1.9 Los costos de operación son más bajos y pueden además -- ser reducidos con diseños que permiten potencias de ventiladores variables. La operación controlada del ventilador toma -- ventaja del hecho de que el aire está predominantemente abajo de su temperatura de diseño.

3.1.10 La capacidad puede generalmente ser aumentada cambiando la potencia del motor o ventilador, o ambos.

3.1.11 Los requisitos de labor de mantenimiento y operación --- son despreciables y los costos son aproximadamente un 25% de -- los del agua.

3.1.12 La limpieza es mas simple y barata y puede ser hecha con equipo poco costoso porque el fluido de proceso está dentro de los tubos, cuando el agua es el medio de enfriamiento, el lado del tubo es reservado para el agua con el fin de facilitar el - mantenimiento del mismo.

3.1.13 Los haces removibles de tubos no son un requisito, así - que tubos largos son prácticos y económicos. Los tubos de 30 ft. son comunes.

3.1.14 El mantenimiento de las torres de enfriamiento es dismi- nuido al menos en proporción a la reducción del uso del agua en la planta.

3.1.15 Las posibles contaminaciones de las corrientes de proceso por el agua (o viceversa) son eliminadas.

3.1.16 Las fugas de un enfriador de aire no permanecen indetec- tadas durante largo tiempo.

3.1.17 El aire caliente de salida puede ser usado para calentar edificios o áreas de proceso en temporadas frías.

3.1.18 La capacidad puede ser aumentada más que lo que es posi- ble con otros intercambiadores (por el simple aumento en el flu- jo de aire) normalmente esto requiere un bajo costo de equipo, pero los costos de operación son más altos.

3.2 Desventajas.

3.2.1 Los enfriadores de aire tienen costos iniciales mas altos para la misma carga térmica (2 ó 3 veces es común) cuando los comparamos únicamente con los cambiadores de agua. Aunque algunos ahorros iniciales han sido logrados en algunos proyectos por usar enfriadores de aire, es necesario incluir los costos de operación y un período de pago razonable para compararlos económicamente.

3.2.2 Los enfriadores de aire requieren más espacio cuando los comparamos con los enfriadores de agua, pero cuando los requisitos de espacio de la torre son considerados, el espacio total es más o menos el mismo. Además, el espacio de piso bajo los cambiadores de aire es a menudo usado para almacenamiento, o las unidades pueden ser instaladas en espacios no usados, tales como la parte superior de las camas de tuberías.

3.2.3 La localización del enfriador de aire es un gran problema debido a sus dimensiones, esto algunas veces representa un aumento en la tubería de proceso, de cualquier forma, el problema es menor que para torres de enfriamiento con agua. Los enfriadores de aire no deben ser colocados cerca de grandes obstrucciones, pues éstas pueden causar la recirculación de aire caliente.

3.2.4 Las temperaturas de diseño son hasta 20°F más altas para el aire de bulbo seco que para el agua (bulbo húmedo).

3.2.5 Las temperaturas del agua provenientes de torres o ríos, etc., varían muy poco. El aire por otra parte, fluctúa amplia y más rápidamente. Esto en ocasiones disminuye los costos de energía pero complica el control. Las lluvias han mostrado -- que disminuyen rápidamente las temperaturas de proceso en unidades sin protección.

También, el calor por radiación del sol debe ser considerado - en algunas áreas, por ésta razón, tanto como 2 ó 3% de superficie adicional es necesaria.

3.2.6 Las aletas que no son protegidas pueden doblarse con tormentas de granizo, aunque las partículas de granizo deben ser más grandes que 1" de diámetro para tener éste efecto.

3.2.7 La corrosión atmosférica es un problema en algunas plantas, y las aletas no pueden ser protegidas fácilmente de ésta clase de daño. Las aletas de cobre fueron comunmente usadas - antes de 1950, pero presentan la desventaja que se corroen severamente en atmósferas de dióxido de azufre. Las aletas de - aluminio son más económicas y han tenido un buen desempeño en casi todas las condiciones.

Cuando el aluminio es inapropiado, aletas de acero son usadas, por ejemplo, en plantas de cloruro de polivinilo.

3.2.8 Las pruebas de funcionamiento son difíciles de llevar a cabo debido a las variaciones del clima.

3.2.9 Un gran número de enfriadores de aire en una planta pue-

de causar aumentos locales de la temperatura del aire de la planta y debe ser considerado para estos diseños.

3.2.10 La Industria de los enfriadores de aire usa 1" de diámetro externo como estandar para los tubos. Otras medidas no son comunes con los fabricantes debido al costo de Ingeniería e Instrumentos necesarios para su fabricación. Esto representa una limitación para la selección óptima.

3.2.11 Los ventiladores hacen ruido, esto es una desventaja en algunas plantas cerca de zonas residenciales. Afortunadamente, varios ventiladores hacen poco más ruido que un solo ventilador, además, es posible usar ventiladores de 2 velocidades que disminuyen el nivel de ruido durante la noche.

3.2.12 El almacenamiento de los tubos aletados es más complicado que el que se tiene con tubos comunes y corrientes.

DESCRIPCION DE EQUIPO

II. DESCRIPCION DE EQUIPO

1. Generalidades

Los cambiadores utilizan aire atmosférico por el lado de los tubos aletados (en muy pocas ocasiones se usan tubos lisos) para enfriar o condensar flúidos en el interior de los tubos. Para que ocurra un enfriamiento efectivo, el aire debe fluir en convección forzada para desarrollar coeficientes de transferencia de calor aceptables.

Las Figs. 5 y 6 ilustran los 2 tipos denominados según el -- movimiento del aire, tiro inducido o forzado.

Las. Figs. 7 y 8 muestran el ensamblaje típico para los haces de tubos con cabezales forjados y manufacturados y tambien -- con placas de cubiertas bridadas.

2. Características.

2.1. Arreglo mecánico de los haces de tubos.- El diseño con-- vencional es el que tiene los haces de tubos horizontales, con uno o más ventiladores encima induciendo el aire a través de los tubos, o los ventiladores abajo para obtener tiro forzado. Los ventiladores usados son los de tipo propela, puesto que --- manejan grandes volúmenes de aire a bajas presiones estáticas. Los cambiadores en posición vertical son más usados para pe-- queñas cargas como en el enfriamiento del agua de las chaqueta de los medios motrices y otras aplicaciones. En los cambia-- dores con posición en A, las secciones son inclinadas y el -- tiro forzado es usado. Otro nuevo diseño de tiro inducido ---

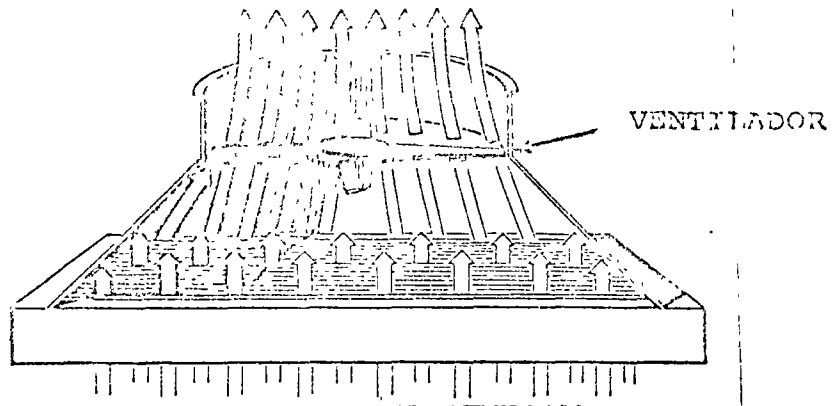


FIG. 5

Enfriador de aire tipo tiro inducido

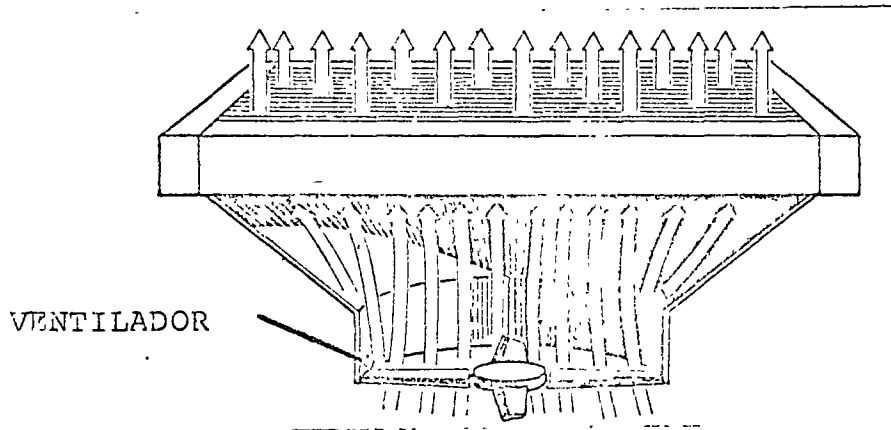


FIG. 6

Enfriador de aire tipo tiro forzado

FIGURA 7
Haz de tubos tí-
pico.

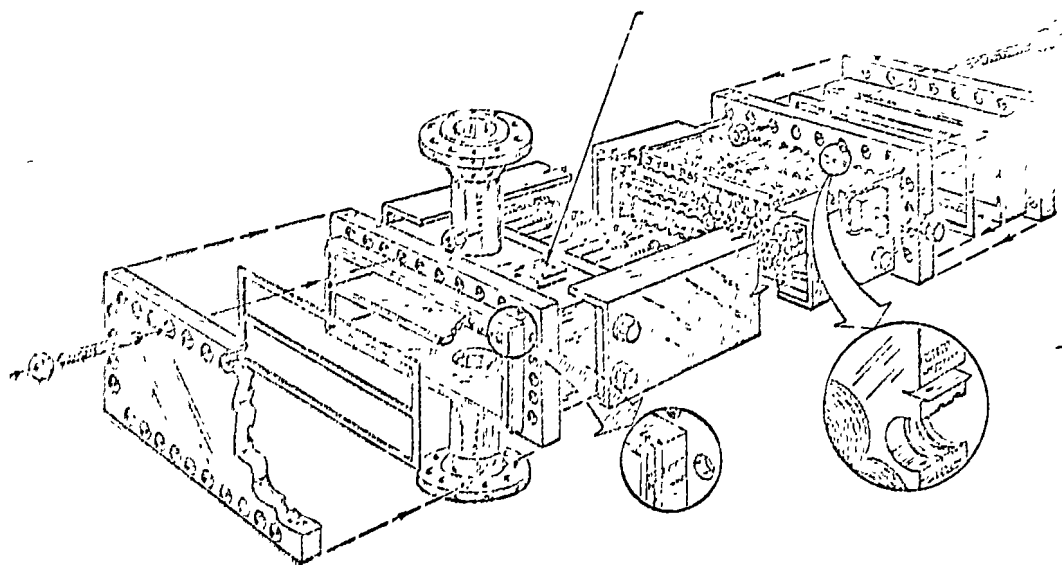
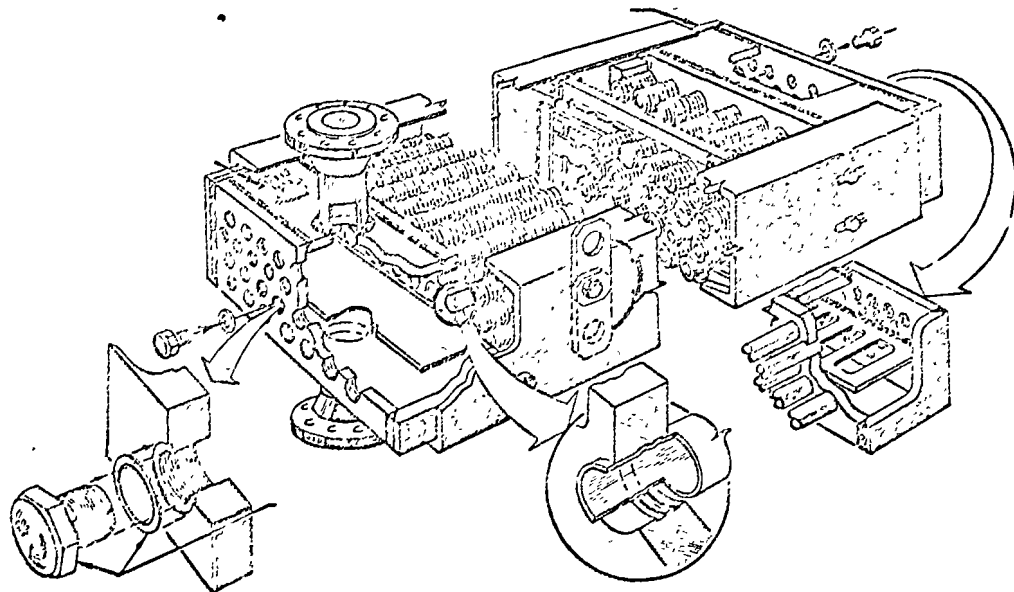


FIGURA 8
Haz de tubos con tapa
del cabezal bridada.

consta de varios haces de tubos erigidos verticalmente y arreglados en un anillo. El ventilador va montado encima del haz de tubos y forma una cámara junto con los haces. En este tipo de arreglo los costos de la estructura y la cámara son --- minimizados, pero el diseño no es adaptable para amplios rangos de dimensiones o velocidades de aire debido a la medida del ventilador y a los límites de capacidad, los pequeños --- flujos de aire y caídas bajas de presión son prácticas solamente con superficies aletadas especialmente desarrolladas para éste diseño.

La operación de los diseños en anillo y en "A", en mayor forma que los haces de tubos horizontales, está sujeta a las vicisitudes del viento a menos que persianas u otro método sean usados para controlar la temperatura del fluido de proceso.

2.2 Tipos de aletas.- Las aletas proporcionan una superficie extra que ayude a compensar la deficiencia en las características de transferencia de calor del aire. Para cualquier carga con un coeficiente alto de proceso, la máxima superficie de aleta por pie cuadrado de tubo será la más económica para coeficientes más bajos, las selecciones óptimas pueden ser -- logradas con menos superficie de aleta.

Desde el punto de vista térmico, la aleta óptima puede ser -- considerada como la que da los coeficientes de transferencia más altos para la mínima cantidad de metal. Esto sucede cuan

do el flux de calor es constante a cualquier sección de la aleta entre la base y el extremo superior de la misma.

Las aletas longitudinales y transversales corresponden a formas altas y angostas con pequeñas áreas de base. Además, los lados de las aletas deben tener curvaturas parabólicas.

2.2.1 Aletas longitudinales son comúnmente usadas en cambiadores de doble tubo o cambiadores de tubos y carcaza sin mamparas, o sea cuando el flujo procede a lo largo del eje del tubo.

Las aletas longitudinales son más comúnmente empleadas en problemas implicando gases y líquidos viscosos.

2.2.2 Las aletas transversales son hechas en una variedad de tipos y son empleadas principalmente para el enfriamiento y calentamiento de gases en flujo cruzado.

Las aletas helicoidales en la Fig. 9a. son clasificadas como aletas transversales y son adheridas en una variedad de formas tales como acanalado e insertado de la aleta, expandiendo el metal del tubo a la forma de la aleta o soldando la aleta continuamente al tubo.

Las aletas tipo disco son también transversales y son generalmente soldadas al tubo como se muestra en la Fig. 9b. y 9c.

Otros tipos de aletas transversales son conqidas como discontinuas, y varias formas tales como la aleta en estrella son mostradas en la Fig. 10.

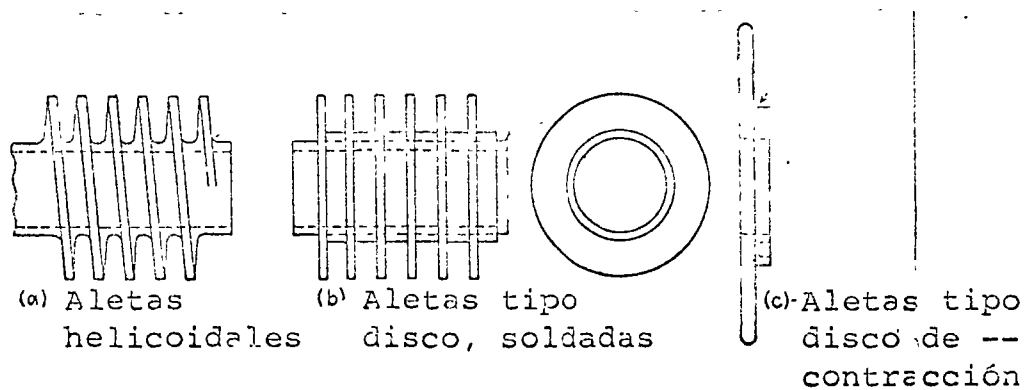


FIGURA 9 a, b y c.

Aletas transversales

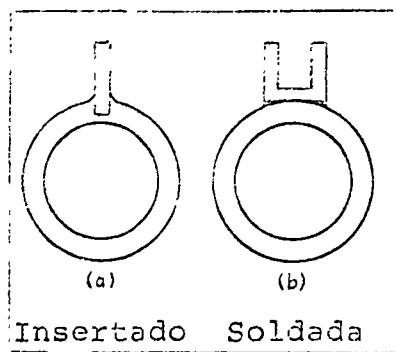


FIG. 10a

Colocación de aletas

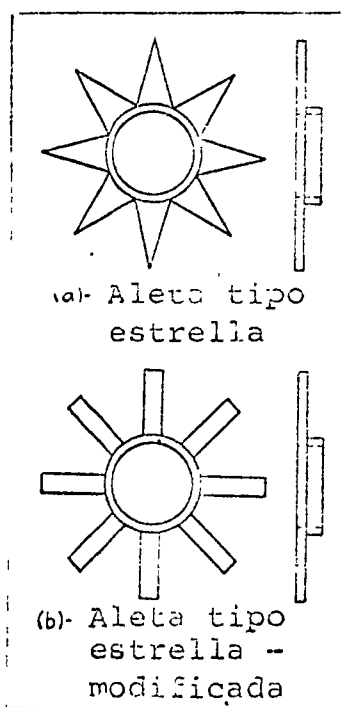


FIG. 10b

Aletas discontinuas

Otro tipo de aletas usadas en los tubos son las que tienen formas de conos, prámides o cilindros, las cuales se extienden de la superficie del tubo y pueden ser usadas para flujo longitudinal y flujo cruzado.

2.2.3 El número usual de aletas por pulgada, cuando son colocadas transversalmente oscila entre 7 y 11:

La superficie aletada puede ser aumentada cambiando la altura de las aletas del usual $1/2"$ a $5/8"$. El espesor de las aletas generalmente varía entre $0.014"$ y $0.019"$.

2.3 Características de los tubos.- Los tubos son generalmente de acero al carbón y admiralty con las aletas de aluminio o cobre según sea el servicio que efectúen.

Las relaciones de tubo aletado a tubo liso varían aproximadamente entre 13:1 a 23:1. Los diámetros comunes son $3/4"$ y $1"$ con aletas de $1/2"$ a $5/8"$ de altas, aunque algunas veces tubos de $1\frac{1}{2}"$ de diámetro y menores son necesarios para un diseño específico.

La longitud de los tubos varía de 5 ft. hasta 30 ft.

La longitud económica de los tubos generalmente está entre los 14 y 30 ft.

El funcionamiento de los tubos es variado para un número fijo de tubos e hileras, cambiando el número de aletas por pulgada lineal.

Cuando el fluido en los tubos produce un bajo coeficiente -

de película, la cantidad de superficie aletada es ajustada -- para proporcionar una área compatible y económica. Una relación alta de superficie aletada externa a superficie lisa, es de pequeño valor cuando los coeficientes interno y externo -- son más o menos los mismos.

El arreglo de los tubos oscila generalmente entre 2" y 2-3/8" en forma triangular a 60°

2.4 Haces de tubos y cabezales.

2.4.1 Haces de tubos.- Los cambiadores de calor enfriados con aire generalmente tienen haces de tubos rectangulares conteniendo varias hileras de tubos alineados horizontal y verticalmente.

Como diseño estandar, los haces de tubos pueden tener hasta 12' de ancho, 40' de largo y hasta 8 hileras de tubos en profundidad. Los haces de tubos pueden ser colocados uno encima de otro, resultando en una profundidad total de hasta 30 hileras de tubos para un servicio en particular.

La profundidad de los haces de tubos individuales pueden ser limitada, o en su caso cabezales divididos horizontalmente -- son requeridos, para servicios teniendo temperaturas altas y muy diferentes para cada paso, puesto que la expansión dicta un límite, uno o más haces de tubos del mismo o diferente ser vicio pueden ser combinados en una misma unidad.

Haces de tubos de diferente profundidad cuando son combinados en una sola unidad, son diseñados para una caída de presión - estática igual, y consecuentemente con diferentes velocidades de aire.

Las unidades teniendo la misma longitud que ancho generalmente tienen un ventilador mientras que unidades más largas tienen 2 o más ventiladores. Un diseñador experimentado intentará - estandarizar una unidad modelo, y tipo dentro de una planta - para minimizar la diferencias del equipo mecánico, tubos, reducir requisitos de espacio, mejorar la apariencia, y simplificar el montaje y localización del equipo.

2.4.2 Cabezales.

2.4.2.1 Tipo caja forjada, con tornillos taponando ambos extremos de los tubos. Los tornillos son generalmente considerados mejores para muchos servicios, esto proporciona además -- acceso individual a cada tubo con el fin de dar limpieza, reajustar el tubo y taponar el tubo en caso de fugas.

2.4.2.2 Tipo caja soldada.- Presenta los mismos rasgos que el anterior.

2.4.2.3 Tipo de placa cubierta.- Este tipo usa empaques y para tener acceso a los tubos es necesario quitar la placa. Este tipo generalmente se usa para servicios incrustantes y obstructores, donde una limpieza frecuente es necesaria.

2.4.2.4 Tipo cabezal multiple.- Este es usado en aplicacio--

nes a altas presiones.

2.5 Tipos de enfriadores según la posición del ventilador --
muchas comparaciones de los dos tipos básicos de cambiadores
de calor enfriados por aire han sido hechas y cada diseño --
tiene sus puntos buenos y malos, pero estos a menudo son ---
académicos, siendo más importante el costo.

Ambos tipos ofrecen ventajas inherentes en sus característi-
cas de diseño y operación.

2.5.1. El tipo de tiro forzado es generalmente preferido pa-
ra enfriamiento a altas temperaturas cuando la temperatura -
del aire de descarga es suficientemente alta para dañar el -
ventilador; a continuación se enumeran una serie de ventajas
de este tipo.

2.5.1.1 El equipo mecánico es más simple y más fácilmente --
accesible.

2.5.1.2 La estructura es más simple.

2.5.1.3 Se adapta fácilmente a cualquier tipo de unidad mo--
triz.

2.5.1.4 Menos energía del ventilador es requerida para el mis-
mo funcionamiento.

2.5.1.5 Las secciones son más fáciles de remover para repara-
ción.

2.5.2 El tipo de tiro inducido permite la descarga del aire
caliente a máxima velocidad para un rápido desalojo del aire.

A continuación se enumeran una serie de ventajas de este tipo.

2.5.2.1 Existe menos recirculación del aire caliente.

2.5.2.2 Hay una mejor distribución del aire sobre el banco de tubos aletados.

2.5.2.3 Las secciones de tubos aletados estan más cercanas al piso y por lo tanto, es mas fácil darles mantenimiento.

2.5.2.4 Da máxima protección de la lluvia, granizo, etc.

2.5.2.5 Requiere menos tubería de conexiones.

2.6 Ventiladores.- Los ventiladores son máquinas con aumentos relativamente bajos de presión, los cuales manejan gases o vapores por medio de aspas rotantes, y cambian la energía mecánica de rotación en presión o trabajo sobre el gas.

El resultado de este trabajo sobre el fluido será en la forma de energía de velocidad o de presión, o una combinación de --- ambas.

El aumento máximo usual de presión de la entrada a la descarga a través de un ventilador, es alrededor de una psi, de cualquier forma, existen ocasiones en que dos y tres psi son desarrolladas.

Los ventiladores son normalmente usados en aplicaciones donde en la succión el gas está a presión atmosférica o solamente a unas pulgadas de agua abajo o arriba de la atmosférica. Esto es debido generalmente a los rasgos de construcción de las aspas de los ventiladores, siendo básicamente fabricados de ho--

jas de metal, las cuales no aguantan altas presiones. Los -- ventiladores son clasificados como:

2.6.1 Centrífugos.

Es diseñado para mover aire o gases sobre un amplio rango de volúmenes y presiones.

2.6.2 De aspa axial.

Es diseñado para mover aire o gases sobre un amplio rango de volúmenes y presiones. La diferencia con el anterior estriba en la construcción.

2.6.3 De tubo axial.

Es diseñado para mover aire o gases sobre un amplio rango de volúmenes a medias presiones, su construcción es similar al anterior.

2.6.4 De propela.

Es diseñado para mover aire o gases sobre un amplio rango de volúmenes a bajas presiones. Este tipo de ventilador es el -- que comunmente se usa en cambiadores de calor enfriados por -- aire.

Este tipo generalmente no opera con tuberías o ductos en la -- succión o la descarga, y maneja el aire o gases desde un espa -- cio abierto a otro.

Las presiones son generalmente muy bajas y los volúmenes de -- penden de las dimensiones, arreglo de las aspas, número de as -- pas y velocidad.

Para servicios generales 4 aspas por ventilador son conside--

radas mínimas, las aspas son generalmente de arreglo fijo hasta de 48 pulgadas de diámetro, con mayores diámetros un ----- arreglo ajustable puede ser usado. Arreglo fijo es usado con aspas hasta de 60 pulgadas de diámetro, cuando las aspas son de aluminio y el ventilador está conectado directamente a la flecha del motor.

Arreglo variable es usado con transmisión por bandas, engranes, etc.

Aspas de aluminio son usadas hasta 300°F, mientras que las de plásticos están limitadas hasta 160° - 180°F

El ruido que produce el aire es generalmente menor con ventiladores de muchas aspas (cuatro o más) que con dos o tres aspas. En general, el ruido no es un problema real cuando es comparado con otra maquinaria de operación y el nivel de frecuencia es muy bajo y no penetrante. El nivel de ruido es generalmente limitado a 75 decibeles máximo a 50 ft. de distancia del ventilador, y la velocidad periférica es limitada a ----- 11,000 - 12,000 ft/min. (= π x diámetro de la aspa en ft x R.P.M.) Esta puede ser más alta para aspas abajo de 48 pulgadas de diámetro.

2.6.4.1 Cálculo de la potencia.- La potencia puede ser calculada, por medio de las siguientes formulas:

$$HP = \frac{(ACFM) (p_t)}{(6,356) (\epsilon_f) (\epsilon_d)} \quad (2.1)$$

6:

$$(HP)_s = \frac{0.0001573 (ACFM) (P_s)}{\epsilon_s} \quad (2.2)$$

Donde:

Pv = Presión de velocidad, generalmente 0.1 pulgadas de agua

Ps = Presión estática

Pt = Pv + Ps

ef = Eficiencia del ventilador, generalmente 0.65

ed = Eficiencia del motor, generalmente 0.95

ACFM = ft³/min. a las condiciones de entrada del ventilador

2.6 Medios motrices.- El ventilador puede ser operado por --
cualquiera de los siguientes medios:

2.6.1 Motor eléctrico directamente conectado o por bandas.

2.6.2 Motor eléctrico de 2 velocidades con bandas o engranes.

2.6.3 Turbina de vapor directamente conectada, por medio de --
engranes.

2.6.4 Motor de gasolina con transmisión por bandas y engranes.

Los engranes deben ser especificados según la "American Gear -
Manufacturer's Association" para asegurar un adecuado factor -
de servicio mínimo, el engrane tipo espiral es más usado que -
el tipo gusano y además más barato. Cuando los engranes son -
usados con unidades de tipo tiro inducido, la temperatura máxima
del aire de salida debe ser limitada por la especificación
de los engranes, que deben ser evaluados a la temperatura del
aire esperada.

Generalmente las transmisiones por bandas no son usadas a más de 30 H.P. Las bandas no deben ser usadas en condiciones donde la temperatura rodeando a la unidad, sea mayor a 160°F, --- con o sin ventilador operando, esto es de particular importancia en condiciones de tiro inducido, donde las bandas estarían en la corriente de salida.

2.7 Control.

El problema principal que se deriva en la operación de un cambiador de calor enfriado con aire es el control de la temperatura de salida del lado de proceso.

El personal familiarizado con enfriadores de agua, algunas veces experimenta algunas dificultades en reorientarse al control --- de las unidades enfriadas con aire.

La temperatura del agua no varía tan ampliamente como lo hace la del aire durante todo el año, y unicamente por "By-Pasear" el agua o la corriente de proceso, es posible obtener un buen control de las condiciones. Especialmente, solo la diferencia verdadera de temperaturas es afectada.

Cuando el aire es el medio de enfriamiento, no solamente su temperatura está sujeta a cambios, sino ambos, el coeficiente de transferencia de calor y la diferencia verdadera de temperatura son afectadas cuando se varía su flujo como medio de control. Como el flujo de aire es reducido, el coeficiente y el L.M.T.D. son disminuidos.

Las temperaturas de salida de proceso pueden ser reguladas -- por una simple válvula de control, por "By - Pasear" el flúido, si es que no existe peligro de congelamiento para el mismo.

Las temperaturas de proceso pueden ser reguladas por cualquiera de las siguientes formas:

2.7.1 Las persianas han sido usadas por muchos años. La operación puede ser manual o automática, las persianas también ofrecen protección contra la lluvia, granizo y en menor grado, contra el sol.

2.7.2 Los motores eléctricos de 2 velocidades tienen muy bajo costo y son muy satisfactorias para muchos servicios.

Un serio inconveniente es la fuerte demanda para el arranque - del motor.

2.7.3 Los motores eléctricos de velocidad variable, turbinas - de vapor o medios motrices hidráulicos son usados, pero no son tan comunes como los otros métodos.

2.7.4 Los ventiladores con arreglo variable, automáticamente - fijados y usando medios motrices de velocidad constante, permiten un control de $\pm 1.5^{\circ}\text{F}$ en el flúido de proceso.

Combinaciones de más que un método de control deben ser evitadas puesto que pueden causar un serio problema en la operación. Cuando se tienen enfriadores con varios ventiladores, pueden - disminuirse los gastos sin afectar seriamente el grado de regulación, usando únicamente un ventilador controlado automática-

mente. A la capacidad más baja del ventilador controlante, -
la señal de control corta uno de los otros ventiladores, la -
regulación puede ser completamente automática, secuenciando -
todos los arrancadores de los ventiladores a una unidad con--
trolante. En cualquier arreglo, es importante que el venti--
lador controlante tenga tanta capacidad como y preferiblemen-
te más que los otros.

A continuación se ilustran tres tipos de control en las figu-
ras 10, 11 y 12.

Control neumático de
Temperatura

Regulador de pre-
sión y filtro

Operador neumático de las
persianas con posisionador

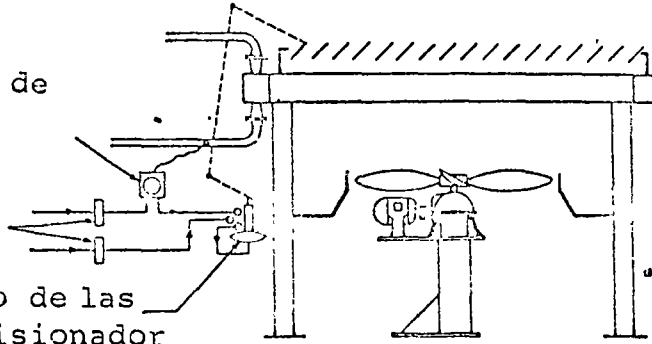


FIGURA 11a

Control neumático de
temperatura

Regulador de pre-
sión y filtro

Relevador neumático

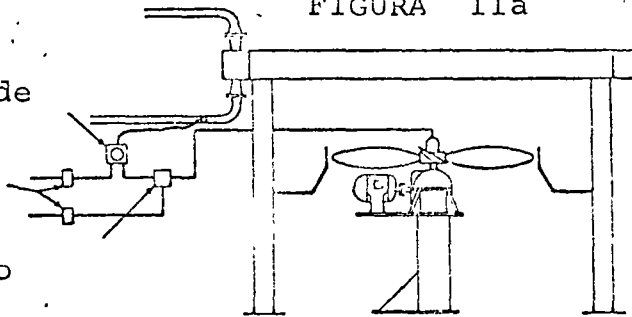


FIGURA 11b

Operador neumático de las
persianas con posicio-
nador

Regulador de pre-
sión y filtro

Control neumático de
temperatura

Relevadores eléctrico
neumático

Controlador del motor

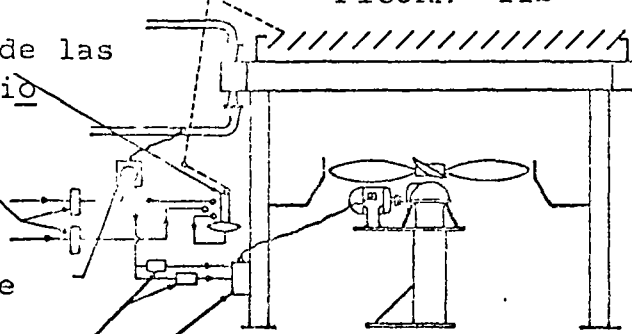


FIGURA 12

TEORIA

1.- PROBLEMA

1. Lado del aire

1.1 Coeficiente del aire.

Del estudio de Briggs y Young se obtiene una ecuación para el coeficiente del lado del aire para tubos aletados:

$$\frac{h_t d_r}{K_a 12} = 0.1378 \left(\frac{d_r G_m}{12 \mu_a} \right)^{0.718} \left(\frac{C_a \mu_a}{K_a} \right)^{1/3} \left(\frac{s}{1} \right)^{0.296} Fr \quad (3.1)$$

Donde h_t es referido a la superficie total expuesta A_t .

Para la exactitud requerida y con el fin de simplificar los cálculos lo más posible, el calor específico, la conductividad térmica y la viscosidad del aire pueden ser tomados a una temperatura promedio, teniendo para el caso en que ésta sea 120°F la siguiente ecuación:

$$h_t = 0.0356 \frac{\left(\frac{G_m}{d_r} \right)^{0.718}}{0.282} \left(\frac{s}{1} \right)^{0.296} Fr \quad (3.2)$$

Si se expresa:

$G_m = 60 p_a v_m$ (3) con $p_a = 0.075$ refiriéndose a las condiciones estandars, substituyendo (3) en (2) y multiplicando por A_t / A_o . Para referir el coeficiente, a la superficie lisa se obtiene:

$$h_a = 0.105 \frac{\left(\frac{v_m}{d_r} \right)^{0.718}}{0.282} \left(\frac{s}{1} \right)^{0.296} \left(\frac{A_t}{A_o} \right) Fr \quad (3.3)$$

Valores de A_t/A_o y $(s/1)^{0.296}$ son dados en la tabla I. en éste caso particular los tubos disponibles son de 1" de diámetro externo para lo cual $(d_r)^{0.282} = 1$.

1.2 Coeficiente de incrustación.- Generalmente no es acostumbrado incluir el coeficiente de incrustación para el lado del aire puesto que es despreciable. Suponiendo que una resistencia de 0.002 es típica para un haz de tubos aletados, si tomamos el caso en que la relación de la superficie total a la superficie lisa sea 20 la resistencia referida al tubo liso será de 0.002/20 o sea 0.0001.

Existen ocasiones en que una cubierta de pintura debe ser aplicada a las aletas. Estos casos presentan un pequeño efecto -- sobre el coeficiente total de transferencia.

En el caso en que la atmósfera sea aceitosa y polvorienta, un depósito puede ser formado sobre las aletas; esto tendría que ser tomado en consideración.

1.3 Caída de presión.

Briggs y Young dan también la ecuación recomendada para la caída de presión del lado del aire, la cual es:

$$\Delta P_a = 0.438 \times 10^{-6} (V_m)^{1.725} R \frac{(0.074)}{P_a} F_s \quad (3.4)$$

$$\Delta P_a = 3.24 R (V_m)^{1.725} F_s / 10^8 P_a \quad (3.5)$$

El factor de seguridad F_s ha sido añadido a la ecuación (3.4) y un mínimo valor recomendable de 1.1 debe ser usado, la razón es que los investigadores han tenido dificultad para encontrar una relación satisfactoria para ΔP_a

La ecuación como queda escrita está de acuerdo solamente --
con las características de aletas y tubos de la tabla I. -
La densidad del aire es evaluada a las condiciones de pre--
sión del lugar donde va a ser instalado el enfriador y a la
temperatura promedio en el haz de tubos.

2. Lado de los Tubos.

2.1 Coeficiente de enfriamiento.- El procedimiento para la
evaluación del coeficiente en el interior de los tubos, es -
el propuesto por Donald Kern para flujos con número de -----
Reynolds mayor de 2100 (régimen turbulento) que utiliza la
siguiente ecuación: (Ref. Bibliográfica) (2)

$$\frac{h_i D}{k_p} = 0.027 (Re)^{0.8} (Pr)^{1/3} \phi_t \quad (3.6)$$

La gráfica de esta ecuación se presenta en la Fig. 13, la --
cual es aplicable para líquidos orgánicos, soluciones acuo--
sas y gases.

2.1.1 Método

2.1.1.1 Es necesario determinar en primer lugar la temperatur
ra calórica de lado de proceso, para lo cual es indispensa--
ble conocer las temperaturas de entrada y salida de proceso
y del aire.

2.1.1.1.1 Calcular ΔT

$$\Delta T = T_1 - T_2 \quad (3.7)$$

2.1.1.1.2 Calcular $\Delta T_c / \Delta T_h$

T A B L A 1

DATOS Y DIMENSIONES PARA ALGUNOS TUBOS ALETADOS DE CAMBIADORES.

ENFRIADOS POR AIRE

TIPO	D.E. DE ALETA d_f	DIAMETRO MEDIO d_r	ALETAS / IN n	ESPEJOR DE ALETA t	ARREGLO DE TUBO p	$\frac{A_T}{A}$	ALETA r_m	$\frac{V_m}{V}$	$\frac{s}{l}$	0.296
E	2.0	1.00	11	0.014	2.1	17.5	0.000590	2.22	0.575	
L	2.0	1.04	10	0.014	2.1	15.6	0.000448	2.27	0.601	
X	2.0	1.10	9	0.019	2.1	13.6	0.000441	2.48	0.625	
E	2.25	1.00	11	0.014	2.375	23.4	0.000736	2.00	0.540	
L	2.25	1.04	10	0.014	2.39	20.9	0.000756	2.02	0.561	
X	2.25	1.10	9	0.019	2.375	18.3	0.000563	2.20	0.582	

$$\frac{\Delta T_c}{\Delta T_h} = \frac{T_2 - t_1}{T_1 - t_2} \quad (3.8)$$

2.1.1.1.3 Con ΔT y API (para el caso de fracc. de petró--
leo) o $U_c = U_d/U_c$ se encuentra en la Fig. 14, K_c .

El valor de K_c corresponde al coeficiente de transferencia
de calor controlante, el cual es supuesto para establecer --
la variación de U con la temperatura.

2.1.1.1.4 Con $\Delta T_c / \Delta T_h$ y K_c se obtiene de la Fig. 15, el
factor F_c , el cual se aplica en la obtención de la tempera--
tura calórica por la ecuación:

$$T_c = T_2 + F_c (T_1 - T_2). \quad (3.9)$$

2.1.1.2 Con el flujo en masa y la densidad ρ_p se obtiene -
el flujo volumetrico v_v .

$$v_v = W / \rho_p \quad (3.10)$$

2.1.1.3 Fijando una velocidad V con el fin de obtener una -
caída de presión cercana a la permisible, se encuentra el --
área de flujo.

$$a_F = \frac{v_v}{v} \quad (3.11)$$

2.1.1.4 Con el área de flujo por tubo a'_F y el área de flujo
total a_F se obtiene el número de tubos:

$$\text{No. de tubos} = \frac{a_f}{a'_F} \quad (3.12)$$

2.1.1.5 Con el área de flujo total y el flujo en masa se ob-
tiene la masa velocidad.

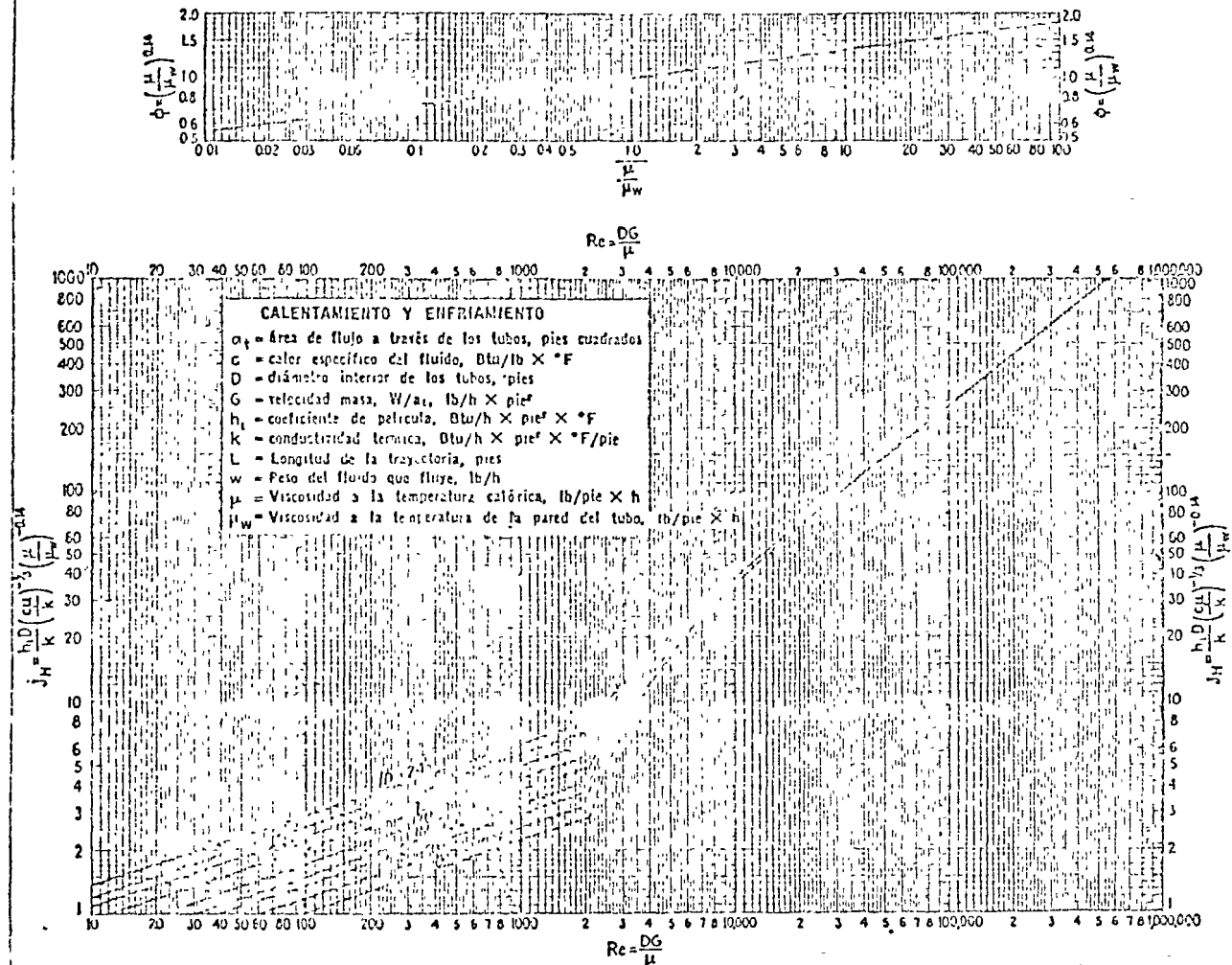


FIG 13 Curva de transferencia de calor lado de tubos. (Adaptada de Sieder y Tate)

$$G_t = \frac{W}{a_t} \quad (3.13)$$

2.1.1.6 Con T_c se obtienen μ_p , K_p , C_p

2.1.1.7 Se calcula el número de Reynolds con:

$$N_{Re} = \frac{D G_t}{\mu_p} \quad (3.14)$$

2.1.1.8 De la Fig. 13 con N_{Re} se obtiene: J_h

$$J_h = \frac{h_i D}{K_p} \left(\frac{C_p \mu_p}{K_p} \right)^{-1/3} \phi_t \quad (3.15)$$

2.1.1.9 Se calcula

$$K_p \left(\frac{C_p \mu_p}{K_p} \right)^{1/3} \quad (3.16)$$

o se obtiene de la Fig. 16 si se trata de fracciones del --
petróleo.

2.1.1.10 Despejando para:

$$\frac{h_i}{\phi_t} = J_h \cdot \frac{K_p}{D} \left(\frac{C_p \mu_p}{K_p} \right)^{1/3} \quad (3.17)$$

2.1.1.11 Se calcula el coeficiente referido al área externa:

$$\frac{h_{i o}}{\phi_t} = \frac{h_i}{\phi_t} \frac{ID}{OD} \quad (3.18)$$

2.1.1.12 Se calcula la temperatura de la pared:

$$t_w = \bar{t} + \frac{h_o/\phi \pm}{h_{io}/\phi_t + h_o/\phi} (T_c - \bar{t}) \quad (3.19)$$

Pero $\phi \approx 1$

FIG. 15

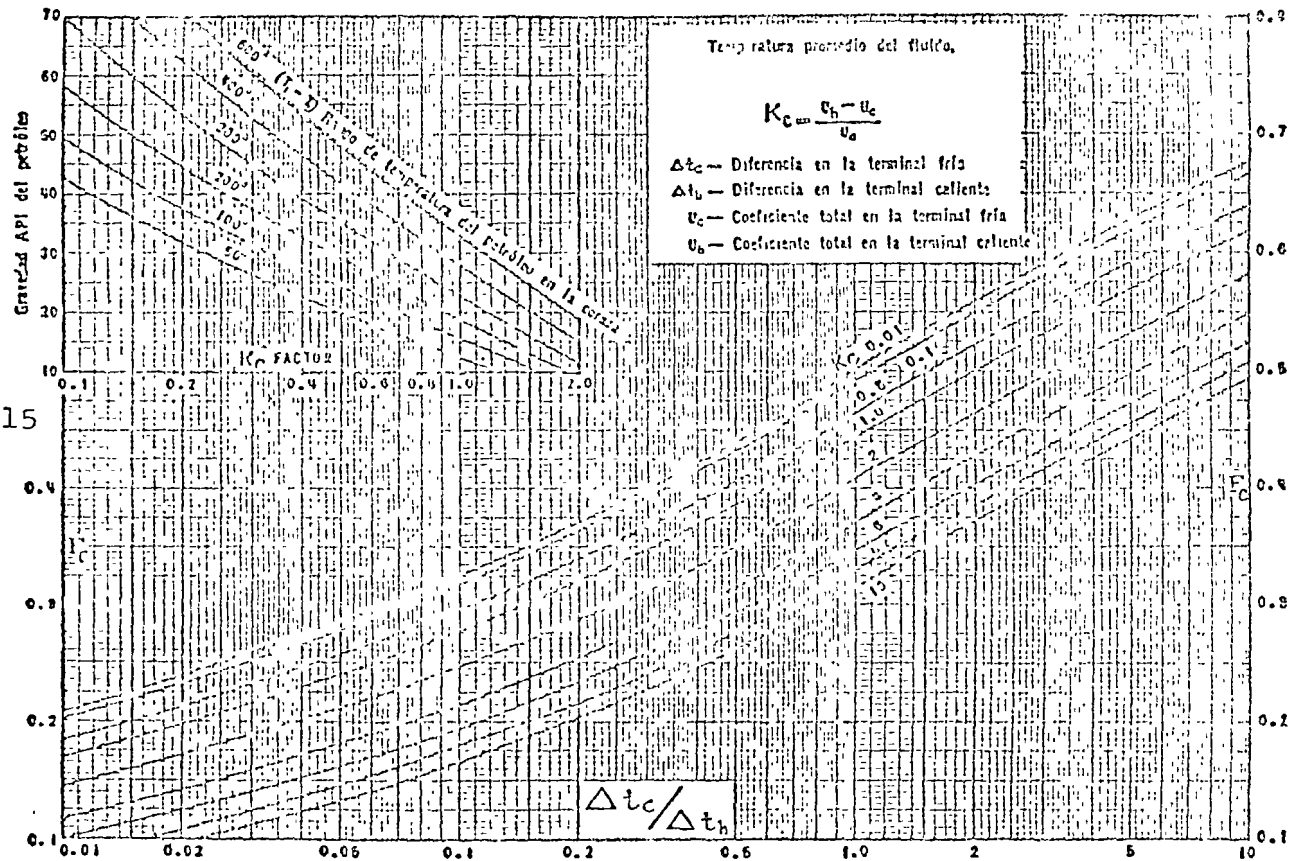


FIG. 14 Factor F de temperatura calórica. (Standards of Tubular Exchanger Manufacturers Association, 2a. ed., New York, 1919)

2.1.1.13 Se calcula μ_w con t_w

2.1.1.14 Se calcula ϕ_t

$$\phi_t = \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (3.20)$$

2.1.1.15 Corrigiendo el coeficiente:

$$h_{io} = \left(\frac{h_{io}}{\phi_t} \right) \phi_t \quad (3.21)$$

2.2 Factor de incrustación.

Valores numéricos de los factores de incrustación para una --
variedad de servicios de procesos, son encontrados en el apen
dice de cualquier libro que trate con intercambiadores de ---
calor.

Los valores enlistados intentan proteger al intercambiador --
de la baja en eficiencia térmica que sufren éstos con el uso.

2.3 Caída de presión.

El procedimiento para la evaluación de la caída de presión -
es el propuesto por Donald Kern, que a su vez se ha basado en
las correlaciones de Sieder y Tate, para obtener los factores
de fricción para flúidos siendo calentados o enfriados en el
interior de tubos.

La caída de presión del lado de los tubos se calcula en 2 par
tes, caída de presión en los tubos y caída de presión en los
retornos:

$$\Delta P_t = P_t + P_r \quad (3.22)$$

En donde.

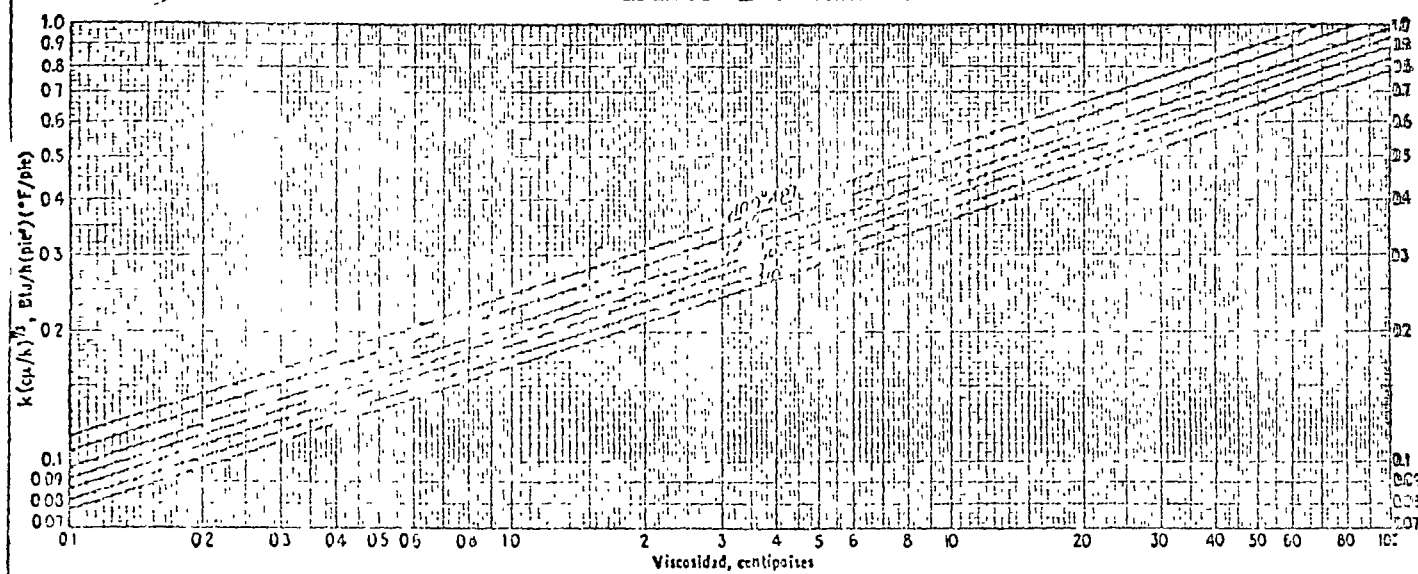


FIG. 16. Valores de $k(c/\mu)^{1/2}$ para hidrocarburos

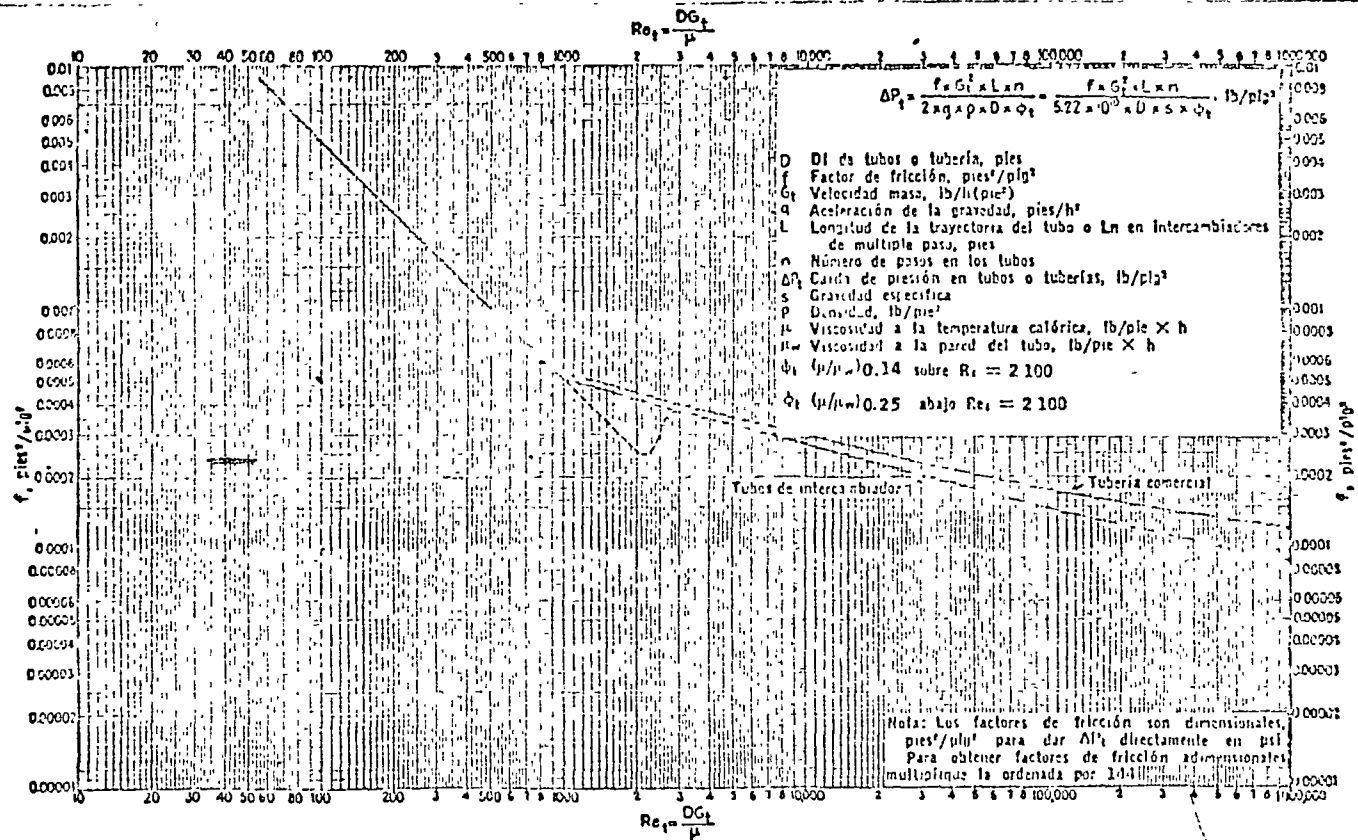


FIG. 17 Factores de fricción, para lado de tubo. (Standards Exchanger Manufacturers Association, 2a. ed., New York, 1949)

$$P_t = \frac{f G_t^2 L n}{5.22 \times 10^{10} d s \phi_t} \quad (3.23)$$

y:

$$P_r = \frac{4n v^2}{2s g} \left(\frac{62.5}{144} \right) \quad (3.24)$$

El factor de fricción f para la caída de presión en los tubos está dado en la Fig. 17 como función del número de Reynolds, y la caída de presión por paso en los retornos en la Fig. 18, como función de la masa velocidad.

2.3.1 Método

2.3.1.1 Calcular el número de Reynolds.

$$N_{Ret} = \frac{G_t D}{v_p} \quad (3.14)$$

2.3.1.2 Determinar el factor de fricción f de la Fig. 17 con el N_{Ret}

2.3.1.2 Calcular P_t con:

$$P_t = \frac{f G_t^2 L n}{5.22 \times 10^{10} d s \phi_t} \quad (3.23)$$

2.3.1.4 Determinar P_r /Paso de la Fig. 18 con la masa velocidad G_t .

2.3.1.5 Calcular P_r con:

$$P_r = \frac{4n v^2}{2s g} \left(\frac{62.5}{144} \right) \quad (3.24)$$

2.3.1.6 Calcular ΔP_t con:

$$\Delta P_t = P_t + P_r \quad (3.22)$$

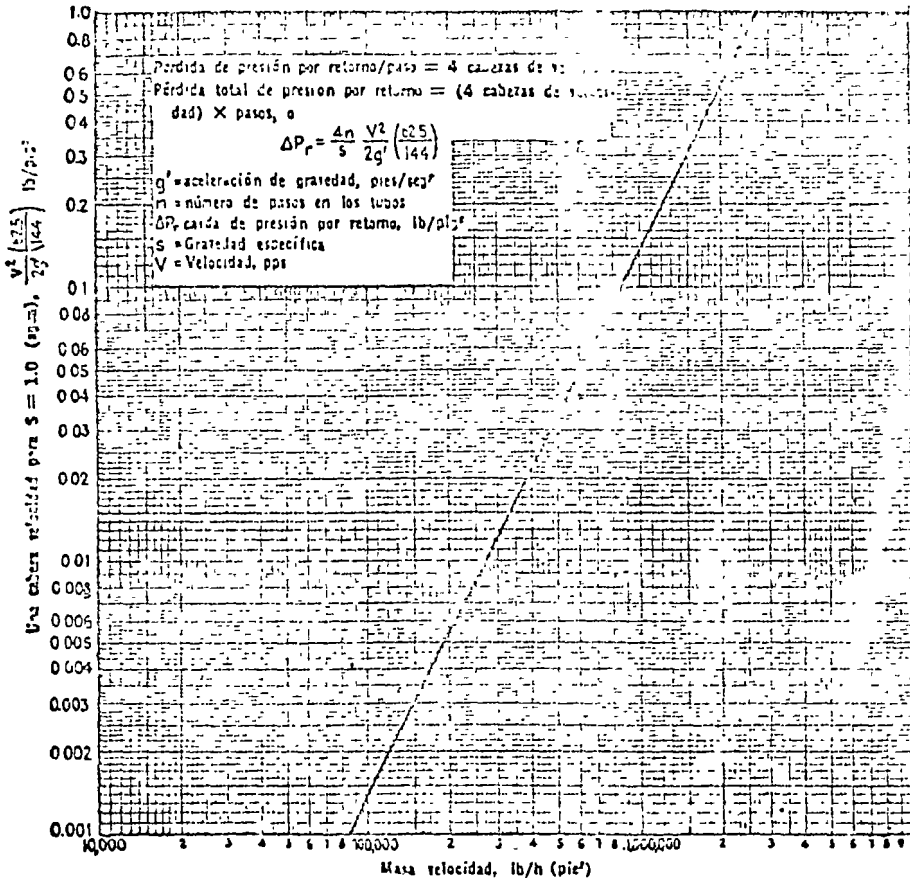


FIG. 18 Pérdida de presión por retorno, lado de tubos

3. Diferencia media logarítmica de temperaturas (LMTD) y factor de corrección (F_t).

3.1 Diferencia media logarítmica de temperaturas L.M.T.D.

Para calcular el L.M.T.D., es necesario conocer las temperaturas de entrada y salida de proceso y del aire.

La formula utilizada es:

$$\text{L.M.T.D.} = \frac{T_1 - t_2 - (T_2 - t_1)}{\ln \frac{T_1 - t_2}{T_2 - t_1}}$$

La razón por la cual se usa el L.M.T.D. para flujo a contra corriente, es que el fluido de proceso, aunque lleva una -- trayectoria en zig - zag, va descendiendo en sentido contra rio al aire cuando se tienen varios pasos en los tubos.

3.2 Factor de corrección

En los cambiadores de calor enfriados por aire, el tipo de flujo que se desarrolla es el cruzado.

Para uno y dos pasos con flujo cruzado, las gráficas desarrolladas por Bowman, Mueller y Nagle son usadas para obtener el factor de corrección.

Para tres pasos Stevens, Fernández y Woolf desarrollaron una gráfica; el método de cálculo para obtener el factor de co-- rrección de estas gráficas es el siguiente:

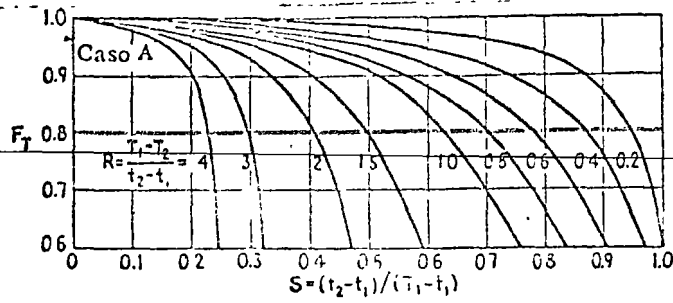
3.2.1. Encuentre.

$$R = \frac{T_1 - T_2}{t_2 - t_1} \quad (3.26)$$

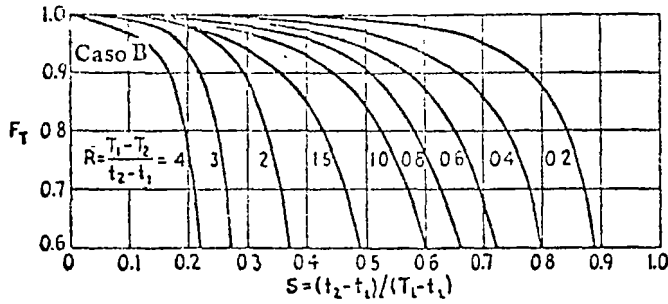
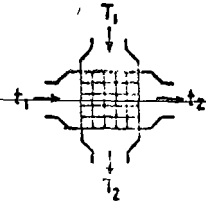
3.2.2 Encuentre

$$S = \frac{t_2 - t_1}{T_1 - t_1} \quad (3.27)$$

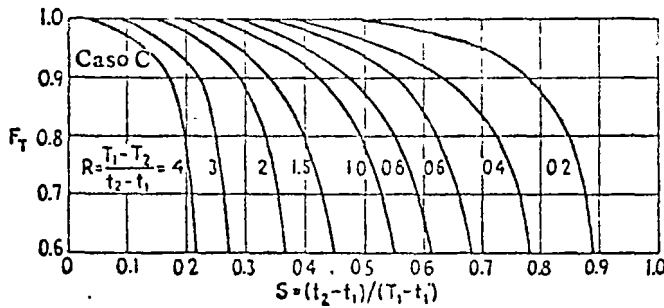
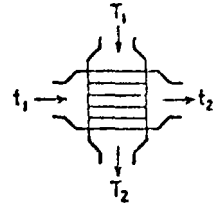
3.2.3 Con R y S y el número de pasos lea F_t de la Fig. 19



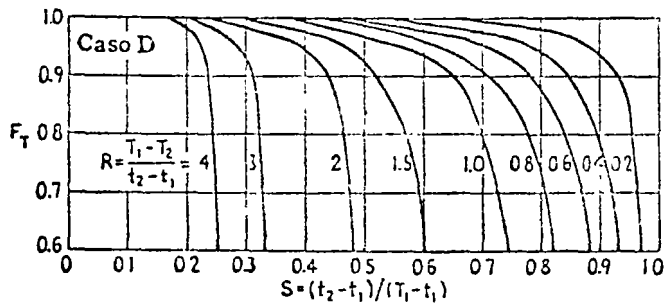
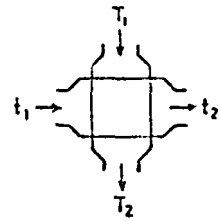
(a) Ambos fluidos sin mezclarse



(b) Un fluido mezclado, otro sin mezclar



(c) Ambos fluidos mezclados



(d) Contracorriente dos pasos, fluido en la coraza mezclado, fluido en los tubos sin mezclar

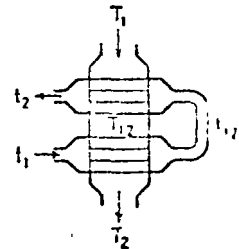


FIG. 19 Factores de corrección y diferencia de temperatura para flujo cruzado. (Bowman, Mueller and Nagle, Transactions of the ASME)

D I S E Ñ O

IV.- DISEÑO

1. Consideraciones de Diseño

Se puede considerar como un buen diseño, aquel que satisfaciendo las necesidades de proceso, representa el costo total mínimo de operación e inversión.

1.1 Temperatura de Diseño del Aire.- La temperatura máxima de diseño debe ser seleccionada para que no se exceda de más que 2-5% del tiempo.

Gráficas de las temperaturas diarias tambien como curvas mostrando el número de horas y el tiempo del año durante el cual cualquier temperatura es excedida son valiosas y muchas veces necesarias para establecer una temperatura de diseño del aire económica.

1.2 La potencia del ventilador, el número de hileras de tubos y las velocidades del aire son los factores a considerar para obtener los costos totales mas bajos.

1.3 El número de hileras de tubos tienen un marcado efecto -- sobre los costos de operación e inversión por lo que rango de hileras entre 4 y 8 debe ser considerado para el diseño económico de un cambiador.

1.4 La potencia del ventilador debe ser evaluada lo más cercana a la realidad, puesto que ésta, junto con los costos de --- mantenimiento, determinan los costos de operación, y a mayor - potencia menor superficie es requerida, pues el coeficiente de

transferencia y la L.M.T.D. son aumentados.

1.5 Los ventiladores tipo propela tienen la característica - de impulsar grandes volúmenes de aire a bajas caídas de presión, por lo que generalmente este tipo de ventilador es usado en los cambiadores de calor enfriados por aire.

Aunque el rango aceptable de caídas de presión es bastante -- pequeño, existen ocasiones donde muy bajas o muy altas caídas de presión son las mas económicas.

1.6 Como en el caso de los cambiadores de tubos y carcaza la máxima longitud práctica de tubos, generalmente dá la selección más barata, debido a que ésto minimiza los requisitos -- de cabezales y las pérdidas de presión en el flúido de proceso.

Los tubos más comunes son de 1" de diámetro externo y son los que se utilizan comunmente para el diseño de un cambiador.

1.7 La geometría de las aletas y los materiales de las mismas son limitados a lo que puede ser económicamente fabricado.

1.8 La resistencia del metal puede ser considerada como la -- suma de las resistencias del tubo y la aleta. En la tabla I - son dadas algunas resistencias de aletas las cuales pueden ser usadas si tubos de 1" de diámetro externo son los considerados en el diseño.

1.9 Las aletas son generalmente de aluminio, a menos que una

atmósfera contaminada pueda perjudicarlas. Para tal caso, -
aletas de acero inoxidable pueden ser usadas.

1.10 El coeficiente de transferencia de calor del aire no -
es afectado por el número de hileras y es mejor en el tipo -
de tiro forzado, debido a que se forma una turbulencia en la
la. hilera de tubos, lo cual no ocurre en el tipo tiro indu-
cido.

1.11 Para una selección óptima el número de hileras de tu--
bos estará entre 2 y 14, pero las consideraciones prácticas
los limitarán a 4, 6 y 8 hileras. Esto es debido a que al -
aumentar el número de hileras, la caída de presión aumenta -
considerablemente y posiblemente ningún ventilador satisfaga
tales condiciones.

2. Procedimiento de Diseño.

2.1 Datos.

2.1.1 Temperatura de fluido de entrada del proceso

2.1.2 Temperatura del fluido de salida del proceso

2.1.3 Temperatura del fluido de entrada del aire del proceso.

2.1.4 Gasto del fluido de proceso

2.1.5 Caída de presión máxima permisible, del lado de proceso

2.1.6 Longitud y diámetro de tubos, tipo y características de
la aleta tomadas de la tabla I.

2.2 Diseño.

2.2.1 Para 4, 6 y 8 hileras y para velocidades de 300, 500, -

700 y 900 ft/min., encontrar h_a , p U.

2.2.2 Con U , R , d_o , P y V encontrar.

$$\frac{LMTD}{\Delta t_a} = \frac{1.08 p V}{d_o \pi U R} \quad (4.1)$$

2.2.3 Con diferentes valores supuestos de t_2 de salida -- del aire, encontrar LMTD y Δt_a y graficar L.M.T.D. F_t Vs. - Δt_a .

2.2.4 Con los valores encontrados en 2.2.2 y utilizando la gráfica construida en 2.2.3 encontrar los valores verdaderos de Δt_a y t_2

2.2.5 Encontrar L.M.T.D.

2.2.6 Con L.M.T.D. Q y U encontrar área lisa:

$$A = \frac{Q}{U \text{ L.M.T.D.}} \quad (4.2)$$

2.2.7 Encontrar flujo de aire y las dimensiones del cambiador por la siguientes ecuaciones:

$$W_a = Q / \Delta t_a \cdot C_a \quad (4.3)$$

$$SCFM = W_a / 60 \rho_s \quad (4.4)$$

$$A_f = SCFM / V \quad (4.5)$$

$$W = A_f / L \quad (4.6)$$

2.2.8 Encontrar la potencia del ventilador por:

$$(HP_s) = \frac{0.0001573 V_1}{\epsilon_s} P_s \quad (4.7)$$

3. consideraciones económicas.

El equipo de enfriamiento para cualquier carga de proceso -- debería ser diseñado para que, en el tiempo de recuperación del costo de la planta, el fluido de proceso sea enfriado -- a un costo total mínimo (operación e inversión) con el fin de obtener máximas ganancias.

Al considerar el diseño de un equipo como un negocio, la -- situación que realmente existe de lo atractivo del riesgo -- del capital, es medido por el capital total incrementado -- retornable sobre un porcentaje comúnmente considerado atrac-- tivo.

La ganancia incremental, la cual es el origen para atraer -- nuevo capital ha sido denominado como "Venture Profit"

La siguiente expresión para "Venture Profit" es similar a -- la presentada por Happel and Aries:

$$V = P - i_m (I + I_w) \quad (4.8)$$

siendo:

$$P = R - eI - t (R - dI) \quad (4.9)$$

reajustando esta última ecuación:

$$P = R - eI - tR + t dI \quad (4.9a)$$

$$P = R (1 - t) - I (e - dt) \quad (4.9b)$$

$$\text{si } R = S - C \quad (4.10)$$

Substituyendo (4.10) en (4.9b)

$$P = (s-c) (1-t) - I (e - dt) \quad (4.11)$$

Substituyendo (4.11 en (4.8)

$$V = (s-c) (1-t) - I (e - dt) - i_m (I + I_w) \quad (4.12)$$

y puesto que:

$$S = I_w = 0$$

$$C_t = (s-c) (1-t) - I (e - dt) - i_m (I) \quad (4.13)$$

o sea

$$C_t = C (1-t) + I (e - dt) + i_m I \quad (4.13a)$$

donde:

P = Ganancias netas

R = Ganancias brutas

e = Depreciación del equipo para propósitos de contabilidad

I = Inversión en equipo

t = Impuestos federales y estatales

d = Depreciación del equipo fijado por el gobierno

s = Ventas

C = Costos de operación

I_w = Capital de trabajo

i_m = Porcentaje mínimo de ganancias sobre un capital invertido.

C_t = Costos totales (operación e inversión)

Para la evaluación de diseños, aquel que muestre los valores -- más altos de "Venture Profit" será el más atractivo.

3.1 Criterio.

El sistema de enfriamiento a instalar será aquel que maximize:

$$V = (S - C)(1 - t) - I(e - dt) - i_m(I + I_w) \quad (4.12)$$

pero $S = I_w = 0$.

lo cual significa que no habrá ventas y por lo cual la anterior ecuación al tomar la forma de la ecuación (6) viene a representar los costos totales, por lo que en este caso, el sistema más adecuado será aquel que proporcione:

$$C_{TMIN} = C(1 - t) + I(e + i_m - dt) \quad (4.14)$$

Estos costos traídos a valor presente proporcionarán el capital necesario para operar el sistema durante el período de tiempo estimado, para lo cual el concepto del "Venture Worth" será considerado.

El "Venture Worth" representa la suma de los valores de "Venture Profit" anuales, pero considerando un interés compuesto durante el tiempo de duración considerado para el equipo.

La ecuación en forma integral para el "Venture Worth" queda:

$$W = \int_0^{\phi} C_t e^{-i\phi} d\phi \quad (4.15)$$

Por lo que el sistema de enfriamiento a instalar será aquel -

que proporcione:

W min.

donde:

C_t = Costo totales (operación e inversión)

θ = Tiempo en años

i = Interés

3.2 Inversión fija

Considerando cambiadores de calor de tubos aletados de las siguientes características:

OD = 1"

ALETA = 0.5"

L = 24'

9 ALETAS/IN

Material del tubo = C.S.

Material de la aleta = Aluminio

Se ha encontrado de los estudios de la "Hudson Corporation"--- que el costo del cambiador de aire puede ser representado por una ecuación de la siguiente forma:

$$I = a n^b A^c \quad (4.16)$$

donde:

n = número de hileras de tubos

A = Area de transferencia de calor.

a, b, c constantes que dependen de los costos del equipo, tipo y características de aleta y tubo, etc.

Para las características anteriormente dadas de aletas y --
tubos y para condiciones de E.E.U.U. La ecuación queda de
la siguiente forma:

$$I = 529 \text{ h} \left(- 0.2151 A - 0.1795 \right) \$/\text{ft}^2 \quad (4.17)$$

Es necesario agregarle a está ecuación los costos directos -
e indirectos que se derivan de un nuevo equipo, para lo cual
los siguientes factores fueron obtenidos de un cuidadoso es-
tudio hecho por H.C. Bauman:

Directos:

Mano de Obra	0.2
Cimientos	0.1
Eléctrico	0.12
Pintura	0.08
Instrumentación	0.1
Estructuras	$\frac{0.03}{0.63}$

Indirectos:

Overhead:	0.25
Ingeniería:	0.13
Contingencias:	$\frac{0.05}{0.43}$

$$L = (1.63) (1.43) = 2.33$$

$$I = 1230 \text{ h} \left(- 0.1795 A - 0.1795 \right) \$/\text{ft}^2 \quad (4.18)$$

$$I = 1230 \text{ h} \left(- 0.2151 A - 0.1795 \right) \$/\text{ft}^2$$

$$I = 1230 \text{ h} \left(- 0.2151 A - 0.8205 \right) \$/\text{ft}^2 \quad (4.18a)$$

De las cotizaciones dadas por una Compañía que fabrica cambiadores de aire en México, el valor por el que hay que multiplicar la ecuación es 1.8 para obtener el valor de la undad en el País. El factor por el que hay que multiplicar la -- ecuación cuando se usan tubos de Admiralty es 1.2 por lo -- que la ecuación quedará:

$$I = 2660 \cdot 1.8 \cdot 0.2151 \cdot A^{0.8205} \cdot \$ \quad (4.19)$$

3.3 Costos de Operación

Los costos de operación incluyen los de energía suministrada al ventilador, y los de mantenimiento.

3.3.1 Costos de energía.- La energía necesaria para enfriamiento con aire es consumida en el medio motriz del ventilador. La mayoría de los ventiladores son accionados por motores eléctricos o turbinas. Las turbinas son generalmente accionadas con vapor o gases calientes. El costo anual de energía para la operación de un cambiador de calor enfriado por aire puede ser representado por:

$$\text{Costo de Energía} = \left(F_p \right) \left(P_a \right) \left(A \right) \quad (4.20)$$

Donde F_p es el costo de la energía eléctrica en el lugar donde será colocado el enfriador, este valor puede variar dependiendo del lugar donde va a ser colocado el equipo.

P_a es el factor de requisitos de energía, el cual se basa - en el área de superficie del cambiador, si un sistema de tiro forzado es construido, la potencia consumida por el ventilador es una función de la temperatura de bulbo seco ambiente. A temperatura y a velocidad del aire constante a través del haz de tubos aletados, P_a es aproximadamente constante. P_a está dado en energía/área.

3.3.2 Costos de mantenimiento.- Los costos de mantenimiento para los enfriadores de aire en México son considerados - aproximadamente como el 3% de los costos de inversión.

Estos costos incluyen: Limpieza interna y externa de los tubos, aceite a la caja de engranes, cojinetes del ventilador.

$$\text{Costo de mantenimiento} = 0.03 \ I \quad (4.21)$$

2.4 Ecuación Final

$$C_t = \left(F_p \ P_a \ A + 0.03 \ I \right) \left(1-t \right) + I \left(e + i_m - dt \right) \quad (4.22)$$

Donde

$$I = 2660 \ n^{-0.2151} \ A^{0.8205} \ \$ \quad (4.19)$$

3.5 "Venture Worth"

$$W = \int_0^{\theta} C_T e^{-i \theta} d \theta \quad (4.15)$$

$$W = \int_0^1 \left[\left(F_p P_a A + 0.03 I \right) (1-t) + I \right. \\ \left. \left(e + i m - dt \right) \right] e^{-i\theta} d\theta \quad (4.23)$$

OPTIMIZACION DE UN CASO ESPECIFICO

OPTIMIZACION DE UN CASO ESPECIFICO

El diseño óptimo será aquel que proporcione los costos totales de operación e inversión mínimos.

Es necesario como 1er. paso transformar las ecuaciones de costos al problema en particular. A continuación se substituyen los valores correspondientes a dichas ecuaciones:

1. Costos de operación

1.1 Costos de energía

$$C_{op} = F_p P_a A$$

Donde $F_p = 0.12 \text{ \$/Kw} - \text{Hr}$

El anterior dato fue aportado por Pemex

y:

$$P_a = \frac{8400 \text{ Hrs}}{1 \text{ año}} \cdot \frac{0.746 \text{ Kw} - \text{Hr.}}{1 \text{ HP}} \cdot \frac{\text{HP}_s}{A}$$

$$P_a = 6260 \frac{\text{HP}_s}{A} \cdot \frac{\text{Kw Hr}}{\text{ft}^2 \text{ año}}$$

Por lo que:

$$C_{op} = 0.12 P_a A \text{ \$ /año}$$

1.2 Costos de mantenimiento

$$C_m = 0.03 I_f$$

El factor 0.03 es corrientemente usado en la Industria.

2. Costos de inversión

$$I_f = \frac{469}{2600} n - 0.2151 A \quad 0.8205$$

3. Ecuación final.

$$\text{si } t = 0.3$$

$$d = 0.1$$

$$i_m = 0.15$$

$$\theta = 10 \text{ años}$$

$$e = \frac{i_m}{e^{i_m \theta} - 1}$$

$$e = \frac{0.15}{e^{(0.15)(10)} - 1} = \frac{0.15}{3.47} = 0.0429$$

$$C_T = C_{Op} (1-t) + I_f (e + i_m - d t)$$

$$C_T = C_{Op} (1-0.3) + I_f (0.0429 + 0.15 - (0.1)(0.3))$$

$$C_T = 0.7C_{Op} + 0.1629 I_f$$

$$C_T = 0.7 (0.12 P_a A + 0.03 I_f) + 0.1629 I_f$$

$$C_T = 0.084 P_a A + 0.1839 I_f$$

$$C_T = 0.084 P_a A + 0.1839 (2660) n - 0.2151 \quad 0.8205 A$$

$$C_T = 0.084 P_a A + 489 n - 0.2151 \quad 0.8205 A$$

4. "Venture Worth"

$$\text{Para } i = 0.15$$

$$\theta = 10 \text{ años}$$

$$W = \int_0^{\theta} C_t e^{-i \theta} d \theta$$

$$W = - \left[\frac{C_t}{i} e^{-i \theta} \right]_0^{\theta} = - \frac{C_t}{i} (e^{-10i} - e^0)$$

$$W = \frac{C_t}{0.15} \left[1 - e^{-1.5} \right] = \frac{C_t}{0.15} (1 - 0.2235)$$

$$W = 5.175 C_T$$

5. Enunciado y resolución

Se requiere enfriar una corriente de aceite cíclico pesado, la cual presenta las siguientes características:

$$T_1 = 400^\circ\text{F}$$

$$T_2 = 150^\circ\text{F}$$

$$W = 186,000 \text{ lb/hr}$$

$$Q = 24,550,000 \text{ Btu/hr}$$

$$i_p = 0.002 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$$

La temperatura de diseño de bulbo seco del aire es:

$$t_1 = 92^\circ\text{F}$$

Las características del tubo aletado de acuerdo con la Tabla I son:

$$\text{OD} = 1"$$

$$\text{ID} = 0.826"$$

$$14 \text{ BWG}$$

$$\text{Longitud} = 24 \text{ ft}$$

$$m_{\text{tubo}} = 0.0003 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$$

$$m_{\text{aleta}} = 0.00059 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu (aluminio)}$$

$$\text{Aleta: Aluminio OD} = 2"$$

$$\text{No. Aletas/In} = 11$$

$$\frac{h_t}{A_o} = 17.5$$

$$\frac{V_m}{V} = 2.22$$

$$\text{PITCH} = 23/32''$$

$$\frac{S}{l}^{0.296} = 0.576$$

El coeficiente del lado de proceso estimado es:

$$h_p = 140 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$\frac{1}{h_p} = \frac{1}{140} = 0.00715 \frac{\text{hr ft}^2 \text{ } ^\circ\text{F}}{\text{Btu}}$$

La resistencia de incrustación del lado del aire puede ser despreciable. La suma de resistencias sin contar la del aire queda:

$$\begin{aligned} r &= \frac{1}{h_p} + r_{m_{\text{tubo}}} + r_{m_{\text{aleta}}} + r_{i_p} \\ &= 0.00715 + 0.0003 + 0.00059 + 0.002 \\ &= 0.01004 \text{ hr ft}^2 \text{ } ^\circ\text{F/Btu} \end{aligned}$$

Usando la ecuación (3.3)

$$h_a = 0.105 \left(\frac{V_m}{d_r} \right)^{0.718} \left(\frac{S}{l} \right)^{0.296} \left(\frac{A_t}{A} \right) F_r$$

Para el tipo de tiro forzado escogido $F_r = 1$ para cualquier caso.

Las características del tubo aletado son substituídas en la anterior ecuación:

$$\begin{aligned} h_a &= 0.105 \left(\frac{V_m}{l} \right)^{0.718} (0.575) (17.5) \\ h_a &= 1.0584 V_m^{0.718} \end{aligned}$$

$$\text{Donde } V_m = 2.22 V$$

Δt_a

100

90

70

60

50

30

20

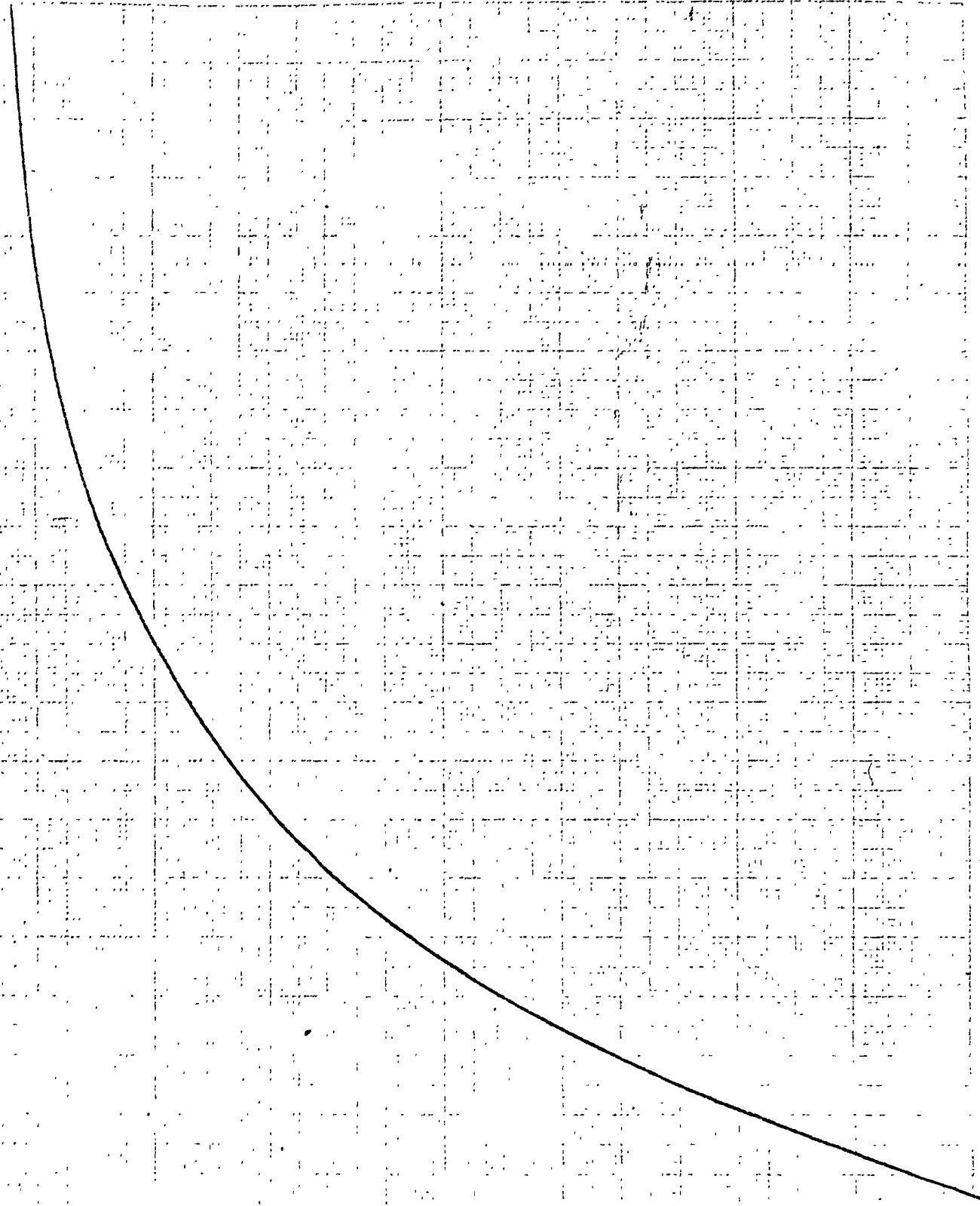
10

1.0

2.0

3.0

$\frac{LMTDF_t}{\Delta t_a}$



Por lo que para el rango de velocidades recomendadas:

V	V _m	1/h _a	1/U	U
300	660	8.77×10^{-3}	0.01881	53.16
500	1110	6.25×10^{-3}	0.01629	61.38
700	1550	4.95×10^{-3}	0.01499	66.71
900	1999	4.02×10^{-3}	0.01406	71.12

Los pasos que siguen a continuación son los dados en el capítulo II.

Para 4 hileras:

$$\Delta p_a = 2.34 \times 10^{-6} (V_m)^{1.725}$$

$$h_a = 1.0584 (V_m)^{0.718}$$

$$\frac{T_m}{\Delta t_a} = 0.1805 \frac{V}{U}$$

V	T _m /Δt _a	Δt _a	t ₂	T _m F _t	T _m F _t U
300	1.07	67.3	159.3	72	3640
500	1.556	51.5	143.5	80.1	4650
700	2.15	39.0	131.0	83.9	5260
900	2.435	34.7	126.7	84.4	5630

V	A.	W _a	S.C.F.M.	A _f	W
300	6740	1.54×10^6	3.425×10^5	1140	47.5
500	5280	1.987×10^6	4.42×10^5	885	36.9
700	4660	2.63×10^6	5.85×10^5	836	34.9
900	4360	2.95×10^6	6.56×10^5	730	30.4

V	ΔP_a	ACFM	ΔP_a ACFM
300	0.172	4.14×10^5	0.712×10^5
500	0.435	5.35×10^5	2.325×10^5
700	0.790	7.08×10^5	5.59×10^5
900	1.150	7.94×10^5	9.11×10^5

V	HP_s ($f = 0.5$)	HP_s ($f = 0.6$)	HP_s ($f = 0.7$)
300	22.43	18.7	16.0
500	73.2	61.0	52.3
700	176	146.8	125.7
900	287	239.5	205.0

V	HP_s/A ($E_f = 0.5$)	HP_s/A ($E_f = 0.6$)	HP_s/A ($E_f = 0.7$)
300	3.34×10^{-3}	2.78×10^{-3}	2.38×10^{-3}
500	1.386×10^{-2}	1.155×10^{-2}	9.9×10^{-3}
700	3.77×10^{-2}	3.15×10^{-2}	2.695×10^{-2}
900	6.57×10^{-2}	5.49×10^{-2}	4.69×10^{-2}

V	P_a ($E_f = 0.5$)	P_a ($E_f = 0.6$)	P_a ($E_f = 0.7$)
300	20.9	17.4	14.9
500	86.8	72.4	62.0
700	236.0	197.3	168.8
900	411.0	344.0	293.5

V	C_{Op} ($E_f = 0.5$)	C_{Op} ($E_f = 0.6$)	C_{Op} ($E_f = 0.7$)
300	11,850	9850	8440
500	38,500	32,100	27,500
700	92,500	77,400	66,100
900	151,000	126,200	107,800

$$C_T = C_{OP} + C_{EQ}$$

V	C_{EQ}	$E_f = 0.5$	$E_f = 0.6$	$E_f = 0.7$
300	450,000	461,850	459,850	458,440
500	370,000	408,500	402,100	397,500
700	335,000	427,500	412,400	401,100
900	316,000	467,000	442,200	423,800

Para 6 hileras

$$\Delta t_a = 3.52 \times 10^{-6} (V_m)^{1.725}$$

$$T_m = 1.0584 (V_m)^{0.713}$$

$$\frac{T_m}{\Delta t_a} = 0.1205 \frac{V}{L}$$

V	$T_m/\Delta t_a$	Δt_a	U_2	$T_m F_t$	$T_m F_t U$
300	0.715	88	173	61.5	3115
500	1.04	67.8	159.8	71.4	4140
700	1.345	57.5	149.5	77.4	4850
900	1.625	49.2	141.2	80.0	5340

V	A	W_a	SCFM	A_F
300	7880	1.19×10^5	2.645×10^5	849
500	5940	1.51×10^5	3.36×10^5	672
700	5060	1.78×10^5	3.96×10^5	565
900	4600	2.08×10^5	4.63×10^5	515

V	W	ACFM	ΔP_a	ΔP_a ACFM
300	35.4	3.2×10^5	0.260	0.832×10^5
500	28.0	4.06×10^5	0.630	2.56×10^5
700	22.5	4.79×10^5	1.190	4.71×10^5
900	21.4	5.60×10^5	1.740	8.05×10^5

V	HP _S (E _f = 0.5)	HP _S (E _f = 0.6)	HP _S (E _f = 0.7)
300	26.2	21.9	18.7
500	80.6	67.3	57.6
700	148.3	124.0	106.0
900	263.5	212.0	181.0

V	HP _S /A (E _f = 0.5)	HP _S /A (E _f = 0.6)	HP _S /A (E _f = 0.7)
300	3.325×10^{-3}	2.78×10^{-2}	2.375×10^{-3}
500	1.36×10^{-2}	1.13×10^{-2}	5.7×10^{-2}
700	2.93×10^{-2}	2.45×10^{-2}	2.09×10^{-2}
900	5.72×10^{-2}	4.61×10^{-2}	3.93×10^{-2}

V	P _a (E _f = 0.5)	P _a (E _f = 0.6)	P _a (E _f = 0.7)
300	20.85	17.4	14.86
500	85.4	70.9	60.8
700	183.5	153.5	131.0
900	358.0	289.0	246.0

V	C _{Op} (E _f = 0.5)	C _{Op} (E _f = 0.6)	C _{Op} (E _f = 0.7)
300	13,800	11,520	9,850
500	42,500	35,350	30,300
700	78,200	65,400	55,800
900	138,200	111,600	95,000

$$C_T = C_{Op} + C_{EQ}$$

V	C_{EQ}	$E_f = 0.5$	$E_f = 0.6$	$E_f = 0.7$
300	430,000	447,800	445,520	443,850
500	340,500	384,000	376,850	371,800
700	302,000	360,200	367,400	257,800
900	277,000	415,200	388,600	372,000

Para 8 hileras

$$\Delta p_a = 4.69 \times 10^{-6} (V_m)^{1.725}$$

$$h_a = 1.0584 (V_m)^{0.718}$$

$$\frac{T_m}{\Delta t_a} = 0.092 \frac{V}{U}$$

V	Δp_a	$T_m / \Delta t_a$	λ_{t_a}	t_2	$T_m F_t$
300	0.346	0.554			
500	0.821	0.794	82.5	174.5	65.5
700	1.580	1.025	69.5	161.5	71.2
900	2.315	1.24	60.5	152.5	75.0

V	U	$T_m F_t$	A	W_a	SCFM
300					
500		3800	6450	1.24×10^6	2.76×10^5
700		4460	5500	1.472×10^6	3.28×10^5
900		5000	4900	1.693×10^6	3.76×10^5

V	f	W	ACFM	Δp_a ACFM
300				
500	552	23	3.34×10^5	3.745×10^5
700	469	19.5	3.97×10^5	6.27×10^5
900	418	17.4	4.55×10^5	10.52×10^5

V	$HP_S (E_f = 0.5)$	$HP_S (E_f = 0.6)$	$HP_S (E_f = 0.7)$
300			
500	86.5	72.1	61.8
700	197.6	164.7	141
900	332	277	237

V	$HP_S/A (E_f = 0.5)$	$HP_S/A (E_f = 0.6)$	$HP_S/A (E_f = 0.7)$
300			
500	1.34×10^{-2}	1.12×10^{-2}	9.58×10^{-3}
700	3.58×10^{-2}	3.0×10^{-2}	2.56×10^{-2}
900	6.78×10^{-2}	5.65×10^{-2}	4.84×10^{-2}

V	$P_a (E_f = 0.5)$	$P_a (E_f = 0.6)$	$P_a (E_f = 0.7)$
300			
500	83.9	70.0	60.0
700	224.5	188.0	161
900	425.0	354	301.5

V	$C_{Op} (E_f = 0.5)$	$C_{Op} (E_f = 0.6)$	$C_{Op} (E_f = 0.7)$
300			
500	45,400	37,900	32,500
700	103,500	87,000	74,500
900	175,000	145,600	124,500

$$C_T = C_{Op} + C_{EQ}$$

V	C_{Op}	$E_f = 0.5$	$E_f = 0.6$	$E_f = 0.7$
300				
500	345,000	390,400	382,900	377,500
700	301,000	404,500	388,000	375,500
900	276,000	451,000	422,600	400,500

De las curvas de costos totales, se encuentra que el diseño óptimo para este caso particular, es aquel que tiene 6 hileras de tubos y una velocidad del aire de 660 ft/min. con un costo total mínimo de \$ 356,000.

Es necesario de cualquier forma, hacer un reajuste del cambiador, - para satisfacer las condiciones del lado de proceso, las cuales -- fueron parcialmente tomadas en cuenta en la optimización del mismo. Los pasos a seguir son los dados en la parte 2 del capítulo III.

Coeficiente de Transferencia

1. $\Delta T = 250^\circ F$

2. $\frac{\Delta T_c}{\Delta T_h} = 0.233$

3. Con ΔT y $\Delta PI = 22$ $K_c = 0.94$

4. Con K_c y $\Delta T_c / \Delta T_h$ $F_c = 0.335$

5. $T_c = 234^\circ F$

6. Con T_c μ_p , K_p , C_p

7. No. de tubos = 864 tubos

C_T

4 HILERAS

600,000

500,000

400,000

300,000

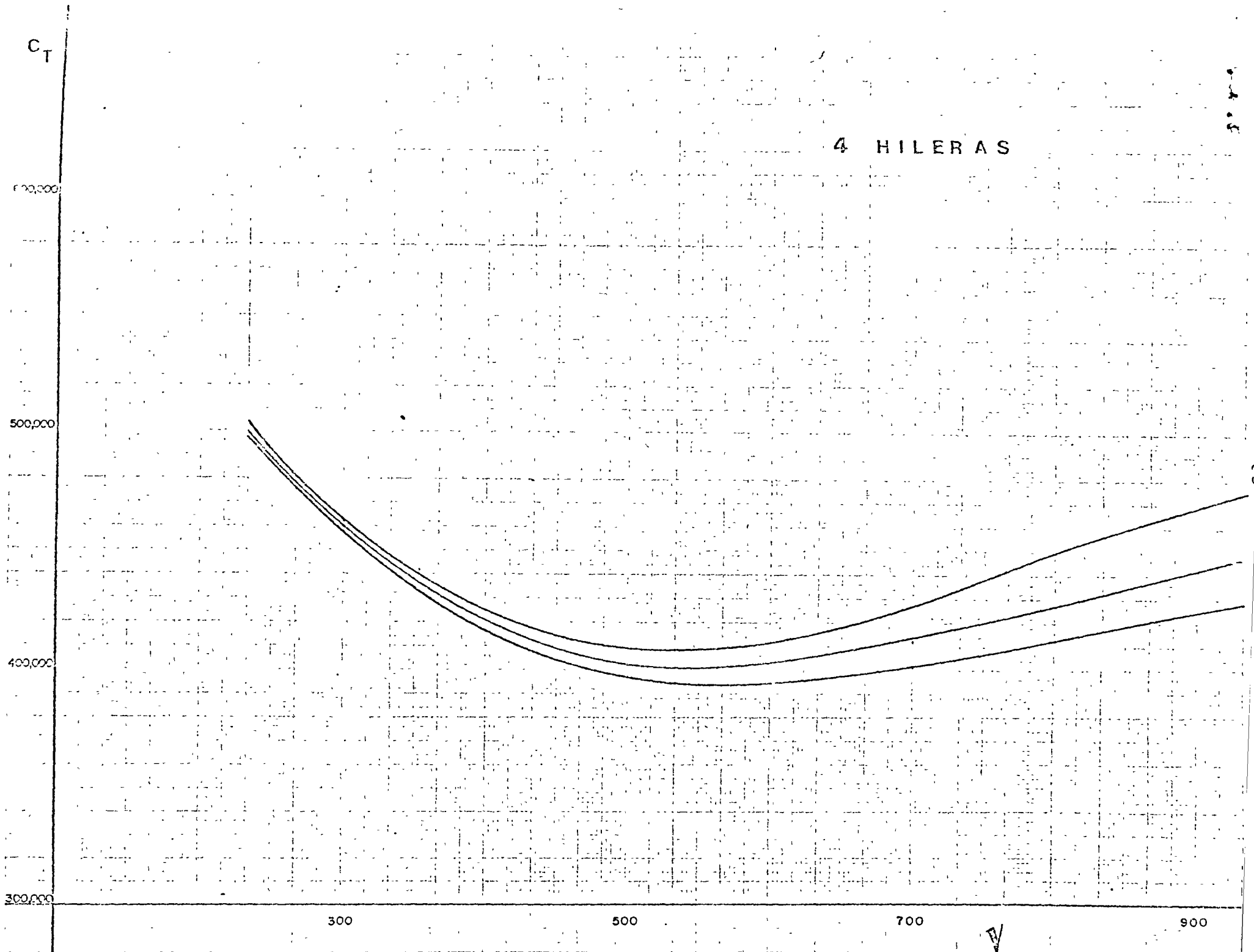
300

500

700

900

V



C_T

6 HILERAS

600,000

500,000

400,000

300,000

300

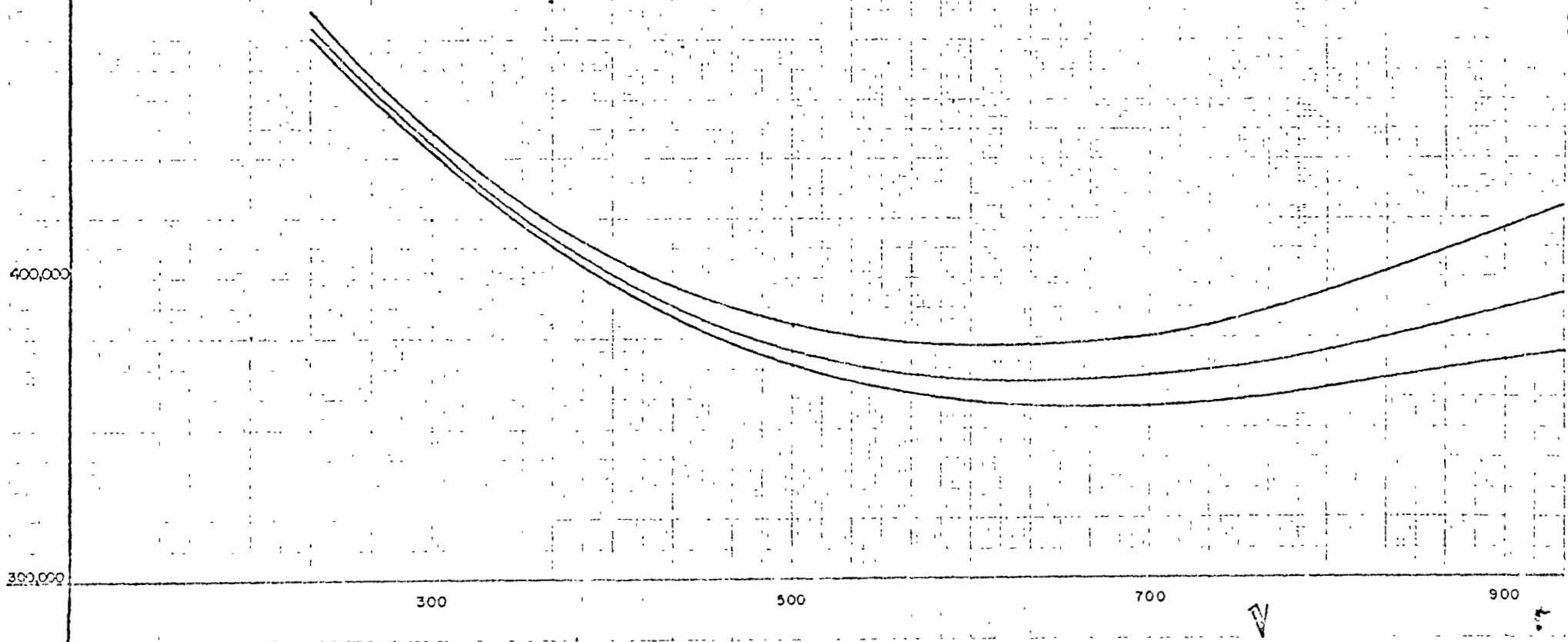
500

700

900

V

98 -



C_T

8 HILERAS

600,000

500,000

400,000

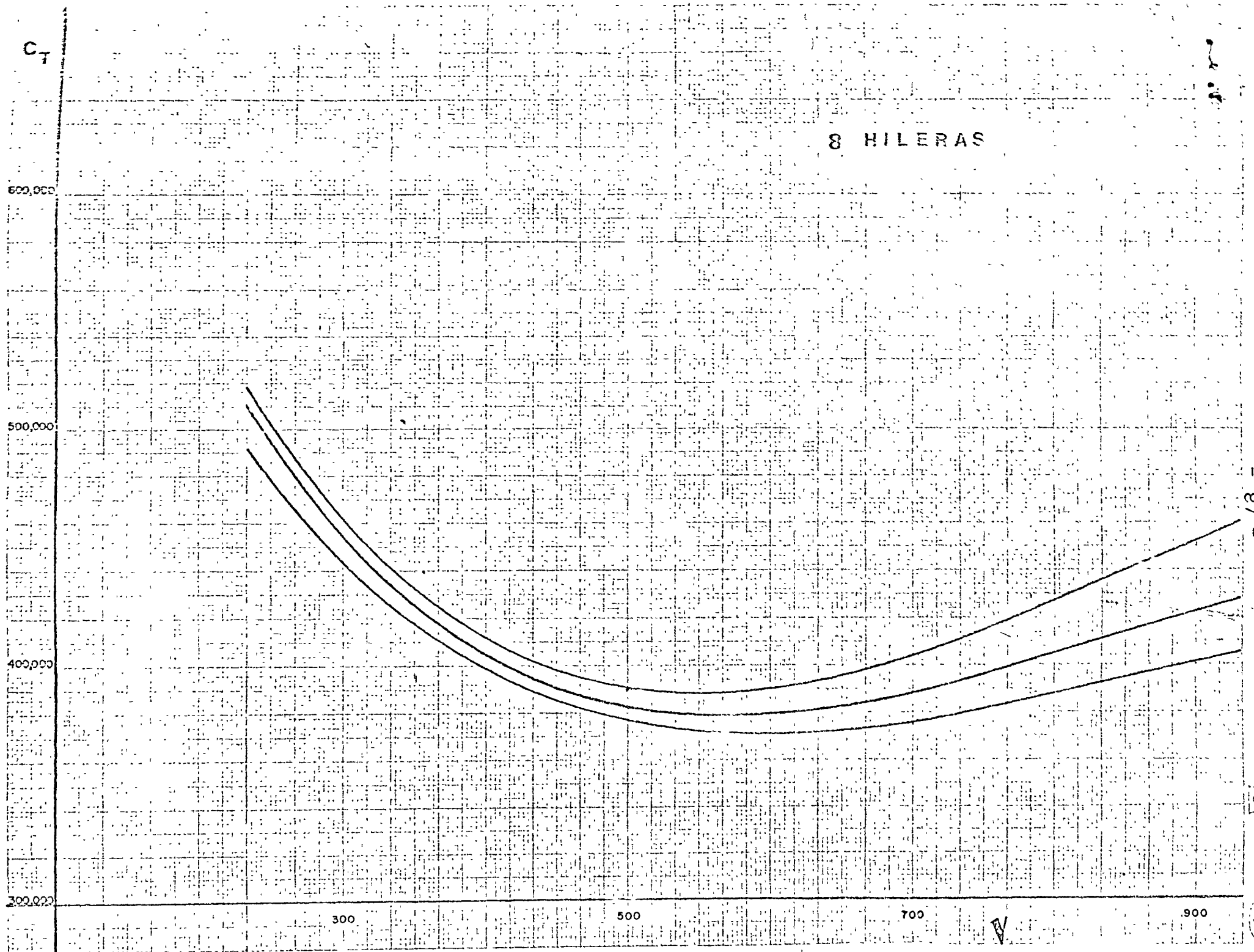
300,000

300

500

700

900



Si el número de pasos es 6

8. Masa velocidad = 340,000 lb/hr ft²
9. $R_c = 10,700$
10. $h_i/\phi = 199$ Btu/hr ft² °F
11. $h_{io} = 150$ Btu/hr ft² °F

El valor del coeficiente de transferencia de calor se encuentra más o menos de acuerdo con el estimado.

Caída de presión

1. $f = 0.00023$
2. $P_t = 10.4$ psi
3. $P_r = 0.446$ psi
4. $\Delta P_t = 10.886$ psi .

El valor de la caída de presión total se encuentra de --- acuerdo con el comunmente tomado como permisible.

CONCLUS IONES

VI. CONCLUSIONES

El sistema de enfriamiento con aire está tomando un lugar importantísimo entre los sistemas de enfriamiento a temperatura ambiente.

Los dos factores más importantes por lo que el aire está desplazando a los otros sistemas de enfriamiento son:

La contaminación ambiental

La escasez de agua apropiada.

Es por lo tanto necesario que el aire sea usado lo más posible como medio de enfriamiento para preservar los recursos naturales y el medio ambiente en un nivel que no sea alarmante para la vida del hombre.

Las variables que generalmente deben ser tomadas en cuenta para la optimización de un sistema de enfriamiento con aire son:

La velocidad del aire, el número de hileras de tubos y el número de pasos por los tubos.

En la optimización desarrollada en el capítulo anterior, la última variable no fue tomada desde un principio, con el fin de simplificar los cálculos, pero en el caso en que una computadora sea usada como medio evaluativo, es recomendable el análisis de las tres variables juntas, Para encontrar el equipo mas económico para una determinada función.

N O M E N C L A T U R A

VII NOMENCLATURA

A_f	Area fasal del cambiador. ft^2
A_o	Superficie total de tubo liso., ft^2
A_t	Area total expuesta de aletas y tubos, ft^2
a_f	Area de flujo total, ft^2
a'_f	Area de flujo por tubo, ft^2
C_a	Calor específico del aire, $Btu/lb^\circ F$
C_p	Calor específico de lado de proceso, $Btu/lb^\circ F$
D	Diámetro del tubo, ft
d_r	Diámetro medio de la aleta, pulgadas.
F_c	Factor para obtener temperatura calórica
F_r	Factor de corrección por el número de hileras en tiro inducido
F_s	Factor de seguridad para la caída de presión del lado del aire
F_t	Factor de corrección sobre el L.M.T.D. para flujo <u>cr</u> zado.
	Factor de fricción
G_m	Masa velocidad máxima del aire a una sección mínima - transversal $lb/seg. ft^2$
G_t	Masa velocidad del lado de proceso, $lb/hr ft^2$
h_a	Coefficiente de transferencia de calor del lado del -- aire. $Btu lb ft^2 \circ F$
h_1	Coefficiente de transferencia de calor del lado de <u>pro</u>

	ceso sin corregir $\text{Btu/hr ft}^2 \text{ } ^\circ\text{F}$
h_o	Coeficiente de transferencia de calor del lado de --- proceso corregido. $\text{Btu/hr ft}^2 \text{ } ^\circ\text{F}$
h_t	Coeficiente de transferencia de calor del lado del -- aire referido a la superficie total externa Btu/hr ft^2 $^\circ\text{F}$
J_h	Factor de la gráfica 13
K_a	Conductividad térmica del aire, $\text{Btu-ft/hr ft}^2 \text{ } ^\circ\text{F}$
K_c	Factor para obtener F_c
K_p	Conductividad térmica de proceso, $\text{Btu-ft/hr ft}^2 \text{ } ^\circ\text{F}$
l	Altura de la aleta, pulgadas
L	Longitud del tubo, ft.
LMTD	Diferencia media logarítmica de temperaturas, $^\circ\text{F}$
n	Número de pasos en los tubos
R	Número de hileras de tubos
S.Gr.	Gravedad específica del fluido de proceso
s	Distancia entre aletas, pulgadas
T_c	Temperatura calórica, $^\circ\text{F}$
t_1	Temperatura de entrada del aire, $^\circ\text{F}$
t_2	Temperatura de salida del aire, $^\circ\text{F}$
T_1	Temperatura de entrada de proceso, $^\circ\text{F}$
T_2	Temperatura de salida de proceso, $^\circ\text{F}$
$\Delta T =$	$T_1 - T_2$

$$\Delta T_c = t_2 - t_1$$

$$\Delta T_h = t_1 - t_2$$

t_w Temperatura de la pared, °F

t_a Temperatura media del aire, °F

V_m Velocidad máxima del aire ft/min.

U_c Coeficiente total de transferencia de calor limpio

U_d Coeficiente total de transferencia de calor sucio

W Ancho del haz de cubos

w Gasto del fluido de proceso

Δp_a Caída de presión del lado del aire, pulgadas en agua

Δp_t Caída de presión total del fluido de proceso, psi.

p_t Caída de presión en la parte recta de los tubos, --
psi.

p_r Caída de presión en los retornos de los tubos, psi.

v Velocidad del fluido de proceso ft/min.

v_v Flujo volumétrico del fluido de proceso ft³/min.

μ_a Viscosidad del aire lb/hr ft.

μ_p Viscosidad del fluido de proceso lb/hr ft.

ρ_a Densidad del aire lb/ft³

ρ_p Densidad del fluido de proceso lb/ft³

β_t Relación de viscosidades

B I B L I O G R A F I A

VIII.- BIBLIOGRAFIA

1. Ludwig E. Ernest "Applied Process Design for Chemical and Petrochemical Plants"
Gulf Publishing Company, Vol. 3, Capítulos 10 y 12, 1965.
2. Kern Q. Donald "Process Heat Transfer"
Mc Graw - Hill, Capítulo 16, 1950.
3. Happel John "Chemical Process Economics"
John Wiley & sons, Capítulos 1 y 2, 1958.
4. Peters & Timmerhaus "Plant Design and Economics for Chemical Engineers"
Mc Graw-Hill, Capítulos 4, 5 y 9, 1968.
5. Cook M. Edward "Comparison of Equipment for Removing Heat from Process - Streams"
Chemical Engineering, mayo 25, 1964.

7. Cook M. Edward "Rating Methods for Selection of Air-Cooled Heat Exchangers" Chemical Engineering, Agosto 3, 1964.
8. C.L. Williams, J.R.
and R.D. Dameron "Which Cools Cheaper:
Water or Air" Hydrocarbon Processing, Vol. -
44, No. 2, Febrero 1965.
9. A.Y. Gunter, and K.V.
Shipes "Hot air Recirculation by air
Coolers" Chemical Engineering Progress
Vol. 68, No. 2, Febrero 1972.
10. C.H. Homer, J.R., and
S. Hopp "Air-Cooled Heat Exchangers" The Petroleum Engineer, abril,
1964.
11. J.C. Chase and H.E.
Degler. "Economics of the Air-Cooled
Heat Exchanger " The Petroleum Engineer, Enero,
1953.

6389/
14424

I

Shell-Side Characteristics of Shell-and-Tube Heat Exchangers

A Simplified Rating System for Commercial Heat Exchangers

By TOWNSEND TINKER,¹ BUFFALO, N. Y.

A simplified rating system for commercial shell-and-tube heat exchangers is developed based on the fluid-flow-pattern concepts derived by the author for the General Discussion of Heat Transfer, London, England, 1951. Recommended heat-transfer and friction characteristics for practical design purposes are presented which are based on previously presented research data and certain findings of the ASME—University of Delaware Heat Exchanger Research Project. Where possible, without serious sacrifice in accuracy, certain resistance-factor approximations and assumptions have been introduced in the interest of practical simplicity for the derivation of formulas of simple form for evaluating the fluid-flow fraction through the crossflow area of the tube bundle. Simple tabulations of heat-transfer and pressure-drop characteristic "rating numbers" may be readily developed from these formulas for various styles or series of heat exchangers having consistent design proportions. These rating numbers permit a rapid rating of the heat-exchanger shell-side performance. The system is responsive to the major effects of unit size, tube size, tube pitch and flow orientation, baffle spacing and cut, and the principal leakage and bypass clearances of the heat-exchanger design. The system developed is accurate well within practical commercial requirements, and is very flexible so that as more accurate resistance factors are developed by research for specific cases, they may be employed in the system for more precise results when desired.

NOMENCLATURE

The following nomenclature is used in the paper:

- A_c = cross-flow area within limits of tube bundle, sq ft = $c_b b D_1$
- A_w = area of one baffle window
- A_m = geometric mean of A_c and A_w = $[(A_c)^{1/2}(A_w)^{1/2}]$
- B_1 = heat-transfer group for flow across tubes

$$= \frac{h_o d_2}{16.1 K} \left(\frac{CZ}{K} \right)^{-1/4} \phi^{1/4}$$
- C = specific heat
- c_b = crossflow area constant
- c_f = friction proportionality factor, $f_s = c_f / \text{Re}_s^{0.5}$
- c_r = constant for crossflow tube rows traversed
- c_u = bypass clearance constant = D_1/D_2

- c_{1b} = baffle-hole clearance constant = $(d_1 - d_2)/d_2$
- c_{1e} = baffle-edge clearance constant = $(D_1 - D_2)/D_1$
- D_1 = shell diameter, in.
- D_2 = baffle diameter, in.
- D_3 = outer-tube limit, bundle diameter, in.
- d_1 = tube-hole diameter in baffle, in.
- d_2 = tube OD, in.
- E_s = shell-side effectiveness
- F_s = fraction of total fluid flowing through A_c of a clean unit
- F_p = modified crossflow friction for service unit pressure-drop calculation
- f_w = friction factor for baffle-window zone
- f_r = friction factor for crossflow through tube bank
- H = baffle cut, in.
- h_o = outside heat-transfer coefficient for entire tube bundle = $h_o E_s$, (Btu)/(hr sq ft deg F)
- h_b = outside heat-transfer coefficient for baffled region of tube bundle, (Btu)/(hr sq ft deg F)
- K = thermal conductivity, (Btu)/(hr sq ft deg F)
- l_1 = total tube length, in.
- l_2 = baffled length of tubes, in.
- l_b = baffle spacing, in.
- M = A_c multiplier for Re_s
- M_w = A_w multiplier to obtain geometric mean = A_c and A_w
- m = $M_w / \sqrt{c_u}$
- N_b = number of baffle spaces
- N_h = rating number for heat-transfer calculation
- N_p = rating number for pressure-loss calculation
- P = tube pitch, in.
- ΔP = over-all shell-side pressure loss, psi
- ΔP_w = baffle-window pressure loss, psi
- ΔP_s = crossflow pressure loss, psi
- Re_s = Reynolds number for heat transfer

$$= \frac{0.03314 l_2 F_s (\text{lb/hr})}{Z M A_c}$$
- Re_p = modified Reynolds number for pressure drop

$$= \frac{0.03314 F_p (\text{lb/hr})}{Z A_c}$$
- S = exchanger size ratio = $\frac{D_1}{P}$
- Y = pressure-loss factor for baffle window = $y'(c_u)^{0.5}$
- y = factor for baffle-window proportions
- x = ratio $\Delta P_w / \Delta P_s$
- $(1+x) = \left(1 + \frac{Y}{S}\right)$ = factor for relative crossflow and window pressure losses
- Z = absolute viscosity of fluid at mean fluid temperature, centipoises

¹ Vice-President, Engineering, Ross Heat Exchanger Division, American Radiator & Standard Sanitary Corporation, Mem. ASME. Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., November 25-30, 1956 of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters August 21, 1956. Paper No. 56-A-123.

- Z_μ = absolute viscosity of fluid at mean tube-wall temperature, centipoises
- $\phi = (Z/Z_\mu)^{0.14}$
- ρ = density, pcf
- Q_{AR} = relative fluid flow associated with baffle-hole leakage
- Q_{BR} = relative fluid flow through tube bundle A_b
- Q_{CR} = relative fluid flow bypassing between bundle and shell
- Q_{LR} = relative fluid flow leaking between baffle and shell
- c = approximate portion of theoretical tube count in tube layout
- c_1 = segmental portion cut from circle to make baffle
- c_2 = portion of circle comprising baffle
- $c_3 = 1 - c_2$
- c_4 = portion of tube bundle diameter representing mean bundle width between baffle windows
- c_5 = portion of steel circumference adjacent to edge of one baffle
- c_6 = velocity heads loss in flow through baffle-tube holes
- c_7 = velocity heads loss for flow across one row of square pitch tubes
- $c_8 = 1.0 f_0$ (square)
- c_9 = velocity heads loss for flow across one row of triangular pitch tubes
- $c_{10} = 1.0 f_0$ (staggered)
- c_{11} = portion of bundle diameter traversed in flowing between successive baffle windows
- c_{12} = friction-factor multiplier for high-velocity C stream
- $$f_r \left[\frac{\alpha \text{ Re} = \left(\frac{V_{rh}}{V_{rb}} \right) \text{ Re}_R}{f_r \text{ at } \text{Re}_R} \right]$$
- c_{13} = velocity heads loss in flow between baffle and shell
- c_{14} = velocity heads loss in flow through baffle window
- Δnoz = loss through shell entrance and exit-nozzle connections, psi

INTRODUCTION

The proposed system for rating commercial heat exchangers is based on available research data for pure cross-flow tube banks and shell-and-tube heat exchangers. The use of pure cross-flow data is recommended with certain modifications as dictated by test experience on the general behavior of actual commercial units.

Expressions for the heat-transfer coefficients and pressure drops are written in a simple form based on certain primary premises that represent the behavior characteristics as accurately as possible without the undue complication that would be required to develop a rigorous Re and flow characteristic for fluid streams through each leakage route and bypass route for fluids of all possible viscosity ranges that might be passed through the exchanger shell.

These primary premises are as follows:

1. Heat Transfer

1.1 Flow Distribution. The relative fluid-flow quantities through the various shell-side flow routes are, for heat-transfer calculations, assumed to be substantially as indicated elsewhere in Table IV. These flow proportions apply to the unit when it is in a sufficiently clean condition that the hydraulic flow distribution through the various flow routes is governed by the original design dimensions and clearances. The fraction of the total fluid-flow

quantity that traverses the crossflow area A_b within baffled region of the tube bundle is designated as

$$F_b = \frac{1}{1 + N_b \sqrt{S}} \quad (3)$$

as defined in Derivation A.

1.2 Crossflow Travel. The cross-flow fluid travel distance between baffles is considered to be from the center to center of baffle windows

$$\left[D_i \left(1 - \frac{H}{D_i} \right) \right]$$

1.3 Baffle-Window Flow. The fluid-flow quantity through the baffle windows is assumed to be the same as the cross-flow stream quantity. This is not an exact relationship in a conical heat exchanger as the window-flow stream is greater than the cross-flow stream due to the merging of certain leakage and bypass streams with the cross-flow stream in the window. This condition, however, is practically compensated for by computing the window area as if a full tube layout extended throughout the entire window. The actual window free area is larger than this owing to the absence of tubes about the peripheral space through which the bypass currents C circulate. It is believed that any inaccuracies introduced by the foregoing assumptions are small relative to the possible error which must be assumed in setting the various resistance factors which determine the distribution of fluid flow through the various routes within the heat exchanger.

1.4 Baffle-Window Cut. Recommended baffle window cuts H/D_i are graduated with the baffle spacing D_b in such a manner that the ratio of window area to crossflow area and thus M_w remains constant as practicable for a given tube arrangement pattern. On this basis each baffle spacing D_b has a single standard window cut H/D_i . To simplify further the rating system, a single M_w value is developed for each baffle spacing and tube arrangement which represents all tube spacings P/d_o from 1.25 to 1.50 with an accuracy of about ± 7 per cent for M_w which could affect the calculated value of h_{os} by about ± 4 per cent.

1.5 Overall Heat-Transfer Coefficient. The effect of crossflow conditions through the baffle window on the overall heat-transfer coefficient is accounted for by basing the heat-transfer coefficient in the window zones on the geometric mean of crossflow and window velocities, and the heat-transfer coefficient in the crossflow zone on the crossflow velocity through the tube bundle. A crossflow area multiplier M is developed in Derivation F which gives this result when the Re_k for heat transfer is expressed as follows:

$$\text{Re}_k = \frac{0.0344 h_2 F_b (\text{lb/hr})}{Z (M A_b)} \quad (4)$$

2. Pressure Drop

2.1 Flow Distribution. The flow distribution used in deriving a clean heat-transfer coefficient applies only when the heat exchanger is in a strictly clean condition. Heat-transfer deviations from this condition are customarily accounted for by the use of appropriate fouling resistance such as published in the Standards of the Tubular Exchanger Manufacturers Association.

When a heat exchanger goes into service, any fouling which takes place can cause a certain amount of choking off of leakage currents, in extreme cases reducing them to practically nothing. This results in more fluid to be transported through the tube nest and bypass routes. Normal fouling also may reduce the free areas through the tube nest and bypass routes, thus further increasing the tube-bundle mass-flow rate and pressure drop while insulating and reducing heat-transfer coefficients. To compensate for these heat-exchanger characteristics due to service fouling, the

* Shell Side Characteristics of Shell and Tube Heat Exchangers," by Townsend, Baker, General Discussion of Heat Transfer, The Institution of Mechanical Engineers and THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, London, England, 1951.

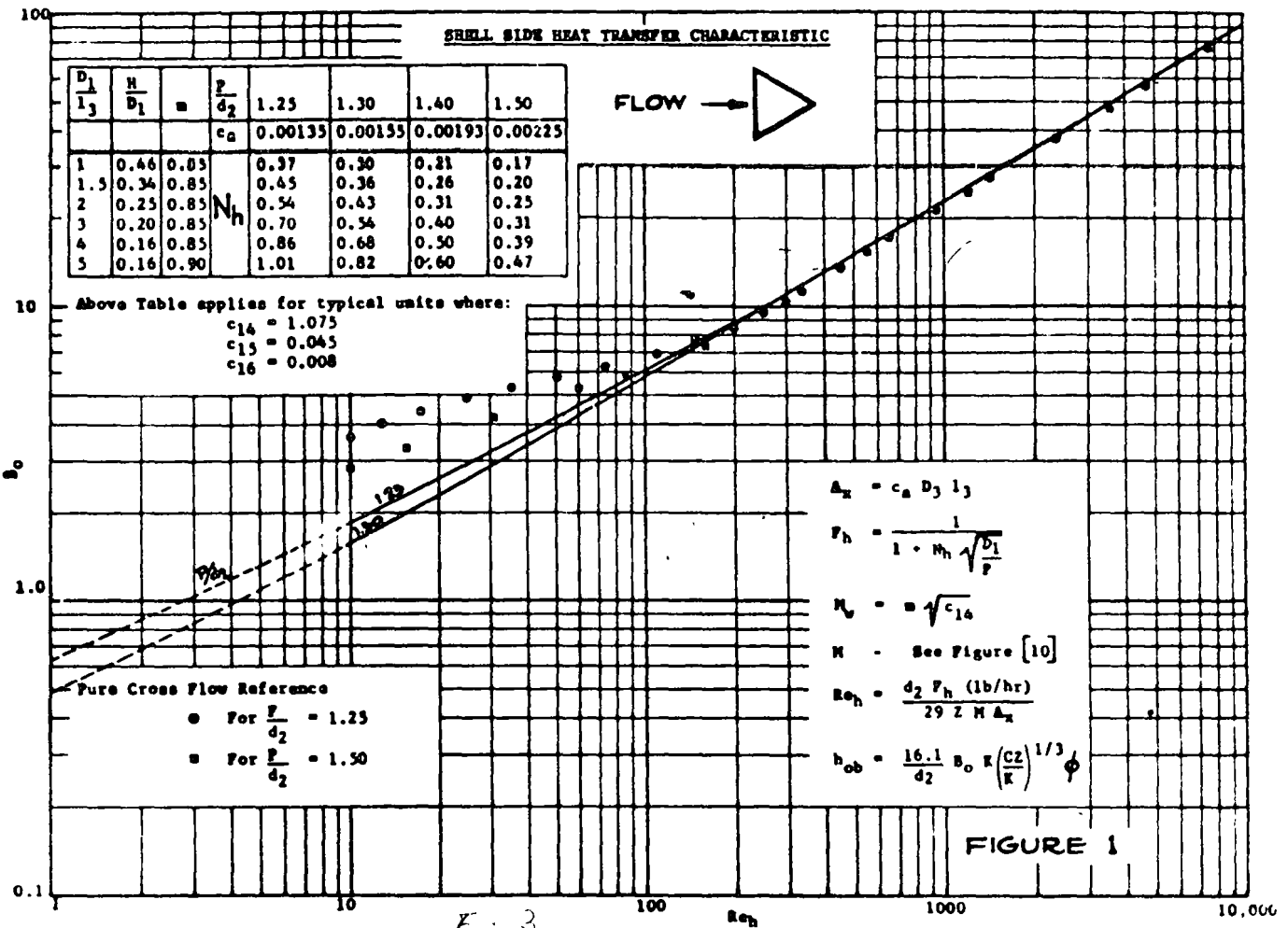


FIG. 1 SHELL-SIDE HEAT-TRANSFER CHARACTERISTICS

modified flow fraction used in pressure-drop rating computations is designated as

$$F_p = \frac{1}{0.80 + \sqrt{p} \sqrt{S}} \quad (13)$$

as defined in Derivation B.

2.2 Crossflow Travel. The crossflow fluid-travel distance between baffles is considered to be from the center to center of baffle windows

$$\left[D_1 \left(1 - \frac{H}{D_1} \right) \right]$$

2.3 Baffle-Window Flow. The fluid quantity through the baffle windows is assumed to be the same as the crossflow stream quantity as outlined under premise 1.3.

2.4 Crossflow Pressure Loss. The crossflow pressure loss is calculated using a crossflow friction factor f_s selected at

$$Re_p = \frac{0.0344 d_2 F_p \text{ (lb/hr)}}{Z A_s} \quad (14)$$

2.5 Baffle-Window Pressure Loss. The window pressure loss over and above the crossflow friction loss to its center is considered as (c_{12}) velocity heads at the geometric mean of window and crossflow velocities. c_{12} is usually taken as 2.0 in the turbulent-flow region. Furthermore, this loss is expressed as a ratio to the cross-

flow friction factor so that under nonturbulent-flow conditions the loss varies in proportion to the crossflow friction factor, thus compensating for frictional drag under viscous-flow conditions

CONCLUSIONS AND RECOMMENDATIONS

Design Formulas The following formulas are recommended for the shell-side design rating of commercial shell-and-tube heat exchangers

$$Re_s = \frac{0.0344 d_2 F_s \text{ (lb/hr)}}{Z (M A_s)} \quad (15)$$

$$h_{ot} = \frac{16.1}{d_2} \left[B_o K \left(\frac{CZ}{K} \right)^{1/3} \phi \right] \quad (16)$$

$$h_o = E h_{ot} \quad (17)$$

$$E_s = \frac{l_2 + 7(l_1 - l_2) \left(\frac{2l_2}{l_1 - l_2} \right)^{0.8}}{l_1} \quad (18)$$

$$Re_p = \frac{0.0344 d_2 F_p \text{ (lb/hr)}}{Z A_s} \quad (19)$$

$$\Delta P = 0.334 c_s S \frac{\lambda_s}{\rho} \frac{l_2}{\phi} \left(\frac{F_p \text{ (lb/hr)}}{10^6 A_s} \right)^2 \left[1 + \frac{F_p}{S} \right] + \Delta(\text{noz}) \quad (20)$$

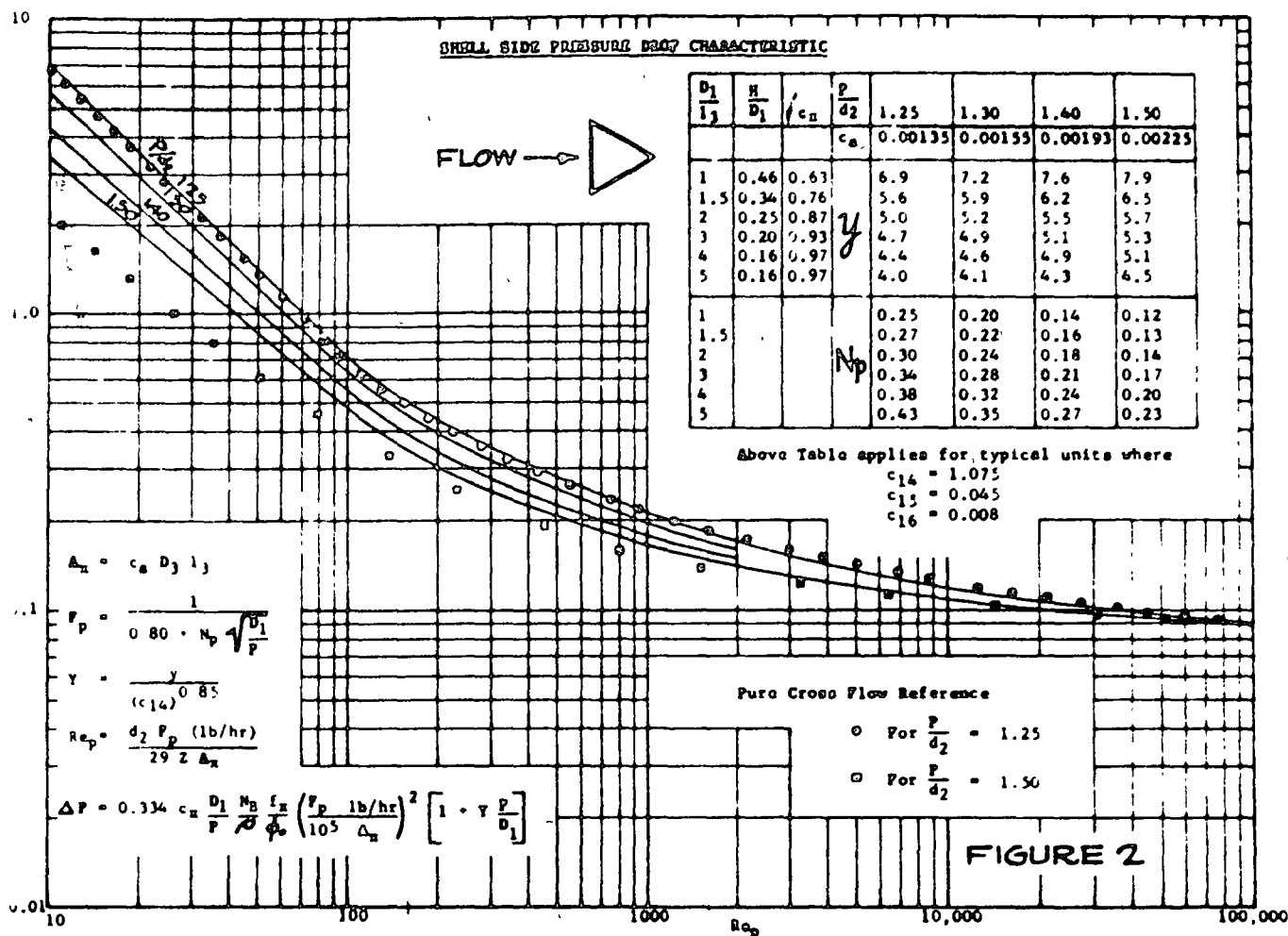


FIG. 2 SHELL-SIDE PRESSURE-DROP CHARACTERISTIC

$$A_n = c_n D_1 l_1$$

$$F_p = \frac{1}{0.80 \cdot N_p \sqrt{\frac{D_1}{P}}}$$

$$Y = \frac{y}{(c_{14})^{0.85}}$$

$$Re_p = \frac{d_2 P_p \text{ (lb/hr)}}{29 Z A_n}$$

$$\Delta P = 0.334 c_n \frac{D_1}{P} \frac{N_B}{\phi} \frac{f_x}{\phi_o} \left(\frac{P_p \text{ lb/hr}}{10^5 A_n} \right)^2 \left[1 + Y \frac{P}{D_1} \right]$$

A_n —see Derivation A

A_p —see Derivation B

Y —see Derivation G

M —see Derivation F

Design Example.—The heat-transfer and pressure-drop characteristics shown in Figs. 1 to 4 are recommended for the shell-side design rating of commercial shell-and-tube heat exchangers.

Dotted reference lines on these figures indicate the approximate findings of the University of Delaware and others for pure cross-flow conditions. It will be noted that the recommended commercial design curves for certain dips and irregularities occurring in the pure crossflow characteristics. This is done because usually in a baffled commercial heat exchanger a number of leakage and parallel-flow streams merge and change flow direction throughout the heat exchanger unit tending to produce eddies that would either wash out certain dips and irregularities or make it practically impossible to predict the exact total flow rate at which the irregularities might appear in the heat-exchanger performance.

Flow Distribution. The flow-distribution formulas developed

herein permit setting up simple tabulations of the heat-transfer and pressure-drop rating numbers N_A and N_p that readily determine the crossflow fractions for a series of heat exchangers of consistent design proportions. Such tabulations, together with constants c_n , c_x , m , and y , tabulated on the design curves, take into account the effect of all practical combinations of tube size, arrangement and pitch, baffle spacing, baffle cut, shell diameter and mechanical clearances for a consistent line of heat-exchanger units.

Design Example—Typical Rating Numbers. Similar lines of floating-head clamping type of shell-and-tube heat exchangers are produced by a number of manufacturers for oil-refining and process industries. One such typical line of equipment has clearance characteristics approximately as shown in Table 1.

TABLE 1

D_1/l_1	H/D_1
1	0.46
1.5	0.34
2	0.25
3	0.20
4	0.16
5	0.16

$$c_{14} = \frac{D_1}{P} = 1.075 \quad c_{15} = \frac{d_1 - d_2}{l_2} = 0.045$$

$$c_{16} = \frac{D_1 - D_2}{D_1} = 0.008$$

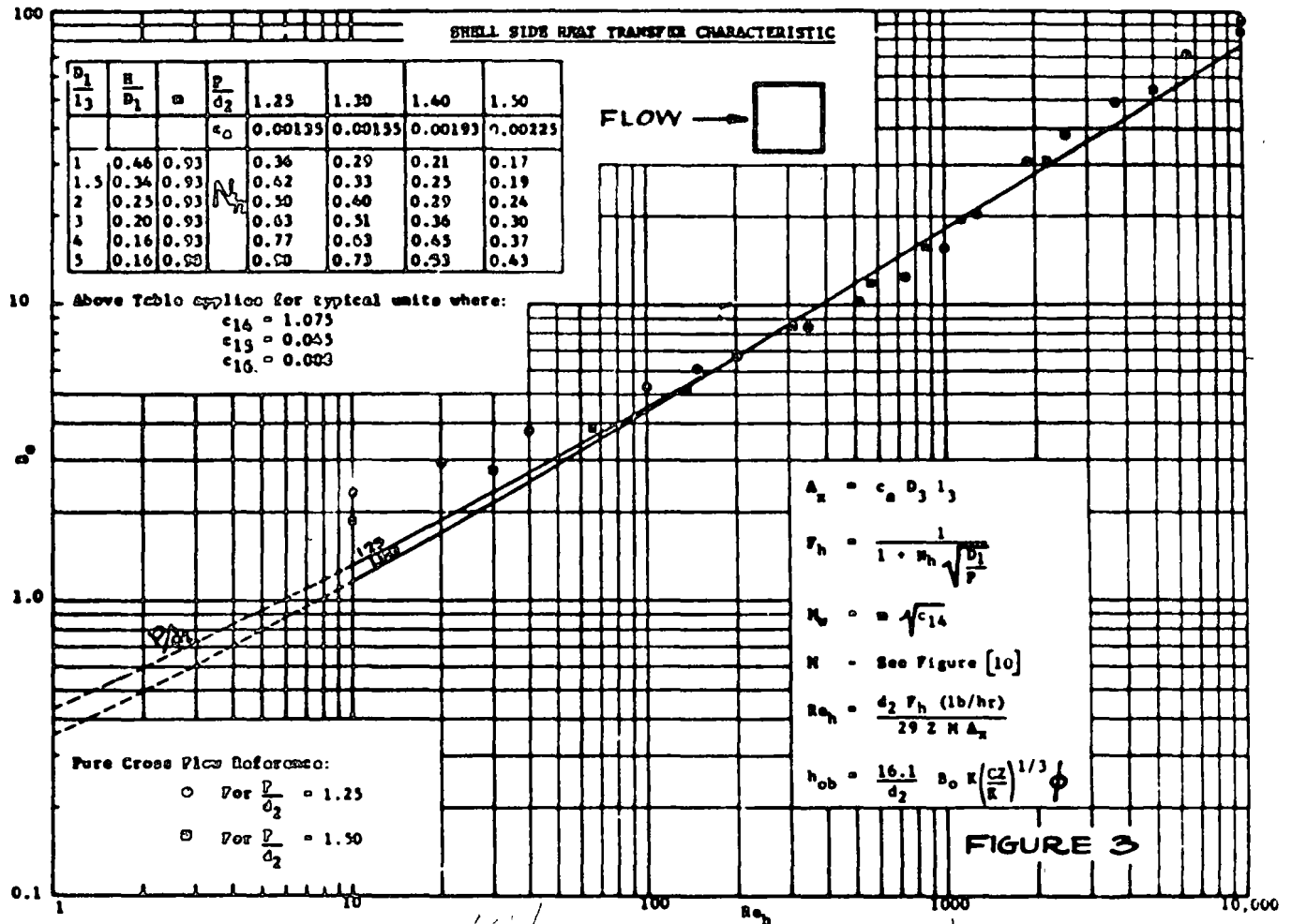


FIG. 3 SHELL-SIDE HEAT-TRANSFER CHARACTERISTIC

A substitution of these values in Equations [10], [11], and [12], and Equations [15], [16], and [17] for various flow orientations and various desired tube pitches gives values of the heat-transfer and pressure-drop rating numbers N_h and N_p , as shown in Figs. 1 to 6, inclusive.

Fixed-Tube-Sheet Exchangers. Rating-number tabulations similar to the foregoing may be produced readily for other series of units which may have a closer tube bundle-to-shell clearance (c_{14}), for example, as is usual for fixed-tube-sheet units. In certain units of this class, the value of c_{14} may be as low as 1.01.

Sealing Strips

Bolted Floating-Head Exchangers. Certain units with pull-through tube bundles and through-bolted floating heads have relatively large bundle-to-shell clearances so that the C bypass current would be large and the flow fraction F_h correspondingly small unless sealing strips, as illustrated in Fig. 7, or other means were used to restrict the C bypass route. Where sealing strips are used, a modified value of c_{14} is recommended to represent the prevailing bypass conditions in view of the following considerations:

(a) The sealing strips usually clear the shell by as much as or more than the baffles so that a direct bypass orifice prevails between the strips and the shell.

(b) Some of the fluid forced through the tube bundle at the sealing strips tends to spread out into the bypass area ($c_{14} = 1$), thus bypassing the tube bundle in the regions between the successive sealing strips.

(c) It is recommended that the mean amount of tube bundle bypassing indicated in (a) and (b) be estimated on the basis of a modified value of

$$c_{14} = \left[1 + \left(\frac{c_{14} - 1}{4} \right) + 1.5c_{14} \right] \quad [17]$$

The rating numbers, N_h and N_p , calculated on the basis of the foregoing value of c_{14} in the first term of Equations [10], [11], [12], [15], [16], and [17] give flow fractions that are consistent with the operating performance of commercial heat exchangers provided with two or more sealing strips on each side of the tube bundle and spaced less than 12 in. apart.

DISCUSSION

Performance Comparisons. A comparison of some of the indicated heat-transfer flow-fraction characteristics of several typical varieties of heat-exchanger designs is shown in Table 2.

It will be noted that the indicated fraction of fluid in crossflow through the tube bundle is higher in fixed-tube-sheet units where the clearance between bundle and shell is smaller than it is in other units having greater bypass clearances. Sealing strips tend to correct this condition by partially eliminating the bypassing.

The tabulation also indicates the effect in larger diameter units of the higher pressure drops per baffle space (from flow across more layers) in driving more fluid through the leakage routes, thus reducing the crossflow fraction through the tube bundle.

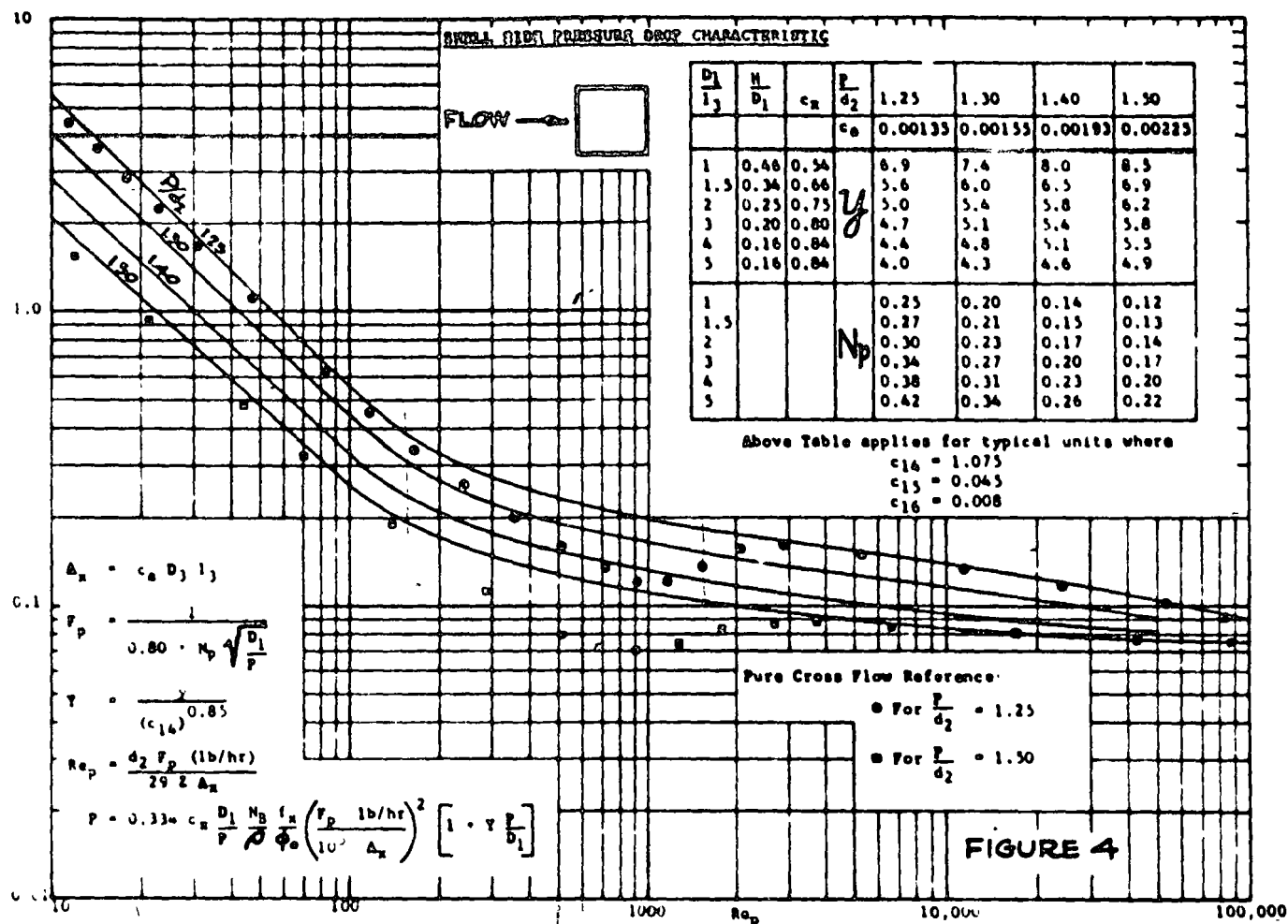


FIG. 4 SHELL-SIDE PRESSURE-DROP CHARACTERISTIC

TABLE 2- PERFORMANCE COMPARISONS

Type of heat exchanger	D_1	h	$\frac{D_1}{h}$	P	d_1	Assumed clearance proportions—			S_A	F_A , per cent
						c_{14}	c_{15}	c_{16}		
Fixed tube sheet—no sealing strips	21	7	3	0.94	0.75	1.04	0.045	0.008	0.53	28.5
Bolted pull-through—floating head with sealing strips	21	7	3	0.94	0.75	1.25	0.045	0.008	0.69	23.5
Clamp ring floating head—no sealing strips	21	7	3	0.94	0.75	1.075	0.045	0.008	0.65	21.4
Clamp ring floating head—no sealing strips	21	7	3	0.94	0.75	1.075	0.045	0.008	0.70	24.2
Clamp ring floating head—no sealing strips	31	10.33	3	0.94	0.75	1.075	0.045	0.008	0.70	20.0
Clamp ring floating head—no sealing strips	42	14	3	0.94	0.75	1.075	0.045	0.008	0.70	17.5

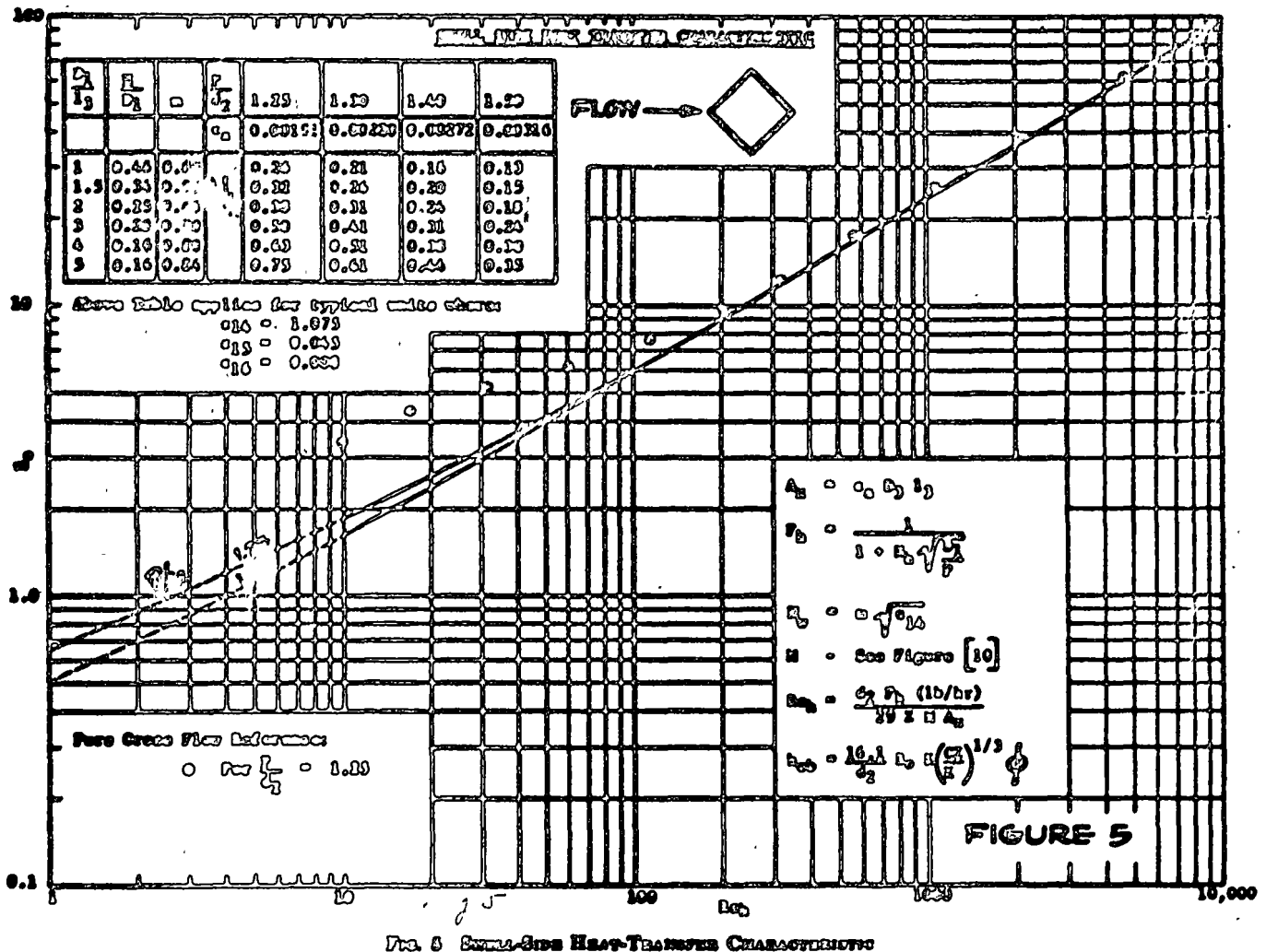


Fig. 8 Small-Side Heat-Transfer Characteristics

Fig. 8 shows a visual comparison of several heat-exchanger layouts. The comparative differences to be expected in the crossflow fractions are practically obvious from inspection. The use of the recommended style of rating numbers quantitatively evaluates the shell-side flow conditions for any specific set of design proportions that may be built into a segmentally baffled heat exchanger.

Accuracy of Simplified Rating System. The author believes the system to be accurate well within practical commercial requirements. It is impractical to attempt to consider all possible minor deviations in flow pattern which might theoretically occur, when estimating each commercial heat-exchanger unit.

Few, if any, rating procedures have been available which attempt to appraise the proper direction and magnitude of performance variations caused by a large range of combinations of different unit diameters, tube size, tube pitch and flow orientation, baffle spacing and cut, together with the principal leakage and bypass clearances of the heat-exchanger unit.

The accuracy of the proposed simplified rating system with respect to the crossflow fraction and heat-transfer coefficient is influenced by the factors given in Table 2.

It is apparent from summary in Table 3 that the physical dimensional assumption introduces no significant error into the system.

The accuracy of the calculated flow-distribution pattern and crowflow fraction is governed largely by the assumed resistance

factories for the various flow routes. These have been selected to line up as well as possible with theory, research, and test performance of a wide range of practical commercial heat-exchanger units.

The analysis presented develops rating-number formulas, for determining crossflow fractions, that are flexible in that more accurate resistance factors may be substituted as developed by research for specific cases to give more precise results when desired.

BASIC REFERENCE DATA

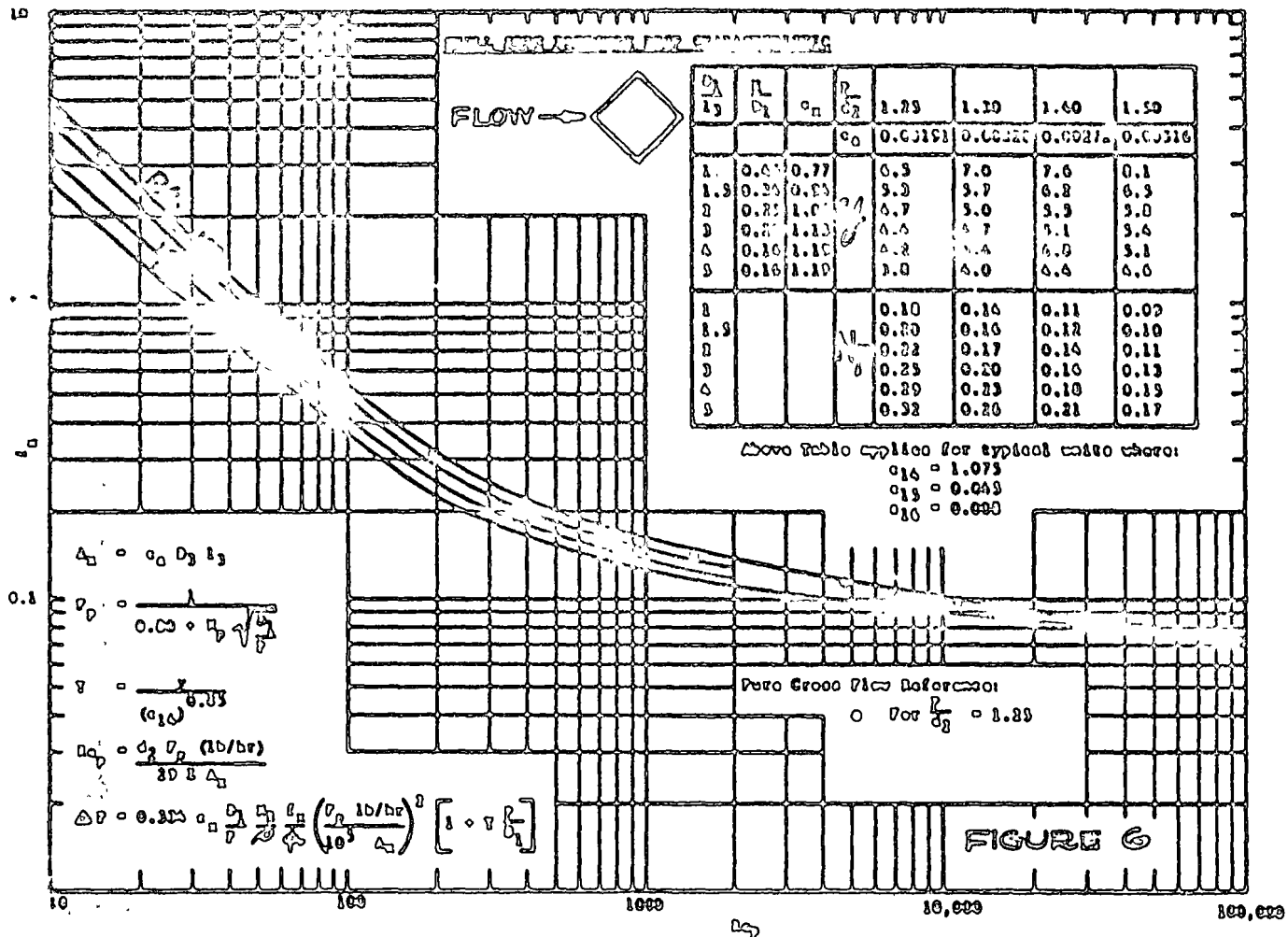
Fluid-Flow Distribution. The principal fluid streams through the shell side of a heat exchanger as shown in fig. 6 of the earlier paper¹ are indicated in Fig. 9. The relative fluid-stream quantities from table 10,¹ amplified to include oriented square pitch and revised on the basis of the bypass resistance factor being reduced to two thirds of the value outlined in Note (a) of the earlier paper,¹ are as shown in Table 4.

DERIVATIONS

A—Crossflow Partition—Heat Transfer

The fraction of the total fluid flowing through the tube-bundle crossflow area of a clean unit may be represented as follows

$$P_0 = \frac{Q_{22}}{Q_{22} + Q_{32} + Q_{42} + Q_{52}} \dots\dots\dots (1)$$



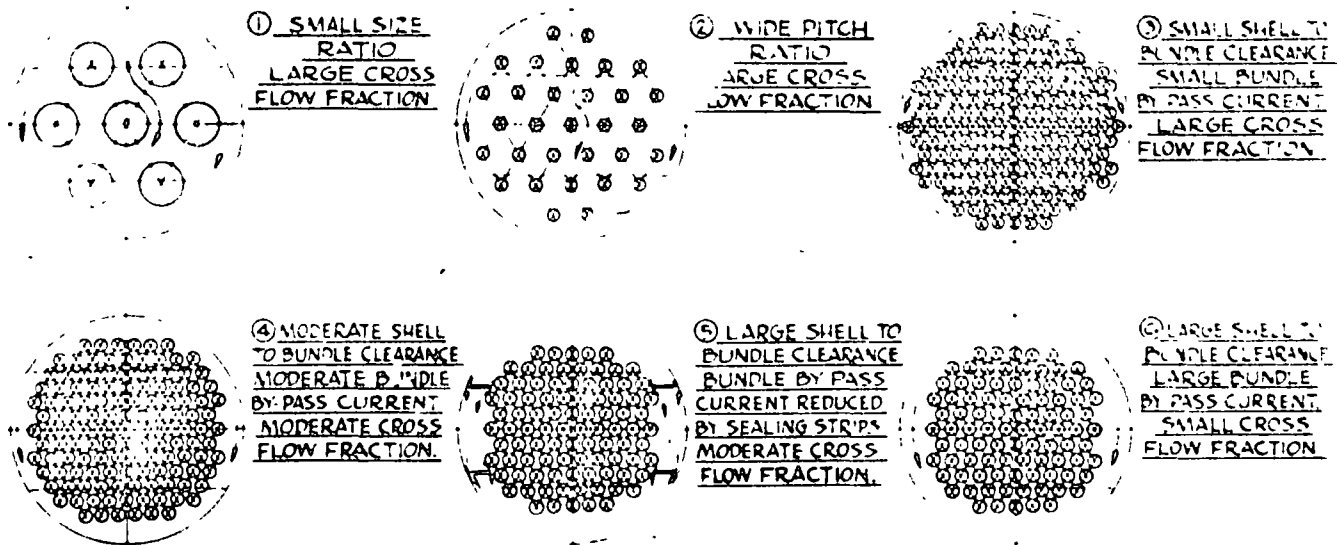


FIG. 8 Heat Exchanger-Performance Comparisons

TABLE 3 Factors Influencing Rating-System Accuracy

Dimensional Assumptions	Approximate maximum error in assumed constant, per cent	Approximate effect (per cent) on calculated values of P_A	Approximate effect (per cent) on calculated values of T_A
Mean tube-bundle width between baffle windows $c_4 = 0.97$	± 3	± 2	± 1
Tube count or layout $c_5 = 0.94$	± 4	± 2	± 1
M_u based on P/d_o at compromise between 1.25 and 1.50	± 7		± 1
Resistance-Factor Assumptions			
Bypass friction-factor multiplier $c_{10} = 0.667$ $(c_{10})^{1/4} = 0.815$	± 18	± 6	± 1
Velocity heads loss through baffle tube holes $c_6 = 4$ (disch. coef. = 0.50) $(c_6)^{1/4} = 2$	± 50	± 5	± 5
Velocity heads loss over one row of tubes $c_7 = 0.6$ $(c_7)^{1/4} = 0.775$	± 20	± 7	± 1
Velocity heads loss between baffle and shell $c_{11} = 3$ (disch. coef. = 0.58) $(c_{11})^{1/4} = 1.73$	± 30	± 8	± 5
Velocity heads loss through baffle window $c_{12} = 2$ ± 50 per cent This affects x ± 50 per cent ($1+x$) about ± 17 per cent ($1+x$) $^{1/4}$ about ± 8 per cent $(1 - \frac{H}{D_h})^{1/4}$ ($1+x$) $^{1/4} = 0.95$	± 10	± 3 ± 1	± 2 ± 2

JANUARY, 1958

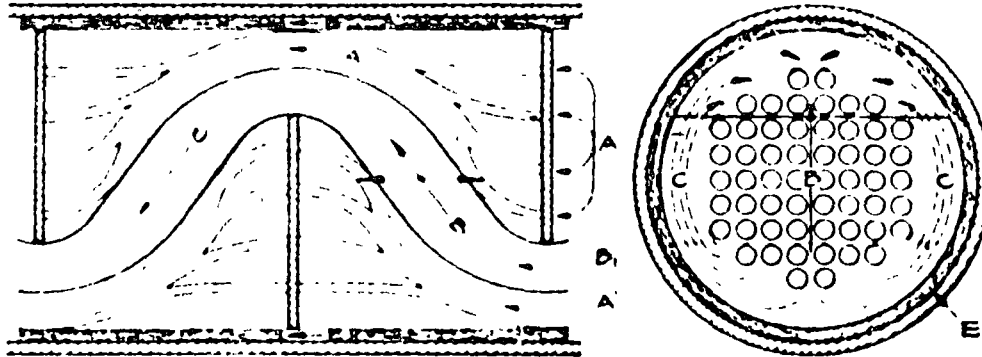


FIG. 9. DIAGRAM OF FLUID STREAMS THROUGH SHELL SIDE OF A HEAT EXCHANGER

Fig. 19

TABLE 1. RELATIVE FLUID-STREAM QUANTITIES

Fluid Stream Identification	Triangular Patch $\rightarrow \nabla$
Q_{AK}	$1.07 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (d_1^2 - d_2^2) \right)$
Q_{BK}	$0.93 \left(\epsilon_0(1+x)^{1/2} d_1 D_1^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{CK}	$0.93 \left(\epsilon_0(1+x)^{1/2} d_2 D_2^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{DK}	$3.11 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (D_1 - D_2) \right)$
	Square Patch $\rightarrow \square$
Q_{AK}	$0.927 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (d_1^2 - d_2^2) \right)$
Q_{BK}	$1.0 \left(\epsilon_0(1+x)^{1/2} d_1 D_1^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{CK}	$1.0 \left(\epsilon_0(1+x)^{1/2} d_2 D_2^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{DK}	$3.11 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (D_1 - D_2) \right)$
	Unfilled Square Patch $\rightarrow \square$
Q_{AK}	$0.927 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (d_1^2 - d_2^2) \right)$
Q_{BK}	$1.18 \left(\epsilon_0(1+x)^{1/2} d_1 D_1^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{CK}	$0.81 \left(\epsilon_0(1+x)^{1/2} d_2 D_2^2 (P - d_1)^{-1/2} (P - d_2) \right)$
Q_{DK}	$3.11 \left(\epsilon_0(1+x)^{1/2} \epsilon_0^{-1/2} (D_1/P)^{1/2} (D_1 - D_2) \right)$

$$F_s = \frac{1}{1 + \frac{Q_{AK}}{Q_{BK}} + \frac{Q_{CK}}{Q_{BK}} + \frac{Q_{DK}}{Q_{BK}}} \quad (1a)$$

substituting the values of Q_{AK} , Q_{CK} , and Q_{DK} from Table 1 for triangular patch and replacing

$d_1^2 - d_2^2$ with $2d_1(d_1 - d_2)$; $(D_1 - D_2)$ with $\epsilon_0 D_1$;

$$F_s = \frac{1}{1 + \left(\frac{D_1}{P} \right)^{1/2} \left[\frac{Q_{AK} P^{1/2}}{Q_{BK} D_1^{1/2}} + \frac{Q_{CK} P^{1/2}}{Q_{BK} D_2^{1/2}} + \frac{Q_{DK} P^{1/2}}{Q_{BK} D_1} \right]} \quad (2)$$

$$\epsilon_0 = \frac{d_1 - d_2}{d_1} \quad (1b)$$

$$F_s = \frac{1}{1 + N_s \sqrt{S}} \quad (3)$$

$$(d_1 - d_2) \text{ with } \epsilon_0 d_1 = (d_1) \text{ with } \left(1 - \frac{H}{D_1} \right), \quad \epsilon_0 = \frac{D_1}{D_2} \quad (1c)$$

$$(D_2 \text{ with } \frac{D_1}{\epsilon_0}, \quad \epsilon_0 \text{ with } \left(1 - \frac{H}{D_1} \right), \quad \epsilon_0 = \frac{D_1 - D_2}{D_1} \quad (1d)$$

where

$$N_s = \left[\frac{Q_{AK} P^{1/2}}{Q_{BK} D_1^{1/2}} + \frac{Q_{CK} P^{1/2}}{Q_{BK} D_2^{1/2}} + \frac{Q_{DK} P^{1/2}}{Q_{BK} D_1} \right] \quad (4)$$

The following expression is obtained

$$N_s = \left[\epsilon_0^{-1/2} (1+x)^{1/4} \left(\frac{P}{P-d_1} \right)^{1/4} + \frac{2 \epsilon_0 \epsilon_0^{-1/2} (D_1/P)^{1/2} \left(1 - \frac{H}{D_1} \right)^{1/2} (1+x)^{1/4}}{\epsilon_0^{-1/2} \epsilon_0^{-1/2} d_1 (P-d_1)^{1/2}} + \frac{1.18 \epsilon_0^{-1/2} \epsilon_0^{-1/2} (1-H) \left(\frac{P}{P-d_1} \right)^{1/4} D_1 \left(1 - \frac{H}{D_1} \right)^{1/4} (1+x)^{1/4}}{\epsilon_0^{-1/2} \epsilon_0^{-1/2} D_1 (P-d_1)^{1/2}} \right] \quad (5)$$

Substituting the following assumed values in Equation [8]

$$c_0 = 0.97 \quad c_1 = 0.94 \quad c_{11} = 3 \quad c_{12} = 0.667 \quad c_6 = 1 \quad c_8 = 0.60 \quad \left(1 - \frac{H}{D_1}\right)^{1/2} (1 + x)^{1/2} = 0.95$$

the following expression is obtained

$$\rightarrow N_h = \left[\frac{(c_{14} - 1)^{1/2}}{(c_{14})^{1/2}} 1.26 \left(\frac{P}{P - d_2} \right)^{1/2} + \frac{c_{15}}{(c_{14})^{3/2}} 0.82 c_1 \frac{D_1}{L} \left(\frac{d_2}{P} \right)^2 \left(\frac{P}{P - d_2} \right) + c_{16} (c_{14})^{1/2} 1.48 \left(\frac{P}{P - d_2} \right) \frac{D_1}{L} \left(1 - \frac{H}{D_1} \right) \right] \quad [9]$$

Evaluating functions of $\frac{P}{d_2}$ and $\frac{D_1}{L}$ in Equation [9] for commonly encountered values of $\frac{P}{d_2}$ and $\frac{D_1}{L}$ gives:

$\frac{P}{d_2} =$	1.25	1.30	1.40	1.50	$\frac{D_1}{L}$	$\frac{H}{D_1}$
$N_h = \left\{ \frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \right.$	14.1	11.3	8.25	6.35		
$+ \frac{c_{15}}{(c_{14})^{3/2}}$	1.44 2.75 4.2 6.7 9.4 11.8	1.15 2.20 3.4 5.4 7.6 9.5	0.80 1.53 2.34 3.7 5.3 6.6	0.59 1.13 1.73 2.75 3.9 4.9	1 1.5 2 3 4 5	0.46 0.34 0.25 0.20 0.16 0.16
$+ c_{16} (c_{14})^{1/2}$	4.0 7.4 11.1 17.8 24.9 31	3.5 6.4 9.6 15.4 21.6 27.0	2.80 5.1 7.8 12.5 17.5 21.8	2.40 4.4 6.7 10.7 15.0 18.7	1 1.5 2 3 4 5	0.46 0.34 0.25 0.20 0.16 0.16

Values of the clearances c_{14} , c_{15} , and c_{16} for a consistent style or series of heat exchangers may be inserted in Equation [10] and the three terms for various values of $\frac{P}{d_2}$ and $\frac{D_1}{L}$ summed up giving a simple table of Rating Numbers for the series of heat exchangers.

In a similar manner the following expression for $\rightarrow \square$ orientation may be obtained.

$\frac{P}{d_2} =$	1.25	1.30	1.40	1.50	$\frac{D_1}{L}$	$\frac{H}{D_1}$
$N_h = \left\{ \frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \right.$	14.1	11.3	8.25	6.55		
$+ \frac{c_{15}}{(c_{14})^{3/2}}$	1.15 2.20 3.3 5.3 7.5 9.4	0.92 1.75 2.72 4.3 6.1 7.6	0.64 1.23 1.86 2.95 4.2 5.3	0.47 0.90 1.38 2.16 3.1 3.9	1 1.5 2 3 4 5	0.46 0.34 0.25 0.20 0.16 0.16
$+ c_{16} (c_{14})^{1/2}$	3.7 6.9 10.4 16.6 23.2 29.9	3.3 6.0 9.0 14.4 20.4 25.2	2.60 4.7 7.3 11.7 16.3 20.4	2.24 4.1 6.2 10.0 14.0 17.5	1 1.5 2 3 4 5	0.46 0.34 0.25 0.20 0.16 0.16

The following expression applies for $\rightarrow \diamond$ orientation.

$$N_h = \left\{ \frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \right. \quad \begin{array}{c} \frac{P}{d_2} = \\ \hline 1.25 \quad 1.30 \quad 1.40 \quad 1.50 \end{array} \quad \left. \begin{array}{c} \frac{c_{15}}{(c_{14})^{3/2}} \\ \hline 0.99 \quad 0.79 \quad 0.55 \quad 0.40 \\ 1.88 \quad 1.51 \quad 1.05 \quad 0.78 \\ 2.87 \quad 2.32 \quad 1.60 \quad 1.19 \\ 4.6 \quad 3.7 \quad 2.53 \quad 1.88 \\ 6.4 \quad 5.2 \quad 3.6 \quad 2.67 \\ 8.1 \quad 6.5 \quad 4.5 \quad 3.4 \end{array} \right\} + c_{16}(c_{14})^{1/2} \quad \begin{array}{c} \frac{D_1}{L_3} \quad \frac{H}{D_1} \\ \hline 1 \quad 0.46 \\ 1.5 \quad 0.34 \\ 2 \quad 0.25 \\ 3 \quad 0.20 \\ 4 \quad 0.16 \\ 5 \quad 0.16 \end{array} \quad [12]$$

B - Modified Cross Flow Fraction - Pressure Drop (See Premise 2.1)

$$F_p = \frac{1}{0.8 + \sqrt{8} [N_p]} \quad [13]$$

Where for $\rightarrow \triangle$ orientation:

$$N_p = \left[\frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \left(\frac{P}{P - d_2} \right)^{3/2} + c_{16}(c_{14})^{1/2} \cdot 1.18 \frac{D_1}{L_3} \left(\frac{P}{P - d_2} \right) \left(1 - \frac{H}{D_1} \right) \right] \quad [14]$$

From which the following values are obtained:

$$N_p = \left\{ \frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \right. \quad \begin{array}{c} \frac{P}{d_2} = \\ \hline 1.25 \quad 1.30 \quad 1.40 \quad 1.50 \end{array} \quad \left. \begin{array}{c} \frac{c_{16}(c_{14})^{1/2}} \\ \hline 3.2 \quad 2.8 \quad 2.24 \quad 1.92 \\ 5.9 \quad 5.1 \quad 4.1 \quad 3.5 \\ 8.9 \quad 7.7 \quad 6.2 \quad 5.4 \\ 14.2 \quad 12.3 \quad 10.0 \quad 8.5 \\ 20 \quad 17.3 \quad 14.0 \quad 12.0 \\ 25 \quad 21.6 \quad 17.4 \quad 15.0 \end{array} \right\} \quad \begin{array}{c} \frac{D_1}{L_3} \quad \frac{H}{D_1} \\ \hline 1 \quad 0.46 \\ 1.5 \quad 0.34 \\ 2 \quad 0.25 \\ 3 \quad 0.20 \\ 4 \quad 0.16 \\ 5 \quad 0.16 \end{array} \quad [15]$$

In a similar manner the following expression for $\rightarrow \square$ orientation may be obtained.

$$N_p = \left\{ \frac{(c_{14} - 1)^{3/2}}{(c_{14})^{1/2}} \right. \quad \begin{array}{c} \frac{P}{d_2} = \\ \hline 1.25 \quad 1.30 \quad 1.40 \quad 1.50 \end{array} \quad \left. \begin{array}{c} \frac{c_{16}(c_{14})^{1/2}} \\ \hline 2.96 \quad 2.64 \quad 2.08 \quad 1.79 \\ 5.5 \quad 4.8 \quad 3.8 \quad 3.3 \\ 8.3 \quad 7.2 \quad 5.8 \quad 5.0 \\ 13.3 \quad 11.5 \quad 9.4 \quad 8.0 \\ 18.5 \quad 16.3 \quad 13.1 \quad 11.2 \\ 23 \quad 20.2 \quad 16.3 \quad 14.0 \end{array} \right\} \quad \begin{array}{c} \frac{D_1}{L_3} \quad \frac{H}{D_1} \\ \hline 1 \quad 0.46 \\ 1.5 \quad 0.34 \\ 2 \quad 0.25 \\ 3 \quad 0.20 \\ 4 \quad 0.16 \\ 5 \quad 0.16 \end{array} \quad [16]$$

The following expression applies for $\rightarrow \diamond$ orientation.

$$N_p = \left\{ \frac{(c_{16} - 1)^{3/2}}{(c_{16})^{1/2}} \right. \quad \left. \begin{array}{cccc} 1.25 & 1.30 & 1.40 & 1.50 \\ \hline 8.1 & 5.9 & 4.7 & 3.8 \end{array} \right\} \times c_{16} (c_{14})^{1/2} \quad \left\{ \begin{array}{cccc} 2.66 & 2.22 & 1.77 & 1.52 \\ 4.7 & 4.1 & 3.2 & 2.8 \\ 7.0 & 6.1 & 5.0 & 4.2 \\ 11.3 & 9.8 & 7.9 & 6.8 \\ 15.8 & 13.7 & 11.1 & 9.5 \\ 19.7 & 17.2 & 13.8 & 11.0 \end{array} \right\} \quad \begin{array}{cc} \frac{D_1}{L_2} & \frac{H}{D_1} \\ \hline 1 & 0.46 \\ 1.5 & 0.34 \\ 2 & 0.25 \\ 3 & 0.20 \\ 4 & 0.16 \\ 5 & 0.16 \end{array} \quad [17]$$

C—Evaluation of c_p

N_p = rows of tubes traversed in each baffle space

$$\begin{array}{lcl} \text{Orientation} & c_p & \\ \rightarrow \triangleright & 1.155 \left(1 - \frac{H}{D_1} \right) \times S & \\ \rightarrow \square & 1.0 \left(1 - \frac{H}{D_1} \right) \times S & \left(\frac{D_1}{P} = S \right) \\ \rightarrow \diamond & 1.414 \left(1 - \frac{H}{D_1} \right) \times S & \end{array}$$

Values of c_p in accordance with the foregoing are tabulated on the friction-factor curve for various values of (D_1/L_2) and their recommended baffle cuts (H/D_1) .

D—Evaluation of c_a

A_w = mean crossflow area within tube bundle between baffle windows (in square feet)

$$\begin{array}{lcl} \text{Orientation} & c_a & \\ \rightarrow \triangleright & \frac{1.0}{144} c_s \left(\frac{P - d_2}{P} \right) \times D_1 L_2 & \\ \rightarrow \square & \frac{1.0}{144} c_s \left(\frac{P - d_2}{P} \right) \times D_1 L_2 & \\ \rightarrow \diamond & \frac{1.414}{144} c_s \left(\frac{P - d_2}{P} \right) \times D_1 L_2 & \end{array}$$

Assume $c_s = 0.97$

Values of c_a in accordance with the foregoing are tabulated on the heat-transfer and friction-factor curves for various flow orientations and tube spacings (P, d_2) .

E—Evaluation of M_w for Determining Geometric Mean of Crossflow and Window Areas

$$A_w = \sqrt{(A_s A_o)} = \left[\sqrt{\left(\frac{A_s}{A_o} \right)} \right] A_s = M_w A_s \quad [18]$$

$$M_w = \left(\frac{A_s}{A_o} \right)^{1/2} = \left\{ \frac{0.786 c_1 D_1^2 \left[1 - \frac{0.786}{c} \left(\frac{d_2}{P} \right)^2 \right]}{144 c_s D_1^2} \right\}^{1/2} \quad [19]$$

Replacing D_1 with D_1/c_{16} and making α = portion of tube lay-out unoccupied by tubes

$$\alpha = \left[1 - \frac{0.786}{c} \left(\frac{d_2}{P} \right)^2 \right] \quad [20]$$

$$M_w = (c_{16})^{1/2} \left[\frac{0.786 c_1 D_1}{144 c_s L_2 \alpha} \right]^{1/2} \quad [21]$$

$$M_w = (c_{16})^{1/2} m \quad [22]$$

where

$$m = \left[\frac{0.786 c_1 D_1}{144 c_s L_2 \alpha} \right]^{1/2}$$

Values of m in accordance with Equation [23] are tabulated on the recommended heat-transfer curves for various values (D_1/L_2) and their recommended baffle cuts (H/D_1) .

F—Heat-Transfer Correction for Baffle-Window Proportions

The heat-transfer coefficient in the window zone is taken to vary with the 0.6 power of the geometric mean of crossflow and window velocities. The heat-transfer coefficient in the crossflow zone is taken to vary with the 0.6 power of the crossflow velocity.

$$h_w \alpha = \frac{1}{(M_w A_s)^{0.6}} = \frac{S_w}{(M_w A_s)^{0.6}} \left(\frac{1 - S_w}{A_s} \right)^{0.6} \quad [24]$$

where S_w represents the fraction of heat-transfer surface in the window zones.

With baffle cuts H, D_1 graduated as recommended, the portion of the heat-transfer surface in the two window zones is approximated by

$$S_w = 0.70 \left(\frac{L_2}{D_1} \right) \quad [25]$$

On this basis Equation [24] reduces to

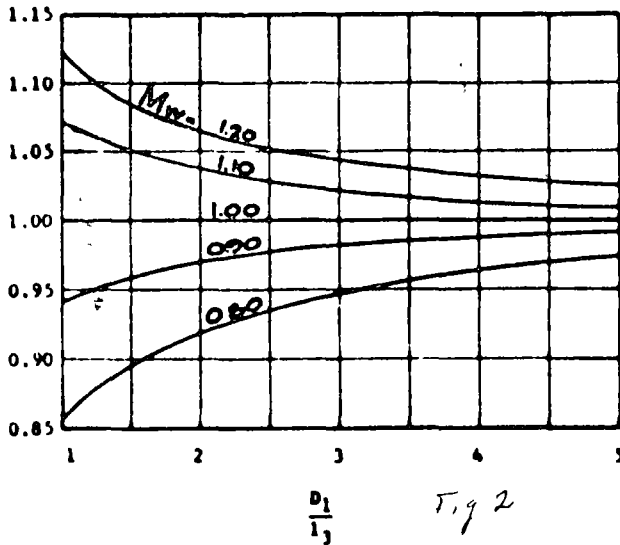
$$M = \left[\frac{1}{1 + \frac{0.70 L_2}{D_1} \left(\frac{1}{M_w^{0.6}} - 1 \right)} \right]^{1.67} \quad [26]$$

The variation of the A_s multiplier (M) with D_1/L_2 and (M_w) in accordance with Equation [26] is shown in Fig. 10

G—Evaluation of Pressure Drop

Equation [15] may be rewritten

$$\Delta P_s = 0.335 c_s \frac{D_1}{P} \frac{N_p}{\rho} \frac{f_s}{\phi} \left(\frac{F_p \text{ lb/hr}}{10 A_s} \right)^2 \quad [27]$$

FIG. 10. M , MULTIPLIER M FOR HEAT TRANSFER

Equation [16] may be rewritten

$$\Delta P = \frac{0.084 c_s V_B}{\rho \phi} \left(\frac{\text{lb/hr } F_p}{10^3 A_s} \right)^2 \quad (28)$$

Combining Equations [27] and [28] with assumption (see premise 2) $V_B = F_p$ and $c_s = 1/c_a$

$$\Delta P = 0.084 c_s S \frac{V_B}{\rho \phi} \left(\frac{F_p \text{ lb/hr}}{10^3 A_s} \right)^2 \left[1 + \frac{1}{Sc_s} \left(\frac{V_B}{1_s} \right)^2 \right] \quad (29)$$

Assume friction-factor corrections for the window zone vary with $Re^{-0.1}$

Then

$$f_w = \frac{c_w}{1} \left(\frac{F_p \text{ lb/hr}}{Z A_s} \right)^{-0.1} \quad (30)$$

$$f_s = \frac{c_s}{1} \left(\frac{F_p \text{ lb/hr}}{Z A_s} \right)^{-0.1} \quad (31)$$

$$\frac{c_w}{c_s} = \left(\frac{1_s}{1_w} \right)^{-0.1} \quad (32)$$

Substituting this value for c_w/c_s in the first term of Equation [29] gives the following

$$\left[\frac{1}{Sc_s} \left(\frac{1_s}{1_w} \right)^2 \right] = \frac{c_w}{c_s} \frac{1}{Sc_s} \left(\frac{A_s}{1_s} \right)^{1.1} \quad (33)$$

Let it be assumed that the pressure loss in the baffle-window zone over and above the crossflow friction to its center varies with the geometric mean of window and crossflow velocities. Substituting the geometric mean area $A_m = A_s^{1/2} A_w^{1/2}$ for A_w , then Equation [33] becomes

$$\left[\right] = \frac{c_w}{c_s} \frac{1}{Sc_s} \left(\frac{A_s}{1_s^{1/2} A_w^{1/2}} \right)^{1.1} = \frac{c_w}{c_s} \frac{1}{Sc_s} \left(\frac{A_s}{A_m} \right)^{2.2} \quad (34)$$

$$\left[\right] = \frac{c_w}{c_s} \frac{1}{Sc_s} \left[\frac{114 c_s D_1}{0.786 c_s D_1} \left[1 - \frac{0.786}{c} \left(\frac{d_1}{p} \right)^2 \right] \right]^{2.2} \quad (35)$$

$$\text{Taking } c_w = 2 \quad c_s = 0.6 \quad D_1 = \frac{D_1}{c_w}$$

α = portion of tube layout unoccupied by tubes

$$= \left[1 - \frac{0.786}{c} \left(\frac{d_1}{p} \right)^2 \right] \quad (21)$$

$$\left[\right] = \frac{1}{Sc_w^{0.8}} \frac{114 c_s D_1}{c_s D_1 \alpha} \left[\frac{114 c_s D_1}{c_s D_1 \alpha} \right]^{0.8} \quad (26)$$

$$\left[\right] = \frac{y}{Sc_w^{0.8}} \quad (27)$$

where

$$y = \frac{4.08}{c_s} \left[\frac{114 c_s D_1}{c_s D_1 \alpha} \right]^{2.2} \quad (28)$$

Representing

$$\frac{y}{(c_w)^{0.8}} = Y \quad (29)$$

then Equation [29] becomes

$$\Delta P = 0.033 c_s S \frac{V_B}{\rho \phi} \left(\frac{F_p \text{ lb/hr}}{10^3 A_s} \right)^2 \left[1 + \frac{Y}{S} \right] + \Delta P_{\text{noz}} \quad (30)$$

Values of y for the various baffle spacings, tube orientations, and pitches in accordance with Equation [38] have been calculated and are tabulated on the recommended friction-factor curves.

REFERENCES

1. "Heat Transfer and Fluid Friction During Flow Across Banks of Tubes—IV. A Study of the Transition Zone Between Viscous and Turbulent Flow," by O. P. Bergelin, G. A. Brown, and S. C. Doberson, Trans. ASME, vol. 74, 1952, pp. 953-960; also University of Delaware Heat Exchanger Research Project Progress Reports.
2. Standards of the Tubular Exchanger Manufacturers Association.

Discussion

K. A. GEMMEL¹: Our understanding of shell-and-tube heat exchanger rating undoubtedly would be far less advanced than it is, were it not for the author's stimulating contributions to the subject, as exemplified by this current paper. The writer believes, nevertheless, that his approach, and that of a companion paper,² contain elements which should be pursued with caution.

The distribution of flow through baffled heat exchangers is hydrodynamically determinate. Equivalence of pressure loss through various flow paths sets the flow proportions beyond question if leakage areas and discharge coefficients are known. Blockage of flow paths by progressive fouling, when it can be estimated, leads to new flow distributions which, however, are still determinate. The same cannot be said for heat-transfer coefficients, upon which more is said later.

Any allowance for the effect of blockage of flow paths on pressure loss should not be thrown empirically into the first term of the denominator of the expression for F_p . Hydrodynamic considerations too lengthy for reproduction here show that the appropriate form is

¹ Chief Engineer, The Griscom-Russell Company, Massillon, Ohio. Mem. ASME.

² "Heat Transfer and Fluid Friction During Flow Across Banks of Tubes—VI. The Effect of Internal Leakages Within Segmentally Baffled Exchangers," by O. P. Bergelin, K. J. Bell, and M. D. Leighton, published in this issue, pp. 53-60.

$$F_p = \frac{(l_b/D_h)}{(l_b/D_h) + \phi(D_h/p)}$$

or

$$F_p = \frac{1}{\left[1 + \frac{D_h}{l_b} \phi\left(\frac{D_h}{p}\right)\right]}$$

This being the case, substitution of 0.8 for 1.0 in the denominator is a purely artificial device for increasing the computed pressure loss above its expected value for nonfouled conditions. The term (l_b/D_h) is essentially a measure of the ratio of cross-flow area to baffle-window area, while the term $\phi(D_h/p)$ includes, among other things, the leakage areas, and is of such a nature that it should approach zero as leakage paths approach complete blockage. Since F_p obviously cannot exceed unity, the proper expression should have any empirical modification for fouling allowance as a multiplier on the second term in the denominator (see also the author's Equation [1a]).

With the foregoing in mind, the writer's basic objection to the author's approach is twofold:

1 Any introduction of empiricism should not be in the expression for F_p , which is determinate on the basis of presently available knowledge, but rather in F_h .

2 There is no sound reason to believe that the complex mechanism of heat transfer in the "cross-flow zones," involving cross-flow, orifice flow, and axial flow, should correlate on one of these mechanisms only; namely, pure crossflow.

The writer's company, like many others in the past, has designed and operated many heat exchangers on the basis of pure orifice flow; i.e., on deliberate leakage through the tube-baffle areas. The shell-side heat-transfer coefficients obtainable with clean fluids are of the same order of magnitude as those for cross-flow baffling with the same pressure loss. Thus, any loss in cross-flow heat-transfer coefficient due to orifice leakage tends to be compensated by an improvement in orifice heat-transfer coefficient. The author's approach, if carried to the extreme of very small baffle windows, would predict coefficients approaching zero, since most of the shell-side flow would pass via orifices. This, obviously, is not what would happen since a respectable orifice-flow heat-transfer coefficient could be obtained.

Inasmuch as the interaction of crossflow and orifice-leakage streams with respect to effect on heat-transfer coefficient is still

indeterminate on the basis of present knowledge, it remains to be demonstrated to the writer that departure from the purely empirical but simple geometric-mean-velocity method of Donahue is warranted. Such departure appears least of all to be warranted when it is in the direction of tying heat-transfer coefficients (in the "crossflow zone") directly and solely to the cross-flow stream velocity.

H. B. NOTTAGE.⁵ It is in order to raise a question concerning the possible limits of application for this most interesting and valuable paper. It is believed that the data basic to this paper were generally obtained with liquids such as water and oil. Would these data also apply for a liquid such as mercury, which does not wet steel surfaces? Also, can any general assurance be given that the rating system will apply generally to liquid metals?

⁵ Project Manager, Propulsion Research Corporation, Santa Monica, Calif. Mem. ARME.

AUTHOR'S CLOSURE

Dr. M. D. Leighton's oral discussion with slides showing comparisons of University of Delaware test results with his performance predictions based on the author's rating system was very timely and convincing. Two heat exchanger units of commercial size were compared, one with large baffle hole and clearances and leakage, and the other with tight fitting baffles and negligible leakage. In the six performance points comparing the heat-transfer coefficient prediction against with test results within seven per cent on the average. The pressure drop values were in all cases somewhat less than the predicted heat exchanger values which, Dr. Leighton stated, appear reasonable since the exchangers were in a very clean condition when tested.

The author's 1951 London shell side papers represent the first work to recognize the importance of the fluid flow pattern on heat exchanger performance and to develop analytical procedures for evaluating the fluid flow distribution on the shell side of a heat exchanger. The present paper employs these techniques to carry the analysis a bit farther and to derive basic formulas for readily calculating the fraction of the total fluid that flows through the cross flow area within the limits of the tube bundle. This flow fraction evaluation is the key to the simple rating system outlined in the paper. An amplification of certain steps outlined at the beginning of the paper under primary premises and in further detail in the derivations, may clarify a number of points of misunderstanding indicated in Mr. Gardner's discussion.

Fluid Flow Fraction

The flow fraction is determined by considering the effect of hydraulic resistances and dimensions for the various flow routes through the exchanger shell have on the relative flow quantities through these various routes. The use of a flow fraction determined in this manner obviously will give much more accurate velocity and performance predictions than are possible with purely empirical methods that neglect the influence of certain clearances, tube arrangements, pitches, and so forth, on the fraction of the total fluid flowing through the tube bundle. The formula for flow fraction (F) determined as outlined is expressed in terms of the basic dimensions and clearances built into the shell side of the heat exchanger and is as indicated in Equations [3] and [9]. This relationship may also be represented as follows:

$$F = \frac{1}{1 + \sqrt{\frac{D_h}{p} \left[\left(f_{CR} f_1 \frac{p}{d_1} \right) + \left(f_{CR} f_2 \frac{p}{d_2} f_3 \frac{D_h}{l_b} \right) + \left(f_{CR} f_1 \frac{p}{d_2} f_3 \frac{D_h}{l} \right) \right]}} \quad [18]$$

The bracket in the denominator is referred to in the paper as the rating number N whose three elements are proportional to the relative magnitude of the leakage and bypass streams as follows:

$f_{CR} f_1 \left(\frac{p}{d_1} \right)$ is proportional to the tube bundle bypass current (Q_{CR})

$f_{CR} f_2 \left(\frac{p}{d_2} \right) f_3 \left(\frac{D_h}{l_b} \right)$ is proportional to baffle tube hole leakage (Q_{AR})

$f_{CR} f_1 \left(\frac{p}{d_2} \right) f_3 \left(\frac{D_h}{l} \right)$ is proportional to the leakage between the baffle edge and the shell (Q_{RR})

Equation [18] can be rewritten in a very simple form to give a clearer picture as follows:

$$F = \frac{1}{1 + \sqrt{S[fQ_{CR} + fQ_{AR} + fQ_{RR}]}} \quad [19]$$

Flow Fraction for Heat Transfer

The numerical value of the flow fraction F_c for a clean unit which may be used in selecting a clean heat-transfer coefficient is obtained by inserting the dimensions of the unit in the denominator of the flow fraction (F_c) formula (Equation (40)) or Equation (41).

Flow Fraction for Pressure Drop

Clean Heat Exchanger. If the exchanger is absolutely clean, the actual dimensions and clearances would govern the fluid flow distribution, and the flow fraction for predicting pressure drop through the tube nest would be the same as that used in selecting the clean heat-transfer coefficient, thus:

$$\text{Clean } F_c = F_p = \frac{1}{1 + \sqrt{S}(\dot{Q}_{c,K} + \dot{Q}_{h,K} + \dot{Q}_{j,K})}$$

Fouled Heat Exchanger. Fouling may be in various forms, distribution, and degree. Here the judgment of the designer comes into play in evaluation for different classes of service. It can be assumed that one or both of the leakage routes (baffle holes, baffle edges) could become partially or totally plugged. For example, if we assume the baffle holes completely plugged with dirt, and the hole leakage area and the flow reduced to zero, then $\dot{Q}_{h,K} = 0$, and the flow-fraction Equation (40) reduces to:

$$F_c = \frac{1}{1 + \sqrt{S}(\dot{Q}_{c,K} + \dot{Q}_{j,K})} \quad (50)$$

If we take this unit fouled, as just suggested, and further consider that dirt coatings on the tubes, fins, rods, and so forth, restrict the area of the remaining flow passages to 80 per cent of normal area due, then, from the standpoint of pressure drop, the unit would act as if the fluid flow rate through the tube bundle had been increased to $\frac{1}{0.80} = 1.25$ times the Equation (50) value and could thus be represented as:

$$\text{Fouled } F_c = \frac{1.25}{1 + \sqrt{S}(\dot{Q}_{c,K} + \dot{Q}_{j,K})} \quad (51)$$

It may be reduced to:

$$\text{Fouled } F_c = \frac{1.0}{0.8 + \sqrt{S} \left[\frac{\dot{Q}_{c,K}}{1.25} + \frac{\dot{Q}_{j,K}}{1.25} \right]} \quad (52)$$

This latter equation is of the form and value suggested by Equations (43) and (44) and is safe for most exchangers with a reasonable amount of fouling.

It is, of course, obvious that no single flow-fraction adjustment could accurately describe bundle flow-fraction and pressure-drop conditions for all different types and degrees of fouling. Considering Equation (48) or Equation (49) in its elements, we have a flexible system where we can predict the effect on the bundle flow rate for any practical or reasonable assumption as to the diminution of any leakage stream or flow area that could result from fouling.

Adjusted flow fractions, as suggested herein, for fouled pressure-drop evaluation are not intended to be used for the selection of fouled heat-transfer coefficients since it is recommended that clean coefficients be used in conjunction with appropriate fouling resistances such as published in the TEMA Standards.

Heat-Transfer Coefficient Premises

The author's approach does not tie the heat-transfer coefficient solely to the cross-flow stream velocity, but in the cross-flow

space between baffles it is assumed the coefficient will agree with the Delaware research values for pure crossflow, and in the baffle window zone it is assumed that the coefficient will correspond to a cross-flow coefficient taken at the geometric mean of the cross-flow velocity between baffles and the parallel-flow velocity through the baffle window. Appropriate multipliers are developed to properly weight the respective transfer surfaces and coefficients of these zones for arriving at the over-all coefficient.

Heat Exchanger Design

The author's analysis did not consider heat exchangers with windowless baffles, but was derived for segmentally baffled heat exchangers designed with the baffle window graduated with baffle spacing to give window areas of the same general magnitude as the cross-flow areas. For these practical design proportions, as specified throughout the paper, the performance predictions line up very satisfactorily with laboratory performance test results.

Table 5 shows clearance constants and baffle window cut proportions for a typical line of clamp ring floating head-type heat exchangers.

TABLE 5

	D_1 in.	H D_1	c_2 (cent away)
$c_1 = \frac{D}{D_1} = 1.075$	1	0.46	0.45
	1.5	0.34	0.30
$c_3 = \frac{d_1 - d}{d_1} = 0.045$	2	0.25	0.20
	3	0.20	0.15
$c_4 = \frac{D_1 - D_2}{D_1} = 0.008$	4	0.16	0.10
	5	0.16	0.10

Table 6 shows typical design dimensions for a 25-in. $\times$ 240-in. heat exchanger of the same type.

TABLE 6

$P = 0.075$	$c_1 = \frac{D_1}{D_2} = \frac{25.00}{23.25} = 1.075$
$d_1 = 0.784$	$c_3 = \frac{d_1 - d}{d_1} = \frac{0.031}{0.75} = 0.045$
$d_2 = 0.75$	$c_4 = \frac{D_1 - D}{D_1} = \frac{0.20}{25.0} = 0.008$
$D_1 = 25.00$	$P = d$
$D = 23.25$	$P = 1.25$
$d_1 = 240$	$\frac{P - d}{P} = 0.20$
$d = 12.5$	
$D = 2.0$	$\sqrt{S} = \frac{240}{12.5} = 19$
$H = 0.25$	$S = \frac{D_1}{P} = \frac{25}{1.25} = 20.0$

The actual rating of the heat exchanger shown in Table 6 is very simple using the rating numbers calculated and tabulated on the design curves of this paper for exchangers of the indicated typical proportions.

Table 7 illustrates this typical rating example.

TABLE 7

Use Fig. 1 and Fig. 2 with tabulated constants
Heat Transfer, Fig. 1

$$\begin{aligned} V_c &= c_1 V_1 D_1 = 0.00135 \times 12.5 \times 23.25 = 0.392 \\ F_c &= \frac{1}{1 + \sqrt{S} \sqrt{S}} = \frac{1}{1 + (0.54 \times \sqrt{20.0})} = 0.264 \\ M_c &= m \sqrt{c_1} = 0.85 \sqrt{1.075} = 0.88 \quad M = 0.96 \text{ (Fig. 10)} \\ R_{c1} &= \frac{d_1 F_c \text{ lb/hr}}{20 M V_c} = \frac{0.75 \times 0.264 \text{ (lb/hr)}}{20 \times 0.96 \times 0.392 \text{ (Z)}} = 0.0181 \quad \text{Z} \end{aligned}$$

Pressure Loss, Fig. 2

$$F_s = \frac{1}{0.80 + N_s \sqrt{S}} = \frac{1}{0.80 + (0.30) \sqrt{26.6}} = 0.427$$

$$Re_s = \frac{d_i F_s \text{ lb/hr}}{29 Z \mu_s} = \frac{0.75 \times 0.427 \text{ (lb/hr)}}{29 \times 0.392 Z} = 0.0282 \frac{\text{(lb/hr)}}{Z}$$

Select f_s at Re_s

$$Y = \frac{y}{(100)^{0.25}} = \frac{5.0}{(1.075)^{0.25}} = 4.7$$

$$\Delta P = 0.334 \times 0.87 \times 26.6 \times \frac{19 f_s}{\rho \phi} \left(\frac{0.427 \text{ lb/hr}}{0.392 \text{ in}^2} \right)^2 \left[1 + \frac{4.7}{26.6} \right]$$

$$\Delta P = \frac{147 f_s}{\rho \phi} \left(\frac{0.427 \text{ lb/hr}}{0.392 \text{ in}^2} \right)^2 (1 + 0.177)$$

If a unit to be rated has clearance constants differing widely from the typical values shown in the example, then rating numbers N_s and N_p for clean and fouled conditions would be calcu-

lated using Equations [10] and [15], and the clean and fouled flow fractions F_s and F_p would then be determined as indicated by Equations [3] and [13].

Liquid Metals

With regard to Mr. Nottage's inquiry concerning the use of the rating system for liquid metal applications, the author believes it would give satisfactory indications of the tubesheet heat transfer.

If the heat-transfer coefficient behavior of the liquid metal with respect to cross-flow and baffle window parallel-flow velocities, agreed with the heat-transfer premises noted in the paper and in the preceding Closure, then heat-transfer predictions for liquid metal should be valid. For pressure-drop considerations it is possible that the flow fraction should be taken much below the clean value than is suggested for heat exchangers in the fouling service.

Heat Transfer and Fluid Friction During Flow Across Banks of Tubes—VI

The Effect of Internal Leakages Within Segmentally Baffled Exchangers

BY O. P. BERGELIN,¹ K. J. BELL,² AND M. D. FRIGHTON³

An experimental study has been made of the leakage past segmental baffles within tubular heat exchangers with no fouling. Orifice coefficients are given which will permit calculation of the amount of leakage if the pressure difference across the baffle is known. For the case of constant flow of the main stream in the exchanger, the effect of internal leakage upon pressure drop and heat transfer is shown by comparing experimental results from exchangers with known internal clearances with the results from an exchanger with no internal clearances. Internal clearances comparable to those encountered in commercial practice were found to lower the heat-transfer coefficient to around 55 per cent and the pressure drop to around 30 per cent of the values without internal leakage. Graphs are presented which will aid in estimating the reduction in pressure drop and heat transfer resulting from internal leakages.

NOMENCLATURE

The following nomenclature is used in the paper:

- A = heat-transfer area, sq ft
- C = orifice coefficient, dimensionless
- C_L = weighted orifice coefficient, $\frac{S_L C_L + S_T C_T}{S_L + S_T}$, dimensionless
- D = hole diameter, ft
- d = tube diameter, ft
- f = friction factor of Chilton and Colburn
- $f = \frac{2g\rho S_L^2 \Delta P_R}{18W_R^2} \left(\frac{\mu}{\mu_c} \right)^{0.14}$, dimensionless
- G = mass velocity, lb/(sq ft)(hr)
- q = dimensional constant (Btu)/(ft²)(lb_m/ft²)(hr)^{0.7}
- h = average heat-transfer coefficient for entire exchanger, Btu/(hr)(sq ft)(deg F)
- K = constant
- L = length, ft
- N = number of rows of tubes in crossflow
- N = effective restrictions in crossflow and window

$$\Delta P = \Delta P_R \left(1 + \frac{\Delta P_R}{\Delta P_R} \right)$$

ΔP_R = pressure difference along main flow streams, psi

δp = average pressure difference across one baffle

$$\delta p = \Delta P_R + \frac{3}{2} \Delta P_R, \text{ psi}$$

S = cross-sectional area for flow, an average of minimum clearances for each row, sq ft

S_B = cross-sectional area for flow within bundle, not including bundle bypass stream, sq ft

S_L = total cross-sectional area for flow, $S_L = S_B + S_P$

S_T = weighted leakage area per baffle $S_L = S_L + S_T$

V = velocity, fpm

V_B = velocity at average minimum area of crossflow section S_L based on no leakage, fpm

$$V_B = \frac{W}{3600\rho S_L}$$

W = weight rate of flow, lb/hr

Z = orifice shape factor = $\frac{\text{thickness of orifice plate}}{\text{width of orifice slot}}$

$$Z = \frac{2L}{D - d}$$

ρ = density, pcf

μ = viscosity, lb/hr ft

Subscripts

- B = for one crossflow section within bundle
- L = with leakage
- NL = nonleakage
- o = overall value for exchanger
- P = bypass stream
- S = between baffle and shell
- s = surface
- T = between baffle and tube
- t = total for crossflow
- W = for one window section

INTRODUCTION

In previous papers, results were reported on the pressure drop and the rate of heat transfer during flow across tube banks (1, 2)⁴ and for flow on the shell side of a cylindrical baffled heat exchanger without appreciable internal leakage (3). The results for the baffled heat exchanger were shown to be consistent with results for flow across tube banks, and a method was proposed for predicting the performance on the shell side of baffled exchangers by making use of the crossflow data of the simple models.

The present paper presents information on leakage through the

⁴ Numbers in parentheses refer to the Bibliography at the end of the paper.

¹ Professor of Chemical Engineering, University of Delaware, Newark, Del.; presently at University of Dacca, Dacca, East Pakistan. Mem. ASME.

² Assistant Professor of Chemical Engineering, Case Institute of Technology, Cleveland, Ohio; formerly, Research Fellow in Chemical Engineering, University of Delaware, Newark, Del.

³ Research Fellow, Chemical Engineering Department, University of Delaware, Newark, Del.

Contributed by the Heat Transfer Division and presented at the Annual Meeting, New York, N. Y., November 25-30, 1956, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

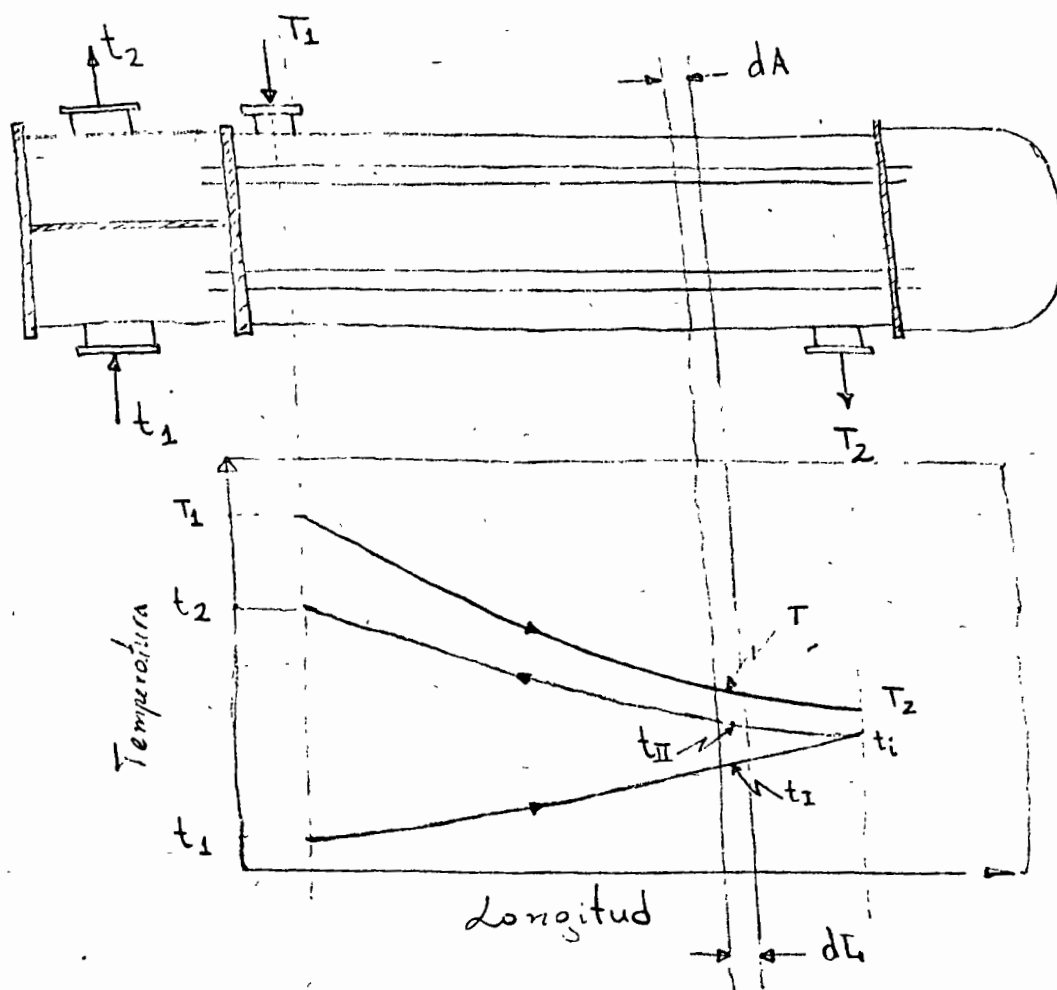
NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, October 2, 1956. Paper No. 56-A-133.

Close para el 8 de Mayo

19

Diferencia de temperaturas para un cambiador

en 2 pas



Balance de calor, y transferencia en base a una ΔT_c

$$Q = UA \Delta T_c = W C_p (T_1 - T_2) = w c_p (t_2 - t_1) \quad \text{--- (1)}$$

$$\Delta T_c = \frac{T_1 - T_2}{\left(\frac{UA}{WC_p} \right)} = \frac{t_2 - t_1}{\left(\frac{UA}{wc_p} \right)} \quad \text{--- (2)}$$

Tomando un elemento diferencial de longitud (dL) en el cambiador de calor, al que corresponderá un elemento diferencial de área (dA), y una transferencia de calor diferencial (dQ).

Balances de calor en el elemento diferencial:

20

$$dq = -WC_p dT = U \frac{dA}{2} (T - t_I) + U \frac{dA}{2} (T - t_{II}) \quad (3)$$

$$WC_p dt_I = U \frac{dA}{2} (T - t_I) \quad (4)$$

$$-WC_p dt_{II} = U \frac{dA}{2} (T - t_{II}) \quad (5)$$

Rearreglando la ecuación (3)

$$-WC_p dT = U dA \left(T - \frac{t_I + t_{II}}{2} \right) \quad (6)$$

$$-\frac{dT}{dA} = \frac{U}{WC_p} \left(T - \frac{t_I + t_{II}}{2} \right) \quad (7)$$

$$\frac{dT}{dA} + \frac{U}{WC_p} \left(T - \frac{t_I + t_{II}}{2} \right) = 0 \quad (8)$$

Definiendo:

$$R = \frac{WC_p}{WC_p} = \frac{T_1 - T_2}{t_2 - t_1} \quad (9)$$

De (9):

$$\frac{1}{WC_p} = \frac{R}{WC_p} \quad (10)$$

Substituyendo (10) en (8)

$$\frac{dT}{dA} + \frac{UR}{WC_p} \left(T - \frac{t_I + t_{II}}{2} \right) = 0 \quad (11)$$

Derivando con respecto al area la ecuación (11) ^{2/}

$$\frac{d^2 T}{dA^2} + \frac{UR}{wC_p} \frac{dT}{dA} - \frac{UR}{2wC_p} \left(\frac{dt_I}{dA} + \frac{dt_{II}}{dA} \right) = 0 \quad (12)$$

De las ecuaciones (4) y (5) :

$$\frac{dt_I}{dA} = \frac{U}{2wC_p} (T - t_I) \quad (13)$$

$$\frac{dt_{II}}{dA} = -\frac{U}{2wC_p} (T - t_{II}) \quad (14)$$

$$\frac{dt_I}{dA} + \frac{dt_{II}}{dA} = \frac{U}{2wC_p} (t_{II} - t_I) \quad (15)$$

Substituyendo (15) en (12)

$$\frac{d^2 T}{dA^2} + \frac{UR}{wC_p} \frac{dT}{dA} - \frac{U^2 R}{(2wC_p)^2} (t_{II} - t_I) = 0 \quad (16)$$

Balance de calor :

$$WC_p (T - T_2) = wC_p (t_{II} - t_I) \quad (17)$$

$$t_{II} - t_I = \frac{T - T_2}{\frac{wC_p}{WC_p}} \quad (18)$$

Substituyendo (9) en (18)

$$t_{II} - t_I = \frac{T - T_2}{R} \quad (19)$$

Substituyendo (19) en (16) :

$$\frac{d^2 T}{dA^2} + \frac{UR}{\omega C_p} \frac{dT}{dA} - \frac{U^2 R}{(2\omega C_p)^2} \frac{T - T_2}{R} = 0 \quad (20)$$

$$\frac{d^2 T}{dA^2} + \frac{UR}{\omega C_p} \frac{dT}{dA} - \frac{U^2}{(2\omega C_p)^2} T + \frac{U^2 T_2}{(2\omega C_p)^2} = 0 \quad (21)$$

Derivando con respecto al área la ecuación (21)

$$\frac{d^3 T}{dA^3} + \frac{UR}{\omega C_p} \frac{d^2 T}{dA^2} - \frac{U^2}{(2\omega C_p)^2} \frac{dT}{dA} = 0 \quad (22)$$

Para resolver ecuación (22)

hacemos $T = e^{rA}$

$$\frac{d^3 T}{dA^3} = r^3 e^{rA}$$

$$\frac{d^2 T}{dA^2} = r^2 e^{rA}$$

$$\frac{dT}{dA} = r e^{rA}$$

$$\left[r^3 + \frac{UR}{\omega C_p} r^2 - \frac{U^2}{(2\omega C_p)^2} r \right] e^{rA} = 0$$

$$r \left(r^2 + \frac{UR}{\omega C_p} r - \frac{U^2}{(2\omega C_p)^2} \right) = 0$$

$$r_1 = 0$$

$$r_2 = -\frac{UR}{2\omega C_p} + \sqrt{\frac{U^2 R^2}{(2\omega C_p)^2} + \frac{U^2}{(2\omega C_p)^2}}$$

$$r_3 = -\frac{UR}{2\omega C_p} - \sqrt{\frac{U^2 R^2}{(2\omega C_p)^2} + \frac{U^2}{(2\omega C_p)^2}}$$

$$r_2 = -\frac{UR}{2\omega C_p} + \frac{U}{2\omega C_p} \sqrt{R^2 + 1}$$

$$r_3 = -\frac{UR}{2\omega C_p} - \frac{U}{2\omega C_p} \sqrt{R^2 + 1}$$

$$r_1 = 0$$

$$r_2 = -\frac{U}{2\omega C_p} (R + \sqrt{R^2 + 1})$$

$$r_3 = -\frac{U}{2\omega C_p} (R - \sqrt{R^2 + 1})$$

$$T = k_1 e^{r_1 A} + k_2 e^{r_2 A} + k_3 e^{r_3 A}$$

$$T = k_1 + k_2 e^{-\frac{UA}{2\omega C_p} (R + \sqrt{R^2 + 1})} + k_3 e^{-\frac{UA}{2\omega C_p} (R - \sqrt{R^2 + 1})} \quad (23)$$

Quand $A \rightarrow \infty$ $T \rightarrow T_2$

$$T_2 = k_1 + k_2 e^{-\infty} + k_3 e^{-\infty} \quad (24)$$

$$T_2 = k_1 \quad (25)$$

Quand A croît de 0 à A $T \rightarrow T_2$

$$T_2 = T_2 + k_2 e^{-\frac{UA}{2\omega C_p} (R + \sqrt{R^2 + 1})} + k_3 e^{-\frac{UA}{2\omega C_p} (R - \sqrt{R^2 + 1})} \quad (26)$$

$$-K_2 C e^{-\frac{UA}{2\omega_{Cp}} (R + \sqrt{R^2 + 1})} = K_3 e^{-\frac{UA}{2\omega_{Cp}} (R - \sqrt{R^2 + 1})} \quad (27)$$

Tomando logaritmos en la ecuación (27)

$$\ln(-K_2) - \frac{UA}{2\omega_{Cp}} (R + \sqrt{R^2 + 1}) = \ln K_3 - \frac{UA}{2\omega_{Cp}} (R - \sqrt{R^2 + 1}) \quad (28)$$

$$\ln\left(-\frac{K_2}{K_3}\right) = \frac{UA}{\omega_{Cp}} \sqrt{R^2 + 1} \quad (29)$$

$$\frac{UA}{\omega_{Cp}} = \frac{1}{\sqrt{R^2 + 1}} \ln\left(-\frac{K_2}{K_3}\right) \quad (30)$$

Derivando la ecuación (23) con respecto al área.:

$$\begin{aligned} \frac{dT}{dA} = & -K_2 \frac{U}{2\omega_{Cp}} (R + \sqrt{R^2 + 1}) e^{-\frac{UA}{2\omega_{Cp}} (R + \sqrt{R^2 + 1})} \\ & - K_3 \frac{U}{2\omega_{Cp}} (R - \sqrt{R^2 + 1}) e^{-\frac{UA}{2\omega_{Cp}} (R - \sqrt{R^2 + 1})} \end{aligned} \quad (31)$$

Despejando $\frac{dT}{dA}$ de la ecuación (11)

$$\frac{dT}{dA} = \frac{UR}{2\omega_{Cp}} (t_I + t_{II}) - \frac{UR}{\omega_{Cp}} T \quad (32)$$

Substituyendo (32) en (31)

$$\begin{aligned} \frac{UR}{2\omega_{Cp}} (t_I + t_{II}) - \frac{UR}{\omega_{Cp}} T = & -K_2 \frac{U}{2\omega_{Cp}} (R + \sqrt{R^2 + 1}) e^{-\frac{UA}{2\omega_{Cp}} (R + \sqrt{R^2 + 1})} \\ & - K_3 \frac{U}{2\omega_{Cp}} (R - \sqrt{R^2 + 1}) e^{-\frac{UA}{2\omega_{Cp}} (R - \sqrt{R^2 + 1})} \end{aligned}$$

Rearreglando ecuación (33)

25

$$R(t_I + t_{II}) - 2RT = -K_2(R + \sqrt{R^2 + 1}) e^{-\frac{UA}{2wC_p}(R + \sqrt{R^2 + 1})}$$

$$-K_3(R - \sqrt{R^2 + 1}) e^{-\frac{UA}{2wC_p}(R - \sqrt{R^2 + 1})} \quad (34)$$

Cuando $A \rightarrow 0$

$$\left. \begin{array}{l} t_I = t_1 \\ t_{II} = t_2 \\ T = T_1 \end{array} \right\} \quad (35)$$

$$R(t_1 + t_2) - 2RT_1 = -K_2(R + \sqrt{R^2 + 1}) - K_3(R - \sqrt{R^2 + 1}) \quad (36)$$

Substituyendo (35) en ecuación (23)

$$T_1 = T_2 + K_2 + K_3 \quad (37)$$

$$T_1 - T_2 = K_2 + K_3 \quad (38)$$

Multiplcando ecuación (38) por $(R + \sqrt{R^2 + 1})$

$$(T_1 - T_2)(R + \sqrt{R^2 + 1}) = K_2(R + \sqrt{R^2 + 1}) + K_3(R + \sqrt{R^2 + 1}) \quad (39)$$

Sumando ecuaciones (36) y (39)

$$R(t_1 + t_2) - 2RT_1 + (T_1 - T_2)(R + \sqrt{R^2 + 1}) = 2K_3\sqrt{R^2 + 1} \quad (40)$$

$$K_3 = \frac{R(t_1 + t_2) - 2RT_1 + (T_1 - T_2)(R + \sqrt{R^2 + 1})}{2\sqrt{R^2 + 1}} \quad (41)$$

De la ecuación (38)

$$-K_2 = K_3 - (T_1 - T_2) \quad (42)$$

$$-\frac{K_2}{K_3} = 1 - \frac{T_1 - T_2}{K_3} \quad (43)$$

$$-\frac{K_2}{K_3} = \frac{K_3 - (T_1 - T_2)}{K_3} \quad (44)$$

Substituyendo (41) en (44)

$$-\frac{K_2}{K_3} = \frac{\frac{R(t_1+t_2) + (T_1-T_2)(R+\sqrt{R^2+1}) - 2RT_1}{2\sqrt{R^2+1}} - (T_1-T_2)}{\frac{R(t_1+t_2) + (T_1-T_2)(R+\sqrt{R^2+1}) - 2RT_1}{2\sqrt{R^2+1}}} \quad (45)$$

$$-\frac{K_2}{K_3} = \frac{R(t_1+t_2) + (T_1-T_2)(R+\sqrt{R^2+1}) - 2RT_1 - 2\sqrt{R^2+1}(T_1-T_2)}{R(t_1+t_2) + (T_1-T_2)(R+\sqrt{R^2+1}) - 2RT_1} \quad (46)$$

$$-\frac{K_2}{K_3} = \frac{R(t_1+t_2) - R(T_1+T_2) - \sqrt{R^2+1}(T_1-T_2)}{R(t_1+t_2) - R(T_1+T_2) + \sqrt{R^2+1}(T_1-T_2)} \quad (47)$$

$$-\frac{K_2}{K_3} = \frac{R(t_1-T_1) - R(T_2-t_2) - \sqrt{R^2+1}(T_1-T_2)}{R(t_1-T_1) - R(T_2-t_2) + \sqrt{R^2+1}(T_1-T_2)} \quad (48)$$

Definiendo:

$$S = \frac{t_2 - t_1}{T_1 - t_1} \quad (19)$$

Dividiendo ecuación (48) entre $t_2 - t_1$

$$-\frac{K_2}{K_3} = \frac{-\frac{R}{S} - R \frac{T_2 - t_2}{t_2 - t_1} - \sqrt{R^2 + 1} R}{-\frac{R}{S} - R \frac{T_2 - t_2}{t_2 - t_1} + \sqrt{R^2 + 1} R} \quad (50)$$

$$-\frac{K_2}{K_3} = \frac{-R - RS \frac{T_2 - t_2}{t_2 - t_1} - \sqrt{R^2 + 1} RS}{-R - RS \frac{T_2 - t_2}{t_2 - t_1} + \sqrt{R^2 + 1} RS} \quad (51)$$

$$-\frac{K_2}{K_3} = \frac{-1 - S \frac{T_2 - t_2}{t_2 - t_1} - S \sqrt{R^2 + 1}}{-1 - S \frac{T_2 - t_2}{t_2 - t_1} + S \sqrt{R^2 + 1}} \quad (52)$$

$$-\frac{K_2}{K_3} = \frac{1 - S \left(\frac{t_2 - T_2}{t_2 - t_1} - \sqrt{R^2 + 1} \right)}{1 - S \left(\frac{t_2 - T_2}{t_2 - t_1} + \sqrt{R^2 + 1} \right)} \quad (53)$$

$$-\frac{K_2}{K_3} = \frac{2 - S \left(\frac{t_2 - T_2}{t_2 - t_1} + \frac{1}{S} - \sqrt{R^2 + 1} \right)}{2 - S \left(\frac{t_2 - T_2}{t_2 - t_1} + \frac{1}{S} + \sqrt{R^2 + 1} \right)} \quad (54)$$

$$\frac{t_2 - T_2}{t_2 - t_1} + \frac{1}{S} = \frac{t_2 - T_2}{t_2 - t_1} + \frac{T_1 - t_1}{t_2 - t_1} \quad (55)$$

$$R+1 = \frac{T_1 - T_2}{t_2 - t_1} + 1 \quad (56)$$

$$R+1 = \frac{T_1 - T_2 + t_2 - t_1}{t_2 - t_1} \quad (57)$$

$$R+1 = \frac{t_2 - T_2}{t_2 - t_1} + \frac{T_1 - t_1}{t_2 - t_1} \quad (58)$$

Substituyendo (58) en (55)

$$\frac{t_2 - T_2}{t_2 - t_1} + \frac{1}{S} = R+1 \quad (59)$$

Substituyendo (59) en (54)

$$-\frac{K_2}{K_3} = \frac{2 - S(R+1 - \sqrt{R^2+1})}{2 - S(R+1 + \sqrt{R^2+1})} \quad (60)$$

Substituyendo (60) en (30)

$$\frac{UA}{\omega C_p} = \frac{1}{\sqrt{R^2+1}} \ln \frac{2 - S(R+1 - \sqrt{R^2+1})}{2 - S(R+1 + \sqrt{R^2+1})} \quad (61)$$

Substituindo 61 em (2)

27

$$\Delta T_c = \frac{t_2 - t_1}{\frac{1}{\sqrt{R^2 + 1}} \ln \frac{2 - S(R+1 - \sqrt{R^2 + 1})}{2 - S(R+1 + \sqrt{R^2 + 1})}} \quad (62)$$

$$\Delta T = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \frac{T_1 - t_2}{T_2 - t_1}} \quad (63)$$

$$F_T = \frac{\Delta T_c}{\Delta T} = \frac{t_2 - t_1}{(T_1 - t_2) - (T_2 - t_1)} \cdot \frac{1}{\frac{1}{\sqrt{R^2 + 1}} \ln \frac{2 - S(R+1 - \sqrt{R^2 + 1})}{2 - S(R+1 + \sqrt{R^2 + 1})}} \quad (64)$$

$$\frac{t_2 - t_1}{(T_1 - t_2) - (T_2 - t_1)} = \frac{\ln \frac{T_1 - t_2}{T_2 - t_1}}{\frac{T_1 - T_2}{t_2 - t_1} - \frac{t_2 - t_1}{t_2 - t_1}} = \frac{\ln \frac{T_1 - t_2}{T_2 - t_1}}{R - 1} \quad (65)$$

$$\frac{T_1 - t_2}{T_2 - t_1} = \frac{T_1 + t_1 - t_2 - t_1}{T_2 + T_1 - T_1 - t_1} = \frac{(T_1 - t_1) - (t_2 - t_1)}{(T_1 - t_1) - (T_1 - T_2)} \quad (66)$$

$$\frac{T_1 - t_2}{T_2 - t_1} = \frac{1 - \frac{t_2 - t_1}{T_1 - t_1}}{1 - \frac{T_1 - T_2}{T_1 - t_1}} \quad (67)$$

$$RS = \frac{T_1 - T_2}{t_2 - t_1} \cdot \frac{t_2 - t_1}{T_1 - t_1} = \frac{T_1 - T_2}{T_1 - t_1} \quad (68)$$

Substituindo (49) y (68) em (67)

$$\frac{T_1 - t_2}{T_2 - t_1} = \frac{1 - S}{1 - RS} \quad (69)$$

Substituindo (69) em (65)

$$\frac{t_2 - t_1}{(T_1 - t_2) - (T_2 - t_1)} = \frac{\ln \frac{1 - S}{1 - RS}}{R - 1} \quad (70)$$

$$\ln \frac{T_1 - t_2}{T_2 - t_1}$$

Substituindo 70 em 64

$$F_T = \frac{\ln \frac{1 - S}{1 - RS}}{R - 1} \quad (71)$$

$$\frac{1}{\sqrt{R^2 + 1}} \ln \frac{2 - S(R + 1 - \sqrt{R^2 + 1})}{2 - S(R + 1 + \sqrt{R^2 + 1})}$$

$$F_T = \frac{\sqrt{R^2 + 1} \ln \frac{1 - S}{1 - RS}}{(R - 1) \ln \frac{2 - S(R + 1 - \sqrt{R^2 + 1})}{2 - S(R + 1 + \sqrt{R^2 + 1})}} \quad (72)$$

APLICACIONES DE LA TRANSFERENCIA DE CALOR EN PLANTAS

TERMoeLECTRICAS.

Ing. I. Becerril Salinas (CFE)

Ing. M. Aguilar Rodríguez (CFE)

I.- EQUIPOS PRINCIPALES Y CAMBIADORES DE CALOR.-

Se describen los ciclos termodinámicos con los equipos y cambiadores de calor que existen en una Planta Termoeléctrica, indicando la localización en diagramas de flujo, describiendo las funciones que realizan, los tipos de transferencia de calor y los problemas más comunes que se tienen. Los principales cambiadores de calor referidos son:

- 1.- Generadores de vapor.
- 2.- Turbogeneradores.
- 3.- Tuberías.
- 4.- Precalentadores de aire (los diferentes tipos)
- 5.- Calentadores de agua de alimentación.
- 6.- Calentadores de combustible.
- 7.- Economizadores.
- 8.- Evaporadores.
- 9.- Condensadores.

II.- DISEÑO, CONTROL Y PRUEBAS DE CONDENSADORES DE VAPOR DE AGUA DE SUPERFICIE.

Comprenderá básicamente la descripción y análisis de los siguientes aspectos:

- 1.- Tipos de condensación (película, goteo y goteo-película).
- 2.- Coeficientes de transferencia térmica.

- 3.- Objetivos.
- 4.- Partes principales y sus funciones.
- 5.- Materiales más comunes empleados en la construcción.
- 6.- Selección de los parámetros principales.
- 7.- Diseño y optimización.
- 8.- Verificación y puesta en servicio.
- 9.- Control y operación.
- 10.- Comportamiento.
- 11.- Pruebas de comportamiento y de control.
- 12.- Forma en que afecta la economía el comportamiento.

III.- APEÑDICES.-

- 1.- Obtención fórmula cálculo presión en el condensador.
- 2.- Folleto de selección y diseño de Allis Chalmers.
- 3.- Procedimiento de pruebas de comportamiento de la --
Pacific Gas & Electric Co. de San Francisco, Calif.
- 4.- Pruebas de Lousiana Power & Light (Unidad de - - -
210 000 KW).
- 5.- Folleto de selección y diseño de Ingersoll Rand.
- 6.- Artículo ASME-Laminar Film Condensation of a Flowing Vapor
on a Horizontal Cylinder at Normal Gravity.
- 7.- Artículo APC-Economic sizing of condensers through the use
of the Digital Computer.

ING. I. BECERRIL S. - Parte I y Parte II puntos 1,2,3,6,7 y 12.

ING. M. AGUILAR R.- Parte II puntos 4,5,8,9,10 y 11.

APLICACIONES DE TRANSFERENCIA DE CALOR EN PLANTAS TERMoeLECTRICAS

En las Plantas Termoeléctricas se tiene la aplicación de todas las formas básicas de transferencia de calor, es decir, conducción, convección y radiación, en los aparatos más importantes y operando a altas -- temperaturas, como los Generadores de Vapor y algunos calentadores de alta presión y temperatura, se tiene la combinación de radiación-convección-conducción, pero en la mayoría de equipos, aparatos e intercambiadores de calor que operan a niveles de temperatura bajos, se tiene únicamente transferencia de calor por conducción y convección.

Los problemas de transferencia de calor más importantes que generalmente se observan en los principales equipos son los siguientes:

Generador de Vapor.-

En el Generador de Vapor, se tiene conversión de energía del combustible por reacciones químicas y debido a la alta temperatura de la flama, superior a 3000°F en los quemadores, y alta temperatura de los -- gases, superior a 2400°F, se tiene un alto porcentaje de transferencia -- térmica por radiación a las paredes de agua, sobrecalentadores y recalentadores de vapor, por radiación y convección natural al medio ambiente, en adición se tienen fenómenos de transferencia térmica por convección- de los gases a las paredes metálicas y por conducción y convección a los fluidos circulando en el interior de las tuberías. Con objeto de aumentar la eficiencia de los generadores de vapor, se trata de transferir el calor de los gases al vapor y agua de alimentación, hasta donde sea práctica

y económicamente recomendable, para lo cual se instalan en el paso de gases los sobrecalentadores y recalentadores, los economizadores y finalmente los precalentadores de aire.

En los generadores de vapor modernos se trata de aumentar la cantidad de calor transferido optimizando la superficie y aumentando los coeficientes de transferencia térmica con mayor turbulencia y condiciones de ebullición nucleada.

Turbogenerador.-

El turbogenerador puede considerarse como una máquina adiabática con respecto al medio ambiente, sin que ello implique que no queda sujeto a varias formas de transferencia de energía térmica, en las zonas más delicadas y sujetas a altos esfuerzos mecánicos y térmicos. El problema más serio en los turbogeneradores se debe a variaciones de las condiciones del vapor en cuanto a presión, temperatura y velocidad que origina diferencias en los coeficientes de transferencia térmica y en las temperaturas de las superficies de los metales, causando con ello esfuerzos térmicos al ocurrir cambios en una zona concentrada. Los efectos de cambios de condiciones del vapor y de cambios de carga dan lugar a variaciones cíclicas de condiciones de temperatura y por lo tanto a variaciones cíclicas de esfuerzos térmicos que disminuyen la vida de los elementos y se manifiestan en forma de fracturas de diversa índole.

Las partes más sujetas a este tipo de esfuerzo y peligro, son los álabes, los rotores, las carcassas en las zonas de alta presión y temperatura, adicionalmente las formas irregulares dan lugar a distribu -

ciones de temperatura que implican esfuerzos térmicos variables y en muchos casos de difícil predicción.

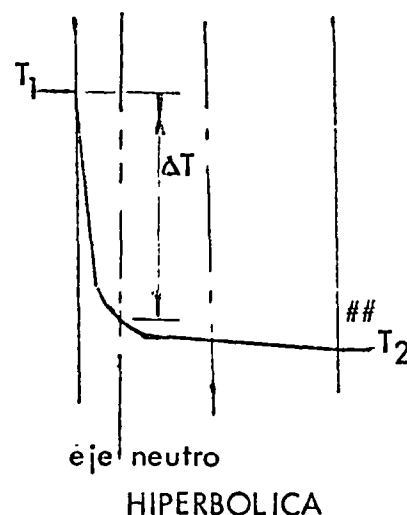
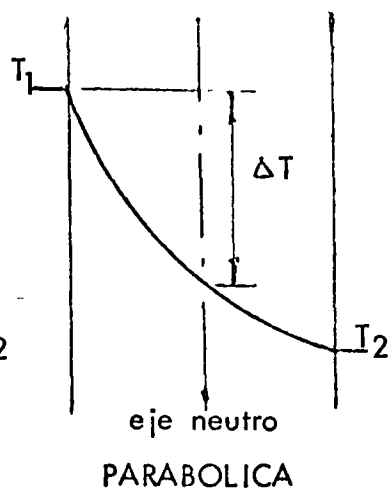
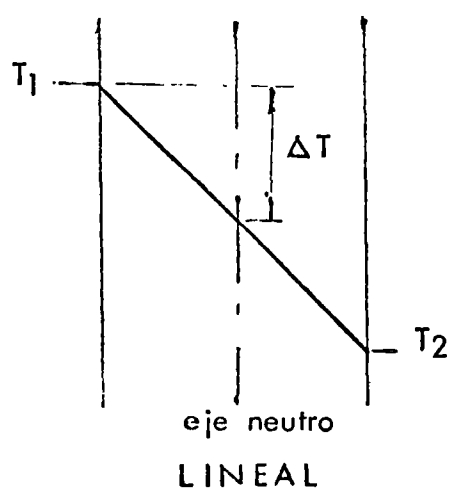
La forma de transmisión de calor en las turbinas es fundamentalmente por convección y conducción en las zonas de contacto con el fluido trabajando, es decir zona de distribución, válvulas de paro y de control, toberas y álabes, conductos de paso y carcassas, con cierta transmisión por radiación al ocurrir cambios bruscos de temperatura. En algunas zonas solo existe transmisión por conducción, como en las bridas de carcassas, pernos de sujeción de carcassas, vástagos de válvulas, en los cuales la distribución de temperaturas es muy irregular.

La variación de temperatura en forma súbita origina altos esfuerzos y una distribución diferente a la que existe en variación de temperatura lineal ó lenta.

$$\Delta T = \frac{T_1 - T_2}{2}$$

$$\Delta T = \frac{2}{3} \frac{T_1 - T_2}{2}$$

$$\Delta T \cong T_1 - T_2$$



TRANSFERENCIA TERMICA POR CONDUCCION EN ESTADO INESTABLE.

Esta es una de las formas de transferencia térmica que afecta más seriamente el diseño, el comportamiento y la vida útil de los equipos.

El cambio de temperatura en función del tiempo está determinada por la ecuación general:

$$\frac{\partial T}{\partial \theta} = \frac{1}{\rho c_p} \left[k_x \frac{\partial^2 T}{\partial x^2} + k_y \frac{\partial^2 T}{\partial y^2} + k_z \frac{\partial^2 T}{\partial z^2} \right] + \dot{q}_T'''$$

La solución de esta ecuación diferencial, para diferentes formas geométricas y condiciones límite ha sido favorita del campo de actividad matemático desde los tiempos de Fourier.

Las series de Fourier y los métodos de Fourier, son hasta la fecha muy utilizados para problemas de esta naturaleza.

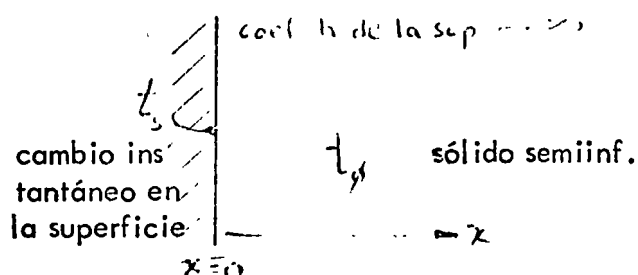
Si no hay generación térmica ni absorción dentro del cuerpo, $\dot{q}_T''' = 0$, la ecuación se reduce a:

$$\frac{\partial T}{\partial \theta} = \alpha \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right] = \alpha \nabla^2 T$$

Esta ecuación tiene diferentes soluciones particulares cuya combinación conduce a una solución de tipo general:

$$T = A + Bx + Cy + Dz + E e^{kx + \lambda y + \mu z + \nu \theta}$$

Que, en un sólido semi-infinito en el cual se tenga una temperatura inicial T_0 uniforme en todo el sólido y esta es cambiada en forma instantánea en la superficie a una temperatura T_s , las condiciones límite para la solución de este tipo de problema son:



$$C.L. 1) \quad t_{0,x} = T_0$$

$$C.L. 2) \quad t_{0,0} = T_s$$

que aplicadas a las soluciones dan como resultado la ecuación de distribución de temperatura.

$$t_{\theta,x} = T_s + (T_0 - T_s) \frac{2}{\sqrt{\pi}} \int_0^{\frac{x}{2\sqrt{\alpha\theta}}} e^{-\eta^2} d\eta$$

donde:

— término denominado "erf" o función de error o integral de probabilidad

$$\eta = \frac{x}{2\sqrt{\alpha\theta}}$$

y la acumulación de calor para un tiempo $\theta = 0$ a $\theta = \theta$, para este caso será:

$$Q_0 = \int_{\theta_0}^{\theta} q_0 d\theta = \frac{2kA(T_s - T_0)}{\sqrt{\pi\alpha}} \sqrt{\theta} \quad [\text{Kcal o BTU}]$$

Existen diversos recursos para la solución de los problemas de transferencia térmica y condiciones inestables en adición al análisis - puramente matemático, tales como las gráficas de Jakob, Perry, etc., que son aplicables a diferentes cuerpos.

También en la literatura relacionada con este tipo de problemas se localizan gráficas para placas planas, cilindros, esferas, sólidos infinitos y semi-infinitos, etc., siendo las más usuales las de Groeber, - las de Heisler, las de Gurney Lurie y las de Hottel.

Tuberías.-

La transferencia de calor en tuberías de alta temperatura - se realiza como radiación y convección forzada en el fluido y película - límite, conducción a través de la pared aislamiento y forro, y convección natural combinada con un mínimo efecto de radiación al medio ambiente.

Precalentadores de aire.-

Los precalentadores de aire son cambiadores de calor de -- contacto directo que operan cíclicamente desde el punto de vista de transferencia de calor, obteniéndose la transferencia térmica por medio de convección forzada y conducción a unos conductos cuyo material almacena calor de los gases calientes y lo cede en la misma forma al aire a menor temperatura.

Cambiadores de calor.-

Los calentadores de agua de alimentación son los elementos del ciclo que hacen posible operarlo en forma regenerativa y permiten elevar su eficiencia.

El fluido caliente es vapor de extracción de la turbina el - cual puede estar sobrecalentado, saturado ó húmedo y circula generalmente por el exterior de los tubos cediendo su energía hasta condensarse y subenfriarse, el fluido a menor temperatura absorbe la energía al circular a través de los tubos.

La transferencia térmica se efectúa por convección forzada, por película y goteo en condensación, por conducción y en grado mínimo por radiación.

Evaporadores. -

En los evaporadores se tiene transferencia térmica en diferentes aparatos, un cambiador de calor de superficie y las cámaras de evaporación, en las cuales al ocurrir desequilibrio termodinámico origina la evaporación, el vapor es condensado por medio de el fluido a menor temperatura.

La transferencia térmica es realizada en las cámaras de evaporación como transferencia de masa evaporando una parte del fluido que posteriormente cede su calor latente en un condensador, con transferencia por película y goteo en condensación, conducción y convección forzada.

Condensadores. -

Los condensadores son en realidad considerados como cambiadores de calor pero su importancia mayor consiste en que son el factor limitante para la eficiencia de los ciclos termodinámicos, por ser la fuente de disipación térmica al medio ambiente.

La forma de transferencia térmica en el condensador es relativamente simple, por condensación en película y goteo en el exterior de los tubos, conducción a través de la tubería y convección forzada en el interior de los tubos por los cuales circula el fluido de enfriamiento.

Transferencia térmica en condensación. -

En los condensadores se tiene generalmente una atmósfera de vapor con bajo grado de humedad en su entrada, el cual se va condensando en el exterior de los tubos por los cuales circula el fluido de en -

friamiento.

La condensación ocurre en virtud de que el vapor entra en contacto con una superficie a una temperatura inferior a la temperatura - de saturación correspondiente a la presión de equilibrio del vapor.

La condensación puede efectuarse por película que cubre la superficie, por goteo y por película y goteo.

El tipo de condensación depende de varios factores físicos - como son las propiedades del fluido, de la velocidad del vapor, de la rugosidad de la superficie, de los depósitos, suciedad o sustancias promotoras en la superficie, de la posición vertical u horizontal y del número de hileras.

El fenómeno de condensación involucrando el transporte simultáneo de calor y masa debe ser considerado un proceso de pasos múltiples.

Inicialmente el vapor debe ser transportado de la masa en fase gaseosa a la superficie de condensación, requiriéndose que exista un movimiento molecular neto normal a la superficie, para un vapor puro esta velocidad tiene pequeño efecto sobre el proceso.

En virtud de que la fase líquida es una región de pequeñas distancias intermoleculares y relativamente altas fuerzas intermoleculares, las moléculas de vapor chocando contra la superficie deben interaccionar con la fase condensada de manera que pierdan gran parte de su energía cinética, siendo capturadas por el líquido y cediendo su energía en forma de calor latente de condensación que es la cantidad de calor transferida

por la superficie fría.

No todas las moléculas que llegan a la superficie son retenidas por el líquido, se ha demostrado experimentalmente que para ciertas sustancias comunes, el 90% o más de las moléculas regresan al estado de vapor.

Una cantidad significativa está involucrada en las fuerzas de atracción que tienden a sujetar las moléculas del líquido a distancias muy pequeñas una de otra, esta energía se manifiesta en una muy pequeña diferencia de temperatura entre el vapor y el líquido del orden de 0.05°F . Esta es precisamente la caída de temperatura que proporciona la fuerza impulsora para mantener el transporte de vapor a líquido.

La forma más usual de condensación es por película y aunque en algunos casos se tiene película y goteo, se prefiere diseñar los aparatos utilizando los coeficientes de transferencia térmica para película pura.

Los coeficientes de transferencia térmica en condensación por goteo son de 4 a 8 veces mayores que los obtenidos en condensación por película.

La base para la determinación de los coeficientes de transferencia térmica fue dada por la derivación de Nusselt para un vapor saturado puro y una superficie fría, considerando una película laminar continua de condensado, efectos de gravedad puros y despreciando el efecto de la velocidad del vapor.

Además de las consideraciones anteriores, se analiza el problema de película en caída libre suponiendo que todo el calor latente es transferido por conducción a través de la película, que no hay dife -

rencia de temperatura en la entre fase de vapor y que la temperatura de la superficie es uniforme, despreciando el calor sensible de la película, entonces:

$$dq = + K dx \frac{\Delta t}{\delta} = h(x) dx \Delta t$$

δ = espesor de la película.

$\Delta t = T_v - T_s$ = Temperatura de saturación del vapor - temperatura de la superficie.

k = Conductividad del líquido

t_p = Temperatura de la película

por lo cual: $h_x = \frac{k}{\delta}$

haciendo el balance de transferencia de calor y masa e integrando para determinar el espesor de la película, se obtiene:

$$\delta = \left(\frac{4 \mu k \Delta t \cdot x}{\rho^2 g h_{fg}} \right)^{1/4}$$

con lo cual es posible determinar el coeficiente local de transferencia térmica por convección:

$$h_x = \frac{k}{\delta} = \left(\frac{k^3 \rho^2 g h_{fg}}{4 \mu \Delta t \cdot x} \right)^{1/4}$$

y el coeficiente promedio para una longitud L .

$$h_L = \frac{1}{L} \int_0^L h_x dx$$

$$h_L = 0.943 \left[\frac{k^3 \rho^2 g h_{fg}}{\mu L \Delta t} \right]^{1/4}$$

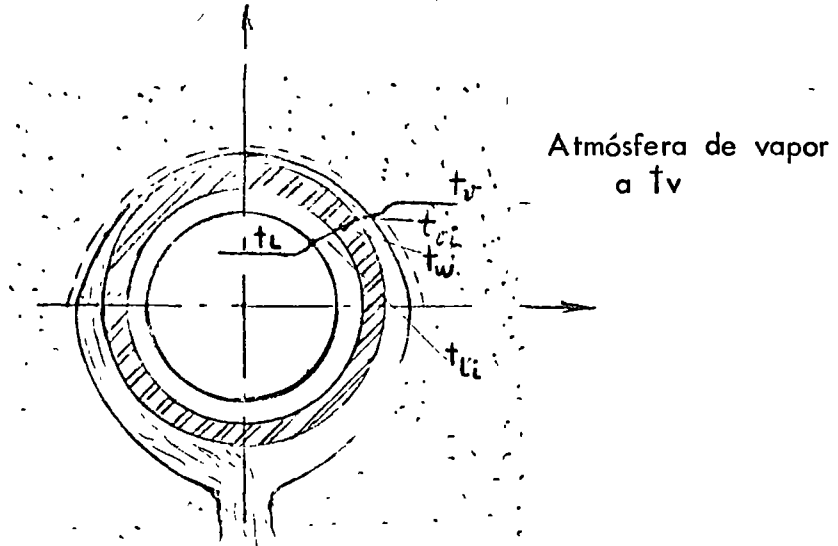
En el Artículo ASME se muestra la expresión del Coeficiente de Transferencia Térmica considerando un cilindro horizontal .

Existen efectos que dependiendo de su magnitud deben ser considerados en la determinación de los coeficientes de transferencia térmica, tales como:

- a. Gases no condensables. - El efecto de 4.5% de aire por volumen, puede hacer disminuir el coeficiente de transferencia térmica de 2300 a $450 \frac{\text{BTU}}{\text{hr-pie}^2 \text{ } ^\circ\text{F}}$
- b. Turbulencia. - La turbulencia origina una mejor transferencia térmica en las partes inferiores de tubos ó placas largas verticales, siendo despreciable su efecto en tubos horizontales.
- c. Velocidad del vapor. - Su efecto puede ser considerable ya que afecta directamente la velocidad y espesor de la película, pudiendo obtenerse valores 10 veces superiores con altas velocidades de vapor en tubos verticales.

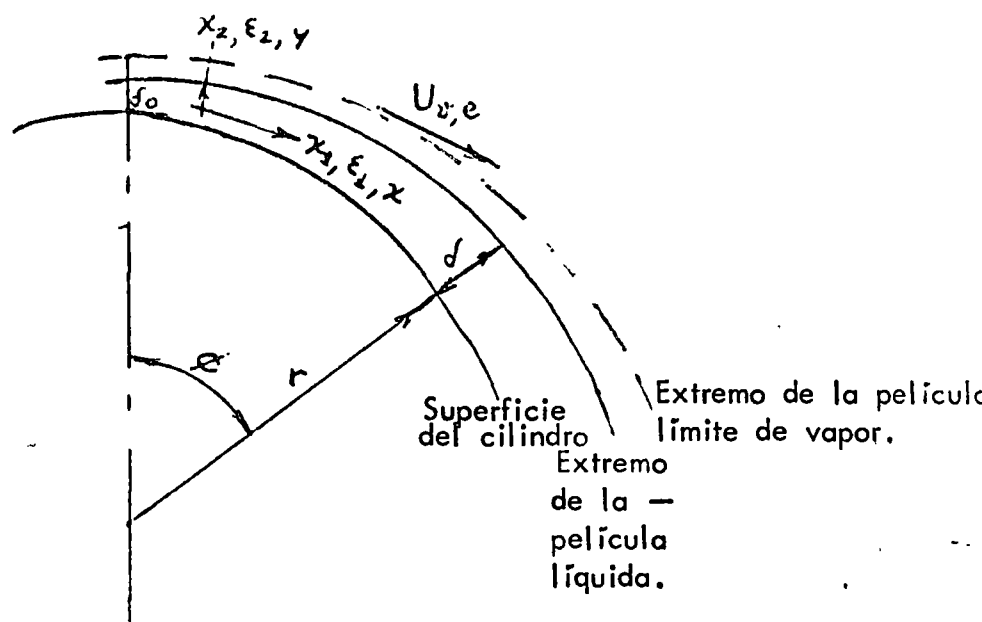
CONDENSACION POR PELICULA EN TUBOS HORIZONTALES. -

La condensación por película en una atmósfera de vapor a presión constante se realiza alrededor de una superficie circular en la siguiente forma:



Con gradientes de temperatura variables dependientes del espesor de la película límite y los coeficientes locales de transferencia térmica.

El análisis detallado de la película límite que se forma en una tubería con película laminar y sujeta a gravedad normal, con velocidades componentes del líquido u_1 y u_2 , se muestra en el Apéndice.



ESQUEMA DEL SISTEMA DE COORDENADAS.

El flujo de vapor saturado a temperatura T_s y velocidad V_v ,
 , la pared exterior del tubo del condensador a temperatura T_w , constan-
 te con el ángulo ϕ .

Utilizando las ecuaciones de gradientes de presión alrededor de la periferia y las limitaciones físicas del fenómeno se pueden determinar las distribuciones de velocidad y temperatura, las cuales aplicadas a los balances de masa y entalpia conducen a la determinación aproximada del espesor de la película límite y del coeficiente teórico de transferencia térmica obteniéndose:

$$h = h (Fr, Re, Pr, k, \lambda, C_p, \Delta t, \phi, r, \phi_o, x.)$$

Se ha determinado que para ángulos hasta 140° existe una discrepancia de menos de 2% entre las predicciones y los experimentos, al aproximarse a los 180° la discrepancia aumenta debido a la violación a las suposiciones de Nusselt.

Las expresiones recomendadas para cálculo de los coeficientes de transferencia térmica, son:

$$\text{Un solo tubo horizontal, flujo laminar}$$

$$h_m = 0.725 \left[\frac{k_f^3 \lambda \rho_f^2 g}{D \mu_f \Delta t} \right]^{1/4}$$

con evaluación de propiedades a $T_f = T_v - \frac{3}{4} \Delta T$

$$\Delta t = T_v - T_s, \quad \lambda = hfg$$

para condensadores en que se tienen N tubos horizontales y el condensado de los tubos superiores fluye al inferior sin salpicamiento,

$$h_m = 1.2 \left[\frac{k_f^3 \rho_f^2 g}{\mu_f^2} \right]^{1/3} N_{Re,f}^{-1/3}$$

$$h_m = 1.51 \left[\frac{k_f^3 \rho_f^2 g}{\mu_f^2} \right]^{-1/3} \left[\frac{4 \omega}{\mu_f^2 L} \right]^{-1/3}$$

$$N_{Re,f} = \frac{4 r_h G}{\mu_f} = \frac{2 \omega}{\mu_f L}$$

Para flúidos en el interior de tubos con Δt moderadas, Pr variando de 0.7 a 120, Re 10000 a 120 000 y $\frac{L}{D} > 60$

$$\frac{h_L D}{k_b} = 0.023 \left(\frac{D G}{\mu_b} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)_b^{0.4}$$

evaluando las propiedades físicas a la temperatura de la masa fluída.

La expresión general de transferencia térmica en función de la diferencia de temperatura y del "coeficiente total de transferencia térmica", es:

$$dq = U_x dA \Delta t_o$$

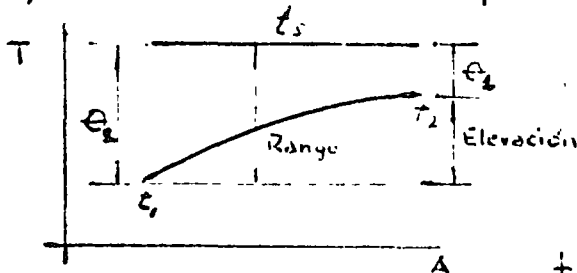
que utilizando la "diferencia de temperatura media logarítmica" conduce a

$$Q = U_m A \Theta_m$$

en la cual U incluye la resistencia a la transferencia térmica de la película del fluído condensado, de la pared del tubo, de la incrustación ó suciedad en el interior del tubo y de la película límite en el interior del tubo.

$$\frac{1}{U} \approx \frac{1}{h_m} + \frac{1}{k_w/\Delta r_w} + \frac{1}{k_i/\Delta r_i} + \frac{1}{h_L}$$

y Θ_m es la diferencia de temperatura media logarítmica.



$$\Theta_m = \frac{\Theta_2 - \Theta_1}{\ln \frac{\Theta_2}{\Theta_1}} = \frac{t_2 - t_1}{\ln \frac{t_s - t_1}{t_s - t_2}}$$

también:

$$t_2 = t_s - \frac{t_s - t_1}{e^{UA/m_a c_{ra}}} \quad \text{y} \quad t_s = t_1 + \frac{t_2 - t_1}{1 - e^{UA/m_a c_{ra}}}$$

OBJETIVOS DE LOS CONDENSADORES

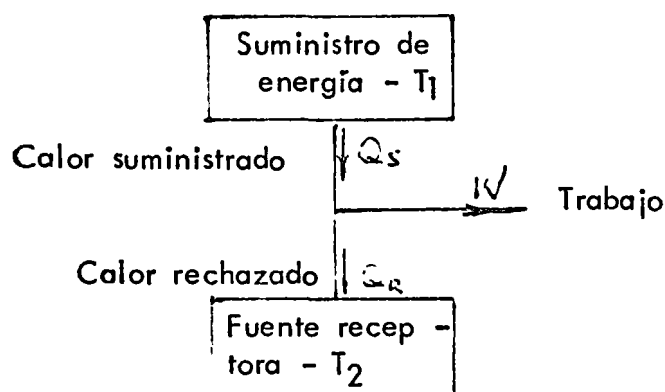
Los condensadores desde el punto de vista termodinámico -- son la fuente receptora de calor, y desde el punto de vista funcional son cambiadores de calor que permiten la utilización práctica del vapor al -- condensarlo y aprovecharlo nuevamente en el ciclo.

Termodinámicamente la limitación más seria que existe a la energía disponible del fluido trabajando, es la temperatura de la fuente receptora, que es la tierra, por medio de sus fuentes de agua ó aire.

De la definición elemental de la eficiencia del ciclo Carnot, que es la limitación de eficiencias en ciclos termodinámicos, mientras menor sea la temperatura a la que se efectúa el rechazo de energía al medio -- ambiente, mayor será la eficiencia del ciclo.

$$\eta = 1 - \frac{T_2}{T_1}$$

por lo cual es importante lograr la menor temperatura de saturación del vapor posible, previa consideración de los parámetros que intervienen en la optimización del sistema de enfriamiento.



$$\oint \delta \dot{W} = \oint \delta \dot{Q}$$

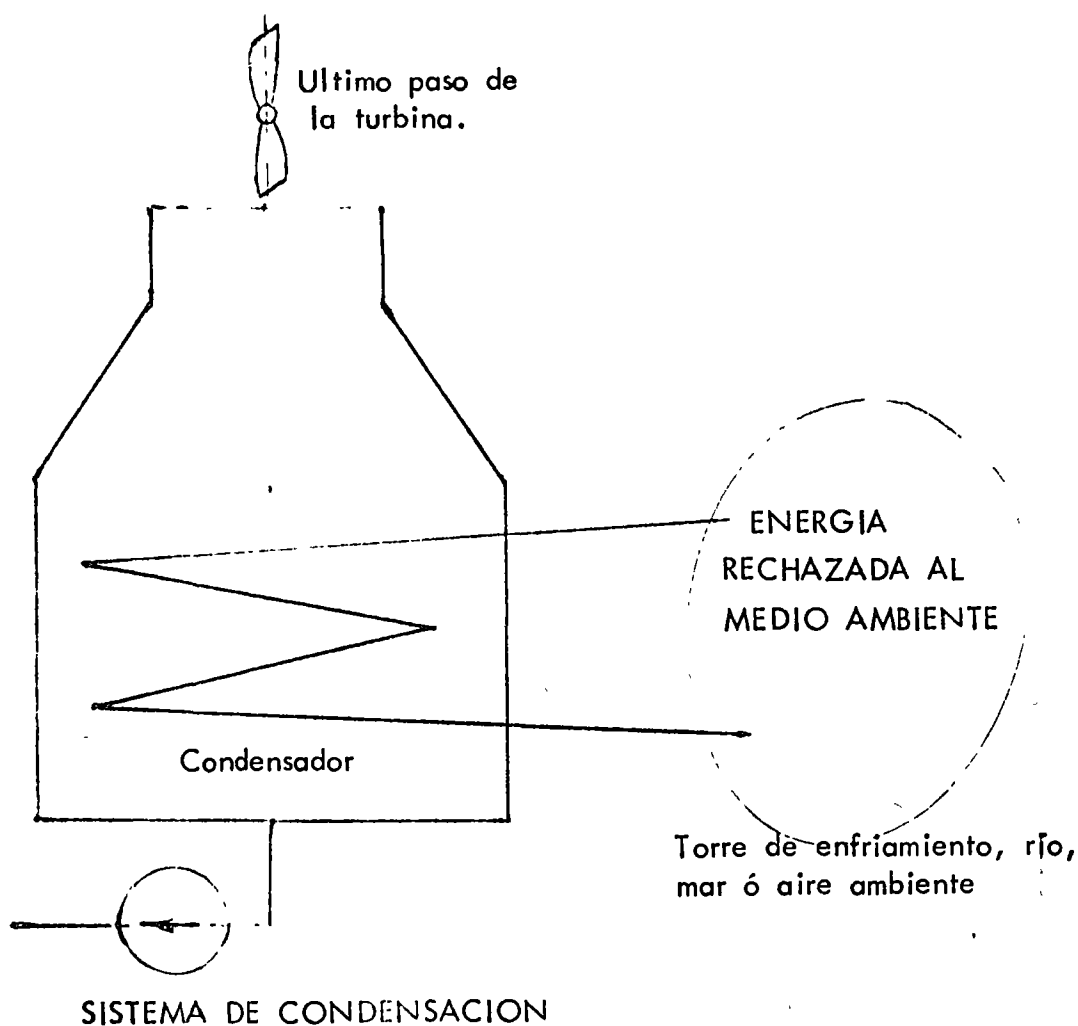
$$\eta = \frac{Q_s - Q_r}{Q_s} = \frac{W}{Q_s}$$

MAQUINA TERMICA.

SELECCION DE LOS PARAMETROS PRINCIPALES.-

Desde el punto de vista económico y de óptimo aprovechamiento de la energía y los energéticos, no es conveniente seleccionar los parámetros de diseño del condensador sin considerar el sistema completo al que está relacionado.

Una optimización de un condensador está relacionada con - la selección de la turbina en lo que respecta a las características de sus últimos pasos y al sistema de enfriamiento con los diámetros y pérdidas en tuberías, parámetros de diseño de la torre de enfriamiento húmeda ó seca, características y consumos de potencia de las bombas de agua de circulación y aparatos para circulación de aire.



Para definir correctamente los parámetros de diseño del condensador se deben considerar todos los elementos técnicos y económicos que intervienen en el sistema de condensación y agua de enfriamiento, es decir:

- 1.- Costo diferencial de la turbina por diferencia en tamaño de álabes y número de salidas de escape.
- 2.- Costo diferencial del consumo térmico unitario (Heat Rate) y de la capacidad en KW.
- 3.- Costo diferencial de tuberías, bombas y torre de enfriamiento u obra de toma.
- 4.- Costo diferencial de los condensadores.

La optimización del sistema debe tomar en cuenta los costos de combustible proyectados y actualizados a valor presente (si se usa este método de evaluación), costos de energía proyectados y actualizados, así como datos confiables sobre variaciones en costo de los tubogeneradores, -- instalaciones completas para obtener el costo unitario por KW, costos de -- tuberías, bombas y tomas de enfriamiento de diferentes tamaños, costos de tuberías y condensadores de diferentes características y tamaños, de tal manera que la utilización y combinación de datos conduzca a la solución dentro de las condiciones óptimas del sistema.

Vacío de diseño del condensador.-

Uno de los parámetros más importantes para el sistema es el vacío de diseño del condensador, el cual está relacionado a las curvas de carga y curvas de temperaturas del medio ambiente con respecto al tiempo, así como los criterios que se tengan en los sistemas sobre reserva de capacidad instalada y disponible.

El vacío del condensador está naturalmente ligado a las -- condiciones físicas del medio ambiente que son el factor límite.

Los parámetros más importantes después del de vacío del -- condensador son los siguientes:

- A.- Número de pasos.
- B.- Diámetro de tubos.
- C.- Longitud de tubos.
- D.- Material de los tubos.
- E.- Velocidad del agua de enfriamiento.
- F.- Factor de limpieza.

A.- NUMERO DE PASOS.-

En la mayoría de instalaciones se tienen condensadores de - 1 o 2 pasos, existiendo en algunas hasta de 3 pasos, también existen condensadores de multipresión en grandes instalaciones.

En general resulta más conveniente instalar condensadores de un solo paso donde se dispone de grandes cantidades de agua y de dos pasos cuando existen limitaciones, aunque debido al mayor flujo y menor potencia necesaria esto debe verificarse mediante el estudio detallado correspondiente.

B.- DIAMETRO DE TUBOS.-

El tamaño del diámetro de tubos es dependiente de los tamaños comerciales disponibles y de su influencia en el comportamiento y costo del condensador, los diámetros pequeños aumentan los coeficientes de transferencia térmica y los diámetros mayores ofrecen mejores condiciones para -

evitar ensuciamiento y disminuyen la potencia eléctrica necesaria.

En general se utilizan los tamaños menores $5/8$ y $3/4$ " en condensadores pequeños, hasta 1500 M^2 y de $3/4$ " y $7/8$ " y 1 " en mayores tamaños.

Los espesores de los tubos son variables, de calibre 14 a 18 BWG, utilizándose en la gran mayoría tubos del calibre No. 18.

C.- LONGITUD DE TUBOS.-

La longitud de los tubos es función de la superficie del condensador y de las limitaciones de espacio en la instalación, en instalaciones modernas de gran tamaño se han utilizado longitudes de tubos hasta de 15m (50 pies) y en muchos casos ha sido necesario colocar los condensadores en dirección longitudinal a los turbogeneradores por las limitaciones de espacio.

La longitud de los tubos tiene una gran influencia en la optimización del sistema por su relación directa con la potencia eléctrica consumida.

D.- MATERIAL DE LOS TUBOS.-

Aunque este concepto se discute detalladamente por separado, deben seleccionarse los materiales cuidadosamente de acuerdo con las condiciones locales de agua de circulación.

La mayoría de los materiales son de aleaciones de cobre, utilizándose más comúnmente para aguas normales de agua fresca dulce, Metal Admiralty (Aleación de cobre 70%, Zinc 29%, Estaño 1%) con algunas variantes del mismo; para aguas saladas y salobres, Latón Aluminio (Aluminum Brass) (Cobre 76%, Aluminio 1.75 a 2.50%, con Hierro, Plomo y Antimonio

en pequeñas cantidades, resto Zinc), también se utilizan aleaciones de - 70-30 y 90-10 Cobre Níquel; en aguas con gran acidez y compuestos derivados de desperdicios de minas se han utilizado aleaciones de acero inoxidable 304, 316 y 347.

E.- VELOCIDAD DEL AGUA DE ENFRIAMIENTO.-

El coeficiente de transferencia térmica mejora con velocidades altas debiendo evaluarse sus ventajas contra las de mayores requerimientos de potencia y en algunos casos reducción de la vida útil de los tubos por erosión.

En la mayoría de los casos se utilizan velocidades de - - 1.85 a 2.44 m/seg. (6.0 a 8 pies/seg.) aunque se ha experimentado con aleaciones de Cobre-Níquel y de Acero inoxidable a velocidades mayores de 3.05 m/seg. (10 pies/seg.).

La velocidad debe optimizarse dentro de las limitaciones de valores convenientes por pérdidas de fricción y erosión.

F.- FACTOR DE LIMPIEZA.-

Debido a la posibilidad de incrustaciones o depósitos en el interior de las tuberías, con objeto de no disminuir la capacidad del condensador y el vacío, después de algún tiempo de operación, se utiliza un factor de limpieza cuyo valor depende de los criterios particulares de diseño, sistemas de tratamiento químico, control del comportamiento del -- condensador y sistemas mecánicos de limpieza que se utilicen.

Los factores usualmente utilizados oscilan del 85% al 95%

DISEÑO

El diseño de un condensador en particular requiere la definición de los parámetros principales descritos anteriormente, aunque los fabricantes utilizan diferentes métodos, la mayoría de ellos converge en la parte básica de ellos, por lo cual utilizaremos para ilustrar los procedimientos de diseño, el establecido por Allis Chalmer.

Datos del sistema.-

Unidad de 3000 KW

Flujo de vapor al condensador = TS = 33 000 lbs/hr.

Agua fresca dulce en abundancia.

Temperatura del agua 80°F

Vacío de diseño recomendado 28" Hg referidas a 30" Barométricas.

Tuberías de 3/4" D.E., Calibre 18 BWG, longitud 14 pies.

Factor de limpieza 85%.

Velocidad 7 pies/seg.

De las tablas en la página 15 se obtienen los siguientes factores:

CONCEPTO	VALOR	TABLA	FACTOR
Temperatura del agua	80°F	A	1.019
Número de pasos	2*	B	0.301
Diámetro de los tubos y calibre	3/4" - 18 BWG	C	1.008
Longitud de tubos	14 pies	D	1.146
Velocidad del agua	7 pies/seg.	E	0.577
Factor de limpieza	85%*	F	0.929
Material	Admiralty*	G	0.000
			4.980

* se seleccionan de acuerdo con las recomendaciones.

De la tabla H, para Factor Total = 4.980

$$X = 0.615$$

De la curva en página 13 o de tablas de vapor

$$T_s = 101.1^\circ \text{ F para vacío} = 28'' \text{ Hg.}$$

$$\text{Rango} = T_s - T_1 = 101.1 - 80 = 21.1^\circ \text{ F}$$

$$\text{Elevación} = \text{Rango} \times X = 21.1 \times 0.615 = 13.0^\circ \text{ F}$$

$$\text{Diferencia terminal} = \text{Rango} - \text{Elevación} = 21.1 - 13 = 8.1^\circ \text{ F}$$

valor aceptable por ser mayor de 5° F .

$$\text{Flujo de agua de enfriamiento } G = \frac{950 \times T_s}{\text{Elev.} \times 500}$$

$$G = \frac{950 \times 33\,000}{13.0 \times 500} = 4825 \text{ GPM}$$

$$\text{Superficie} = \frac{KGL}{V}$$

de la tabla de constantes para tubos, página 5

$$K = 0.377$$

$$S = \frac{KGL}{V} = \frac{0.377 \times 4825 \times 14}{7} = 3640 \text{ pies}^2$$

Es usual el realizar el diseño y la optimización por medio - de computadoras con la programación correspondiente a la cual se incorporan todos los conceptos que intervienen en la optimización del sistema completo, como se ilustra en el artículo de la American Power Conference.

De esta manera se pueden hacer variar los diferentes conceptos, es decir el vacío del condensador, diámetro y calibre de tubos, velocidad del agua, número de pasos, diferencia terminal de temperatura y longitud y hacer el cálculo del resto de los conceptos que intervienen en el sistema como sean influenciados por el condensador, hasta determinar la so-

lución más conveniente dentro de la zona óptima.

INFLUENCIA DE LOS CONDENSADORES EN LA ECONOMIA DE LA PLANTA.-

Como se mencionó en los objetivos de los condensadores, la temperatura de saturación que equivale a la temperatura de la fuente receptora fría, tiene influencia directa en la eficiencia del ciclo termodinámico.

La influencia del condensador en el ciclo termodinámico se refleja en el consumo térmico unitario del turbogenerador y se expresa como

$$\text{unidades de calor /Kw-hr} \left[\frac{\text{Kcal}}{\text{Kw-hr}} \text{ o } \frac{\text{BTU}}{\text{Kw-hr}} \right]$$

los fabricantes de turbogeneradores proporcionan una serie de curvas a diferentes cargas de presión de escape-consumo térmico unitario, otras para la carga en KW, con las cuáles es posible calcular el efecto directo del condensador en la eficiencia del ciclo y en la capacidad.

El costo de 1 BTU/KW-hr se calcula de acuerdo con los costos de combustible proyectados y actualizados, las tasas de interés y vida de las unidades seleccionadas.

El costo de 1 KW de capacidad se calcula con base en el costo total de la instalación que deba suplir la deficiencia.

Como ejemplo de los costos de deficiencias de un condensador para una unidad de 300 000 KW se tienen los siguientes:

Por cada BTU/Kw-hr adicional por deficiencia en el CTU garantizado	\$62,500.00 M.N.
Por cada Kw adicional en demanda equivalente o cada KW de deficiencia en capacidad	1,125.00 M.N.
Por cada Kw adicional de energía auxiliar equivalente	2,840.00 M.N.

Para una unidad de 300 000 KW con carga promedio de -- 225 000 KW el efecto de una deficiencia en presión absoluta de 0.1" Hg origina un aumento de 0.2% del consumo térmico unitario o sea aproximadamente 16 BTU/KW-hr con un costo evaluado a valor presente de - - - - \$1,000,000.00 M.N., lo cual refleja la gran importancia de la optimización de los sistemas y el correcto funcionamiento de los condensadores.

En adición a lo anterior debe evaluarse el comportamiento garantizado de pérdida en los tubos y cajas de agua que pueden originar consumo adicional de energía y también verificarse la efectividad en la extracción de gases no condensables que tiene un serio efecto en la transferencia térmica.

introduction

surface condensers *performance calculations*

The purpose of this manual is to bring together those variables effecting the heat transfer performance calculations for steam surface condensers in an orderly fashion to simplify their use. The many necessary constants that are tabulated herein are drawn from various handbooks and are in complete accord with the Heat Exchange Institute, Standards for Steam Surface Condensers, Fifth Edition (1965).

In addition to the basic condenser performance calculations there are curves plotted to reflect possible trends with changes in only one parameter at a time, such as tube diameter or water velocity.

Finally, guidelines are given to help the user apply the performance calculations to specific installations.

formulae

Condenser performance calculations are made up of two parts, the heat transfer function to be performed and the physical characteristics of the condenser necessary to perform the function.

Assume there is a heat load of (Q) BTU/hr to be condensed at an absolute pressure of (P) " Hg using cooling water at a temperature of (T_1) °F. Then consider the basic temperature relations in a surface condenser by referring to Figure 1.

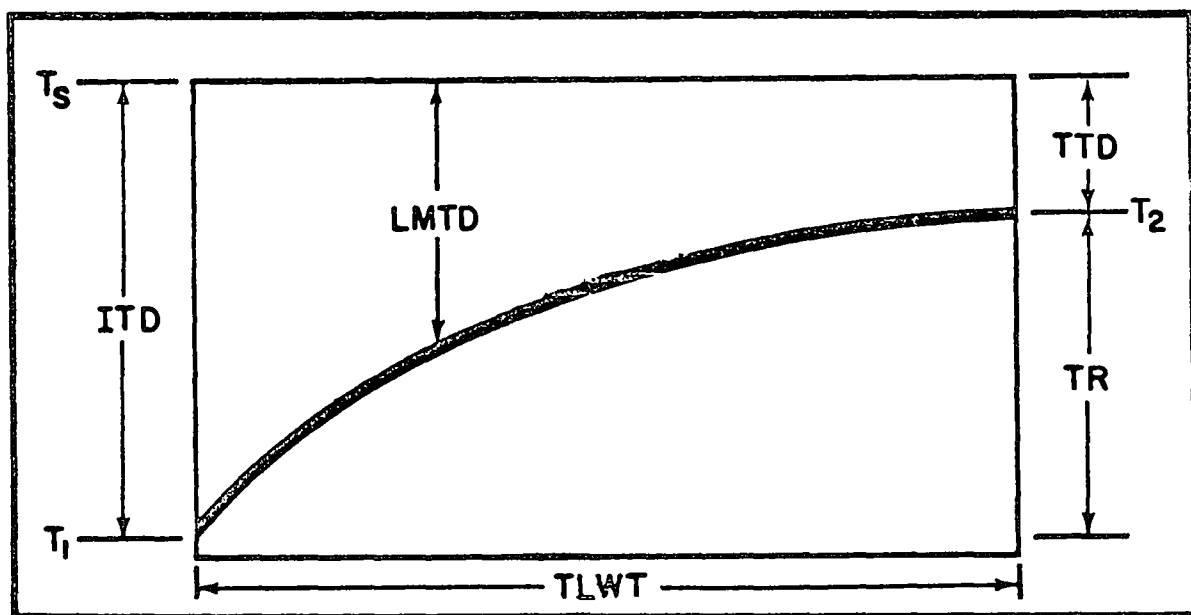


Figure 1

In a saturated steam mixture at an absolute pressure of (P) " Hg there is a corresponding steam temperature of (T_s) °F. The difference between the steam temperature (T_s) and the cooling water temperature (T_1) is defined as the initial temperature difference (ITD).

As the cooling water progresses down the length of the condenser tubes, its temperature is continuously elevated until the water exit at a temperature of (T_2) °F. The difference between inlet and exit cooling water temperatures is called the temperature rise (TR).

The ratio of the temperature rise (TR) to the initial temperature difference (ITD) is defined as (R).

The difference between the steam temperature (T_s) and the cooling water exit temperature (T₂) is known as the **terminal temperature difference (TTD) or approach (A)**.

From fundamental heat transfer relations we define the **Log Mean Temperature Difference (LMTD)** as:

$$\frac{(ITD) - (TTD)}{\ln \left(\frac{ITD}{TTD} \right)} \text{ which is the same as } \frac{(TR)}{\ln \left(\frac{1}{1-R} \right)} \quad \text{eq (1)}$$

By definition we denote the denominator of eq (1) as (K) and then

$$R = 1 - \frac{1}{e^k} \text{ (Table VI).}$$

The required cooling water flow (GPM) can be calculated from the heat balance:

$$\text{GPM} = \frac{Q}{500 (TR)} \quad \text{eq (2)}$$

where: 500 lb/hr = 1 GPM

By a study of the heat transfer characteristics of condenser tubes we can write the following relationship:

$$K = \frac{(L) (NP) \cdot a \cdot b \cdot f \cdot \% \text{ clean} / 100}{\sqrt{V}} \quad \text{eq (3)}$$

- where:
- L = Effective tube length, ft.
 - NP = Number of passes of cooling water
 - a = Constant varying with tube diameter and wall thickness (Table III)
 - b = Tube material correction factor (Table IV)
 - f = Cooling water inlet temperature correction factor (Table V)
 - % clean = Tube overall cleanliness factor
 - V = Velocity of cooling water within the tubes, ft/sec

For a typical surface condenser calculation a summary of the foregoing formulae in their normal order of use is as shown below:

- (a) P corresponds to T_s (Table I)
- (b) (T_s) - (T₁) = ITD
- (c) $K = \frac{(L) (NP) \cdot a \cdot b \cdot f \cdot \% \text{ clean} / 100}{\sqrt{V}}$ eq (3)
- (d) K corresponds to R (Table VI)
- (e) TR = (R) · (ITD)
- (f) $\text{GPM} = \frac{Q}{500 (TR)}$ eq (2)
- (g) $\text{NT} = \frac{\text{GPM}}{g \times v} \times (NP)$

- where:
- NT = Number of tubes (total)
 - g = GPM/tube at 1.0 fps (Table II)
 - m = sq. ft./lineal ft. of tube (Table II)
 - S = sq. ft. of surface area (total)

(h) S = (NT) (L) x m

For general interest, you will note that the overall coefficient of heat transfer (U) is not apparent in the foregoing equations. Actually the relation $U = C \sqrt{v} \times b \times f \times \frac{\% \text{ clean}}{100}$ has been absorbed into the form of equation (3). Values of (U) for any condition can be calculated by:

$$U = K \times \frac{\text{GPM} \times 500}{S} \quad \text{eq (4)}$$

NOTE: In accordance with the Heat Exchange Institute Standards for Steam Surface Condensers, the terminal temperature difference (TTD) should not be less than 5°F.

sample problems

(Refer to inside back cover for an example of a work sheet.)

A. CALCULATION OF SURFACE AND COOLING WATER REQUIREMENTS

Given:

- Q = 1000 x 10⁶ BTU/hr heat load
- P = 2.00'' Hga
- T₁ = 75°F inlet cooling water
- V = 7.00 fps

Condenser to be single pass with 85% clean 1'' O.D. - 18 BWG Admiralty tubes having a 50' - 0'' effective tube length.

Then:

- T_s = 101.14° F (Table I)
- ITD = 101.14 - 75 = 26.14°F
- K = $\frac{50 (1) (.06914) (1.0) (1.025) \left(\frac{85}{100}\right)}{\sqrt{7.00}} = 1.138$
- R = .680 (Table VI)
- TR = (.680) (26.14) = 17.78
- GPM = $\frac{1000 \times 10^6}{500 (17.78)} = 112,486$
- NT = $\frac{112,486}{(1.992) (7.00)} (1) = 8,067$
- S = 8,067 (50) (.2618) = 105,600 Sq. Ft.

NOTE: (ITD) - (TR) = TTD = (26.14) - (17.78) = 8.36°F > 5°

B. CALCULATION OF TUBE LENGTH, SURFACE AND GPM FOR A FIXED TR

Given:

- Q = 1,000 x 10⁶ BTU/hr
- P = 1.50'' Hga
- T₁ = 60°F
- V = 8.00 fps
- TR = 20°F (or that T₂ shall not exceed 80°F)

Condenser to have 100% clean 1'' O.D. - 22 BWG Type 304 Stainless Steel tubes.

Then:

- T_s = 91.72°F (Table I)
 - ITD = 91.72 - 60 = 31.72°F
 - R = $\frac{TR}{ITD} = \frac{20}{31.72} = .630$
 - K = .994 (Table VI)
- Rewrite equation (3)

$$(L) (NP) = \frac{K \sqrt{v}}{a \cdot b \cdot f \cdot \% \text{ clean} / 100} = \frac{.944 \sqrt{8.00}}{(.06313) (.79) (.960) (1.00)} = 58.7 \text{ ft.} \quad \text{eq (5)}$$

Thus, with the fixed temperature rise there is only one total length of cooling water travel possible when the velocity and diameter are fixed. The condenser can be two pass with 29.35 ft. effective tube length or single pass with 58.7 ft. effective tube length. (Note that this was ascertained independent of the heat load Q .)

Then:

$$\begin{aligned} \text{GPM} &= \frac{1000 \times 10^6}{500 (20)} = 100,000 \\ \text{NT} &= \frac{100,000}{(2.182) (8.00)} (1) = 5,728 \\ \text{S} &= 5,728 (58.7) (.2618) = 88,026 \text{ Sq. Ft.} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{GPM} \\ \text{NT} \\ \text{S} \end{aligned}} \right\} \text{Single pass 58.7' tubes}$$

or

$$\begin{aligned} \text{NT} &= \frac{100,000}{(2.182) (8.00)} (2) = 11,456 \\ \text{S} &= 11,456 (29.35) (.2618) = 88,026 \text{ Sq. Ft.} \end{aligned} \quad \left. \vphantom{\begin{aligned} \text{GPM} \\ \text{NT} \\ \text{S} \end{aligned}} \right\} \text{Two pass 29.35' tubes}$$

C. CALCULATION OF PRESSURE FOR A GIVEN SURFACE

Given:

$$\begin{aligned} \text{S} &= 300,000 \text{ Sq. Ft.} & \frac{7}{8}'' \text{ O.D. - 18 BWG Aluminum Brass tubes} \\ \text{GPM} &= 175,000 & \text{Two pass, 36'-0'' effective tube length} \end{aligned}$$

Calculate the pressure (P) for a heat load of $2,500 \times 10^6$ BTU/hr for a cooling water temperature 80° F with 85% cleanliness.

$$\begin{aligned} \text{NT} &= \frac{300,000}{(.2291) 36} = 36,374 \\ \text{V} &= \frac{175,000 (2)}{36,374 (1.478)} = 6.51 \text{ ft/sec} \\ \text{K} &= \frac{36 (2) (.08153) (.97) (1.045) (.85)}{\sqrt{6.51}} = 1.983 \\ \text{R} &= .862 \text{ (Table VI)} \\ \text{TR} &= \frac{2500 \times 10^6}{175,000 (500)} = 28.57 \\ \text{ITD} &= \text{TR} / \text{R} = \frac{28.57}{.862} = 33.14 \\ \text{T}_s &= \text{ITD} + \text{T}_1 = 33.14 + 80 = 113.14^\circ \text{F} \\ \text{P} &= 2.84'' \text{ Hga (Table I)} \end{aligned}$$

$$\text{NOTE: ITD} - \text{TR} = 33.14 - 28.57 = 4.57^\circ < 5^\circ \text{ TTD}$$

Since the calculations result in a TTD less than 5° , then the example should continue as:

$$\begin{aligned} \text{TR} &= 28.57^\circ \\ \text{ITD} &= \text{TR} + 5 = 28.57 + 5 = 33.57^\circ \\ \text{T}_s &= \text{ITD} + \text{T}_1 = 33.57 + 80 = 113.57 \\ \text{P} &= 2.88'' \text{ Hga} \end{aligned}$$

D. PERFORMANCE CURVES

The previous example represents the sequence that is used to calculate condenser performance curves that plot condenser pressure, "Hga against condenser heat load, BTU/hr, for one or more inlet cooling water temperatures. Care must be taken to use $\text{TR} + 5^\circ = \text{ITD}$ for every point that calculates to a TTD less than 5° F .

Since the pressure that will exist when there is zero heat load is not readily calculable, values of these zero load pressures for any inlet water temperature have been established in curve form. (Ref: Curve "B", Fig. SF-3, Pg. 8, 1965 Fifth Edition, H.E.I. Standards for Steam Surface Condensers).

Further, at combinations of low condenser pressures and low cooling water temperatures, the effect of air in the air-vapor mixture leaving the "nose" of the condenser will influence the performance of the unit. Thus the performance curve is altered to reflect the effect of the air partial pressure below a certain pressure called the cutoff point. Values of these cutoff points for any inlet water temperature have been established in curve form. (Ref: Curve "A", Fig. SF-3, Pg. 8, 1965 Fifth Edition, H.E.I. Standards for Steam Surface Condensers.) Therefore, pressures for light load and/or low inlet water temperatures can only be determined by actually drawing the performance curve.

E. MULTI-PRESSURE PERFORMANCE

Whether two or three or more pressure levels are to be considered, the calculations must treat the condenser as a series of individual condensing compartments, each operating at its own absolute pressure, with a series cooling water circuit. Thus the inlet cooling water temperature to the second and succeeding compartments is equal to the exit water temperature from the first or preceding compartment.

Consider the condenser calculated for Example "A" will be oriented with its tubes parallel to the shaft of a Tandem Compound Four Flow Turbine-Generator unit having two low pressure turbine exhaust openings. Then divide the 50 ft. tube bundle into two compartments of 25 ft. length, each to serve one of the L. P. turbine exhausts. The same 105,600 Sq. Ft. condenser will perform as follows with the slightly incorrect assumption that the heat load (Q) will be equally divided to 500×10^6 BTU/hr into each compartment. Actually the steam flow rate in lb/hr will be divided into essentially two equal parts, but the enthalpies of the two halves will differ slightly due to the absolute pressure difference.

$$K_{1*} = \frac{25 (1) (.06914) (1.0) (1.025) \left(\frac{85}{100}\right)}{\sqrt{7.00}} = 0.569$$

$$R_1 = .434$$

*NOTE: Subscripts 1 and 2 reflect the First and Second Compartments respectively.

$$TR_1 = TR_2 = \frac{TR}{2} = \frac{17.78}{2} = 8.89^\circ$$

$$ITD_1 = \frac{TR_1}{R_1} = \frac{8.89}{.434} = 20.48$$

$$T_{s1} = 75 + 20.48 = 95.48^\circ\text{F and } P_1 = 1.69'' \text{ Hga}$$

$$T_1 + TR_2 = 75 + 8.89^\circ = 83.89^\circ \text{ inlet to second compartment}$$

$$K_2 = \frac{25 (1) (.06914) (1.0) (1.056) \left(\frac{85}{100}\right)}{\sqrt{7.00}} = 0.586$$

$$R_2 = .443$$

$$ITD_2 = \frac{TR_2}{R_2} = \frac{8.89}{.443} = 20.07$$

$$T_{s2} = 83.89 + 20.07 = 103.96^\circ\text{F and } P_2 = 2.18'' \text{ Hga}$$

Thus the first L. P. turbine operates at 1.69'' Hga and the second at 2.18'' Hga at the design condition. This mode of operation would have to be evaluated for the specific turbine to establish the heat rate saving that could result compared to a single pressure unit operating at 2.00'' Hga.

Further analysis of the proper multi-pressure selection must be made as described in Ingersoll-Rand Form 9707. Assuming there is a heat rate saving with the above two pressure conditions, there also will be a smaller condenser surface, or a lower cooling water flow, or a combination of the two, which will give the same heat rate in multi-pressure operation as exists with the single pressure selection operating at 2.00'' Hga. Using the single pressure cooling water flow, the surface can be reduced by using a shorter tube length, and the water side friction will be reduced lowering the total pumping power required. Examples of some of the possibilities for the foregoing application are tabulated in Figure 2.

	Reference Single Pressure	Reduced Heat Rate	Same Heat Rate		
			Reduced S, GPM & BHP	Reduced GPM & BHP	Reduced S & BHP
BTU/hr ÷ 10 ⁶	1000	500/500	500/500	500/500	500/500
"Hga	2.00	1.68/2.18	1.73/2.25	1.73/2.26	1.75/2.25
Sq ft	105,500	105,600	101,394	105,600	97,152
GPM	112,486	112,486	108,108	105,576	112,486
Velocity, fps	7.00	7.00	7.00	6.57	7.00
TR	17.78	8.89/8.89	9.25/9.25	9.47/9.47	8.89/8.89
*Friction, ft	12.13	12.13	12.13	11.34	11.33
†BHP	897	897	864	833	871

*Refer to HEI Standards for Steam Surface Condensers, Fifth Edition, 1965.
†Assume 15' external friction and 86% efficient circulating pumps

Figure 2

A further refinement in the economic evaluation would be to study other than a 25 ft. - 25 ft. division of the 50 ft. single pressure tube length.

trends with changes in individual parameters

Three fundamental figures make up the major economic influence on any condenser selection for comparison to another for the same heat transfer function. They are the total square feet of condensing surface, the total GPM of cooling water and the total BHP of pumping power required to move that GPM through the specific cooling water circuit. Oddly enough, the three variables are always in a form of balance. When a lesser amount of cooling water is desired there will be an increased amount of surface or pump power (or a combination of each up or down) to perform the same heat transfer function. Examples of this basic relationship and other less important concepts are reflected in the following graphs showing the influence of varying one parameter at a time.

A. TUBE DIAMETER

Figure 3 reflects the influence of different tube diameters for a given heat transfer function using a constant tube length and cooling water velocity. With increasing tube diameters there will be less square feet of condensing surface and more cooling water required. The relative change in cooling water pumping is shown for an assumed siphon water circuit and cooling tower water circuit, both of which are common today.

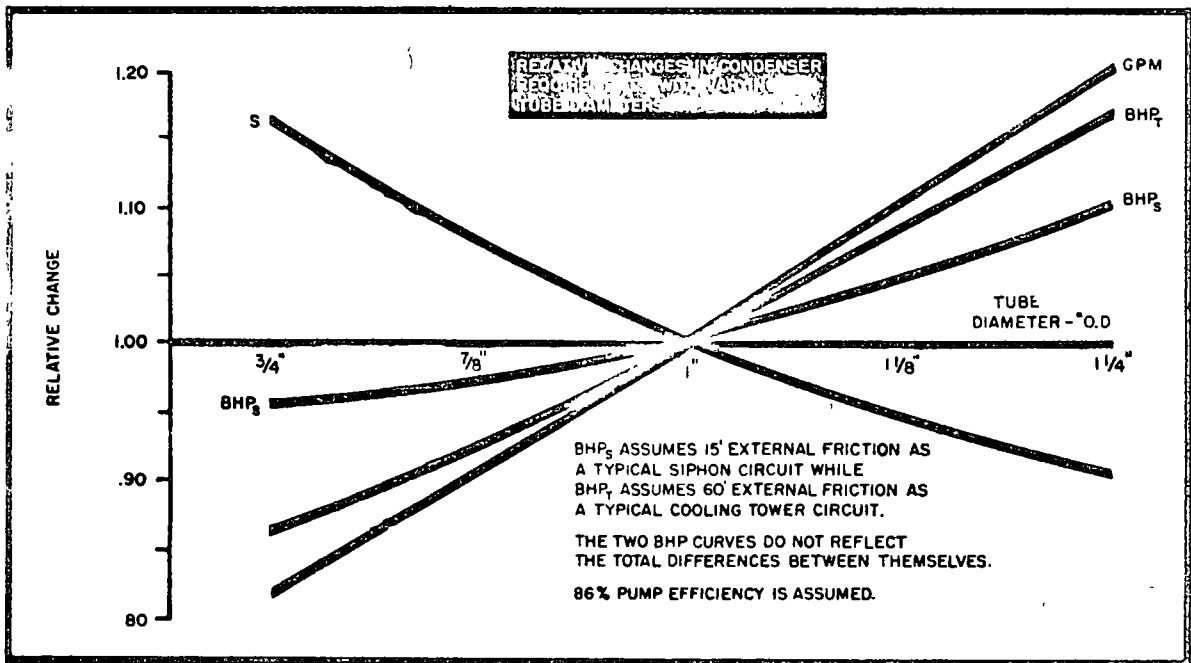


Figure 3

B. VELOCITY OF COOLING WATER

Figure 4 shows the effect of changing the velocity of the cooling water within a given tube diameter for a fixed tube length, all to perform the same heat transfer function. With higher velocities, the required square feet of condensing surface is reduced, but the cooling water flow is increased and along with it the condenser water side friction and overall circulating water pumping power. Again, both an assumed siphon and cooling tower circuit are shown.

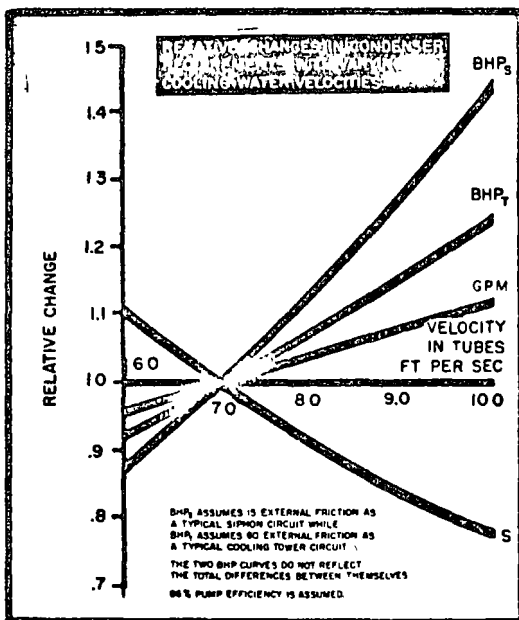


Figure 4

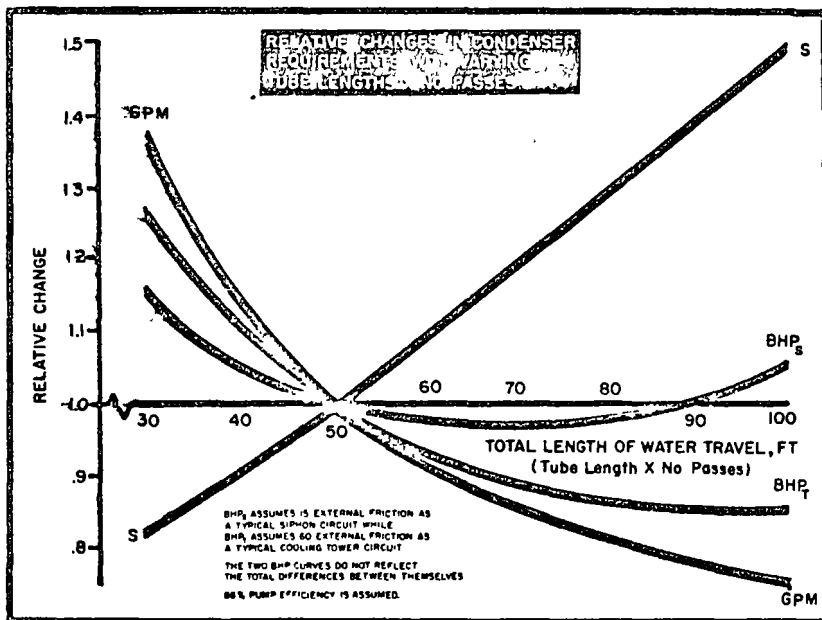


Figure 5

C. TOTAL LENGTH OF WATER TRAVEL

Figure 5 depicts the relative changes in the three basic parameters S, GPM and BHP as the total length of cooling water travel (tube length times number of passes) is increased all for the same heat transfer function and a constant water velocity. With increasing TLWT the condenser cooling water friction drop eventually exceeds the assumed external system friction in the siphon circuit analysis, and the BHP pumping power reaches a minimum and then begins to increase.

D. TUBE MATERIAL

For general information a tabulation of the condenser requirements for one heat transfer job using 95 percent clean Type 304 stainless steel 22 BWG tubes compared to 85 percent clean admiralty 18 BWG tubes is shown in Figure 6. Using the admiralty selection as a reference, the tabulation shows the percent changes of the individual requirements with stainless steel for the following conditions:

- Same velocity, GPM, and TR but with a longer tube and increased surface and pump power.
- Same tube length and velocity but with increased GPM, surface and pump power.
- Same tube length and surface but with increased GPM, pump power and a higher velocity.

85 PER CENT CLEAN 18 BWG ADMIRALTY vs 95 PER CENT CLEAN 22 BWG 304 STAINLESS STEEL			
Per Cent Change Using Stainless Steel			
	A	B	C
Surface	+13.6	+4.8	+17.5
GPM	0	+14.8	+17.5
T. L.	+24.7	0	0
Velocity	0	0	+6.9
Temperature rise	0	(-)13.0	(-)15.0
BHP _s	+5.8	+12.8	+19.7
BHP _t	+2.2	+12.9	+18.2

Figure 6

An examination of this specific example shows that while the use of stainless steel tubing for surface condensers may be desirable from a material life viewpoint on the steam and/or water side or from a first cost analysis, still there exists a performance difference in the amounts of condensing surface, cooling water flow and pumping power to perform a given heat transfer function. One of the most important factors in a decision of this type is the experienced tube cleanliness for admiralty tubes at a specific jobsite.

guidelines

Straight forward condenser calculations are fairly simple to do, but the proper application of the many arrangements of tube length, diameters and orientation requires some experience. Comments reflecting some of this experience may be of assistance.

Whether by legislation or plant design, the maximum allowable temperature rise across a given surface condenser is often stated as part of the design problems. This automatically fixes the minimum GPM of cooling water to be used with a given heat load completely independent of the condenser tube diameter, length or velocity.

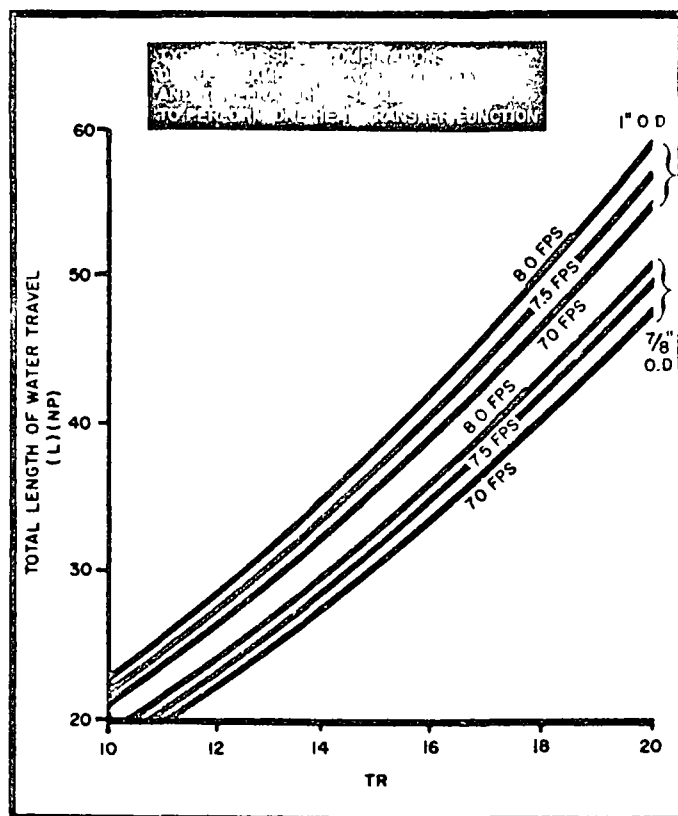


Figure 7

Consider Example B of this manual again. Further, consider the case where a maximum tube length has been established and the combinations of tube diameters, velocities and lower TR's are to be investigated for a given heat transfer job. Then equation (5) would be utilized for a range of velocities for each tube diameter to be considered. This analysis would be repeated for each TR under consideration and in every instance a total length of water travel (L) (NP) is calculated. The results can be plotted with (L) (NP) versus TR for each tube O.D. and velocity studied. Figure 7 is an example of such an analysis and shows the many combinations possible to perform one heat transfer duty at a glance.

Many references have been made to condenser tube diameters, materials, velocity levels and cleanliness. Condenser performance calculations must be guided by the user's experience with the cooling water available for each specific application.

Material selection falls into two basic groups, fresh water and salt water. Admiralty, Arsenical Copper and Type 304 Stainless Steel are common for fresh water applications with Copper Nickel, Aluminum Brass and sometimes Type 316 Stainless Steel for salt water.

Design cleanliness can vary from 70 percent to 90 percent for the copper alloy tubes with 85 percent most common, and from 80 percent to 100 percent for the stainless steel tubes with the majority 90 percent or higher.

The copper alloy tubes are most often 18 BWG wall thickness with some 19 BWG applications in recent years. The stainless steel tubes are usually 22 BWG wall thickness with few exceptions. (Air cooler and periphery are sometimes of 20 BWG to provide a corrosion allowance or factor of safety.)

Water velocity in the tubes will range from 6.0 fps to 8.0 fps for copper alloy tubes with fresh water, and there is a tendency to limit velocities to a maximum of 6.5 fps to 7.0 fps with salt water. For stainless steel tubes the velocity has no real maximum limit, and there is some evidence indicating that a minimum velocity may be desirable to maintain better cleanliness. Applications at 7.0 fps up to 10.0 fps have been made.

When a range of velocities is permissible for any given installation, the economic evaluation including pumping power will generally indicate the optimum. Similarly, when there is more than one allowable tube diameter, the economic study will indicate the proper selection.

It is worth noting that the condenser requirements are more directly related to the ITD than a given design pressure. For example, a unit designed for 2.50" Hga using 80° F cooling water has essentially the same ITD of 28.7° F as a unit designed for 1.50" Hga using 63° F cooling water; and the total condenser requirements are identical.

Similarly, the designer should realize that a two pass condenser with 30' tubes will have the same surface and water requirements as a single pass condenser with 60' tubes, assuming other variables are held equal. With this in mind, it is understandable that a single condenser shell with 60' long tubes installed parallel to the turbine shaft of a tandem compound four-flow turbine may offer first cost economies over a twin-shell, two-pass condenser unit with 30' long tubes perpendicular to the turbine shaft.

There are some cases that result in a total length of water travel as economically proper, but so long that the terminal temperature difference is less than 5°. This problem can be eliminated by multi-pressure operation should its application prove compatible with the proposed equipment arrangement. The TTD of Example A (single pressure) is 8.36° while those of Example E (multi-pressure) are 11.59° and 11.18° for the first and second compartments respectively. It is apparent that longer tube lengths are possible with multi-pressure operation than one pressure for like terminal temperature differences.

conclusion

The surface condenser represents a large investment for a utility and has a direct influence on sizable related equipment such as circulating water pumps, water line sizes, and motors. The proper selection of the condenser requirements involves a complete knowledge of the actual conditions under which the equipment must operate, including the performance characteristics of the low pressure turbines to be served.

The final result of any specific analysis should be based upon a thorough study of all possible combinations following the principles described in this manual.

TABLE I
SATURATED STEAM TEMPERATURES

Absolute Pressure In. Hg.	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	Absolute Pressure In. Hg.
.2	34.56	35.78	36.96	38.09	39.18	40.23	41.23	42.22	43.17	44.08	.2
.3	44.96	45.83	46.67	47.48	48.28	49.05	49.80	50.53	51.25	51.96	.3
.4	52.64	53.31	53.98	54.62	55.25	55.88	56.48	57.08	57.66	58.24	.4
.5	58.80	59.35	59.90	60.44	60.96	61.48	62.00	62.49	62.99	63.47	.5
.6	63.96	64.43	64.90	65.35	65.81	66.26	66.70	67.13	67.56	67.99	.6
.7	68.40	68.82	69.23	69.63	70.03	70.43	70.81	71.20	71.58	71.96	.7
.8	72.33	72.70	73.06	73.42	73.78	74.13	74.48	74.83	75.17	75.51	.8
.9	75.85	76.18	76.51	76.83	77.15	77.47	77.79	78.11	78.42	78.73	.9
1.0	79.03	79.33	79.64	79.94	80.23	80.52	80.81	81.10	81.39	81.67	1.0
1.1	81.95	82.23	82.51	82.78	83.06	83.33	83.60	83.87	84.13	84.39	1.1
1.2	84.65	84.91	85.17	85.43	85.68	85.93	86.18	86.43	86.68	86.92	1.2
1.3	87.17	87.41	87.65	87.89	88.12	88.36	88.59	88.83	89.06	89.28	1.3
1.4	89.51	89.74	89.97	90.19	90.41	90.63	90.85	91.07	91.29	91.50	1.4
1.5	91.72	91.93	92.14	92.35	92.56	92.77	92.98	93.19	93.39	93.60	1.5
1.6	93.80	94.00	94.20	94.40	94.60	94.80	95.01	95.20	95.39	95.59	1.6
1.7	95.78	95.97	96.16	96.35	96.54	96.72	96.91	97.10	97.28	97.46	1.7
1.8	97.65	97.83	98.01	98.19	98.37	98.55	98.73	98.91	99.09	99.26	1.8
1.9	99.44	99.61	99.78	99.96	100.13	100.30	100.47	100.64	100.81	100.97	1.9
2.0	101.14	101.31	101.47	101.64	101.80	101.97	102.13	102.30	102.46	102.62	2.0
2.1	102.78	102.94	103.10	103.25	103.41	103.57	103.73	103.88	104.04	104.19	2.1
2.2	104.34	104.50	104.65	104.80	104.95	105.11	105.26	105.41	105.55	105.70	2.2
2.3	105.85	106.00	106.15	106.29	106.44	106.58	106.73	106.87	107.02	107.16	2.3
2.4	107.31	107.45	107.59	107.73	107.87	108.01	108.15	108.29	108.43	108.57	2.4
2.5	108.71	108.84	108.98	109.12	109.25	109.39	109.52	109.66	109.79	109.92	2.5
2.6	110.06	110.19	110.32	110.46	110.59	110.72	110.85	110.98	111.11	111.24	2.6
2.7	111.37	111.49	111.62	111.75	111.88	112.01	112.13	112.26	112.38	112.51	2.7
2.8	112.63	112.76	112.88	113.01	113.13	113.25	113.37	113.50	113.62	113.74	2.8
2.9	113.86	113.98	114.11	114.23	114.35	114.46	114.58	114.70	114.82	114.94	2.9
3.0	115.06	115.19	115.30	115.41	115.52	115.64	115.76	115.88	116.00	116.11	3.0
3.1	116.22	116.33	116.45	116.57	116.68	116.79	116.90	117.00	117.11	117.22	3.1
3.2	117.35	117.46	117.57	117.69	117.80	117.90	118.01	118.12	118.22	118.33	3.2
3.3	118.45	118.56	118.67	118.78	118.89	118.99	119.10	119.20	119.30	119.41	3.3
3.4	119.52	119.62	119.72	119.83	119.93	120.04	120.15	120.25	120.35	120.46	3.4
3.5	120.56	120.66	120.77	120.87	120.98	121.07	121.17	121.28	121.39	121.49	3.5
3.6	121.58	121.68	121.78	121.88	121.98	122.08	122.18	122.28	122.37	122.47	3.6
3.7	122.57	122.66	122.76	122.85	122.96	123.06	123.15	123.25	123.35	123.45	3.7
3.8	123.55	123.63	123.72	123.82	123.92	124.02	124.10	124.20	124.30	124.40	3.8
3.9	124.49	124.59	124.69	124.78	124.87	124.97	125.05	125.14	125.23	125.32	3.9
4.0	125.42	125.51	125.60	125.70	125.79	125.88	125.97	126.05	126.13	126.23	4.0
4.1	126.33	126.41	126.50	126.60	126.70	126.78	126.87	126.96	127.05	127.13	4.1
4.2	127.22	127.31	127.40	127.50	127.59	127.67	127.76	127.85	127.93	128.02	4.2
4.3	128.10	128.18	128.26	128.35	128.44	128.52	128.60	128.69	128.78	128.86	4.3
4.4	128.95	129.03	129.11	129.20	129.28	129.37	129.45	129.53	129.62	129.70	4.4
4.5	129.79	129.87	129.95	130.03	130.11	130.19	130.28	130.36	130.44	130.52	4.5
4.6	130.61	130.69	130.77	130.85	130.93	131.01	131.09	131.17	131.25	131.34	4.6
4.7	131.42	131.50	131.58	131.66	131.74	131.82	131.90	131.98	132.06	132.14	4.7
4.8	132.22	132.30	132.38	132.46	132.53	132.61	132.69	132.77	132.85	132.93	4.8
4.9	133.00	133.09	133.17	133.25	133.33	133.41	133.49	133.57	133.65	133.73	4.9

TABLE II

Tube Size O.D.	Sq. Ft. of Area Per Ft. of Length (m)	Value of Constant-C In $U = C \sqrt{V}$	GPM PER TUBE, AT 1.00 FT. PER SEC. VELOCITY (g) TUBE WALL GAUGE - BWG													
			12	13	14	15	16	17	18	19	20	21	22	23	24	
3/8"	0.1636	267	0 406	0 463	0 516	0 566	0 600	0 634	0 680	0 716	0 754	0 770	0 793	0 809	0 826	
3/4"	0.1964	267	0 693	0 768	0 835	0 899	0 941	0 984	1 041	1 086	1 132	1 152	1 179	1 200	1 220	
7/8"	0.2291	263	1 057	1 149	1 231	1 308	1 359	1 420	1 478	1 532	1 586	1 610	1 642	1 666	1 691	
1"	0.2618	263	1 497	1 606	1 703	1 794	1 853	1 913	1 992	2 054	2 117	2 145	2 182	2 209	2 237	
1 1/8"	0.2945	259	2 014	2 140	2 251	2 356	2 424	2 492	2 582	2 653	2 725	2 756	2 798	2 829	2 861	
1 1/4"	0.3272	259	2 607	2 751	2 876	2 994	3 071	3 148	3 249	3 328	3 409	3 443	3 490	3 525	3 560	

TABLE III

VALUE OF (a)

TUBE SIZE O.D.	TUBE WALL GAUGE - B.W.G.												
	12	13	14	15	16	17	18	19	20	21	22	23	24
5/8"	0.2154	0.1086	0.1494	0.1343	0.1457	0.1378	0.1285	0.1220	0.1159	0.1134	0.1102	0.1075	0.1057
3/4"	0.1513	0.1366	0.1256	0.1166	0.1114	0.1066	0.1008	0.0966	0.0926	0.0910	0.0889	0.0874	0.0854
7/8"	0.1140	0.1049	0.0979	0.0921	0.0887	0.0854	0.0815	0.0787	0.0760	0.0748	0.0734	0.0723	0.0713
1"	0.0920	0.0857	0.0809	0.0768	0.0743	0.0720	0.0691	0.0670	0.0650	0.0642	0.0631	0.0623	0.0616
1 1/8"	0.0758	0.0713	0.0678	0.0648	0.0630	0.0612	0.0591	0.0575	0.0560	0.0554	0.0545	0.0539	0.0533
1 1/4"	0.0650	0.0616	0.0589	0.0566	0.0552	0.0538	0.0522	0.0509	0.0497	0.0492	0.0486	0.0481	0.0476

TABLE IV

TUBE MATERIAL CORRECTION FACTOR (b)

TUBE WALL GAUGE - B.W.G.							
Tube Materials	12	14	16	18	20	22	24
Admiralty Metal	0.87	0.92	0.96	1.00	1.02	1.04	1.06
Arsenical Copper	0.87	0.92	0.96	1.00	1.02	1.04	1.06
Aluminum	0.87	0.92	0.96	1.00	1.02	1.04	1.06
Aluminum Brass	0.84	0.90	0.94	0.97	1.00	1.02	1.03
Aluminum Bronze	0.84	0.90	0.94	0.97	1.00	1.02	1.03
Muntz Metal	0.84	0.90	0.94	0.97	1.00	1.02	1.03
90 - 10 Cu - Ni	0.74	0.80	0.85	0.90	0.94	0.97	0.99
70 - 30 Cu - Ni	0.64	0.71	0.77	0.82	0.87	0.90	0.93
Cold-Rolled Low Carbon Steel	0.74	0.84	0.86	0.91	0.95	0.98	1.00
Stainless Steels Type 410/430 Type 304/316 Type 329	0.59	0.65	0.70	0.76	0.82	0.85	0.88
	0.49	0.56	0.63	0.69	0.75	0.79	0.83
	0.54	0.60	0.65	0.69	0.74	0.76	0.78
Titanium (Tentative)				0.71	0.77	0.81	0.85

TABLE V

INLET CIRCULATING WATER CORRECTION FACTOR (f)

Inlet Water-°F	f	Inlet Water-°F	f	Inlet Water-°F	f	Inlet Water-°F	f	Inlet Water-°F	f
30	0.550	45	0.747	60	0.915	75	1.025	90	1.075
31	0.562	46	0.760	61	0.925	76	1.029	91	1.078
32	0.574	47	0.772	62	0.934	77	1.033	92	1.080
33	0.586	48	0.785	63	0.942	78	1.037	93	1.083
34	0.601	49	0.797	64	0.951	79	1.041	94	1.085
35	0.615	50	0.810	65	0.960	80	1.045	95	1.088
36	0.628	51	0.822	66	0.970	81	1.048	96	1.090
37	0.641	52	0.833	67	0.978	82	1.051	97	1.092
38	0.655	53	0.844	68	0.986	83	1.054	98	1.095
39	0.668	54	0.855	69	0.993	84	1.057	99	1.097
40	0.683	55	0.865	70	1.000	85	1.060	100	1.100
41	0.694	56	0.875	71	1.005	86	1.063		
42	0.707	57	0.885	72	1.010	87	1.066		
43	0.720	58	0.895	73	1.015	88	1.069		
44	0.733	59	0.905	74	1.020	89	1.072		

TABLE VI
VALUES OF R CORRESPONDING TO K

K	.000	.001	.002	.003	.004	.005	.006	.007	.008	.009	K
.25	221	222	223	224	224	.225	226	.227	.227	228	.25
.26	229	230	230	231	232	233	234	234	235	236	.26
.27	237	237	238	239	240	240	241	.242	243	243	.27
.28	244	245	.246	246	247	.248	249	.249	.250	251	.28
.29	252	252	.253	.254	255	.255	.256	.257	.258	258	.29
.30	259	260	261	262	262	.263	264	.264	265	266	.30
.31	267	267	268	.269	.270	.270	271	.272	.272	273	.31
.32	.274	.275	.279	.276	277	.278	.278	.279	.280	.280	.32
.33	281	.282	.283	.283	284	285	.285	.286	.287	.288	.33
.34	288	289	290	.290	291	292	293	.293	294	.295	.34
.35	.295	296	.297	297	298	.299	300	300	.301	.302	.35
.36	302	303	304	.304	305	.306	306	.307	.308	309	.36
.37	309	310	311	311	312	313	.313	.314	.315	.315	.37
.38	316	317	.317	.318	319	320	320	.321	.322	.322	.38
.39	323	324	.324	325	326	326	327	.328	.328	.329	.39
.40	330	330	.331	332	332	333	.334	.334	.335	.336	.40
.41	.336	337	338	338	339	340	340	.341	.342	.342	.41
.42	343	344	.344	345	346	346	.347	.348	.348	349	.42
.43	.350	350	351	351	352	353	.353	.354	.355	.355	.43
.44	356	.356	.357	.358	358	359	.359	.360	.361	.361	.44
.45	362	.363	363	.364	365	365	366	.367	367	.368	.45
.46	369	.369	370	371	371	372	.373	.373	374	.374	.46
.47	375	.376	.376	377	378	378	.379	.379	380	.381	.47
.48	381	.382	383	384	384	385	385	.386	387	.387	.48
.49	387	.388	.389	389	390	390	391	.392	.392	393	.49
.50	.394	.394	395	395	396	397	.397	.398	398	.399	.50
.51	400	400	401	.401	402	403	403	.404	404	405	.51
.52	406	406	407	407	408	409	409	.410	.410	411	.52
.53	.411	.412	413	414	414	415	415	.416	.417	.417	.53
.54	417	418	.418	419	.420	420	421	.421	.422	423	.54
.55	423	.424	.424	.425	425	426	427	.427	.428	428	.55
.56	.429	.429	430	.430	431	432	432	.433	433	434	.56
.57	.435	435	436	436	437	437	438	.438	439	440	.57
.58	440	441	441	.442	442	443	443	.444	445	445	.58
.59	.446	446	447	447	448	449	449	.450	450	451	.59
.60	451	452	452	453	453	454	.454	455	455	456	.60
.61	457	457	458	458	459	459	460	.460	461	.461	.61
.62	462	.463	.463	464	464	465	465	.466	466	467	.62
.63	467	468	468	469	470	470	471	.471	472	472	.63
.64	.473	.473	474	474	475	475	476	.476	.477	.477	.64
.65	.478	479	.479	480	480	481	481	.482	482	483	.65
.66	483	484	484	485	485	486	486	.487	487	488	.66
.67	488	489	489	490	490	491	491	.492	.492	493	.67
.68	493	494	494	495	495	496	496	.497	497	498	.68
.69	.498	499	499	500	500	501	501	.502	502	503	.69
.70	.503	.504	504	505	505	506	506	.507	507	508	.70
.71	508	.509	509	510	510	511	511	.512	.512	513	.71
.72	513	514	514	515	515	516	516	.517	517	518	.72
.73	518	519	519	520	520	521	.521	.522	.522	522	.73
.74	523	.523	524	524	525	.525	.526	.526	527	527	.74
.75	528	528	529	529	529	530	.530	531	531	532	.75
.76	532	533	533	534	534	535	.535	.536	.536	537	.76
.77	537	537	538	538	539	539	540	.540	541	541	.77
.78	542	542	543	543	543	544	544	.545	545	546	.78
.79	546	547	547	548	548	548	549	.549	550	550	.79
.80	551	551	552	552	552	553	553	.554	554	555	.80
.81	.555	556	556	.557	557	557	558	.558	559	559	.81
.82	560	560	560	561	561	562	562	.563	563	564	.82
.83	564	564	565	565	566	566	567	.567	567	568	.83
.84	568	569	.569	570	570	570	571	.571	572	572	.84
.85	.573	.573	573	574	574	575	.575	.576	.576	576	.85
.86	577	.577	578	578	578	579	.579	.580	.580	581	.86
.87	581	581	.582	582	583	.583	.584	.584	.584	585	.87
.88	585	586	586	586	587	.587	.588	.588	.588	589	.88
.89	.589	590	.590	591	591	.591	.592	.592	.593	593	.89
.90	593	594	.594	595	595	.595	596	.596	.597	597	.90
.91	598	598	598	599	599	600	600	.600	601	601	.91
.92	602	602	602	603	603	603	604	.604	.605	605	.92
.93	605	606	606	607	607	608	608	.608	609	609	.93
.94	609	610	610	611	611	611	612	.612	.613	613	.94
.95	613	614	614	614	615	615	.616	.616	.616	617	.95
.96	617	617	618	618	619	619	620	.620	.620	620	.96
.97	621	621	622	622	622	.623	.623	.624	624	624	.97
.98	625	625	625	626	626	.627	.627	.627	628	628	.98
.99	628	629	629	630	630	630	631	.631	.632	.632	.99
K	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09	K
1.0	632	636	639	643	647	650	.654	657	.660	664	1.0
1.1	667	670	674	677	680	683	.687	690	693	696	1.1
1.2	699	702	705	708	711	714	.716	719	722	725	1.2
1.3	728	730	733	736	738	741	.743	746	748	751	1.3
1.4	753	756	758	760	763	765	.768	770	772	775	1.4
1.5	777	779	781	784	786	788	.790	792	794	796	1.5
1.6	798	800	802	804	806	808	.810	812	814	816	1.6
1.7	817	819	821	823	825	826	.828	830	.831	833	1.7
1.8	.835	836	838	840	.841	843	.844	846	.847	.850	1.8
1.9	850	852	853	855	856	858	.859	861	.862	.863	1.9
2.0	865	866	867	869	870	871	.873	874	875	876	2.0
2.1	878	879	880	881	882	884	.885	886	887	888	2.1
2.2	.889	890	891	893	894	895	.896	897	898	899	2.2
2.3	900	901	902	903	904	905	.906	907	907	908	2.3
2.4	.909	910	911	.912	913	914	.915	915	916	917	2.4
2.5	918	919	920	920	921	922	.923	924	.924	.925	2.5

APPENDIX

Design of condenser monitoring equipment described was based on the solution of two fundamental simultaneous equations, viz., the HEI heat transfer through the condenser tubes, equation (1), and the circulating water enthalpy increase, equation (2).

$$q = US\Theta_m \text{ Btu/hr} \quad (1)$$

where the condenser heat transfer coefficient, U , in Btu/hr/sq ft/deg F = $CF_1F_2F_3\sqrt{V}$

$$q = 3600 \rho AVC_p(t_o - t_i) \text{ Btu/hr} \quad (2)$$

Combining (1) and (2)

$$CF_1F_2F_3\sqrt{V}S\Theta_m = 3600 \rho AVC_p(t_o - t_i) \quad (3)$$

but,

$$\Theta_m = \frac{t_o - t_i}{\ln \frac{T_s - t_i}{T_s - t_o}} \quad (4)$$

hence,

$$\begin{aligned} CF_1F_2F_3\sqrt{V}S \frac{t_o - t_i}{\ln \frac{T_s - t_i}{T_s - t_o}} \\ = 3600 \rho AVC_p(t_o - t_i) \end{aligned} \quad (5)$$

$$\text{or, } \ln \frac{T_s - t_i}{T_s - t_o} = \frac{CF_1F_2F_3S}{3600 \rho AC_p} \cdot \frac{\sqrt{V}}{V} \quad (6)$$

$$\text{Let } \frac{CF_2F_3S}{3600 \rho AC_p} = K \quad (7)$$

$$\text{Since } \ln a \equiv b \quad \text{Log}_e a \equiv b \quad a = e^b \quad (8)$$

$$\text{then } \frac{T_s - t_i}{T_s - t_o} = e^{-K \frac{F_1}{\sqrt{V}}} \quad (9)$$

Solving for t_o

$$t_o = -(T_s - t_i)e^{-K \frac{F_1}{\sqrt{V}}} + T_s \quad (10)$$

Substituting (10) in (2), letting $C_p = 1$

$$q = 3600 \rho AV (T_s - t_i) \left(1 - e^{-K \frac{F_1}{\sqrt{V}}}\right) \quad (11)$$

Solving for T_s

$$T_s = \frac{q}{3600 \rho AV \left(1 - e^{-K \frac{F_1}{\sqrt{V}}}\right)} + t_i \quad (12)$$

$$\text{but, } P_s = f(T_s) \quad (13)$$

$$\text{Hence, } P_s = f \left[\left(\frac{q}{3600 \rho AV \left(1 - e^{-K \frac{F_1}{\sqrt{V}}}\right)} \right) + t_i \right] \quad (14)$$

NOMENCLATURE

q = Condenser duty or heat load in Btu/hr

C = Condenser tube heat transfer constant

F_1 = Heat transfer correction factor for circulating water inlet temperature

F_2 = heat transfer correction factor for tube material and wall thickness

F_s = Tube cleanliness factor	C_p = Constant pressure specific heat of circulating water, Btu/lb
V = Velocity of circulating water through condenser tubes, ft/sec	t_i = Circulating water inlet temperature, F
S = Total exposed tube surface in condenser, sq ft	t_o = Circulating water outlet temperature, F
Θ_m = Logarithmic mean temperature difference, F	T_s = Steam temperature at condenser inlet corresponding to absolute pressure, F
ρ = Average circulating water density, lb/cu ft	P_s = Absolute pressure at condenser inlet, in. Hg
A = Total effective condenser tube cross-sectional area, sq ft	

INTRODUCTION

The following are the primary factors which affect condenser performance:

1. Cooling water inlet temperature.
2. Cooling water flow rate.
3. Air leakage into the steam space.
4. Air ejector efficiency.
5. Cleanliness of the steam and water-side heat transfer surfaces.
6. Heat transfer area.
7. Plugging of tubes by debris.

Of the above factors the cooling water inlet temperature is independent of control. Cooling water flow rate may be regulated by throttling the butterfly valve in the condenser discharge lines. Air leakage into the steam space, air ejector efficiency and plugging of tubes by debris are largely matters of proper maintenance. Cleanliness of the steam and water-side heat transfer surfaces may be controlled to some extent by employing chlorine treatment to remove algae films on the water side. The heat transfer area can be controlled by replacing plugged tubes.

Condenser performance deteriorates with any adverse change in the factors enumerated above. Day-to-day deterioration, such as fouling, would be indicated by a rise in the water box differential pressure and a subsequent increase in the back pressure. Long-term deterioration would involve a gradual decrease in the heat transfer coefficient as determined under non-plugged conditions. This procedure is primarily concerned with the long-term decrease in the over-all heat transfer coefficient. Day-to-day difficulties are to be considered as maintenance problems.

The equations to be used in determining the actual heat transfer coefficient are the following:

$$U = \frac{q \times \ln \frac{\theta_1}{\theta_2}}{A \times (t_2 - t_1)} \quad (1)$$

$$CF = \frac{U}{U_0} \quad (2)$$

where

q = steam load, BTU/hr

θ_1 = $t_v - t_1$

θ_2 = $t_v - t_2$

t_v = saturation temperature corresponding to back pressure

t_2 = cooling water exit temperature, °F

t_1 = cooling water inlet temperature, °F

U = over-all heat transfer coefficient, BTU/hr x sq ft x °MD

MD = log mean temperature difference, °F; $MD = \frac{t_2 - t_1}{\ln \frac{\theta_2}{\theta_1}}$

A = outside tube surface area, sq.ft

CF = cleanliness factor

U_o = design heat transfer coefficient, corrected to test conditions, BTU/hr sq ft °MD

The steam load, q , cannot be accurately measured in a condenser test. It will therefore be necessary to determine this value in some indirect manner. A plot of steam load vs. gross generation (BTU/hr vs. MW), based on design heat balances, may be used for test conditions with sufficient accuracy. In this manner the steam load may be found from the gross generation. This determination introduces a maximum error of 2% in the steam load value.

The design coefficient of heat transfer, U_o , appearing in Equation (2), must be determined for actual test conditions in order that the calculation of the cleanliness factor, CF , is meaningful. Specifically, U_o must be found for the actual CW velocity. The equation is:

$$U_o = U_B \times F_T \times F_{TM} \times F_C$$

Where

U_o = Over-all heat transfer coefficient

U_B = basic heat transfer coefficient, from curve. (Fig. 1)

F_T = temperature correction factor, from curve. (Fig. 1)

3

F_{TM} = tube material and gauge correction factor, from table. (Fig. 1)

F_C = cleanliness factor, set at unity in determining U_o .

PURPOSE OF THE TEST

The purpose of the test is to determine the following:

1. Cooling water velocity, V .
2. Over-all heat transfer coefficient, U .
3. Cleanliness factor, CF .

TEST CONDITIONS

All required instrumentation should be checked and calibrated before the test. Special attention should be given to the pitot tube calibration, as it is extremely desirable that this measurement be reliable. (An error of ± 1 fps in the CW velocity determination yields a CF value as much as 10% in error.)

For the duration of the test the condenser should be as free from fouling, air leakage, and air ejector malfunctionings as general maintenance procedures will permit.

The test should be conducted at full load and last approximately two hours, or as long as it is necessary to make the proper pitot tube traverses. Before commencement of the test, load conditions must be allowed to stabilize.

REQUIRED INSTRUMENTATION AND DATA

1. Cooling water inlet and exit temperatures ($^{\circ}F$), determined with thermocouple probe. (See Appendix)
2. Cooling water velocity (ft/sec), determined with pitot tube. (See Appendix)
3. Gross generation (MW), determined with kilowatt-hour meter.
4. Back pressure (Inches Hg absolute), determined with Zimmerli gauge.
5. Number of active tubes.

6. Water box differential pressure (Inches Hg), determined with mercury manometer.

The required reading accuracy of measurements are as follows:

1. Temperature to the nearest 0.1 °F.
2. Generation to the nearest 0.1 MW.
3. Differential pressure to the nearest 0.01" Hg.

Note: Since a pitot tube measurement is so time consuming an operation, it may be found impractical to make the necessary traverses during each test of the condenser. Therefore, the pitot tube measurement should only be included in special tests (e.g., those after overhauls). For daily testing the water box differential pressure nomograph determination of velocity will be considered sufficient. This should be compared with the more accurate pitot tube velocity measurement whenever possible.

SAMPLE CALCULATIONS (Heat Transfer Coefficient and Cleanliness Factor)

Morro Bay Unit #2, Test performed June 7, 1961.

Data:

$$t_1 = 52.0 \text{ }^{\circ}\text{F}$$

$$t_2 = 66.3 \text{ }^{\circ}\text{F}$$

water box differential pressure = 8.07 Inches Hg

gross generation = 169.967 MW

back pressure = 1.163 Inches Hg absolute

number of active tubes = 9528

effective tube length = 29.78 ft.

tube OD = 7/8 inch - 18 BWG

tube material = aluminum brass

Calculation:

Cooling water velocity may be determined from the pitot tube measurements in the conventional manner. As a check, or in the event a pitot traverse cannot be made, this velocity should be found from the enclosed nomograph, (Fig. 2), from the known water box differential pressure:

Water box differential pressure reading = 8.07" Hg

Ambient temperature = 65.0 °F

Correction factor (from Fig. 3) = 0.92325

Corrected differential pressure = 7.45" Hg

Entering nomograph (Fig. 2) at 7.45" Hg, and for 29.78 ft., 7/8" OD, 18 BWG tubes, the cooling water velocity is read as 6.02 ft/sec

Temperature of condensation existing in the condenser corresponding to back pressure (from Table 1) $t_v = 83.7$ °F

$$\theta_1 = t_v - t_1 = 83.7 - 52.0 = 31.7 \text{ °F}$$

$$\theta_2 = t_v - t_2 = 83.7 - 66.3 = 17.4 \text{ °F}$$

$$\frac{\theta_1}{\theta_2} = \frac{31.7}{17.4} = 1.822 \quad \ln \frac{\theta_1}{\theta_2} = 0.5999$$

$$U = \frac{q \ln \frac{\theta_1}{\theta_2}}{A(t_2 - t_1)} \quad (1)$$

$$q \quad @ \quad 169.967 \text{ MW} = 747 \times 10^6 \frac{\text{BTU}}{\text{HR}} \quad \left(\text{from curve for Morro Bay \#2 steam load vs gross generation} \right)$$

$$A = \pi \times \text{outside tube dia} \times \text{number of tubes open} \times \text{length of tubes}$$

$$A = \pi \times \frac{7/8}{12} \times (29.78) \times (9523) = 64,998 \text{ sq ft}$$

$$U = \frac{747 \times 10^6 \times 0.5999}{64,998 \times 14.3}$$

$$U = 482 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{MD}$$

$$\text{Design } U_o = U_B \times F_T \times F_{TM} \times F_C$$

for 52° inlet temp. and 6.02 ft/sec CW velocity:

$$U_o = 647 \times 0.86 \times 0.96 \times 1.00$$

$$U_o = 534 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{MD}$$

$$CF = \frac{482 \times 100}{534} = 90.26\%$$

The cost of deteriorated performance incurred as a result of higher than expected condenser back pressure may be determined as in the following example:

Data:

Generation = 60 MW, deteriorated back pressure = 1.8" Hg

From Input-Output (EB/hr vs. MW) curve, fuel input is 113.3 EB/hr

$$\text{Heat rate} = \frac{113.3}{60} = 1.888 \text{ EB/MWhr}$$

From curve in condenser section of plant data book the design back pressure at actual inlet water temperature and approximate CW flow is 1.5" Hg. This figure may be used as a bench mark against which test performance may be compared. Actual back pressure was 1.8" Hg. From the manufacturer's heat rate correction curve a rise in back pressure to 1.8" Hg increases the steam rate by 0.9%. The heat rate will also rise by 0.9%, to 1.009 (1.888) = 1.905 $\frac{\text{EB}}{\text{MWhr}}$. This corresponds to a fuel rate of

(60) x (1.905) = 114.3 $\frac{\text{EB}}{\text{hr}}$. Therefore, the increase in back pressure caused a fuel rate rise of 114.3 - 113.3 = 1.0 EB/hr.

RESULTS OF TEST

1. The coefficient of heat transfer, U, and the cleanliness factor, CF, should be plotted against elapsed time. In this manner the over-all deterioration of the condenser with time may be observed.

2. The cost, in EB/hr, incurred as a result of deteriorated back pressure should be plotted against elapsed time.
3. The cooling water velocity, V , should be plotted against elapsed time.

REFERENCES

ASME Power Test Code - Steam Condensing Apparatus. .

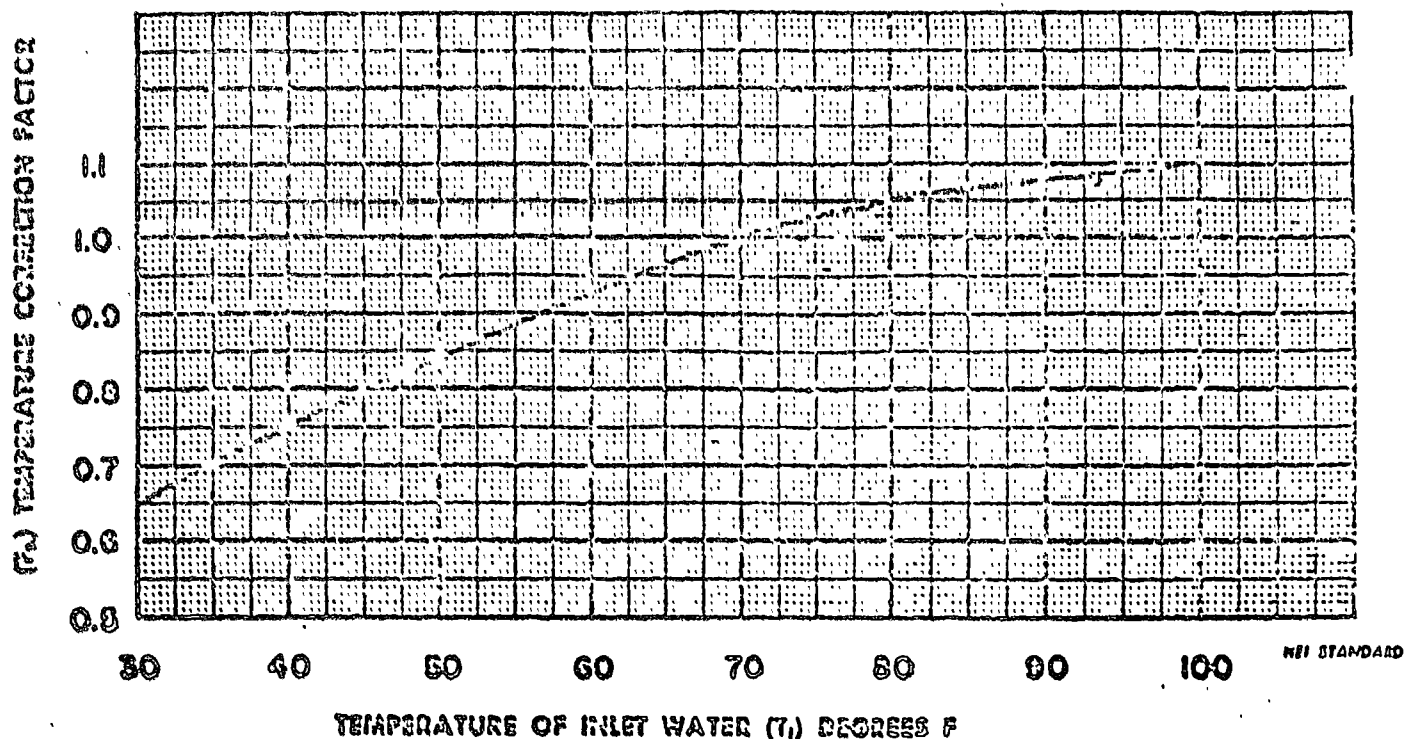
"Proportioning of Surface Condensing Equipment", by Allis-Chalmers Mfg. Co., Manufacturer's Bulletin 19B8201.

"Standards for Steam Surface Condensers", by Heat Exchange Institute, 1955, Fourth Edition.

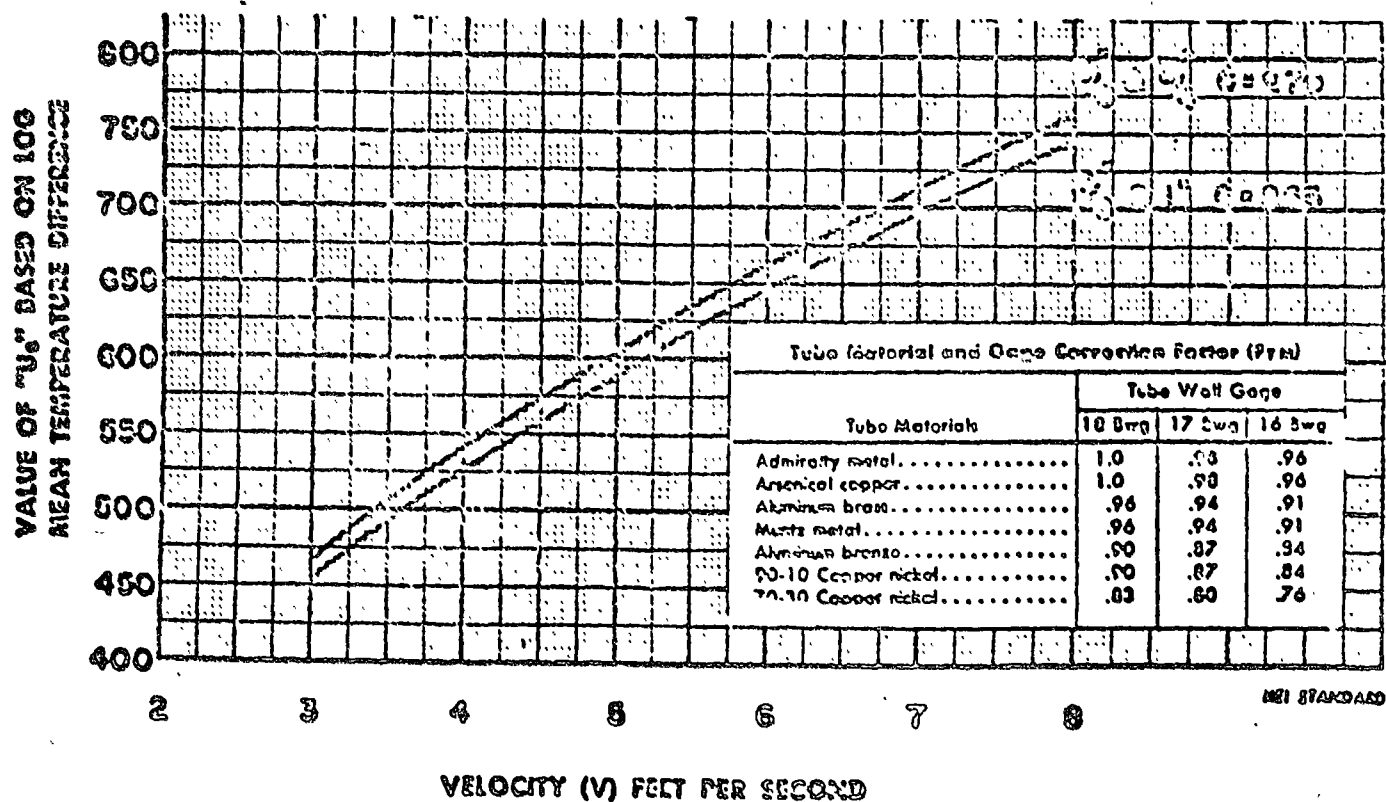
Westinghouse Steam Condenser Calculator and Instruction Manual.

Temperature Correction Factor (F_t)

8



Heat Transfer at 70 Degrees (U_h)



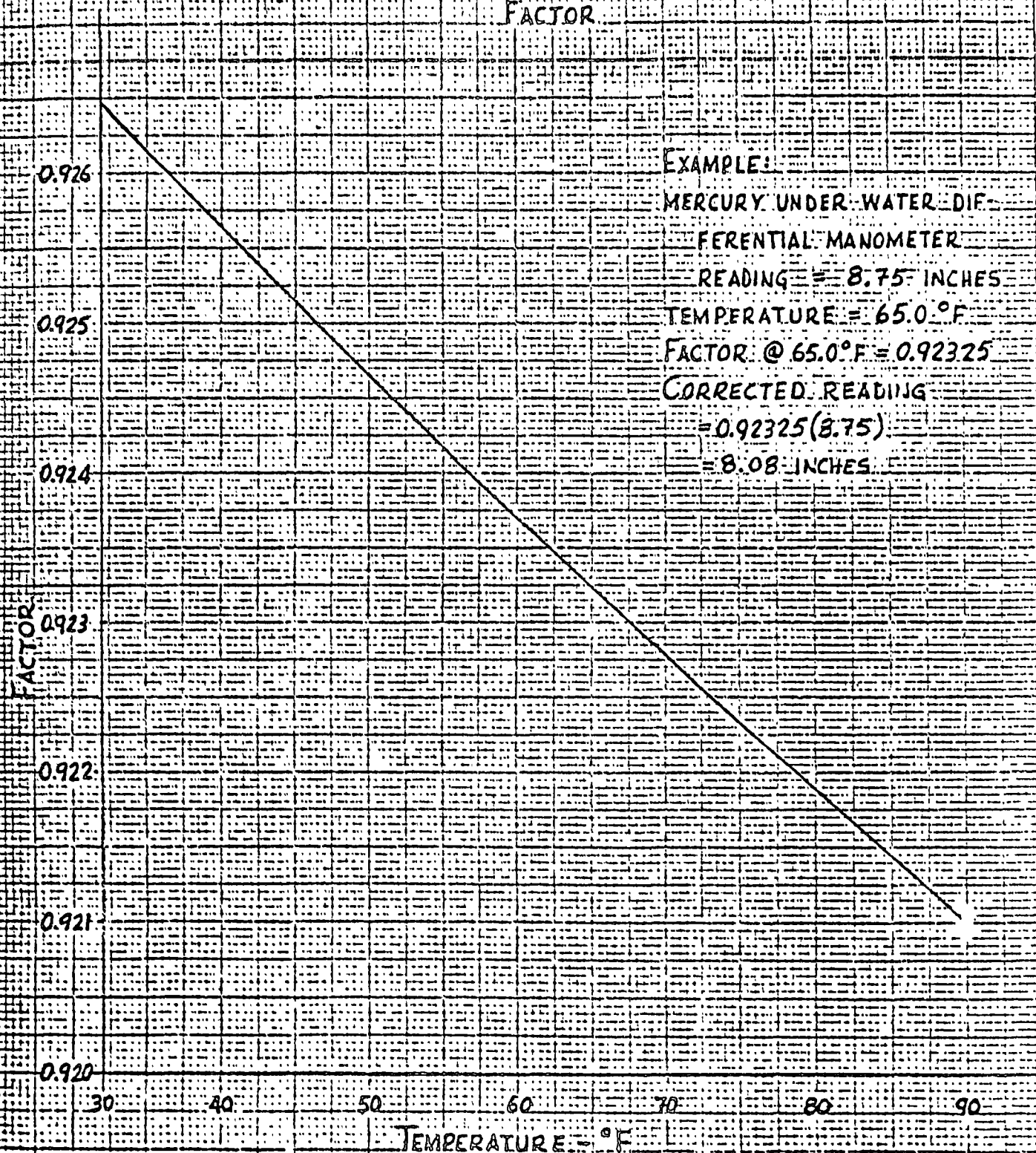
Note 1: Heat transfer coefficients (U_h) based on C x V^{1.8} and on new, clean, bright and oxide free tubes.

Note 2: Curves apply to condensers serving steam turbines.

Note 3: For condensers serving steam engines, use 85% of above values.

FIG. 1

MERCURY COLUMN CORRECTION AND CONVERSION FACTOR



EXAMPLE:

MERCURY UNDER WATER DIFFERENTIAL MANOMETER

READING = 8.75 INCHES

TEMPERATURE = 65.0 °F

FACTOR @ 65.0 °F = 0.92325

CORRECTED READING

= 0.92325(8.75)

= 8.08 INCHES

* CORRECTED FOR WATER LEG AND CONVERTED TO MERCURY AT 32 °F

FIG. 3

PITOT TUBE

10

POWER PLANT _____
UNIT _____
EQUIPMENT _____
TEST DATE _____

DIUS				LOCATION	TRAV. #1		TIME		TRAV. #2		TIME	
No	Radius in.	Sta	Radius in.	INCHES Decimal	Horizontal d. in.	$\sqrt{d}$	Vertical d. in.	$\sqrt{d}$	Horizontal d. in.	$\sqrt{d}$	Vertical d. in.	$\sqrt{d}$
1	.025											
2	.076											
3	.134											
4	.194											
5	.258											
6	.330	1	.051									
7	.408	2	.163									
8	.500	3	.293									
9	.613	4	.453									
0	.776	5	.684									
INTERPOINT (Not in Averages)												
11	1.224	6	1.316									
2	1.367	7	1.547									
3	1.500	8	1.707									
4	1.592	9	1.837									
5	1.670	10	1.949									
6	1.742											
7	1.806											
8	1.866											
9	1.922											
0	1.975											

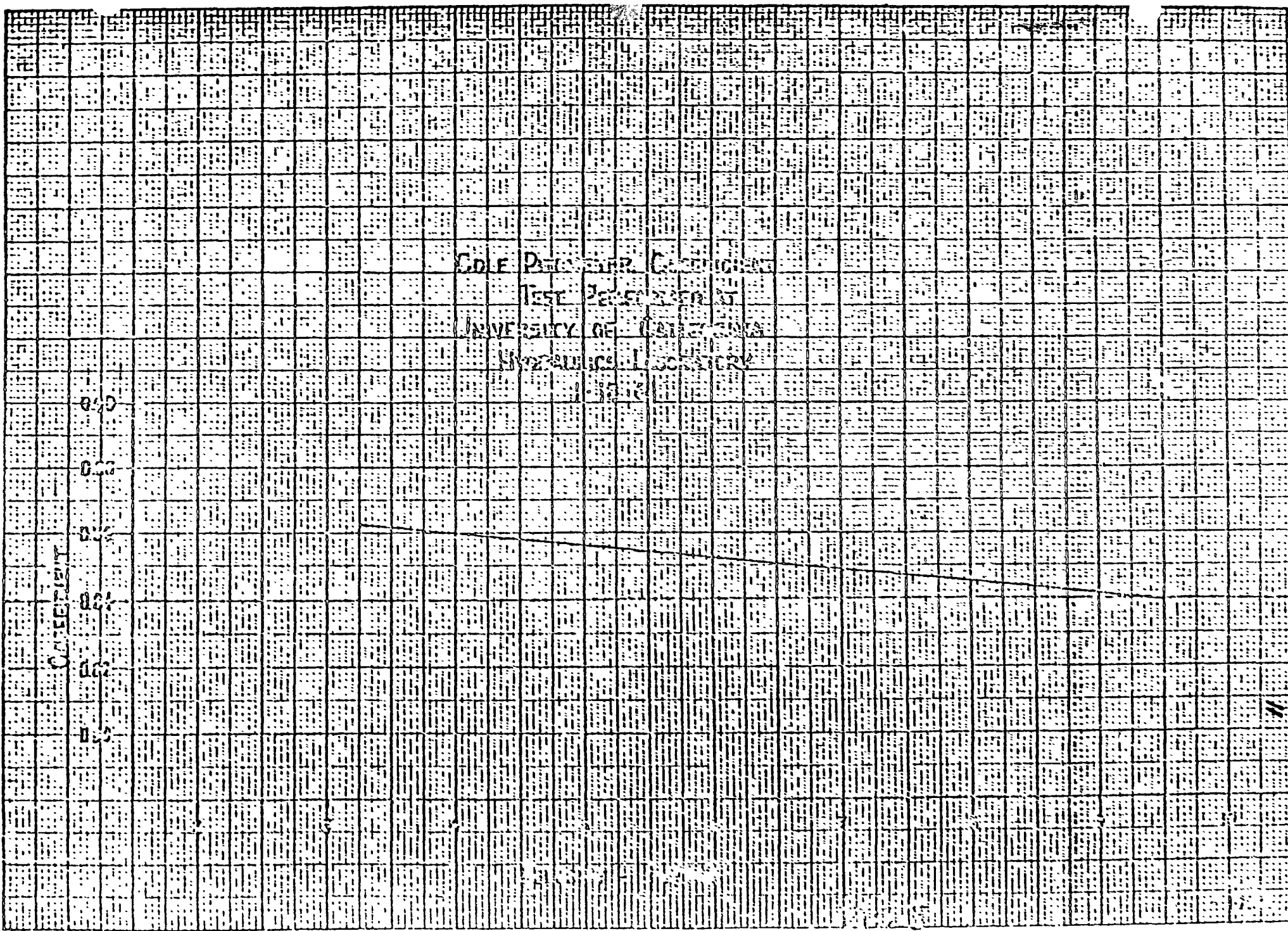
INTERPOINT CHECK RDGS.				Pump
No	Time	d	$\sqrt{d}$	P

Total Avg.	Total Avg. $\sqrt{d}$	Total Avg.	Total Avg. $\sqrt{d}$

Pipe Size, Inches		Mano Fluid	Sp. gr.
Nom.	OD		S

Area Coefficient	Area = .00545 ID ² , sq. ft	$\sqrt{(S-1)}$ if manometer fluid heavier than H ₂ O
A =		

gpm = 1040 x C x A x $\sqrt{(S-1)}$ x $\sqrt{d}$



COLLEGE DEPARTMENT OF CHEMISTRY
TEST DEPARTMENT
UNIVERSITY OF CALIFORNIA
RIVERSIDE, CALIFORNIA

CALIBRATION

0.00
0.02
0.04
0.06
0.08
0.10

APPENDIX

CIRCULATING WATER FLOW AND TEMPERATURE MEASUREMENT

A. Pitot Tube Methods

The rate of flow in pipe lines is frequently measured by means of a Pitot Tube or Pitometer. Their use in measuring rate of flow is essentially an integration of the product of velocity and the area through which velocity exists.

One method of obtaining rate of flow from velocity is to divide the pipe cross section into a number of equal annular areas and to measure the average velocities through these areas by placing the velocity-measuring device at points where these average velocities are assumed to exist. These points are taken to be at the midpoints of the areas, i.e. at points where circles divide these areas in half. This assumes the velocity to vary linearly over the areas considered which (in turbulent flow) is obviously more true near the center of the pipe than near the walls. This assumption does not cause serious error, however, if a large number of annular areas are taken. In general, the velocity distribution is not symmetrical about the pipe center and the average velocity through an annular area is taken to be the numerical average of the two velocities measured in this area. This means that the average velocity in the pipe line is given by the following expressions:

$$V_{\text{pipe}} = \frac{V_1 + V_2 + V_2 + \dots + V_n}{n} \quad (\text{where } n \text{ is the number of velocity measurements taken})$$

This is a simple arithmetical average of velocities existing at special points on the diameter of the pipe line.

Based on this method a standard data sheet (Fig. 4) is suggested for measurement of velocity through a tube. The inside radius, R, multiplied by the various factors in Col. 2 indicate the centers of equal areas. (Col. 3) At these points the manometer deflection of the Pitometer, d, (Col. 7) should be read and recorded.

By averaging the values for $\sqrt{d}$, the velocity can be calculated by the following:

$$V = C \sqrt{\frac{2g}{12}} \sqrt{S-1} \sqrt{d} = C \times 2.317 \times \sqrt{S-1} \times \sqrt{d}$$

where S = gravity of indicating fluid
(water = 1.0)

d = deflection in inches
of indicating fluid

c = Pitometer coefficient

Since the value of C for the Cole Pitometer is velocity dependent, a number of trial solutions must be made in order to arrive at the correct velocity figure. As a first approximation a value for C of 0.860 should be assumed and the corresponding velocity calculated. Then the value of C corresponding to the calculated velocity should be read from the Pitometer calibration curve. (Fig. 5) The calculation should then be repeated with the new value of C until agreement exists between the assumed C and the value of C from the calibration curve to better than ± 0.0005 . The flow is given by: $Q = 449.83 \times \text{velocity} \times \text{area}$
where, Q is flow in GPM.

B. Temperature Measurement

For installations having facilities for inserting thermocouple probes, the temperature of circulating water entering and leaving the condenser should be made by temperature traverses of the pipe lines. The procedure for making these traverses is similar to that outlined above with measurements made at the center of equal areas. Flowing temperature is determined as the velocity weighted average of the individual readings.

Where means for temperature traverses are not available, it is necessary to use existing fixed temperature wells with high accuracy measuring devices. The use of calibrated thermocouples, thermistors or laboratory thermometers with 0.1°F graduations have proved satisfactory for making these measurements.

8-18-69

ECONOMIC SIZING OF CONDENSERS THROUGH THE USE OF THE DIGITAL COMPUTER

MARION J. ARCHBOLD
General Mechanical Engineer

FRANK V. MIHOLITS
Engineer, Office of Manager of Engineering

AMI LEIDNER
Engineer, Mechanical and Building
Engineering Department

CONRAD E. PERSON
Methods Analyst, Program Division Revenue
Accounting Department
Commonwealth Edison Company, Chicago

To properly size a condenser, consideration must be given to using many variables such as tube lengths and diameters, heat transfer surfaces and water velocities, as well as turbine loads and circulating water temperatures. To calculate just a few condenser sizes requires a very large amount of labor and a certain amount of short cutting to bring the work within practical bounds. Even though the work is conscientiously done, the size selection is often under question.

It has been felt that, for the large units which we are now installing, the long-hand evaluation is no longer satisfactory. We realized that for a large (300 Mw) unit even a small improvement in condenser vacuum would result in a considerable saving in operating cost during the 35 year life span of the unit, and it was decided that a more rigorous and refined evaluation should be employed in determining the optimum condenser size. It was for this reason that the Commonwealth Edison Company made use of a digital computer in the selection of several of their recent condensers.

The use of a digital computer still involves a considerable amount of time and labor, but it does increase the accuracy and permits the study to cover a great many more sizes and variables.

This paper discusses some of the more important and interesting aspects in the use of a digital computer for such a study. The conventional formulae¹ used by condenser engineers have been placed in the computer to obtain the total operating and investment cost for the 35 year life of the condenser. The condenser with the lowest summation of costs is then selected.

COMPUTER SELECTION

There are a number of digital computers available throughout the country. The IBM 650 is probably the most widely used computer for engineering problems. This was the one first selected by our computer specialists, but it proved too small for this application.

The problem was reprogrammed and then solved by the IBM 705 computer. Due to the large number of variations in our first study it was necessary to solve for 2860 separate condensers. Besides requiring a considerable amount of time by the computer specialists it took about 30 machine hours. At \$200 per hour, this came close to \$6000 and this amount is difficult to justify.

To reduce this cost on the next similarly sized unit, those variables which were considered to lie outside the range of the problem were eliminated. The number of condensers solved was reduced to 650, or roughly one-fourth the number of the first study, and this resulted in a comparable

saving in computer hours. To reduce the cost still further on future studies, the 704 should probably be considered. It is understood that the 704 should solve a similar problem in less than an hour.

For those who are not familiar with digital computers some explanation is in order. The 705 has been designed for accounting or business purposes, with a large input and output, but a small computation section. The 650 and 704 are primarily intended for engineering problems. Both of these computers have small input and output sections with a large computation section, but the 704 can take a much larger problem than the 650 and, therefore, this type of computer would be ideal for solving condenser problems.

ECONOMICS OF A CONDENSER

The function of a condenser is to produce an economical back pressure on the turbine exhaust. The back pressure is dependent upon a number of variables, the more important ones being condenser surface, circulating water velocity and temperature, and steam flow to the condenser. Since the heat rate of the steam cycle and the capability of the unit are very sensitive to the condenser back pressure, an economic value can be put on the heat rate and capability in terms of fuel cost and plant investment, respectively. Therefore, it is the proper combination of all these factors which determines the optimum condenser size.

The following remarks discuss only those items which are unusual, or are treated in a manner other than is normally practiced.

COST OF COAL

Since there was some doubt that one could peer into the future and forecast the cost of fuel with accuracy for the

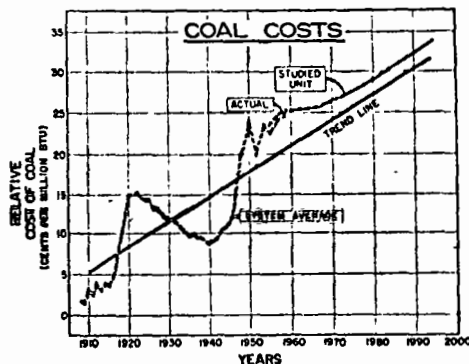


Fig. 1—Past coal prices and projected future prices, showing trend line.

life of the equipment, until recently all calculations for new equipment were based on the present-day fuel cost.

By charting the past 50 year coal cost history, it became possible to develop a projected fuel price. Figure 1 shows this chart, the trend line, and the projected coal cost. The trend line in solid black has been determined by use of the standard trend formula which is by the "sum of the least squares method."

Since the transportation of fuel is such a large proportion of the total fuel cost, it was necessary to adjust the average cost according to the station location. The curve for the unit in the study starts at the actual present-day station fuel cost and gradually assumes a position parallel to the trend line.

Recent units have been evaluated by using the projected fuel cost.

CAPABILITY EVALUATION

Economic studies show that large exhaust annuli are justified for the circulating water temperatures and load factors experienced on most of the Edison units that have recently been purchased. Units with large exhaust annuli show an appreciable increase in capability with a decrease in back pressure.

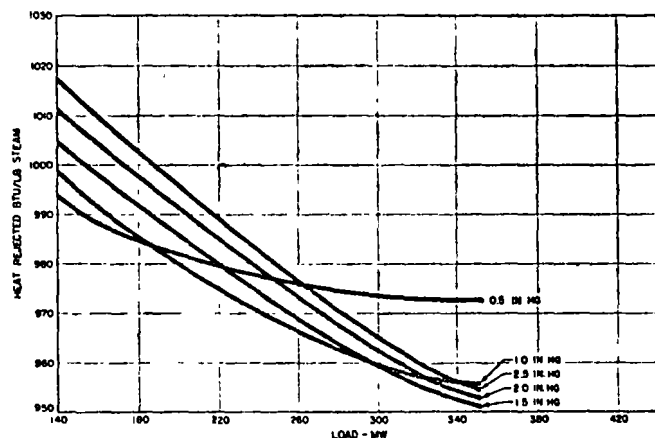


Fig. 2—Heat rejected versus load.

A table showing the capabilities of the turbine with wide open throttle at various back pressures is used to evaluate the capability gains at full plant cost. Some may question the advisability of using the full plant cost and may prefer to use some adjusted or incremental cost. It is well that this item be given serious consideration, since the condenser size is quite sensitive to the evaluated capacity gain. In this study capability gains have been calculated at the summer circulating water temperature.

CIRCULATING WATER TEMPERATURE

The nine-year average circulating water temperature (1950 to 1958, inclusive) for each month of the year was obtained from the station records.

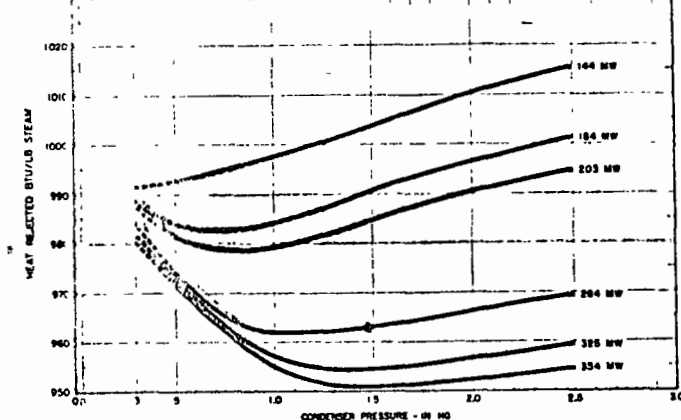
Past practice has been to average the monthly temperatures and then to use the yearly average. It was felt, however, that this might not give an accurate determination of the total fuel consumed by the unit. Since some of the monthly averages were very close, those that were close were averaged and placed into groups as follows:

	deg F
December, January, February and March	38.87
April and November	45.62
May, June and October	54.50
July, August and September	65.92

HEAT REJECTED

In sizing condensers, engineers normally use 950 or 970 Btu per pound of steam as the heat rejected to the cooling water. It is questionable whether this figure is representative of the actual heat rejected for the loads and back pressures at which the unit would operate. In this study the heat rejected was obtained from the turbine engineers for two loads and five turbine exhaust pressures. By plotting heat rejected versus load (Fig. 2) with back pressure as a parameter, the heat rejected at all loads was obtained. Using these curves, a new set of curves were plotted showing heat rejected versus back pressure, Fig. 3, with the economic load points as a parameter. From this final set of curves, the heat rejected was obtained at each of the selected load points for all back pressures. These figures were then

Fig. 3—Heat rejected versus condenser pressure.



used in the calculations in place of the 950 or 970 Btu per pound of steam.

the above throttle flows by using the following equation:

CONDENSER FLOW

To determine the condenser flow, the throttle flow is obtained at some given back pressure (usually 1 or 1.5 in. Hg) and at one or more load points, and these should be stated in the turbine proposal. In this study, the throttle flow at rated load and back pressures between 0.3 and 3.5 in. Hg was determined from the following equation:

$$\text{Throttle flow} = \frac{\text{nominal rating of turbine}}{\text{throttle steam rate}} \times \text{correction factors}$$

The correction factors (Reference 2, Section 7.2, Figs. 7.1a to 7.1f) are for the final feedwater temperature, exhaust annulus area, back pressure, rated generator output, temperature of feedwater leaving the heater at the reheat point, and feedwater temperature rise on the heater above the reheat point.

The condenser flow at all loads and back pressures can be determined from

$$\text{Condenser flow} = \frac{\text{throttle flow at rated load}}{\text{percent of turbine load}} \times \text{correction factors}$$

The correction factors (Reference 2, Section 7.2, Figs. 7.3a and 7.3b) are for the final feedwater temperature at rated generator output and exhaust pressure, and percent of rated generator output. This information was then tabulated and put in the computer program.

LOAD DURATION OR CAPACITY FACTOR

The load duration or capacity factor of the unit is usually the subject of much discussion. For this study the annual capacity factors for the system were reviewed, and the average of these factors was adjusted to reflect the effect that the cost of coal and the operation of the "Automatic Dispatch System" (ADS) might have on the loading of the unit under study. The ADS allocates the load to the various units on the system in ac-

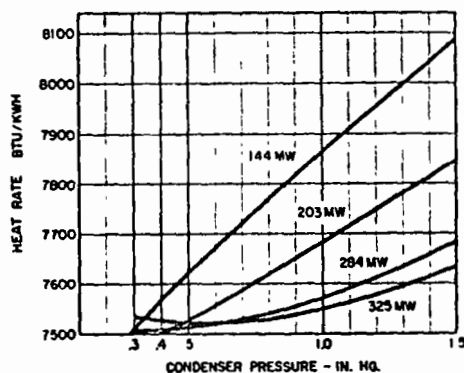


Fig. 4—Turbine heat rate versus condenser pressure.

cordance with the relative cost of the output, and this takes into account the efficiency of the unit, fuel cost, line losses, etc.

For this study it was the judgment of the engineers that the capacity factor over the 35 year life of the unit would be 48 percent.

Using this capacity factor, the loading of the unit for the four loads and three periods selected would be as shown in Table I.

TURBINE HEAT RATE

Figure 4 shows the turbine heat rate plotted against condenser pressure with load as a parameter. The heat rates, as read from these four curves, were tabulated at intervals of 0.1 in. Hg and then placed in the computer program. The curves for the high loads are quite flat and they show little gain at the lower back pressures. The curve at rated load (325 Mw) shows a slight increase in heat rate as the pressure drops.

For units of similar characteristics, it has been the Operating Department's practice to reduce the amount of circulating water for such conditions, thereby improving the heat rate and saving the

pumping costs. Trial calculations indicated that one-pump operation was economical at loads of 325 and 284 Mw for the average circulating water temperatures of 38.87 and 45.62 F, respectively.

COMPUTER PRINTOUT

The application of the digital computer to this problem makes it possible to obtain a large quantity of results which would not be available if longhand and slide rule methods were used. The real problem in reference to printout is to determine in advance the results desired, and then to set the program to obtain them.

Actually, the problem is being programmed to determine the condenser size which will produce the lowest overall cost. To accomplish this, it is only necessary to bring out the summation of all costs and a combination of figures that will identify the condenser size and proportions.

On the first condenser problem, we brought out a rather meager amount of information. Therefore, it was difficult to analyze results, note trends, and do a good job of answering the many questions which our colleagues raised.

The printout on the latest study gives the engineer considerably more information with which to work. The information obtained is as follows:

Identifying Information

1. Condenser surface.
2. Cooling water velocity.
3. Tube length.
4. Tube diameter.

Calculate total cost:

1. Two-pump operation, capability at 100 percent of plant cost.
2. Combination one- and two-pump operation, capability at 100 percent of plant cost.

TABLE I
LOADING OF UNIT

LOAD		1st Period of 10 Years,	2nd Period of 10 Years,	3rd Period of 15 Years,
Mw	Percent	hrs/yr	hrs/yr	hrs/yr
325	100	700	440	0
284	87.5	3330	2150	790
203	62.4	2890	2670	1840
144	44.5	970	1490	2450

3. Two-pump operation, capability at 90 percent of plant cost.
4. Combination one- and two-pump operation, capability at 90 percent of plant cost.

Additional Information

1. Condenser cost (shell plus tube cost).
2. Cost of tubes.
3. Weight of tubes.
4. Friction head of condenser and tunnel.
5. Pumping horsepower (two pumps).
6. Gallons per minute used, two-pump operation.
7. Gallons per minute used, one-pump operation.
8. Total investment cost (present valued).
9. Total pumping cost for three periods (present valued).
10. Fuel cost for each of three periods (present valued).

Two-pump Operation and Combination One- and Two-Pump Operation

11. Condenser vacuums for all load points, cooling water temperatures for two-pump and combination one- and two-pump operation.
12. Flow of 70 F cooling water.

13. Capability (Mw) with 70 F cooling water.
14. Present value capability evaluation at 90 and 100 percent of plant cost.
15. Total cost for each pumping arrangement excluding capability (present valued).

DISCUSSION OF THE COMPUTER PROGRAM

The problem was programmed (Fig. 5) for an IBM 705 digital computer, Model II. The first part of the program uses approximately 20,000 characters of high-speed core storage and one magnetic tape for output. In addition, two magnetic tapes are used for check-point information so that the program may be interrupted at any time or may be restarted in case of machine failure. The program's second part uses one input tape, the output of the first part, and one output tape.

Due to the variations in the decimal point, fixed point scaling was impractical. The calculations therefore, are made in the floating decimal point mode in which the computer itself keeps track of the decimal point. Each floating point number consists of two parts: a mantissa, a number between 0.1 and 1, and an exponent which is the power of 10 that restores the mantissa to its true fixed point form by multiplication. In the com-

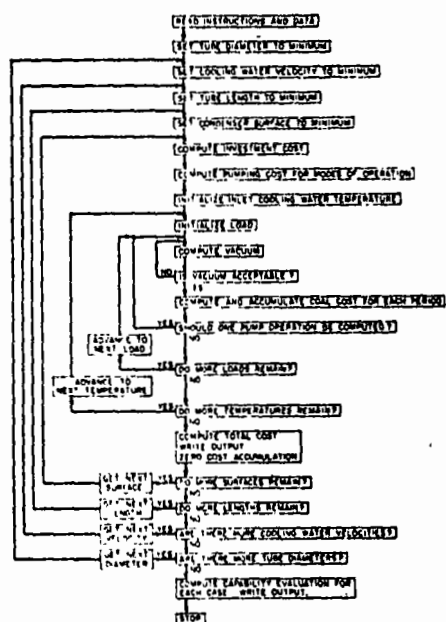


Fig. 5—Computer flow diagram.

puter representation the number 50 is added to the exponent in order that the exponent will always be positive.

Examples:

Fixed Point

—12.345678

.00012345678

Floating Point

Form

—12345678 × 10⁹

.12345678 × 10⁻³

Computer Form

5212345678

4712345678

The first two digits are the exponent; the last eight digits are the mantissa. The sign of the number is superimposed over the least significant digit.

The calculations were divided into two

parts and the program split into two sub-programs. The first computes the total cost without capability evaluation; the second computes total cost including capability credit.

The computer flow diagram, Fig. 5, illustrates the logical order of the program steps. The numbers to the right refer to the numbered descriptions as follows:

1. The program begins with the condenser having the minimum values of the four basic variables: tube diameter, cooling water velocity, tube length, and condenser surface.
2. First, the investment cost is calculated by summing up the computed costs in the following order: (1) shell, (2) erection, (3) erection supervision, and the cost of (4) tubes, (5) circulating water pumps, (6) circulating water pump motors, and (7) pipe, valves, fittings, and screens. The cost of the circulating pumps involves water flow (gpm), and tunnel and condenser head loss.
3. Next, pumping cost is calculated for each of the three periods. Each period is divided into two separate parts, the first covers two-pump operation at all times and the second covers one-pump operation at the two highest loads with the two lowest circulating water temperatures as well as two-pump operation at all other loads and temperatures.
4. The most complicated and time consuming section of the program is the calculation of the condenser vacuum and the coal cost. Starting with the minimum circulating water temperature and the highest load, the pro-

- gram finds the corrected heat transfer coefficient. Computation of the vacuum is done by an iteration process. Assuming an arbitrary initial condenser pressure of 1.0 in. Hg, the program computes a new value for the condenser pressure based on the condenser flow and heat rejected at the assumed condenser pressure. If this new value is not within 0.01 in. Hg of the old value, the new value is used to compute another vacuum and the process is continued until the final vacuum satisfies the test. Normally, about three iterations are necessary. The heat rejected per pound, the total steam flow to the condenser, and the vacuum for the steam temperature are all stored in table form. Linear interpolation between the tabular values was considered sufficiently accurate.
5. After the vacuum is found, the turbine heat rate is computed from the table and the appropriate factors of boiler efficiency, hours, etc., are selected and the coal cost calculated for each of the three periods.
 6. The vacuum and the coal cost calculation is now computed for each temperature and load combination and for each pump operation; the costs are then accumulated for each period and each operation mode.
 7. Lastly, the costs are summed to get the total cost without capability. At this point the results of the problem are written.
 8. The computer will now advance to the next larger condenser surface area and perform a similar calculation. After the largest condenser surface area is reached, the surface is reset to its minimum value and the tube length is increased. This process is continued until each combination of the various condenser surface areas, tube lengths, cooling water velocities, and tube diameters are considered.
 9. In the second part of the program, the vacuum for the maximum load condition is computed at 70 F water temperature, and by using the results of the first section of the program the total cost is adjusted by the capability credit.

ACTUAL OUTLINE OF DATA AND FORMULA FOR THE CONDENSER EVALUATION AS SUBMITTED FOR PROGRAMMING ON THE IBM 705 DIGITAL COMPUTER

THE PROGRAM

To determine the economic sizing of a condenser for a 300 Mw unit, the following variables were used:

1. Nine condenser surfaces, ranging from 150,000 to 230,000 sq ft in 10,000 sq ft increments.
2. Eight tube lengths varying from 38 to 52 ft in increments of 2 ft.
3. Three tube diameters: $\frac{9}{8}$, $\frac{7}{8}$ and 1 in.
4. Three cooling water velocities: $6\frac{1}{2}$, 7 and $7\frac{1}{2}$ fps.

To determine the optimum condenser size the total present valued cost of each size was calculated, covering a 35 year period. This included the carrying charge on investment, coal, and pumping cost, and capability evaluation.

The investment cost is comprised of the cost of the shell, tubes, erection, erection supervision, circulating water pumps and motors, spring supports and support jacks, circulating water pipes, valves and traveling screens.

Items which cost the same for all condenser sizes, such as steam jet air evactors, hotwell pumps, etc., were not included in the investment costs.

The pumping cost covered the cost of the energy required to overcome the total resistance of the circulating water system, including the intake and discharge tunnels, screens, condenser, piping and valves.

The cost of fuel was computed by using four loads, namely: 325, 284, 203, and 144 Mw; four circulating water inlet temperatures: 38.87, 45.62, 54.5, and 65.92 F.

For one condenser size and one circulating water inlet temperature, the heat transfer rate U , from steam to cooling water, is determined for a particular load and at an assumed 1 in. Hg vacuum according to the Heat Exchange Institute Standards. The weight of steam in pounds per hour entering the condenser and the heat rejected per pound of steam to the cooling water are determined. Using these factors the computer calculates the vacuum. If the difference between the assumed 1 in. Hg condenser pressure and the calculated pressure is 0.01 in. Hg or less, the calculated pressure is accepted for the particular load and water temperature. If this difference is greater than 0.01 in. Hg the calculated pressure becomes, in turn, the assumed pressure and the process is repeated until the accepted pressure is obtained. For every pressure, at a particular load, the heat rate is found from the correction curves. The heat rate multiplied by the total number of kilowatthours and by the cost per Btu of coal results in the cost of coal for a particular load and circulating water inlet temperature.

The circulating water pumping cost is calculated separately for two cases:

1. Both pumps operating at all loads and circulating water inlet temperatures.
2. Both pumps operating as above except for the two upper loads and two lower circulating water inlet temperatures, when only one pump operates.

For each vacuum the turbine will have a different capability evaluation. The 315 Mw load at $3\frac{1}{2}$ in. Hg is used as the capability base and is subtracted from the number of kilowatts corresponding to the computed vacuum for each alternate. This difference, multiplied by the investment cost per kilowatt determines the value of the additional output of the turbine with respect to the above base. Since it was considered that maximum demand on the system would occur when the circulating water temperature was at 70 F, capability differences were evaluated only for this temperature. Realizing that, due to the overhaul schedule of other generating capacity, the circulating water might be a limitation on capability at other times of the year, the capability evaluation was made at both 90 and 100 percent of the summer peak capability. The two percentages gave the differences between the two capability charges.

The power used by the circulating water pumps was subtracted from the capability of the machine in determining the capability evaluation.

Determine:

- I. Investment cost, present valued
- II. Pumping cost, present valued, two-pump operation
- IIa. Pumping cost, present valued, one- and two-pump operation
- III. Coal cost, present valued, two-pump operation
- IIIa. Coal cost, present valued, one- and two-pump operation
- IV. Capability evaluation, present valued
- V. Total cost, present valued, two-pump operation *only*:

$$I + II + III - IV = V$$

Va. Total cost, present valued

For two-pump operation at all loads and temperatures, except for one-pump operation for the following:

325 Mw at 38.87 and 45.62 F

284 Mw at 38.87 and 45.62 F

It was determined that the unit would probably run with one circulating pump at these two loads and at all temperatures up to and including 45.62 F.

Given Values

Condenser surface, S 150,000, 160,000, 170,000, ..., 230,000 sq ft

Cooling water velocity, V 6.5, 7.0 and 7.5 fps

Actual tube lengths 38, 40, 42, 44, ..., 52 ft

Effective tube lengths, L 37.75, 39.75, 41.75, 43.75, ..., 51.75 ft

Diameter of tube 0.750, 0.875 and 1.000 in.

I. Investment Cost, Present Valued

Investment cost, present valued = $(A+B+C+D+E+F+G+H) \times \text{factor } X$

The factor represents the present value of the carrying charge for the 35 year life of the equipment. It includes insurance, state and local taxes, depreciation, interest, equity return, and Federal income tax. Since most of the above items vary from year to year and from one plant location to another, it is necessary to check this factor each time a condenser or unit is calculated.

A. Shell Cost

<i>Cost per sq ft</i>	<i>Length of Tube, ft</i>
$\$3.06 \times 1.15$	38
3.01×1.15	40
2.96×1.15	42
2.90×1.15	44
2.84×1.15	46
2.78×1.15	48
2.73×1.15	50
2.69×1.15	52

The factor 1.15 reflects the increased book price of the condenser shell since the publication of these figures up to the date of this condenser study.

<i>Diameter of Tube, in.</i>	<i>Diameter Factor</i>
1.000	1.00
0.875	0.95
0.750	0.90

Shell cost = condenser surface $\times$ cost per sq ft $\times$ diameter factor

B. Erection Cost

Condenser surface, $S \times \$0.85 \times 1.15 =$ erection cost

C. Erection Supervision Cost

Condenser surface, $S \times \$0.13 \times 1.15 =$ erection supervision cost

D. Cost of Tubes

<i>Diameter, in.</i>	<i>Square Feet per Lineal Foot</i>	<i>Pounds per Lineal Foot</i>	<i>*Base Price, \$ per lb</i>
0.750	0.1963	0.419	\$0.6812**
0.875	0.2291	0.493	0.6754
1.000	0.2618	0.568	0.6633

Sq ft per lineal ft $\times$ effective length = area per tube, sq ft

Condenser surface $\div$ area per tube = number of tubes

Number of tubes $\times$ actual length of one tube = total tube length, ft

Total tube length $\times$ 1.05 lb per ft = total weight, lb ***

Base price + extra price (if above 30 ft) = price of tubing, \$ per lb

Price of tubing $\times$ total weight = cost of tubes

E. Circulating Pump Cost

Cost of the pump is determined for each case according to the following formula:

$$\text{Cost of the pump} = 0.5533 \times \text{gpm} (1 + \text{factor})$$

The 0.5533 is based on a hook value \$503 per 1000 gpm + 10 percent for price increase.

The *factor* in the formula represents the deviation from the standard head as follows:

<i>Head, ft</i>	<i>Factor</i>
15-25	$0.075 + 0.04 \times (25 - \text{required head})$
26-40	$0.005 \times (40 - \text{required head})$
41-55	$0.005 \times (\text{required head} - 40)$
56 and up	$0.075 + 0.04 \times (\text{required head} - 55)$

*Over 30 foot tubes up to and including 40 ft add \$0.08 per lb -

Over 40 foot tubes up to and including 50 ft add \$0.10 per lb

Over 50 foot tubes up to and including 60 ft add \$0.12 per lb

**Published prices

***The actual weight per foot of condenser tubes is higher than the published theoretical weight for commercial tubes of the same alloy. Five percent has been added to the published weight to account for this increase in weight.

Total pumping head = friction head in tubes + water box resistance + circulating water pipe, valves and expansion joint resistance + traveling screens resistance + tunnel resistance

Tube and water box resistance per Heat Exchange Institute Standards are as follows:

FRICTION HEAD IN TUBE (Feet of Water per Linear Foot)			
	Tube OD, in.		
Velocity, fps	0.750	0.875	1.000
6.5	0.350	0.283	0.234
7.0	0.400	0.325	0.269
7.5	0.454	0.370	0.304

WATER BOX RESISTANCE (Feet of Water)	
Velocity, fps	Resistance
6.5	1.265
7.0	1.418
7.5	1.570

The circulating water pipe is assumed to be sized for a velocity of about 10 fps, resulting in an average head loss of approximately 0.6 ft per 100 linear ft for all flows. The average head loss in fittings, valves and expansion joints is found to be approximately 2.4 ft for all flows. The circulating water system head loss, excluding the condenser tubes and water box, is, therefore,

$$2.4 + 0.6 = 3 \text{ ft, based on an expected 100 feet of pipe.}$$

The tunnel and screens resistance is formulated as follows:

$$\text{Tunnel and screens resistance} = 2.5 \times 10^{-11} \times (\text{gpm})^2, \text{ ft}$$

The above equation was developed by the Company hydraulic engineer.

The circulating water flow is calculated as follows:

$$\text{Circulating water flow} = \frac{\text{Condenser surface} \times \text{velocity}}{\text{Effective length} \times K}$$

K is a constant depending on the tube diameter, length and gage, and number of passes.

FOR SINGLE-PASS CONDENSER, 18 Bwg	
Tube OD, in.	K
0.750	0.188
0.875	0.155
1.000	0.131

F. Circulating Pump Motors Cost

$$\text{Horsepower} = \frac{\text{Total resistance} \times \text{gpm} \times 8.3368}{33,000 \times 0.906 \times 0.85}$$

8.3368 = lb per gal at 60 F

0.906 = Motor efficiency

0.85 = Pump efficiency

$$= \frac{\text{Total resistance} \times \text{gpm} \times 8.3368}{25,413.3}$$

Cost of circulating pump motors = \$50.00 × horsepower

G. Springs and Support Jacks

Cost of jacks = \$.078 × 1.15 × condenser surface

= \$.08970 × condenser surface

Cost of bolts = \$.031 × 1.15 × condenser surface

These costs were not included in this calculation since this condenser was mounted solidly upon concrete and the expansion was taken care of by a "dog bone" expansion joint.

H. Cost of Pipes and Valves

<i>Circulating Flow Range, in gpm</i>	<i>Cost of Pipes, Valves, Fittings and Screens</i>
150 000-150 900	\$ 88 564
151 000-179 000	113 500
180 000-199 000	114 500
200 000-209 000	128 000
210 000-299 000	130 000

II. Pumping Cost, Present Valued, Two-Pump Operation

Pumping cost, (1) (2) (3) (4)

Period I = Hp × 0.746 × CP × 7890 × Factor A

Pumping cost, (1) (2) (3) (4) (4)

Period II = Hp × 0.746 × CP × 6750 × Factor A × Factor B

Pumping cost, (1) (2) (3) (4) (4)

Period III = Hp × 0.746 × CP × 5080 × Factor C × Factor D

(1) kw/hp (2) cost of power \$/kwhr (3) hr/yr

(4) present value factors

$$\left. \begin{array}{l} \text{Total Pumping Cost,} \\ \text{Two-Pump Operation} \end{array} \right\} = \begin{array}{l} \text{Pumping Cost, Period I + Pumping Cost,} \\ \text{Period II + Pumping Cost, Period III} \end{array}$$

IIa. Pumping Cost, Present Valued, Combination One- and Two-Pump Operation

One-pump operation at 325 Mw at 38.87 and 45.62 F

284 Mw at 38.87 and 45.62 F

Two-pump operation for all other loads and temperatures.

The number of operating hours per year during Period I is 700 hours at 325 Mw and 3330 hours at 284 Mw. The circulating water inlet temperatures of 38.87 and 45.62 F are the monthly average temperatures for four- and two-month periods of the year.

The number of operating hours with one pump during Period I:

325 Mw

$$\begin{array}{ccccccc} 700 & \times & \frac{4}{12} & + & 700 & \times & \frac{2}{12} = 700(0.500) \\ \text{hr/yr} & & \text{fraction of} & & \text{hr/yr} & & \text{fraction of} \\ & & \text{year with} & & & & \text{year with} \\ & & 38.87 \text{ F water} & & & & 45.62 \text{ F water} \end{array}$$

284 Mw

$$\begin{array}{ccccccc} 3330 & \times & \frac{4}{12} & + & 3330 & \times & \frac{2}{12} = 3330(0.500) \\ \text{hr/yr} & & \text{fraction of} & & \text{hr/yr} & & \text{fraction of} \\ & & \text{year with} & & & & \text{year with} \\ & & 38.87 \text{ F water} & & & & 45.62 \text{ F water} \end{array}$$

Total number of operating hours with one pump:

$$0.500(700 + 3330) = 4030 \times 0.500 = 2015 \text{ hr}$$

Based on the characteristic curves of circulating pumps on an existing unit of similar size, the horsepower of one pump operating alone is estimated to be about 0.4 of the horsepower of both pumps operating together. Hence, the equivalent number of hours when using the horsepower of both pumps as a factor will be:

$$2015 \times 0.4 = 806 \text{ hr}$$

The total number of equivalent hours in this case will be:

$$7890 - 2015 + 806 = 6681 \text{ hr}$$

The total number of operating hours per year during Period I is 7890.

Similarly, for Period II:

325 Mw

$$440 \text{ (hr/yr)} \times 0.500$$

284 Mw

$$2150 \text{ (hr/yr)} \times 0.500$$

Total number of operating hours with one pump:

$$(2150 + 440)0.500 = 2590 \times 0.500 = 1295 \text{ hr}$$

$$\text{Equivalent number of hours} = 0.4 \times 1295 = 520 \text{ hr}$$

The total number of equivalent hours are:

$$6750 - 1295 + 520 = 5975 \text{ hr}$$

For Period III:

325 Mw

0

284 Mw

$$790 \text{ (hr/yr)} \times 0.500$$

Total number of operating hours with one pump:

$$790 \times 0.500 = 395 \text{ hr}$$

Equivalent number of hours:

$$0.4 \times 395 = 158 \text{ hr}$$

Total equivalent hours:

$$5080 - 395 + 158 = 4843 \text{ hr}$$

$$\begin{array}{l} \text{Pumping cost,} \quad (1) \quad (2) \quad (3) \quad (4) \\ \text{Period I} = \text{Hp} \times 0.746 \times \text{CP} \times 6681 \times \text{Factor A} \end{array}$$

$$\begin{array}{l} \text{Pumping cost,} \quad (1) \quad (2) \quad (3) \quad (4) \quad (4) \\ \text{Period II} = \text{Hp} \times 0.746 \times \text{CP} \times 5975 \times \text{Factor A} \times \text{Factor B} \end{array}$$

$$\begin{array}{l} \text{Pumping cost,} \quad (1) \quad (2) \quad (3) \quad (4) \quad (4) \\ \text{Period III} = \text{Hp} \times 0.746 \times \text{CP} \times 4843 \times \text{Factor C} \times \text{Factor D} \end{array}$$

(1) kw/hp (2) cost of power, \$/kwhr (3) hr/yr (4) present value factor

$$\left. \begin{array}{l} \text{Total Pumping Cost,} \\ \text{Combination One-} \\ \text{and Two-Pump} \\ \text{Operation} \end{array} \right\} = \begin{array}{l} \text{Pumping Cost, Period I} + \text{Pumping Cost,} \\ \text{Period II} + \text{Pumping Cost, Period III} \end{array}$$

III. Cost of Coal, Present Valued

The cost of coal is to be calculated twice, first for two-pump operation and later for a combination one- and two-pump operation. The cost of coal is to be calculated for each period for each of the four loads, and each of the circulating water inlet temperatures and then totalled as follows: Cost of coal, present valued = turbine heat rate $\times$ kw $\times$ hr per year for the respective load and period $\times$ fraction of year for the respective circulating water inlet temperature $\times$ reciprocal of boiler efficiency $\times$ reciprocal of operating efficiency $\times$ cost of coal per Btu $\times$ present value factor.

Two-Pump Operation—The condenser vacuum must be known in order to determine the heat rate

The following formulae and tables are used to calculate the condenser vacuum for two-pump operation:

Circulating Water Inlet Temperature	Temperature Correction Factor, F_t
38.87	0.739
45.62	0.800
54.50	0.881
65.92	0.971
50.66	0.842
70.00	0.993

Tube Material and Gage Correction Factor (F_{tm}) = 1.000

Tube Cleanliness Factor (F_c) = 0.850

(Determined from station records on condenser performance)

Values For U_b —Basic Heat Transfer Rate
(Btu per sq ft per F per hr)

Tube OD, in.	Velocity, fps		
	6.5	7.0	7.5
0.625 and 0.750	687	713	739
0.875 and 1.000	671	696	722

Corrected Heat Transfer Rate (U) = $U_b \times F_t \times F_{tm} \times F_c$

$$\text{Logarithmic Mean Temp. Diff.} = MD = \frac{\text{heat rejected per lb} \times \text{condenser flow}}{U \times \text{condenser surface}}$$

Use Table II to determine the heat rejected per pound.

The data in Table II were obtained from Fig. 3. Figure 3 is the result of cross plotting the data on Fig. 2.

Use Table III to determine the condenser flow.

$$\text{Temperature Rise} = T_r = \frac{\text{heat rejected per lb} \times \text{condenser flow}}{500 \times \text{circulating water flow, gpm}}$$

TABLE II
TYPICAL HEAT REJECTED TO CONDENSER
(Btu per lb of steam)

Pressure In. Hg Abs.	325 Mw	284 Mw	203 Mw	144 Mw
0.3	982.0	984.3	988.2	991.5
0.5	972.4	974.0	981.5	992.6
1.0	956.6	961.7	979.0	997.5
1.5	954.4	962.8	984.7	1003.3
2.0	956.7	966.0	990.3	1010.0
2.5	959.3	969.3	994.6	1015.3

(Extend Table to 3.5 in. Hg)

These rejection rates were worked up from figures supplied by the turbine engineers. Their figures were plotted against the back pressures supplied and then cross plotted to obtain the rejection rates at 0.1 in. Hg increments at the loads set up in the scope of the work. (See Figs. 2 and 3)

TABLE III
TYPICAL CONDENSER FLOW—TOTAL STEAM
(lb per hr)

Pressure In. Hg Abs.	325 Mw	284 Mw	203 Mw	144 Mw
0.3	1 369 417	1 194 598	864 384	621 580
0.5	1 382 945	1 206 399	872 923	627 720
1.0	1 410 000	1 230 000	890 000	640 000
1.5	1 429 768	1 247 245	902 478	648 973
2.0	1 444 334	1 259 951	911 672	655 584
2.5	1 451 623	1 266 310	916 273	658 893
3.0	1 463 072	1 276 297	923 500	664 090
3.5	1 468 275	1 280 836	926 784	666 451

TABLE IV
TYPICAL HEAT RATES AT VARIOUS BACK PRESSURES AND LOADS
(Btu per kw-hr)

Pressure In. Hg Abs.	325 Mw	284 Mw	203 Mw	144 Mw
0.3	7536	7506	7469	7515
0.5	7521	7512	7524	7623
1.0	7550	7573	7682	7865
1.5	7635	7683	7846	8086
2.0	7742	7789	7997	8300
2.5	7837	7883	8135	8474

These calculations are from correction factors given in the turbine contract for this turbine. This table should be in 0.1 in. Hg increments and extended to 3.5 in. Hg

$$\text{Range} = \frac{-T_r e^{T_r/MD}}{1 - e^{T_r/MD}}$$

Vacuum Temperature = T_v = circulating water inlet temperature + range

Terminal Differential = TD = range - T_r = $T_v - T_2$

The computer calculated the terminal difference by the range minus the temperature rise since these data were already stored in the machine. The circulating water outlet temperature (T_2) was not calculated by the machine.

If TD is less than 5 F then correct T_v as follows:

$$T_v = \text{circulating water inlet temperature} + 5 + T_r$$

The corresponding vacuum at the temperature T_v is placed in the computer program in tabular form for temperatures from 42 to 126 F in 1 F increments in accordance with Reference 3.

If the calculated vacuum is within 0.01 in. Hg of the assumed vacuum the turbine heat rate can be obtained from Table IV.

The turbine heat rate data in Table IV were determined by applying vacuum correction factors to the contract or given data at 1 in. Hg.

	Contract or Given Data at 1 in. Hg			
<i>Mw</i>	325	284	203	144
Throttle Flow, lb per hr	2 070 000	1 770 000	1 225 000	830 000
Gross Turbine Heat Rate, Btu per kw hr	7550	7573	7682	7865

Exhaust Pressure, in. Hg	Throttle Flow, lb per hr			
	2 300 000	2 000 000	1 500 000	1 000 000
0.5	-0.20	-0.45	-1.40	-2.60
1.0	0	0	0	0
1.5	1.00	1.20	1.75	2.50
2.0	2.40	2.60	3.35	4.85
2.5	3.75	3.85	4.80	6.90

The data in the vacuum correction factor table are plotted to obtain Fig. 6. The turbine engineers throttle flow at 325, 284, 203, and 144 Mw at 1 in. Hg is drawn as a vertical line on Fig. 6. The vacuum correction as percent of heat rate can now be read from Fig. 6 at 0.5, 1.0, 1.5, 2.0, and 2.5 in. Hg exhaust pressures at the 1 in. Hg throttle flows. Using these corrections the turbine heat rate at this throttle flow can be calculated for 0.5, 1.0, 1.5, 2.0, and 2.5 in. Hg. These data are plotted in Fig. 4 as heat rate versus vacuum with throttle flow at the load points as a parameter.

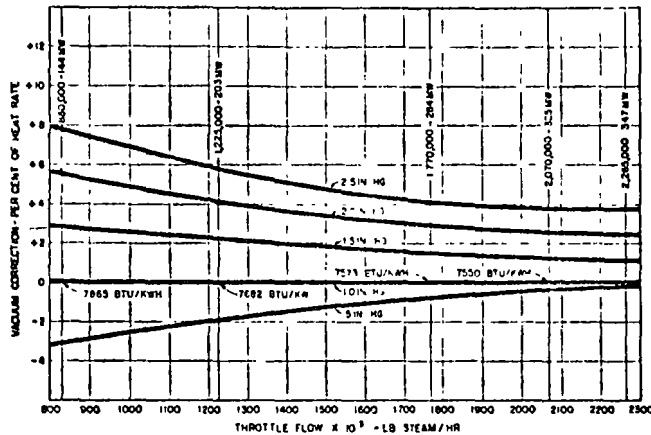


Fig. 6—Vacuum correction curves.

While this article was being written it was realized that the heat rate curve, Fig. 4, was drawn for constant load at the various back pressures, whereas the heat rate was actually obtained for constant throttle flow at the various back pressures. The heat rate is correct at 1 in. Hg but needs some additional modification as the vacuum deviates from that pressure. Since the hours of operation of the unit were set up on the basis of constant load versus back pressure, the turbine specifications should be written to obtain the vacuum correction factors on the basis of constant load. It is believed that, for this particular study, the circulating water inlet temperature was low enough that the condenser selection was unaffected.

The data in Table V are used to determine the remaining items in the cost of coal calculation for the first period.

Example:

Cost of coal for Period I at 325 Mw, 38.87 F circulating water inlet temperature

$$\begin{aligned}
 & \text{Turbine Heat Rate} \times 325,000 \times 700 \times 0.3333 \\
 & \quad \text{(from Table IV at} \quad \text{(kw)} \quad \text{(hr/yr,} \quad \text{(fraction of} \\
 & \quad \text{the calculated} \quad \text{Table V) year, Table I)} \\
 & \quad \text{vacuum)} \\
 \text{Cost of Coal} = & \times \frac{1}{0.8941} \times \frac{1}{0.93} \times \text{CC-1} \times \text{Factor 4} \\
 & \quad \text{(1/blr. eff.,} \quad \text{(operating} \quad \text{(cost of coal} \quad \text{(present} \\
 & \quad \text{Table V)} \quad \text{eff., Table V)} \quad \text{per Btu)} \quad \text{value} \\
 & \quad \quad \quad \quad \quad \quad \quad \quad \quad \quad \quad \quad \quad \text{factor)}
 \end{aligned}$$

Combination One- and Two-Pump Operation—For the combination of one- and two-pump operation the vacuum should be calculated in the same manner except that for the two loads and two circulating water inlet temperatures of the one-pump operation. For one-pump operation the circulation water flow will be approximately 65 percent of flow with two-pump operation. The water velocity through the tubes will be decreased proportionately for one-pump operation.

The basic heat transfer rate table and temperature rise formula must be modified. All the other formulae and tables will remain the same.

Value For U_{11} —Basic Heat Transfer Rate
(Btu per sq ft per F per hr)

Tube OD, In	One-pump Operation, Velocity fps		
	*4.225	4.550	4.875
0.625 and 0.750	556	576	580
0.875 and 1.00	540	561	596

*The velocities of 4.225, 4.550 and 4.875 were calculated on the basis of 65 percent of the two-pump operation velocities of 6.5, 7.0 and 7.5 fps.

$$\text{Temperature Rise (for one-pump operation)} = \frac{\text{heat rejected per lb} \times \text{condenser flow}}{0.65 \times 500 \times \text{circulating water flow, gpm (two-pump operation)}}$$

The coal cost calculation for combination one- and two-pump operation is the same as for two-pump operation when the above modifications are added.

IV. Capability Evaluation

$$\text{Capability Evaluation} = \text{Capability Credit for Turbine} + \text{Capability Charge for Circulating Water Pump Power Requirements}$$

IVa. Capability Credit for Turbine

Table VI shows the turbine capability used in this study in 0.5 in. Hg increments. These figures were calculated from the vacuum correction data and the following formula:

$$kw = \frac{\text{maximum capability at 1 in. Hg}}{100 - \text{percent change in heat rate}}$$

In this study:

$$kw \text{ at 0.5 in. Hg} = \frac{347,000}{100 - .32} = 348,110 \text{ kw}$$

$$\begin{aligned} \text{Capability Credit at 100 percent and 0.5 in. Hg} &= (\text{Computed kw} - \text{Base kw}) \times \frac{PC}{\text{at 2.5 in. Hg}} \times \frac{\text{Factor } X}{(\$ \text{ per kw plant Cost (Present value factor)})} \\ &= (348,100 - 334,393) \times PC \times \text{Factor } X \end{aligned}$$

$$\begin{aligned} \text{Capability Credit at 90 percent and 0.5 in. Hg} &= .90 (348,100 - 334,393) \times PC \times \text{Factor } X \end{aligned}$$

IVb. Capability of Circulating Water Pump Power Requirements

$$\begin{aligned} \text{Capability Charge for Circulating Water Pump Power at 90 and 100 percent} &= \text{Pump Horsepower} \times PC \times \text{Factor } X \end{aligned}$$

352 Volume XXII—Proceedings of the American Power Conference, 1960

V. Total Cost, Present Valued, Two-Pump Operation Only

The total cost is calculated by the following:

$$V = I + II + III - IV$$

Va. Total Cost, Present Valued, Combination One- and Two-Pump Operation

The total cost is calculated by the following:

$$Va = I + IIa + IIIa - IV$$

TABLE V
COAL COST—PERIOD I, 10 YEARS

River Temp.	Load Mw	Hr/Yr	Fraction of Year	$\frac{1}{\text{Blr. Eff.}}$	$\frac{1}{\text{Op. Ratio}}$
	325	700		$\frac{1}{0.8941}$	
38.87 F	284	3330	0.3333	$\frac{1}{0.8953}$	$\frac{1}{0.93}$
Jan., Feb.	203	2890		$\frac{1}{0.8975}$	
March, Dec.	144	970		$\frac{1}{0.8990}$	
	325	700		$\frac{1}{0.8941}$	
45.62 F	284	3330	0.1666	$\frac{1}{0.8953}$	$\frac{1}{0.93}$
Apr., Nov.	203	2890		$\frac{1}{0.8975}$	
	144	970		$\frac{1}{0.8990}$	
	325	700		$\frac{1}{0.8941}$	
54.50 F	284	3330	0.2500	$\frac{1}{0.8953}$	$\frac{1}{0.93}$
May, June, Oct.	203	2890		$\frac{1}{0.8975}$	
	144	970		$\frac{1}{0.8990}$	
	325	700		$\frac{1}{0.8941}$	
65.92 F	284	3330	0.2500	$\frac{1}{0.8953}$	$\frac{1}{0.93}$
July, Aug. Sept.	203	2890		$\frac{1}{0.8975}$	
	144	970		$\frac{1}{0.8990}$	

Coal Cost/Btu = CC-1

Present Value = Factor A

Similar tables can be made for Periods II and III by supplying the respective hours per year, coal cost per million Btu, and present value factor for each period.

STUDY RESULTS

The final results of the calculations performed on 650 separate condensers are covered in Figs. 7, 8 and 9.

Figure 7 shows the change in total cost as the condenser surface area is varied while the tube length (44 ft), water velocity (6.5 fps), and tube diameter ($\frac{3}{4}$ in.) are held constant. This curve shows that the investment for additional surface area can be made on the basis of fuel and capability savings as the area is increased from 150,000 to 220,000 sq ft. For condensers larger than 220,000 sq ft the total cost increases; therefore, the investment cannot be justified. A curve of this shape is obtained for each tube diameter, tube length, and water velocity.

Figure 8 shows the change in total cost in a series of loops as the surface area, tube length and water velocity are varied, and the tube diameter is held constant at $\frac{1}{2}$ in. In general, the trend is for a decrease in total cost as the water velocity is increased from 6.5 fps to 7.5 fps. Operating experience will determine the possible range of the water velocities. At 6.5 fps there is a sharper increase in cost for longer tube lengths than at 7.5 fps. The amplitude of the loops decreases for higher water velocities.

TABLE VI
TYPICAL CAPABILITY AT VARIOUS
BACK PRESSURES

Pressure, in. Hg Abs.	kW
0.5	348 110
1.0	347 000
1.5	343 190
2.0	338 536
BASE 2.5	334 393

This table should be in 0.1 in. Hg increments and extended to 3.5 in. Hg.

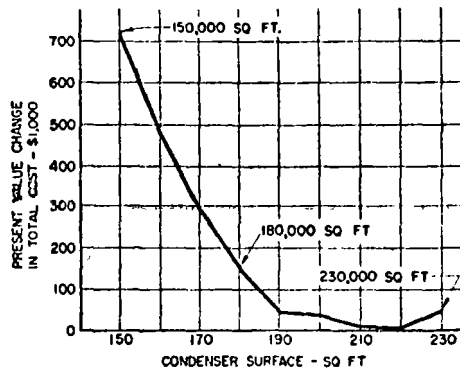


Fig. 7—Present value change in total cost versus condenser surface area for 6.5 fps water velocity, 0.750 in. tube diameter and 44 ft tube length.

Figure 9 shows the change in total cost in a series of loops as the surface area, tube length, and the tube diameter are varied when the water velocity is held constant at 7 fps. In general, the trend is for a decrease in total cost as the tube diameter is increased from $\frac{3}{4}$ in. to 1 in. The $\frac{3}{4}$ in. tube diameter shows a sharper increase in cost for longer tube lengths than with the 1 in. tube. The amplitude of the loops decreases for larger tube diameters.

The study shows that a 180,000 sq ft condenser with a 40 ft tube length, 1 in. tube diameter, and a 7.5 fps water velocity would be the most economical. The 7.5 fps water velocity was considered to be in the range of possible tube erosion and was, therefore, eliminated. The next most favorable condenser with a lower velocity (7.0 fps) was the one having a surface area of 180,000 sq ft, a 40 ft tube length, and a 1 in. tube diameter. The 40 ft tube length resulted in a space limitation.

Upon careful consideration of all these limitations, the decision was made to purchase a 180,000 sq ft condenser having a 44 ft tube length, a 7 fps water velocity and a 1 in. tube diameter.

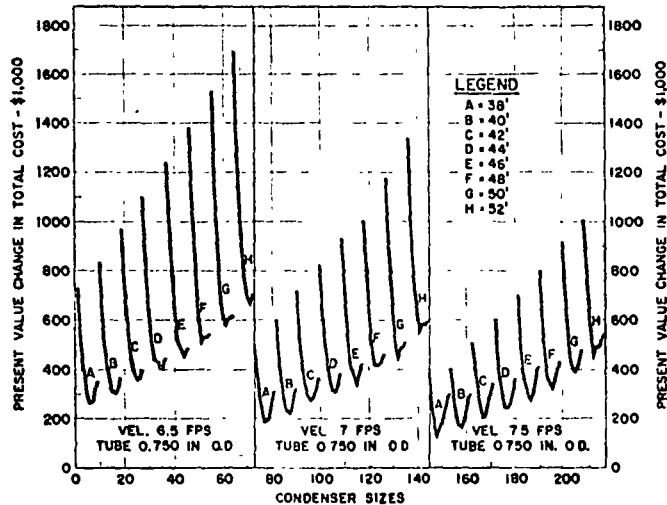


Fig. 8—Present value change in total cost versus condenser surface area for 0.750 in. tube diameter.

DESIRABLE IMPROVEMENTS FOR THE STUDY

1. To develop the tables covering heat rejected to the condenser, turbine heat rate, unit capability, and condenser flow a considerable amount of long-hand calculations were necessary prior to giving this information to the computer programmers. It would be very desirable if the tables could be developed directly by the computer from the given values.
2. The effect of the cut-off point on the calculated back pressure at low circulating water temperatures has not been incorporated in the study due to the inability of programming a formula for it. Since the amount of cut-off correction is directly related to the amount of air leakage, and since most of the condensers on the system operate with less than 3 to 5 cfm air leakage, it was reasoned that the effect of the cut-

off would be negligible if the spare pumps were also placed in service at such times when a single vacuum pump limited the condenser performance. It would, however, be desirable to have a method of inserting the cut-off formula in the program.

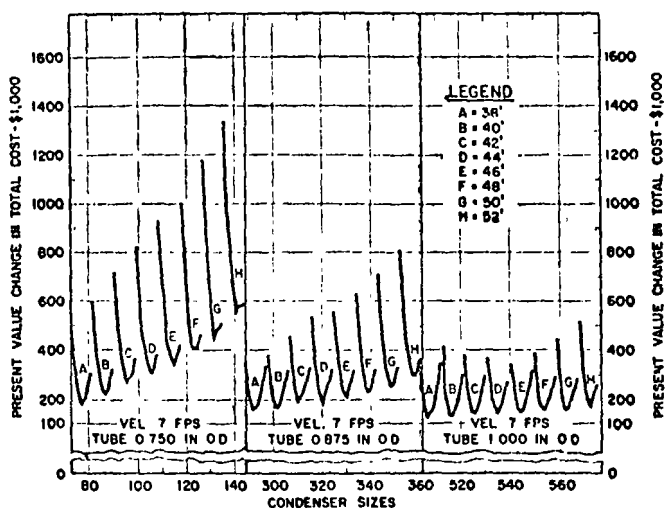
3. A considerable amount of work is involved in reviewing the print-out and plotting the results to determine the most economical condenser. It would be very helpful if the computer could plot curves of the results or list the condensers in the order of their relative economics.

CONCLUSIONS

The results of this study show that the use of a digital computer for a large condenser selection is fully justified.

To obtain optimum range of variables and maximum use of calculation refinements it is necessary to select a computer with ample memory and rapid calculating speed.

Fig. 9—Present value change in total cost versus condenser surface area for 7 fps water velocity.



The program used by Edison should be further expanded to reduce the long-hand calculations to a minimum.

A judicious selection of printout material will aid the designers in their study and understanding of the effects of the condenser variables.

It is important that turbine heat rates, vacuum corrections, condenser flows, and heat rejected be obtained early in the design schedule and that they be accurate. To accomplish this objective, these items should be carefully delineated in the turbine specifications.

Once a condenser problem is well programmed for the computer it is relatively easy to modify it for other condenser studies.

REFERENCES

1. Heat Exchange Institute, "Standards for Steam Surface Condensers," 4th Ed. New York: Heat Exchange Institute, 1955.
2. Bartlett, R. L., "Steam Turbine Performance and Economics." New York: McGraw-Hill Book Co., Inc., 1958.
3. Keenan, J. H. and Keyes, F. G., "Thermodynamic Properties of Steam." New York: John Wiley & Sons, Inc., 1936.

PREPARED DISCUSSIONS

M. K. DREWRY

Chief Engineer of Power Plants
Wisconsin Electric Power Co.
Milwaukee, Wisconsin

One can be quite sure that no exception will be taken to the Authors' conclu-

sion that "the use of a digital computer for a large condenser is fully justified." Appreciation is due the Authors for sharing their studies with others of the power industry.

Somewhat related is the need of including variable turbine characteristics with condenser variables when the tur-

bine and condenser are being integrated for the optimum overall plant results. A large condenser is obviously uneconomic for a turbine with relatively small exhaust area, and vice versa. Both turbine and condenser specifications seem to deserve previous studies wherein the economic sizes of turbine exhaust and condenser surface are determined by equa-

tions that integrate the two. Relatively larger turbine exhausts and condensers than often used presently would seem probably resultant, as study of the slight economy and capability gains with lower exhaust pressure suggested in Fig. 2.

Computer procedures as illustrated by this paper afford good promise of aiding the limitation of power costs.

W. B. LEE

Application Engineer
Allis-Chalmers Manufacturing Company
Milwaukee, Wisconsin

The selection of condensing equipment through evaluation of economic factors pertaining to a given power plant by use of a digital computer is a subject on which there have been no published data. The Authors are to be congratulated for their pioneering work contributing to this problem.

The evaluation has taken into consideration a number of factors that are seldom available to a condenser manufacturer but are important in the solution of the problem. Turbine heat rates and correction factors for various loads and vacuum are generally available only after the turbine and cycle are completely designed. Load duration and capacity factors are difficult to obtain. Water temperatures can be predicted, but method of operation, i.e., one pump or two pump operating periods for various loads and water temperatures, are difficult to predict.

The greatest difficulty in sizing a condenser economically is the ability to accurately predict the method of operation throughout the life of the equipment and

determine the factors to apply. If any of the assumptions or factors used in the evaluation of the condensers are incorrect, the analysis could be completely wrong and the improper sized unit purchased, resulting in increased costs to the owners. This, however, is a risk that cannot be avoided.

Our company has used a 704 calculator for several years in helping solve condenser selections. Our program has been set up to size, price and calculate power requirements of condensers. As each utility has their own ideas on operation and as turbine data as well as plant operating data are difficult to obtain, we have not set up the program to include this information. Our actual time requirement on the 704 is only a fraction of the requirement given by the Authors, due to the method of calculation described above. We have taken the customer's evaluating data, together with the 704 calculations and, through manual calculations and experience, solved the problem. At present, we have not had as complete information on which to base an evaluation as presented here.

We have presented the data from our calculating program to the customer's engineers and allowed them to apply evaluating data as applied to local conditions. Space requirements, as well as

circulating water system, must be considered. We have used data from our calculation program and assumed factors generally accepted or previously used by the industry to determine an economic size of condenser for bid purposes.

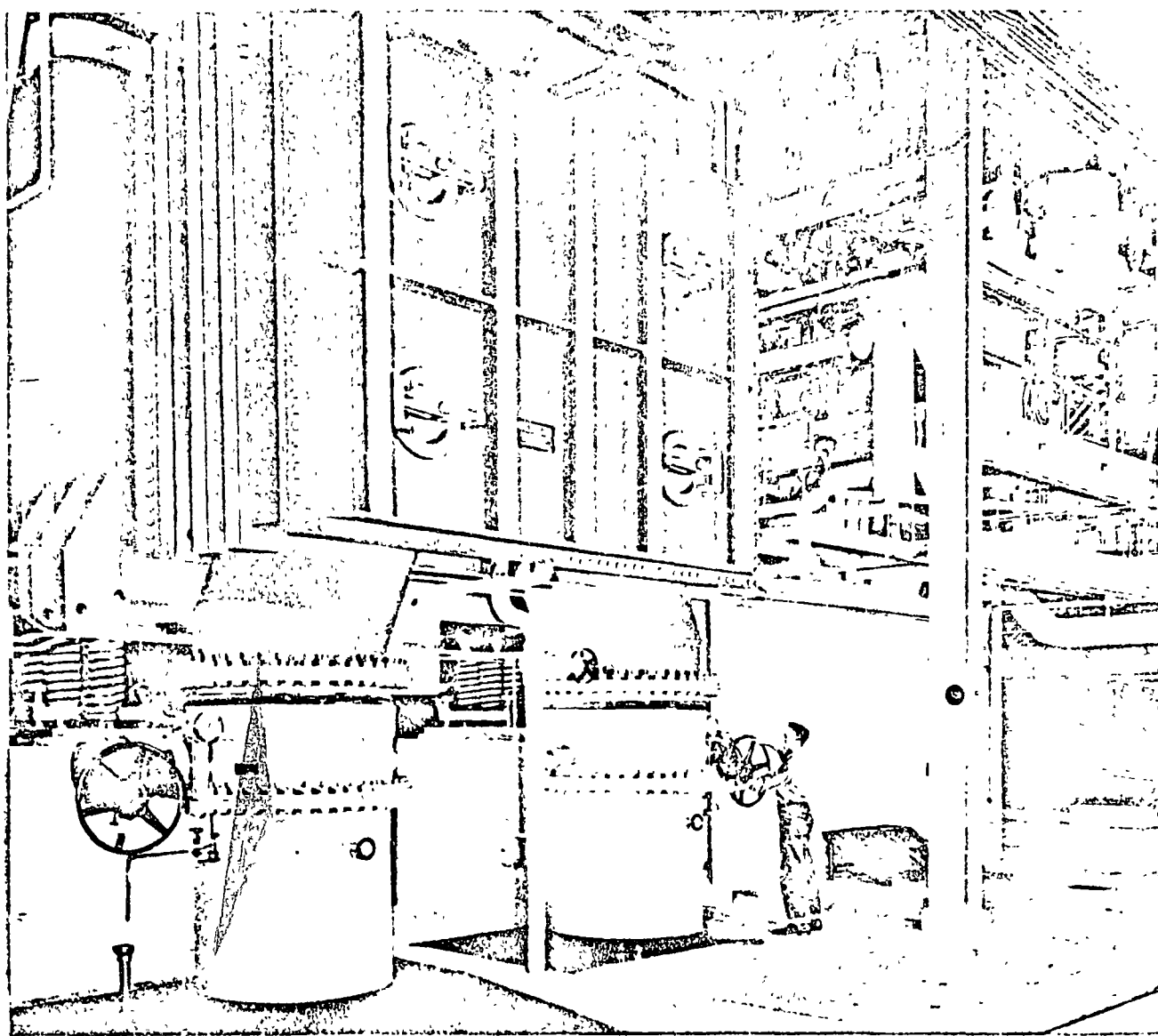
We have plans to set up a second or additional program in which we can take evaluating data and a selected group of condensers from the first program and complete a condenser selection. We are somewhat handicapped in setting up this program as it must be flexible enough to

be able to take various evaluating data from a great number of companies and use any part to evaluate the system.

The programming must be constantly revised to take care of price changes and methods to obtain correct information. For example, tube pricing today allows the use of 40 foot tubes without price penalty whereas, previously, an addition of \$.08 per pound applied. Today, pricing of tubes has been placed on a per foot basis whereas previously it has been on a per pound basis.

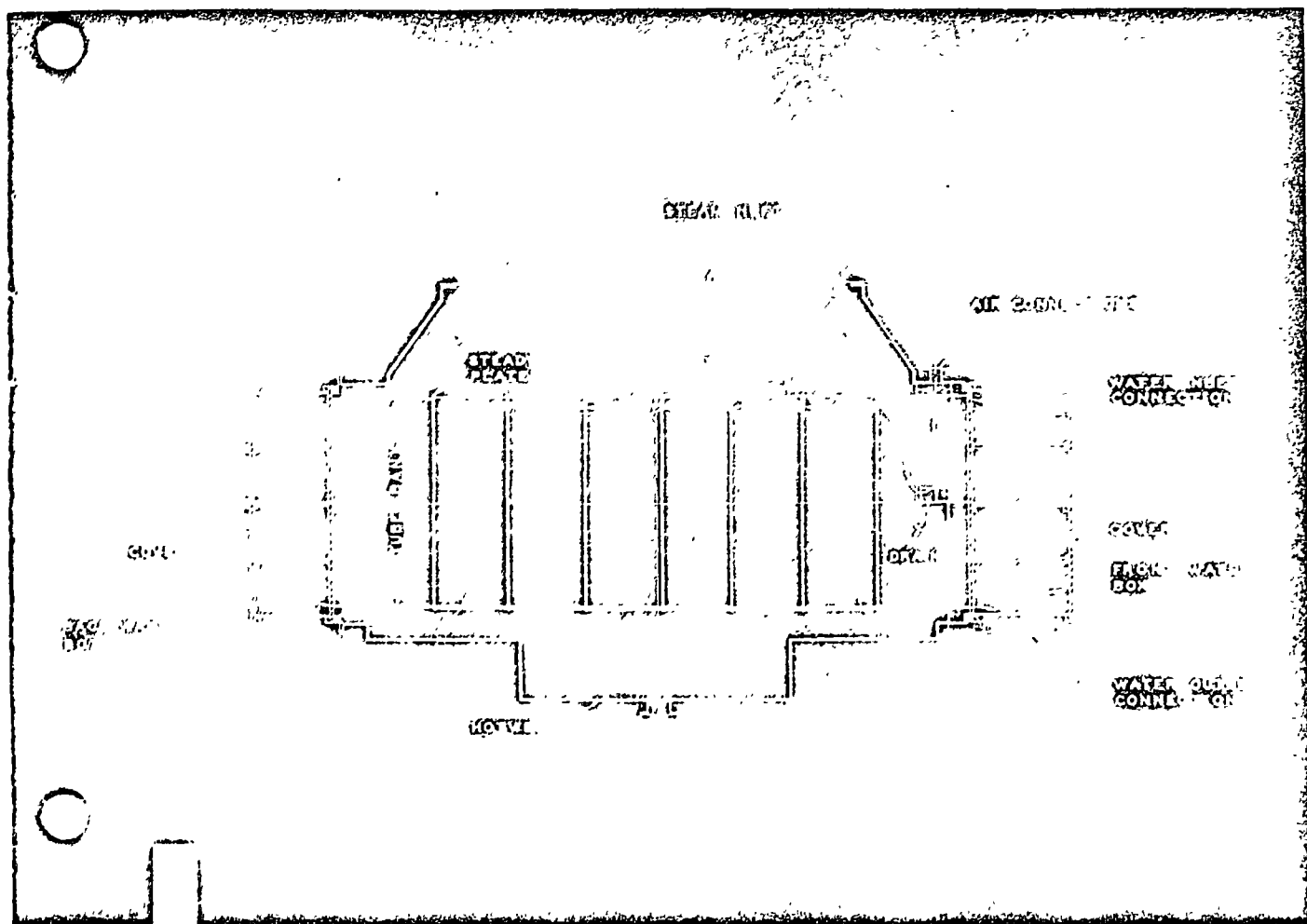
PROPORTIONING OF SURFACE CONDENSING EQUIPMENT

CURRENT TRENDS *in power plant development*



This book is designed to give you the latest data and technical
information on problems of power plant engineering.

ALLIS-CHALMERS - MILWAUKEE, WISCONSIN



Condenser Nomenclature and Definitions

Steam Load (TS) — Pounds of steam per hour entering condenser. Heat rejected to cooling water usually assumed as 950 Btu per pound of steam. (1000 Btu for engine service.)

Condenser Vacuum (Inches of Hg) Sub-atmospheric pressure representing the difference between atmospheric pressure and the static pressure within the condenser. Usually expressed in reference to 30" barometer.

Absolute Pressure (Inches of Hg) Difference between atmospheric pressure and condenser vacuum.

Vacuum Temperature (T_s) — The saturation temperature corresponding to the static pressure of the steam entering the condenser.

Range — Difference between vacuum temperature (T_s) and cooling water inlet temperature (T_1).

Terminal Difference (TD) — Difference between vacuum temperature (T_s) and cooling water outlet temperature (T_2).

Rise — Difference between cooling water outlet and inlet temperatures. ($T_2 - T_1$).

Logarithmic Mean Temperature Difference (MD) Average difference in temperature between cooling water in its

passage through the condenser and of the vacuum steam temperature expressed by the formula—

$$MD = \frac{T_s - T_1}{\text{Log}_e \left(\frac{T_s - T_1}{T_s - T_2} \right)}$$

Condenser Heat Transfer Coefficient (U) — Rate of heat transfer from steam to cooling water expressed in Btu per hour per square foot of condenser surface per degree F logarithmic mean temperature difference.

Condenser Surface (S) — Total condenser surface measured on the outside of tubes and including internal or external air cooler surface expressed in square feet.

Cleanliness Factor — Percent of new and clean heat transfer coefficient.

Cooling Water Velocity (V) — Average velocity of the cooling water through the condenser tubes expressed in feet per second.

Condenser Friction — Pressure loss from circulating water inlet to outlet expressed in feet of water.

Cutoff — Minimum duty point (steam load) at which condenser vacuum is not effected by limitations due to removal equipment.

FACTORS ENTERING INTO THE PROPORTIONING AND SELECTION OF SURFACE CONDENSERS

3

STEAM LOAD—Since the amount of steam for which to design the condenser is not an obvious figure, very careful consideration must be given to this steam load in order to choose an amount consistent with both present and future plans of turbine operation. In general, the unit should be designed to handle the amount of steam it will receive with the turbine operating at full load. In some cases it may be desirable to plan for overload conditions. Usually the condenser is designed to receive the throttle flow steam to the turbine less the amount extracted for feedwater heating or other purposes. In the case of extraction turbines, care should be taken not to select a condenser too small to handle the steam load if the amount of extracted steam should be reduced at some future time.

CONDENSER VACUUM—The condenser should be designed for the highest possible vacuum consistent with the quantity and temperatures of available cooling water, as well as other considerations. The relationship between turbine performance and exhaust vacuum must be considered along with initial condenser costs. For a given cooling water temperature the higher the design vacuum the larger the condenser; and at some point a further improvement in vacuum and the corresponding increase in condenser size will not be justified. The correct vacuum for which to design will involve a careful evaluation study. However, as a "rule-of-thumb," the following relationship between vacuum and cooling water temperature represents good practice:

Cooling Water Temperature (F)	Design Vacuum (inches)	
60.....	29	} Hg ref 30 bar.
70.....	28½	
80.....	28	
85.....	27½	
90.....	27	
95.....	26½	

COOLING WATER—Cooling water may be supplied from such natural sources as rivers, lakes, wells, or the ocean, or it may come from a cooling tower or spray pond. However, from any source the temperature may vary considerably during the course of the year. Therefore, a careful study of the temperature variations of the cooling water during the course of the year is important, for it is necessary to settle on a single temperature as a representative design figure. This should be neither the maximum nor the minimum temperatures, but rather an in-between point which takes into consideration the degree of vacuum which will be obtained at the extreme seasons of the year. As a "rule-of-thumb" it would be permissible to design any condenser for 28 inch vacuum with 80 F cooling water on the theory that the resulting condenser will perform a vacuum consistent with the actual water temperature available.

The temperature rise of the cooling water as it passes through the condenser is not limited in the case of the natural water except by virtue of the degree of vacuum desired. Actually, a rise of around 15 degrees is considered good practice. In general, the rise should be somewhat lower with cooling towers and spray ponds as compared to natural lakes, rivers, etc.

NUMBER OF PASSES—Most condensers are built with either one or two cooling water passes, although in rare instances additional passes may be used. The choice of either the one or two pass type should be a matter of careful evaluation. In general, the following conditions will apply:

If the amount of cooling water is at all limited, a two pass condenser will probably be the best selection, since a single pass unit uses more cooling water than a two pass unit. This is one of the reasons why a two pass type is almost always used in conjunction with cooling towers or spray ponds.

Although the single pass condenser requires more cooling water than the two pass unit, the distance traveled within the condenser is less, so the BHP required to pump the water through the condenser only may be about the same for either type of condenser. The amount of pumping head external to the condenser consequently becomes a point of considerable importance. If this external head is appreciable, the single pass condenser using the larger amount of water will be at a disadvantage.

Single pass condensers are generally the type used very near to a large supply of natural water that does not fluctuate in elevation. Condensers relying on cooling towers, spray ponds, or natural water that fluctuates in elevation, or units located away from these sources are generally of the two pass type.

A single pass condenser is generally somewhat smaller in physical size than an equivalent two pass unit, and this is very often an advantage.

If local conditions are not stipulated, a two pass condenser should be considered for estimating purposes.

TUBE DIAMETER—The selection of the proper tube size is based entirely upon the conditions surrounding the particular installation, and involves a full knowledge of the cooling water characteristics as regards quantity available, temperature and fouling.

Condenser tubes are available in sizes of ¾, ¾, ¾, or 1 inch diameters, although the ¾ inch size is infrequently used. Heat transfer rates and water quantity requirements favor the use of smaller tube diameters, while cooling water friction and fouling characteristics favor the larger diameters. In general, ¾ inch tubes are used in small condensers and ¾ inch or 1 inch sizes in larger units. The following table can be used as a guide in selecting tube diameters:

Condenser Surface
(sq ft)

Tube Diameter (inch)
Two Pass Single Pass

500 to 10,000	$\frac{3}{4}$	$\frac{3}{4}$
10,001 to 20,000	$\frac{3}{4}$ or $\frac{1}{2}$	$\frac{3}{4}$ or $\frac{1}{2}$
20,001 to 50,000	$\frac{1}{2}$	$\frac{1}{2}$
50,001 and up	$\frac{1}{2}$ or 1	$\frac{1}{2}$ or 1

Condenser tubes are available in various wall thicknesses, but 18 BWG (.049 inch) is almost universally accepted as standard.

TUBE LENGTH — The length of condenser tubes increases as condensers become larger. The trend is towards the use of the longest tubes permissible.

The space available for the installation of the condenser, as well as for removal of the tubes, largely govern the selection of the tube length. However, the power required to pump the cooling water through the tubes often places a definite limit on the tube length. Proper proportions to secure effective longitudinal distribution of the steam, which involves the size of the steam inlet, shell diameter, and number of tube support plates also have an important bearing on the maximum length of tube which can be employed.

On page 20, listed under "Recommended Condenser Surfaces and Tube Proportions" are the lengths available for each of the tube diameters for each standard condenser surface. Any of the four lengths shown will result in a well proportioned unit, but in general, the two shorter lengths are most suitable for two-pass condensers, and the two longer lengths for single-pass units. To estimate the approximate amount of condenser surface, figure 10 pounds of steam per square foot of surface for single-pass condensers, and 8 pounds per square foot for two-pass units.

TUBE MATERIAL—Condenser tubes are usually of a copper alloy. Page 20 lists the most frequently used materials and the chemical composition of each. The selection of a proper material requires a careful study of local water conditions.

The most commonly used material, particularly for fresh water installations is Admiralty Metal (70% copper, 29% zinc, 1% tin).

For salt water and brackish waters aluminum brass and the copper nickel alloys are most frequently used.

COOLING WATER VELOCITY—The normal range of velocity of the cooling water through condenser tubes is from 6 to 8 feet per second. The selection of a definite figure is dependent upon water conditions as well as consideration of the friction loss and brake horsepower requirements. Heat transfer is improved with increased velocity but the friction loss also increases. Higher velocities may also have a detrimental effect on tube life due to increased corrosion and abrasive effects. Lower velocities are generally specified for salt water service.

In general, it would be good practice to use a 7 feet per second cooling water velocity for two pass condensers and 7½ feet per second for single pass condensers.

TUBE CLEANLINESS FACTOR—The rate of heat transfer referred to in condenser calculations is based upon commercially clean tubes. In actual operation the tubes cannot be maintained at this standard of cleanliness. Consequently, as tubes become dirty the heat transfer falls off and vacuum is decreased.

It is customary for users to specify a tube cleanliness percent which can be reasonably maintained. Although cleanliness is entirely dependent on the cooling water characteristics, good practice sets a value of 85% of that of clean tubes.

CONDENSER TUBE CONSTANTS

Outside Tube Dia. Inches	Gauge BWG	Tube Wall Thickness Inches	Value for "K" — No. of Tube Passes		
			Single	Two	Three
$\frac{1}{2}$	18	.049	.240	.481	.721
	17	.058	.258	.515	.773
	16	.065	.272	.545	.817
	15	.072	.289	.577	.866
	14	.083	.317	.634	.951
$\frac{3}{4}$	18	.049	.188	.377	.565
	17	.058	.199	.399	.598
	16	.065	.208	.417	.625
	15	.072	.218	.436	.654
	14	.083	.235	.470	.705
1	18	.049	.155	.310	.465
	17	.058	.162	.325	.487
	16	.065	.168	.337	.505
	15	.072	.175	.350	.525
	14	.083	.186	.372	.558
1½	18	.049	.131	.263	.394
	17	.058	.137	.274	.411
	16	.065	.141	.283	.424
	15	.072	.146	.292	.438
	14	.083	.154	.308	.461

CALCULATING CONDENSER PERFORMANCE

Formulae

The following formulae are used to calculate the performance of a given condenser.

FORMULA 1. $V = \frac{G \times L \times K}{S}$

V = Cooling water velocity-ft/sec.

L = Effective length of tubes measured between tube sheets — ft.

G = Cooling water — gallons per minute

K = Constant from table on page 5.

S = Condenser surface — sq. ft.

FORMULA 2. $U = U_B \times F_T \times F_{TM} \times F_C$

U = Corrected heat transfer —
Btu/hr/sq ft/degree MD.

U_B = Basic heat transfer rate from curve, page 7.

F_T = Temperature correction factor from curve, page 7.

F_{TM} = Tube material and gauge correction factor from table, page 7.

F_C = Tube cleanliness %. 85%

FORMULA 3. $MD = \frac{950 \times TS}{U \times S}$

MD = Logarithmic mean temperature difference

TS = Total steam condensed — lbs per hour.

FORMULA 4. Temperature Rise = $\frac{950 \times TS}{500 \times G}$

Note (a) Using the calculated Rise and MD look up the Range from the curves on pages 9-12.

FORMULA 5. $T_s = T_1 + \text{Range}$

T_s = Vacuum temperature

T₁ = Initial cooling water temperature

Note (b) Using calculated Vacuum temperature (**T_s**) look up the corresponding Vacuum from curve on page 13.

FORMULA 6. $TD = T_s - T_2$

TD = Terminal difference, degree F

T₂ = Outlet cooling water temperature

Note (c) TD must always equal 5 degrees F or more. If it calculates to be less, **T_s** must arbitrarily be increased to obtain a 5 degree TD and the vacuum decreased accordingly.

Note (d) Under certain conditions, particularly at light loads, the performance of a condenser may be limited by the air removal equipment. Calculated condenser performance should be checked for possible limitation with the "Absolute Pressure Limit Curve" on page 8.

SAMPLE CONDENSER PERFORMANCE CALCULATION

Examples

Given a 50,000 sq ft Single Pass Condenser equipped with ¾ in O.D. × 18 BWG × 28 ft effective length Admiralty tubes. What vacuum would be expected when condensing 535,000 pounds of steam per hour and utilizing 82,000 gpm of 85 F cooling water? Consider tubes 85% clean.

FORMULA 1. Velocity (**V**) =
 $\frac{82,000 \times 28 \times .155}{50,000} = 7.12 \text{ ft/sec}$

FORMULA 2. Heat transfer (**U**) =
 $701 \times 1.068 \times 1.0 \times .85 = 636 \text{ Btu}$

FORMULA 3. Log mean temp. diff. (**MD**) =
 $\frac{950 \times 535,000}{636 \times 50,000} = 16.0 \text{ F.}$

FORMULA 4. Temperature Rise =
 $\frac{950 \times 535,000}{500 \times 82,000} = 12.4 \text{ F}$

Range (from curve on page 9) = 23.0 F

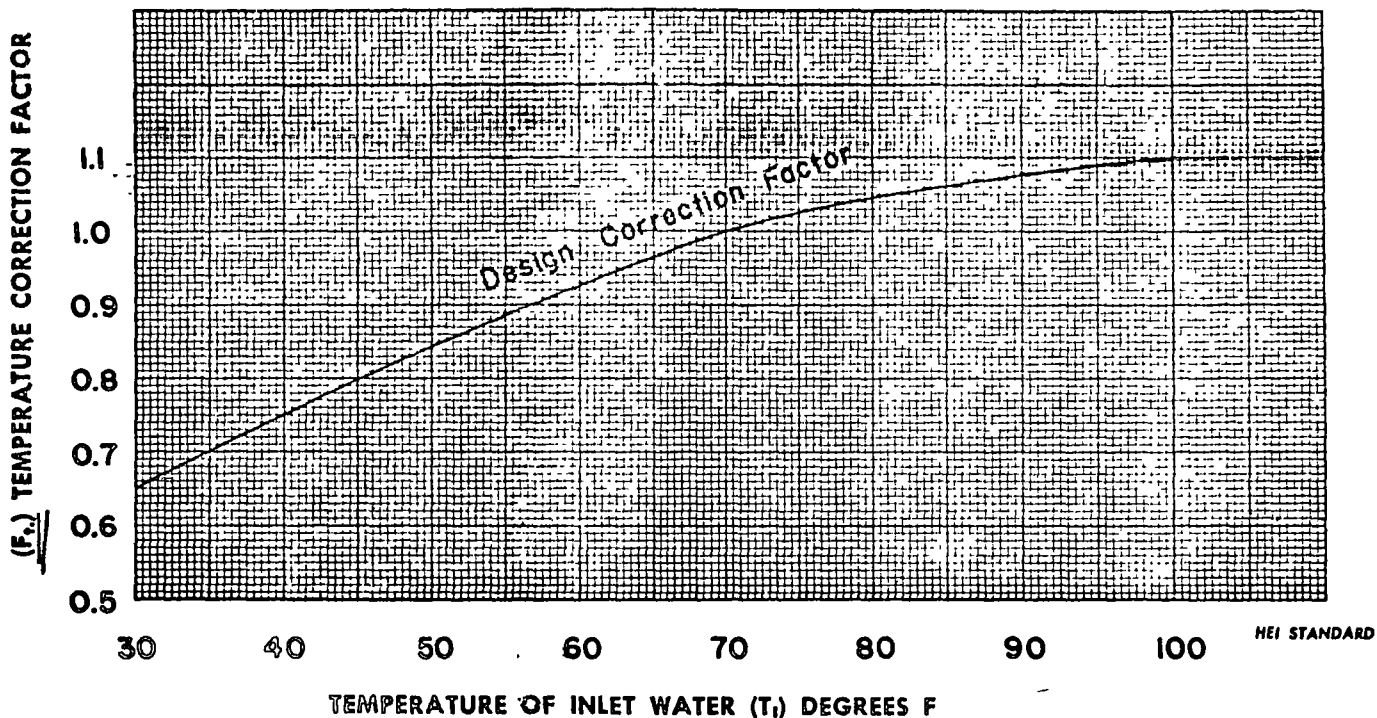
FORMULA 5. Vacuum temperature (**T_s**) =
 $85 + 23.0 = 108 \text{ F}$

Answer Vacuum (from curve on page 13) =
27.55" Hg ref. 30" barometer

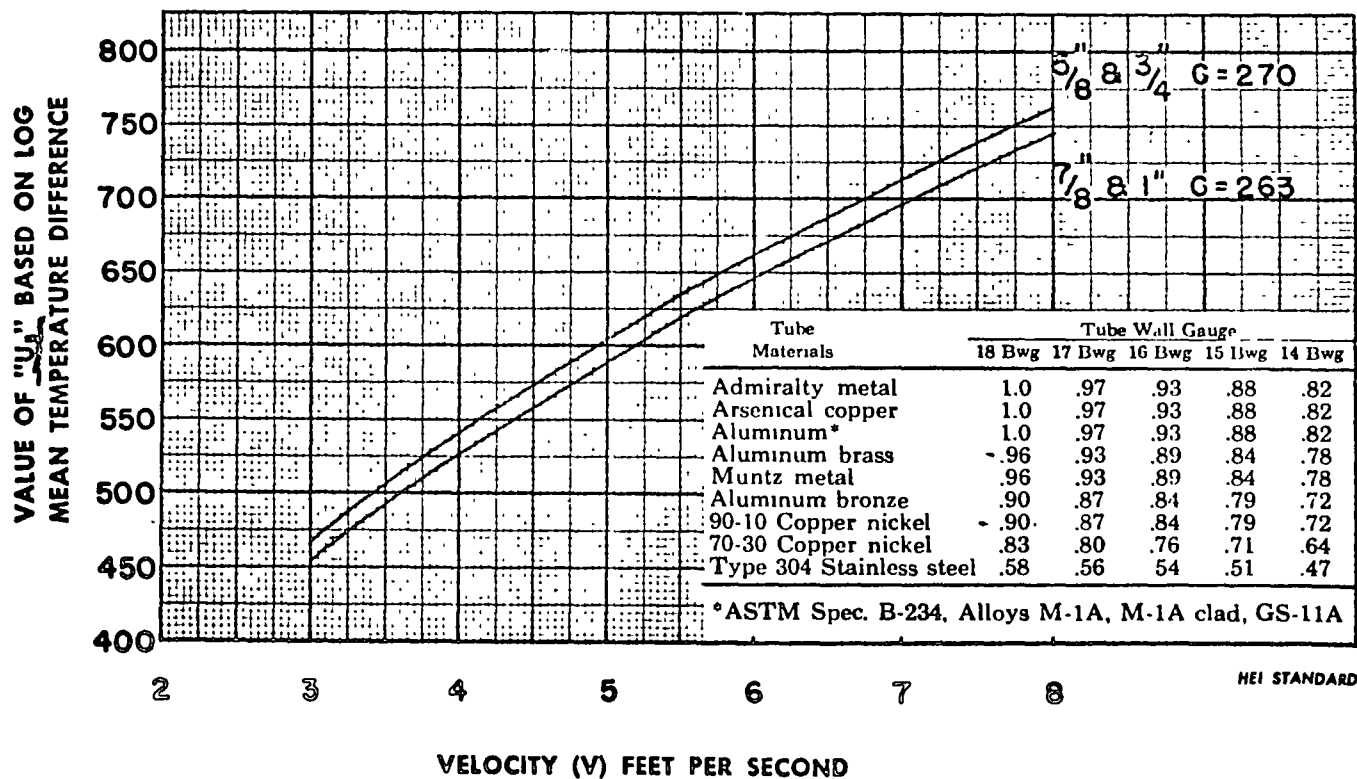
FORMULA 6. Terminal difference (**TD**) =
 $108 - (85 + 12.4) = 10.6 \text{ F}$

Note: From Curve A on page 8 the maximum cutoff vacuum for 85 F cooling water is $(30 - 1.59) = 28.41$ Hg ref. 30 Barometer and therefore the condenser performance is not limited by air removal equipment and the above calculations are correct.

Temperature Correction Factor (F.)



Heat Transfer at 70 Degrees (U_R)



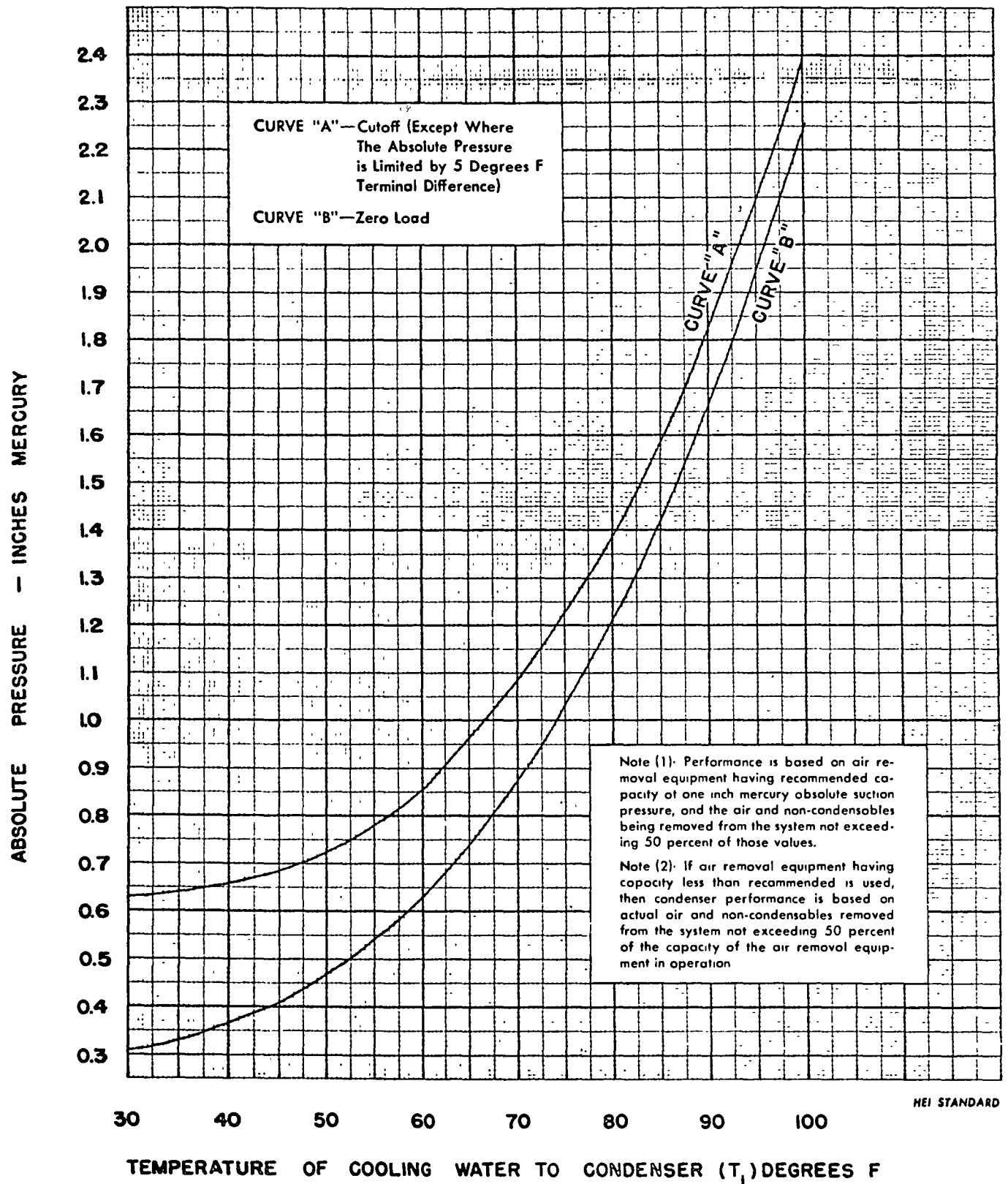
Note 1: Heat transfer coefficient (U_b) based on $C \sqrt{VEL}$ and on new, clean, bright and oxide free tubes.

Note 2: Curves apply to condensers serving steam turbines.

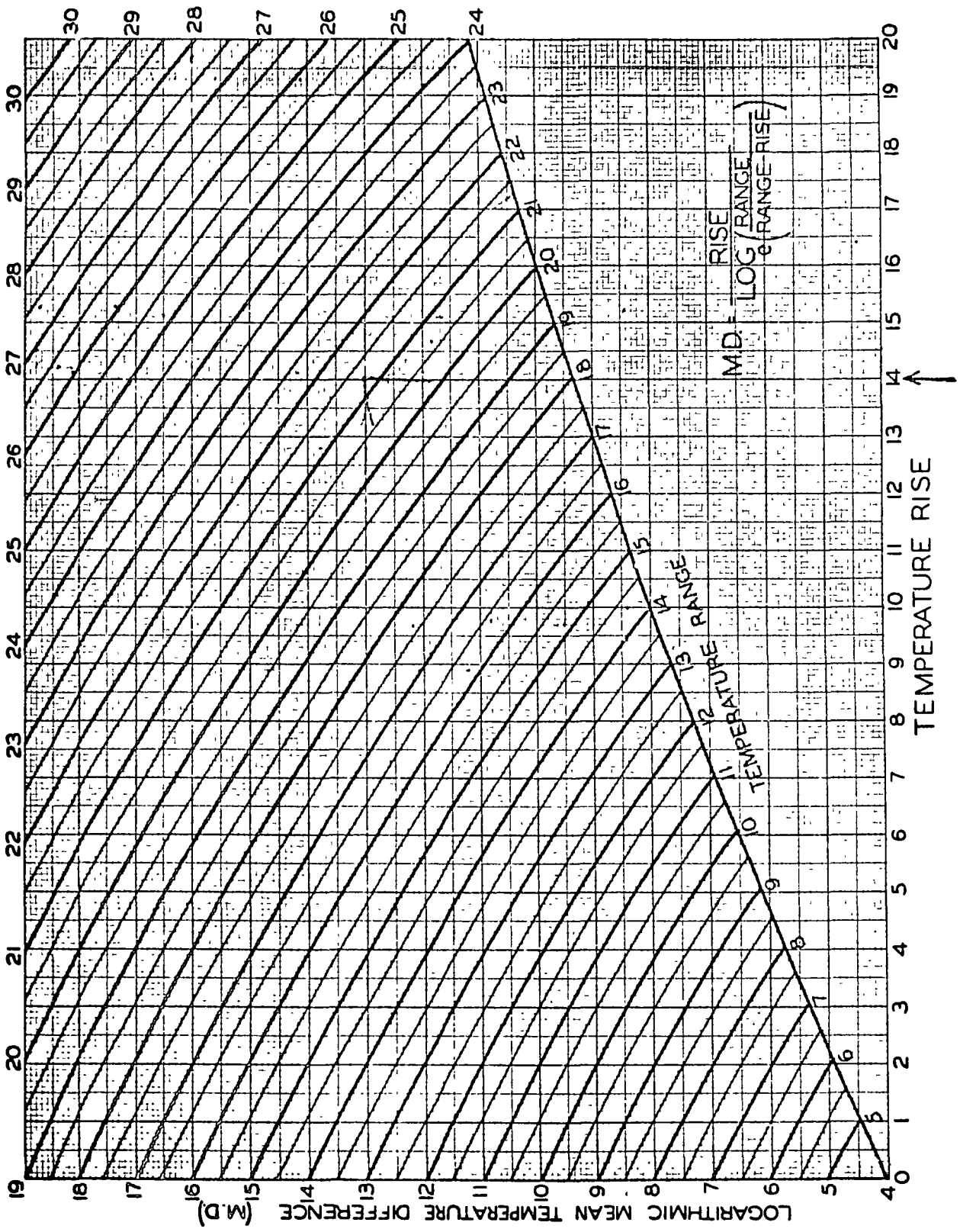
Note 3: For condensers serving steam engines, use 65% of above values.

Absolute Pressure Limit Curve For Steam Turbine Service

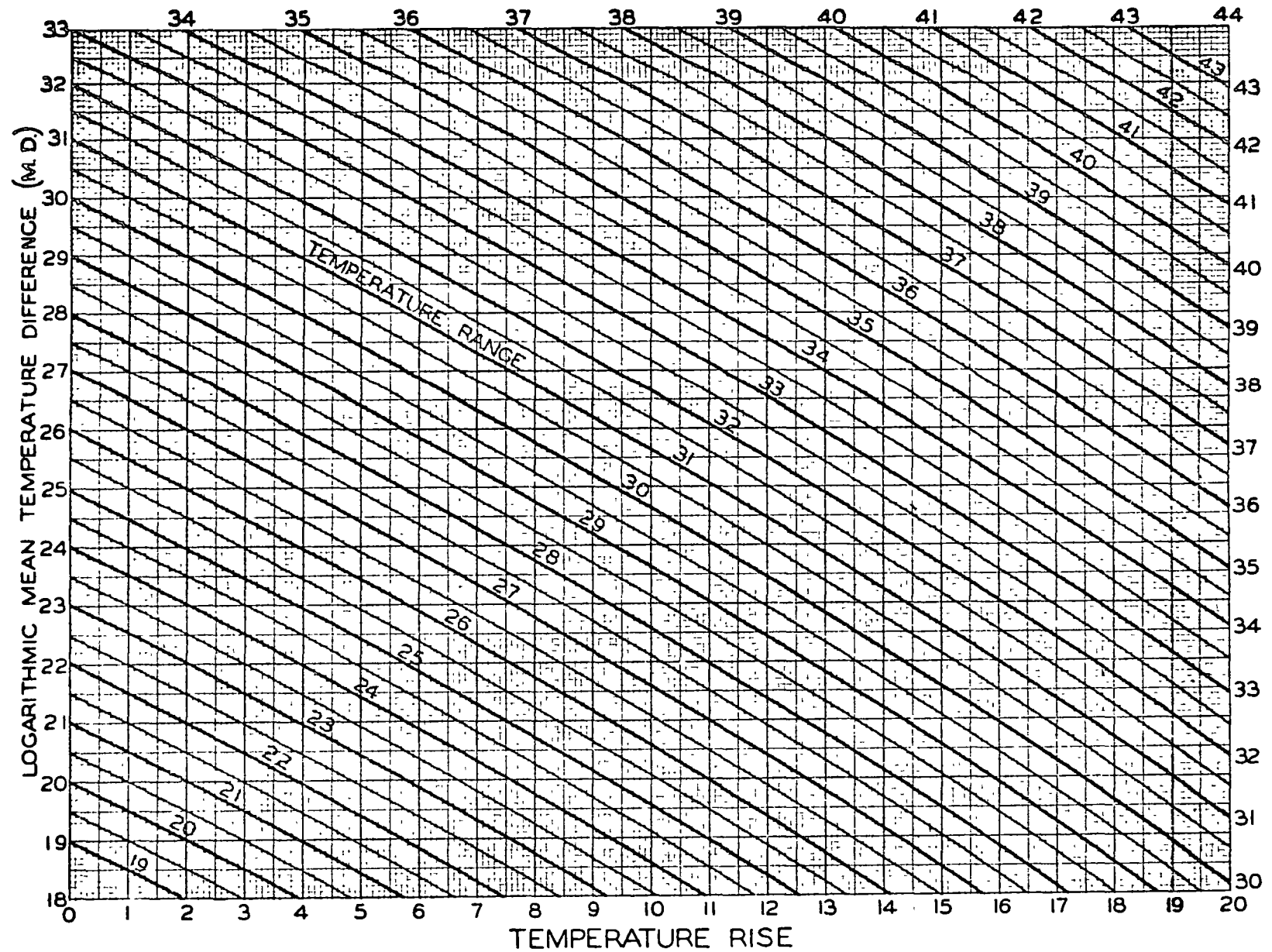
7



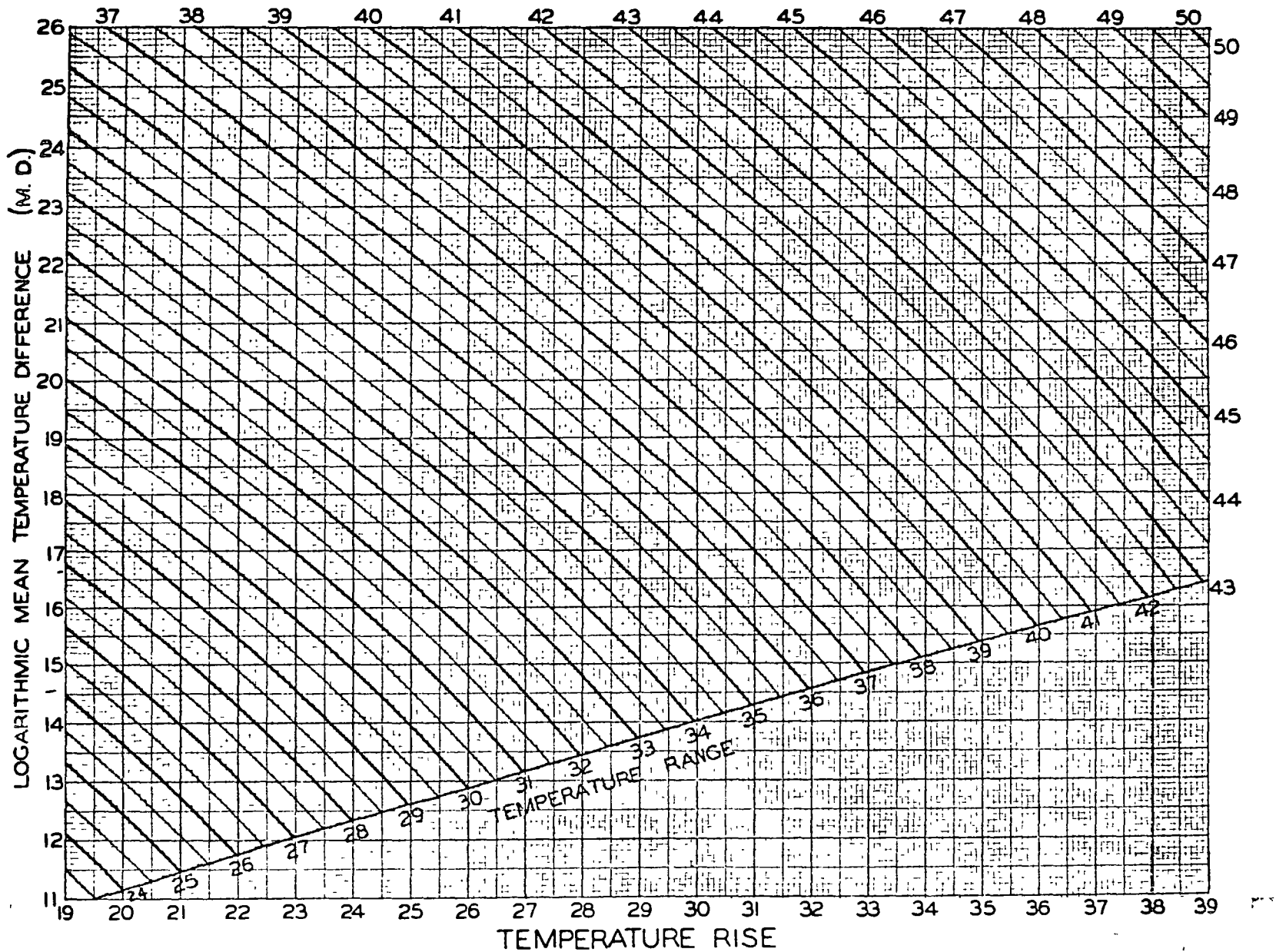
Relation Between Temperature Rise, Range, and Logarithmic Mean Temperature Difference



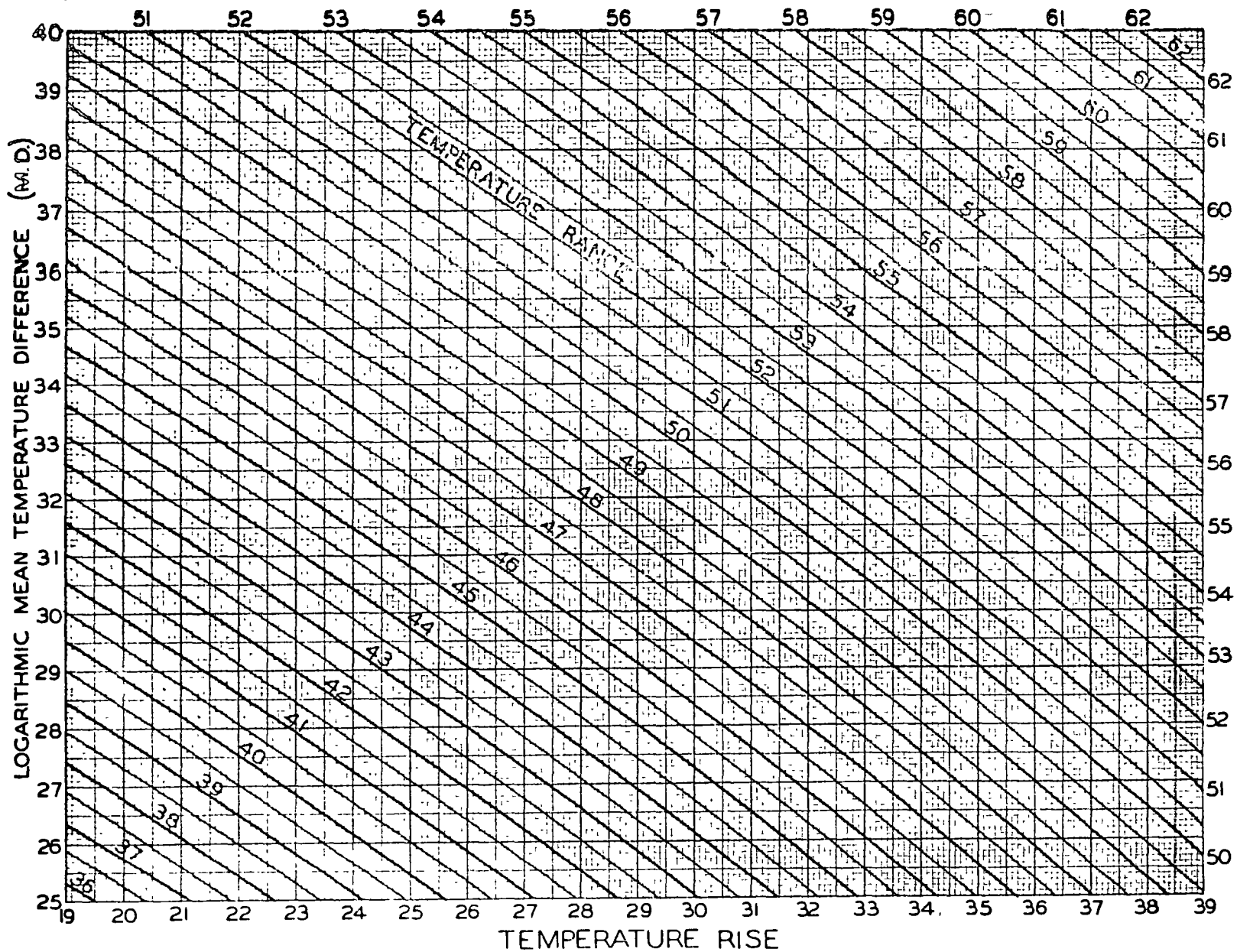
**Relation Between Temperature Rise, Range,
and Logarithmic Mean Temperature Difference**



Relation Between Temperature Rise, Range, and Logarithmic Mean Temperature Difference

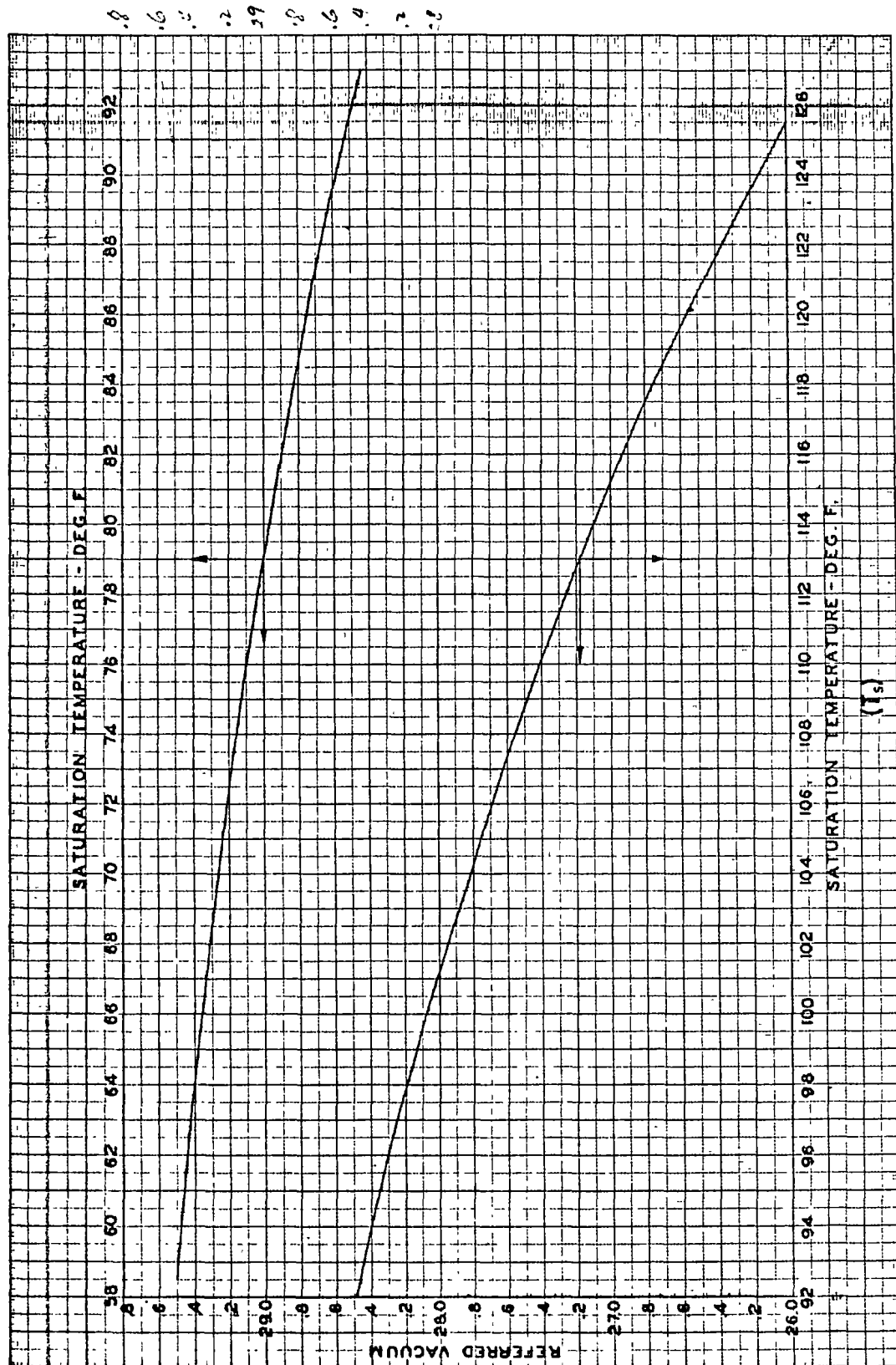


Relation Between Temperature Rise, Range, and Logarithmic Mean Temperature Difference



Relation Between Temperature and Vacuum

12



On page 6 the basic formulae are given for calculating the performance of a specific condenser. However, the power plant engineer is very often called upon to determine the most suitable size condenser to do a job, and the use of these formulae to calculate the required square footage could only be done by the "cut and try" method. To eliminate such a tedious process a simple tabular system has been devised for proportioning condensers. An example illustrating the use of this method, and the necessary tables, are contained on this and the next page. It should be kept in mind that the performance of any condenser proportioned by this method should be checked by the basic formulae as shown on page 6.

On page 4 and 5 the many variables which enter into the design of a condenser were discussed. The proper choice in each case will depend upon local conditions, space limitations, personal preferences, and other good engineering considerations. Where complete information is lacking we have listed suggestions for "good practice" to obtain preliminary condenser selections.

SAMPLE PROBLEM

To illustrate the use of this proportioning system, assume a condenser is required to serve a 3000 kw turbo generator unit, the steam load to the condenser being 33000 lb per hour. Fairly clean water in abundant quantity is available at 80 F.

From page 4 design for a vacuum of 28 in. Hg referred to 30 in. barometer which is consistent with 80 F cooling water.

Also from pages 4, 5 and 19 choose 3/4 in. O.D. 18 BWG admiralty tubes 14 ft long as representing good practice.

Assume a two pass condenser, 85% clean tubes, and 7 ft/sec water velocity.

Based on this information both known and assumed, set up the tables to determine the amount of surface required.

	Value	Table	Table Factor
Cooling water temperature	80 F	A	1.019
Number of condenser passes	2	B	301
Tube dia. & gauge	3/4 in. x 18 BWG	C	1 008
Tube length	14 ft	D	1 146
Cooling water velocity	7 ft/sec.	E	577
Tube cleanliness %	85%	F	.929
Tube material	Admiralty	G	0
	Total		4.980

X = .615 (from table H) $T_s = 101.1^\circ\text{F}$ from page 13 curve for 28" Hg vacuum.

Range = 101.1 - 80 = 21.1 F

Rise = Range $\times$ X = 21.1 $\times$.615 = 13.0 F

TD = 21.1 - 13.0 = 8.1 F (within 5 F limit)

Cooling water (G) = $\frac{950 \times TS}{\text{Rise} \times 500}$

$$\frac{950 \times 33000}{13.0 \times 500} = 4825 \text{ gpm}$$

$$\text{Surface (S)} = \frac{KGL}{V} =$$

$$\frac{.377 \times 4825 \times 14}{7} = 3640 \text{ sq ft}$$

Note: Referring to page 19 it is noted that 3640 sq ft is not a standard size condenser and it will be necessary to use either 3500 sq ft or 3750 sq ft. To arrive at one of these standard surfaces means a change in one or more of the design variables and it is usual practice to alter the velocity and, if necessary, the tube length.

In this example it appears that the velocity may be increased slightly to bring the surface requirement down to standard. On this basis the table is again set up.

	Value	Table	Table Factor
Cooling water temperature	80 F	A	1.019
Number of condenser passes	2	B	301
Tube dia. & gauge	3/4 in. x 18 BWG	C	1 008
Tube length	14 ft	D	1 146
Cooling water velocity	7.4 ft/sec.	E	565
Tube cleanliness %	85%	F	.929
Tube material	Admiralty	G	0
	Total		4.968

X = .605

Range = 101.1 - 80 = 21.1 F

Rise = Range $\times$ X = 21.1 $\times$.605 = 12.78 F

TD = 21.1 - 12.78 = 8.32 F (within limit)

Cooling water (G) = $\frac{950 \times TS}{\text{Rise} \times 500} = \frac{950 \times 33000}{12.78 \times 500} = 4900 \text{ gpm}$

$$\text{Surface (S)} = \frac{KGL}{V} =$$

$$\frac{.377 \times 4900 \times 14}{7.4} = 3500 \text{ sq ft}$$

Now check these condenser proportions using the regular performance formulae from page 6.

Velocity (V) = $\frac{4900 \times 14 \times .377}{3500} = 7.4 \text{ ft/sec}$

Heat transfer (U) = $732 \times 1.045 \times 1.0 \times .85 = 653 \text{ Btu}$

Log mean temp. diff. (MD) = $\frac{950 \times 33000}{653 \times 3500} = 13.73 \text{ F}$

Rise = $\frac{950 \times 33000}{500 \times 4900} = 12.8 \text{ F}$

Range = 21.1 F

Vacuum temperature = 80 + 21.1 = 101.1 F

Vacuum = 28 in. Hg. referred to 30 in. barometer

Terminal difference (TD) = $101.1 - (80 + 12.8) = 8.3 \text{ F}$

Maximum "cut off" vacuum = $30 - 1.385 = 28.615 \text{ in. Hg}$ referred to 30 in. barometer

This check shows that the results obtained from the proportioning tables are correct.

Condenser Solution Tables

14
TABLE C—Tube Diameter and Gauge Factor

Gauge	3/8"	1/2"	3/4"	1"
18 BWG.	1.113	1.008	.911	.840
17 BWG.	1.135	1.021	.923	.848
16 BWG.	1.150	1.032	.931	.854

TABLE A—Temperature Factor

Temp. Degree F	Factor	Temp. Degree F	Factor
50	.925	80	1.019
55	.946	85	1.025
60	.966	90	1.033
65	.984	95	1.037
70	1.000	100	1.041
75	1.011		

TABLE B—Pass Factor

Number of Passes	Factor
1.....	0.000
2.....	0.301
3.....	0.477

TABLE D—Tube Length Factor

Tube Length (ft)	Factor	Tube Length (ft)	Factor
6	0.778	24	1.380
8	0.903	26	1.415
10	1.000	28	1.447
12	1.079	29.75	1.473
14	1.146	32	1.515
16	1.204	34	1.531
18	1.255	36	1.556
20	1.301	38	1.580
22	1.342	39.75	1.599

TABLE E—Velocity Factor

Velocity of water ft/sec	.0	.1	.2	.3	.4	.5	.6	.7	.8	.9
3	.761	.754	.747	.741	.734	.728	.722	.716	.710	.704
4	.698	.694	.688	.683	.678	.673	.669	.664	.659	.655
5	.651	.646	.642	.638	.634	.630	.626	.622	.618	.615
6	.611	.607	.604	.600	.597	.594	.590	.587	.584	.581
7	.577	.574	.571	.568	.565	.562	.560	.557	.554	.551
8	.548	.546	.543	.540	.538	.535	.533	.530	.528	.525
9	.523	.521	.518	.516	.513	.511	.509	.507	.504	.502

TABLE F—Cleanliness Factor

100%	95%	90%	85%	80%	75%	70%	65%
1.000	.978	.954	.929	.903	.875	.845	.813

TABLE G—Tube Material Factor

	Gauge	18 BWG	17 BWG	16 BWG
Admiralty	0	0	—0.010	
Arsenical Copper	0	0	—0.010	
Aluminum (ASTM Spec. B-234, Alloys M-1A, M-1A Clad, GS-11A).....	0	0	—0.010	
Aluminum Brass.....	—0.018	—0.019	—0.029	
Muntz Metal	—0.018	—0.019	—0.029	
Aluminum Bronze	—0.047	—0.052	—0.058	
90 10 Copper Nickel.....	—0.047	—0.052	—0.058	
70 30 Copper Nickel	—0.081	—0.088	—0.102	
Type 304 Stainless Steel.....	—0.237	—0.240	—0.248	

TABLE H—X Factor (From Table Factor Total)

Table Factor	.00	.01	.02	.03	.04	.05	.06	.07	.08	.09
4.5	.271	.276	.282	.287	.293	.299	.304	.310	.316	.322
4.6	.328	.335	.341	.347	.354	.360	.367	.374	.380	.387
4.7	.394	.401	.408	.416	.423	.430	.438	.445	.453	.460
4.8	.468	.476	.484	.491	.499	.507	.515	.524	.532	.540
4.9	.548	.557	.565	.573	.581	.590	.598	.607	.615	.624
5.0	.632	.641	.649	.658	.666	.674	.683	.691	.699	.708
5.1	.716	.724	.732	.741	.748	.757	.764	.772	.780	.788
5.2	.795	.803	.810	.817	.824	.831	.838	.845	.851	.858
5.3	.865	.871	.878	.885	.892	.898	.905	.912	.918	.925

TABLE J—Tabular Interpolation

Diff.	.001	.002	.003	.004	.005	.006	.007	.008	.009
5	1	1	2	2	3	3	4	4	5
6	1	1	2	2	3	4	4	5	5
7	1	1	2	3	4	4	5	6	6
8	1	1	2	3	4	5	6	6	7
9	1	2	3	4	5	5	6	7	8
10	1	2	3	4	5	6	7	8	9
11	1	2	3	4	5	7	8	9	10

SURFACE CONDENSERS

General Data

CONDENSER FRICTION

The circulating water pressure loss measured from inlet to outlet connections is a summation of the hydraulic loss through the condenser tubes and the condenser water boxes. The magnitude of this loss is dependent upon the water velocity through the tubes, the size and length of the tubes, and the number of circulating water passes for which the condenser is designed. The amount of friction loss through any condenser may be read from curve on page 18.

NUMBER OF CONDENSER TUBES

The number of tubes in any condenser may be calculated by the following formula:

$$\text{Number of tubes} = \frac{\text{Condenser surface (S)}}{\text{Tube length (L)} \times \text{external surface per linear foot}}$$

TUBE CHARACTERISTICS

O.D. of Tubing (inches)	8WG	Thickness (inches)	Inside Diameter	Surface External, Sq ft per linear foot	Length in ft for 1 Sq ft surface	Water Gpm at 1 ft per sec velocity
3/4	16	.065	.495	.1636	6.112	.60
	17	.058	.509	.1636	6.112	.63
	18	.049	.527	.1636	6.112	.68
	20	.035	.555	.1636	6.112	.75
1	16	.065	.620	.1963	5.094	.94
	17	.058	.634	.1963	5.094	.98
	18	.049	.652	.1963	5.094	1.04
	20	.035	.680	.1963	5.094	1.13
1 1/2	16	.065	.745	.2291	4.367	1.36
	17	.058	.759	.2291	4.367	1.41
	18	.049	.777	.2291	4.367	1.48
	20	.035	.805	.2291	4.367	1.59
2	16	.065	.870	.2618	3.817	1.85
	17	.058	.884	.2618	3.817	1.91
	18	.049	.902	.2618	3.817	1.99
	20	.035	.930	.2618	3.817	2.12

PERFORMANCE GUARANTEES

1. The performance of a condenser is guaranteed for one specific set of conditions only, termed the "Design Point." For all other operating conditions as indicated on curves or tabulations, performance is expected but not guaranteed.
2. All condenser performance calculations are based upon 5 F or higher terminal difference. If calculations show less than a 5 F terminal difference then the terminal difference shall be arbitrarily increased to 5 F and the expected vacuum decreased accordingly.

3. The standard guaranteed oxygen content of the condensate is 0.03 CC per litre maximum.

4. Under conditions as listed below the oxygen content of the condensate may be guaranteed not to exceed .01 cc per litre.

(a) Condenser pressures not lower than the values shown on the curve on page 19.

(b) Air and non-condensables removed from the system not exceeding 25 per cent of the capacity of the vacuum pump equipment in operation. This capacity is that of the vacuum pump equipment at the operating suction pressure and at a temperature 7.5 F below the saturation temperature corresponding to this suction pressure.

However, for circulating water inlet temperatures of less than 50 F, the vacuum pump equipment in operation shall have a minimum capacity of 3 cfm free dry air at 0.5 inches Hg absolute suction pressure and a temperature 7.5 F below the saturation temperature corresponding to this suction pressure.

(c) Zero air leakage directly into the condensate below the condensate level in the hotwell.

(d) Arrangement and location of all entrance points into the condenser for water, vapor or other gases, subject to the approval of the Company.

(e) Total water introduced into the condenser shell, at a temperature lower than the inlet steam temperature, not more than 5 per cent of the steam being condensed.

(f) Method of sampling and measuring the oxygen in accordance with the Standards of Heat Exchange Institute—"Method and Procedure for the Determination of Dissolved Oxygen."

5. The temperature of the condensate in the condenser hotwell will be equal to the temperature corresponding to the vacuum within the condenser at half steam load and above.

6. Condenser performance is based on air and non-condensables removed from the system not exceeding 50 per cent of the capacity of the air removal equipment in operation.

MATERIALS OF CONSTRUCTION

Water Boxes and Water Box Covers

Cast Iron—ASTM—Spec A-48 Class 30

Steel—ASTM Spec A 285 Grade C Flange Quality.

Shell Plate

Steel—ASTM Spec A 285—Grade C Flange Quality.

Tube Sheets

Rolled Muntz Metal—ASTM Spec B-171.

Tube Support Plates

Steel—ASTM Spec A285 Grade C Flange Quality or free Machining Steel Plate

Tubes

ASTM Spec B111

SHELL PLATE THICKNESS

The thickness of condenser shell plate shall be in accordance with the following:

Cross Sectional Area of Shell—sq in	Steel Plate Shell Thickness—Inches
Up to 2,460.....	$\frac{3}{8}$
2,461 to 5,670.....	$\frac{1}{2}$
5,671 to 11,500.....	$\frac{5}{8}$
11,501 to 29,000.....	$\frac{3}{4}$
29,001 and over.....	$\frac{7}{8}$

TUBE SHEET THICKNESS

The thickness of tube sheets shall be in accordance with the following:

Total Area of Tube Sheet—sq in	Thickness of Tube Sheet—Inches
Up to 1,964.....	$\frac{3}{8}$
1,965 to 3,739.....	$\frac{1}{2}$
3,740 to 8,495.....	$\frac{5}{8}$
8,496 to 30,791.....	$\frac{3}{4}$
30,792 and over.....	$\frac{7}{8}$

TUBE LENGTH

The effective tube length is the distance between inside faces of tube sheets. The overall length of tubes will be the effective length plus the thickness of the two (2) tube sheets.

HYDROSTATIC TESTS

1. Condenser shells are tested by completely filling with water.
2. Condenser water boxes are tested at a hydrostatic pressure five pounds more than the design pressure but in no case less than 25 pounds per square inch, measured at the bottom of the water box.
3. Shells and water boxes of the smaller condensers which are completely tubed and assembled at the factory are hydrostatically tested before shipment. Larger condensers assembled and tubed in the field are hydrostatically tested after completion of erection.

STEAM ENGINE SERVICE

Condensers for use with steam engines should be sized with a 65% heat transfer factor to cover the oil film on the outside surface of the tubes. The usual Tube Cleanliness Factor depending on local cooling water conditions should also be applied.

In steam engine service the heat rejected to the cooling water is usually assumed as 1000 BTU per pound of steam.

ATMOSPHERIC RELIEF VALVES

The size of atmospheric relief valves used for normal operation of condensing turbines or engines are recommended below.

Maximum Steam Flow Pounds Per Hour	Sizes of Atmospheric Relief Valves—Inches	
	For Protection*	For Maximum Non-Condensing Operation*
Up to 7,500	6	8
7,501 to 11,800	8	10
11,801 to 17,000	8	12
17,001 to 20,000	8	14
20,001 to 23,100	10	14
23,101 to 30,200	10	16
30,201 to 38,200	12	18
38,201 to 45,000	12	20
45,001 to 47,200	14	20
47,201 to 62,000	14	24
62,001 to 68,000	16	24
68,001 to 82,000	16	30
82,001 to 106,000	18	30
106,001 to 120,000	18	
120,001 to 170,000	20	
170,001 to 250,000	24	
250,001 to 380,000	30	
380,000 to 550,000	36	

*The sizes listed "For Protection" are normally used under ordinary condensing operation. However, if it is desired to operate the turbine temporarily non-condensing at its maximum non-condensing capacity, the sizes listed under "For Maximum Non-Condensing Operation" should be used.

VACUUM PUMP CAPACITY

To insure adequate removal capacity it is recommended that the capacity of the vacuum pump be not less than indicated below.

Maximum Steam Condensed Pounds Per Hour	CFM—70 F Free Dry Air	
	Serving Turbines	Serving Engines
Up to 25,000	3.0	6.0
25,001 to 50,000	4.0	8.0
50,001 to 100,000	5.0	10.0
100,001 to 250,000	7.5	15.0
250,001 to 500,000	10.0	
500,001 and up	12.5	

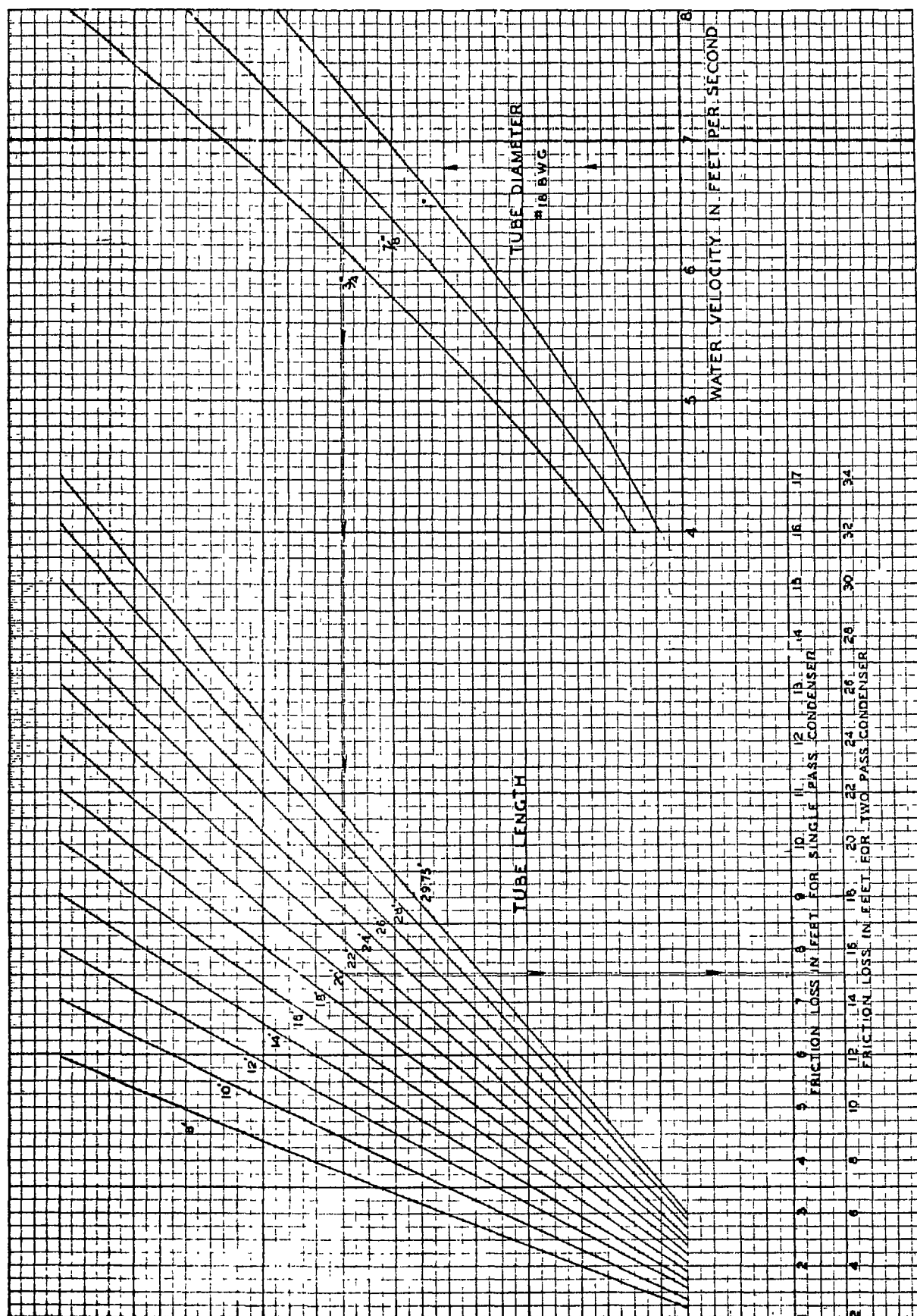
RAPID EVACTORS FOR STEAM SPACE

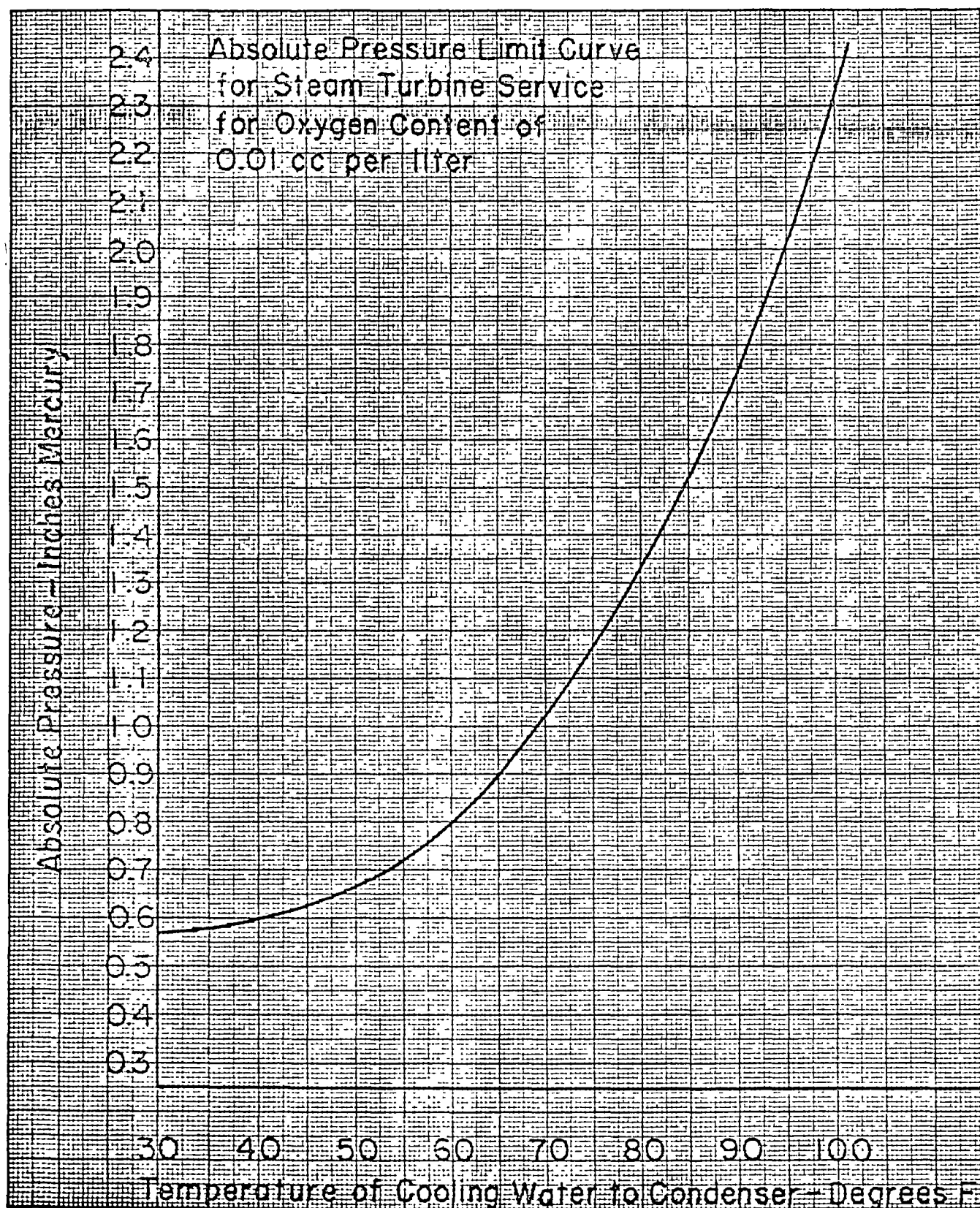
The capacity of the rapid evacuator is dependent on several variables but in most cases the following capacities will be satisfactory.

Maximum Steam Condensed Pounds Per Hour	CFM—70 F Free Dry Air At 15 in. Hg. Abs. Pressure
Up to 75,000	150
75,001 to 250,000	300
250,001 to 600,000	600
600,001 and up	900

Condenser Circulating Water Friction Through Water Boxes and Tubes

17





CHEMICAL COMPOSITION OF TUBES

19

Alloy	Copper, per cent	Tin per cent	Aluminum per cent	Nickel, ② per cent	Lead, max, per cent	Iron, max, per cent	Zinc per cent	Manganese, max, per cent	Arsenic per cent	Antimony per cent	Phosphorus per cent
Antz metal.....	59.0 to 63.0				0.3	0.07	remainder	...		..	..
Admiralty metal—											
Type A.....	70.0 to 73.0	0.90 to 1.20			0.075	0.06	remainder	...		..	..
Type B.....	70.0 to 73.0	0.90 to 1.20			0.075	0.06	remainder	...	0.02 to 0.10	..	..
Type C.....	70.0 to 73.0	0.90 to 1.20			0.075	0.06	remainder	...		0.02 to 0.10	..
Type D.....	70.0 to 73.0	0.90 to 1.20			0.075	0.06	remainder	...		..	0.02 to 0.10
Red brass.....	84.0 min.				0.075	0.06	remainder	...		..	..
Aluminum brass ③											
Type B ④.....	76.0 min.		1.75 to 2.50		0.075	0.06	remainder	...	0.02 to 0.10	..	..
Type C.....	76.0 min.		1.75 to 2.50		0.075	0.06	remainder	...		0.02 to 0.10	..
Type D.....	76.0 min.		1.75 to 2.50		0.075	0.06	remainder	...		..	0.02 to 0.10
Aluminum bronze.....	remainder		5.0 to 6.5		0.10	0.10		...	0.35 max.	..	..
70-30 copper-nickel ⑤	remainder	1.50 max.		29.0 to 33.0	0.05	0.6	1.0 max.	1.0		..	..
80-20 copper-nickel—											
Type A.....	remainder	1.0 max.		19.0 to 23.0	0.05	0.6	1.0 max.	1.0		..	..
Type B.....	remainder	1.0 max.		19.0 to 23.0	0.05	0.6	3.0 to 6.0	1.0		..	..
90-10 copper-nickel.....	remainder			9.0 to 11.0	0.05	1.0	1.0 max.	1.0		..	..
Copper.....	99.90 min.							...		..	0.04 max.
Arsenical copper....	99.40 min.							...	0.15 to 0.50	..	..

① Silver counting as copper.

② Cobalt counting as nickel.

③ Total other impurities, 0.30 per cent, max.

④ Type A aluminum brass has been discontinued.

⑤ When specified by the purchaser the iron content shall be 0.25 to 0.60 per cent.

(Note: Reproduced from ASTM Specification No. B 111-51.)

RECOMMENDED CONDENSER SURFACES AND TUBE PROPORTIONS

HEI STANDARD

Surface Square Feet	EFFECTIVE TUBE LENGTHS—FEET				Surface Square Feet	EFFECTIVE TUBE LENGTHS—FEET			
	¾ In. OD	¾ In. OD	¾ In. OD	1 In. OD		¾ In. OD	¾ In. OD	¾ In. OD	1 In. OD
100	6- 8-10	6- 8-10			7750		16-18-20-22	16-18-20-22	16-18-20-22
200	6- 8-10	6- 8-10			8000		16-18-20-22	16-18-20-22	16-18-20-22
300	6- 8-10	6- 8-10			8500		16-18-20-22	16-18-20-22	16-18-20-22
400	6- 8-10	6- 8-10			9000		16-18-20-22	16-18-20-22	16-18-20-22
500	6- 8-10-12	6- 8-10-12			9500		16-18-20-22	16-18-20-22	16-18-20-22
600	6- 8-10-12	6- 8-10-12			10000		16-18-20-22	16-18-20-22	16-18-20-22
700	6- 8-10-12	6- 8-10-12			11000		16-18-20-22	16-18-20-22	16-18-20-22
800	6- 8-10-12	6- 8-10-12			12000		16-18-20-22	16-18-20-22	16-18-20-22
900	6- 8-10-12	6- 8-10-12			13000		16-18-20-22	16-18-20-22	16-18-20-22
1000	8-10-12-14	8-10-12-14			14000		16-18-20-22	16-18-20-22	16-18-20-22
1100		8-10-12-14			15000		18-20-22-24	18-20-22-24	18-20-22-24
1200		8-10-12-14			16000		18-20-22-24	18-20-22-24	18-20-22-24
1300		8-10-12-14			17000		18-20-22-24	18-20-22-24	18-20-22-24
1400		8-10-12-14			18000		18-20-22-24	18-20-22-24	18-20-22-24
1500		8-10-12-14			19000		18-20-22-24	18-20-22-24	18-20-22-24
1750		8-10-12-14			20000		20-22-24-26	20-22-24-26	20-22-24-26
2000		10-12-14-16			22500		20-22-24-26	20-22-24-26	20-22-24-26
2250		10-12-14-16			25000		20-22-24-26	20-22-24-26	20-22-24-26
2500		10-12-14-16			27500		20-22-24-26	20-22-24-26	20-22-24-26
2750		12-14-16-18			30000		22-24-26-28	22-24-26-28	22-24-26-28
3000		12-14-16-18	12-14-16-18		32500		22-24-26-28	22-24-26-28	22-24-26-28
3250		12-14-16-18	12-14-16-18		35000		22-24-26-28	22-24-26-28	22-24-26-28
3500		12-14-16-18	12-14-16-18		37500		22-24-26-28	22-24-26-28	22-24-26-28
3750		12-14-16-18	12-14-16-18		40000		22-24-26-28	22-24-26-28	22-24-26-28
4000		12-14-16-18	12-14-16-18		42500		22-24-26-28	22-24-26-28	22-24-26-28
4250		12-14-16-18	12-14-16-18		45000		22-24-26-28	22-24-26-28	22-24-26-28
4500		12-14-16-18	12-14-16-18		47500		22-24-26-28	22-24-26-28	22-24-26-28
4750		12-14-16-18	12-14-16-18		50000		24-26-28-30*	24-26-28-30*	24-26-28-30*
5000		14-16-18-20	14-16-18-20	14-16-18-20	55000		24-26-28-30*	24-26-28-30*	24-26-28-30*
5250		14-16-18-20	14-16-18-20	14-16-18-20	60000		24-26-28-30*	24-26-28-30*	24-26-28-30*
5500		14-16-18-20	14-16-18-20	14-16-18-20	65000		24-26-28-30*	24-26-28-30*	24-26-28-30*
5750		14-16-18-20	14-16-18-20	14-16-18-20	70000		24-26-28-30*	24-26-28-30*	24-26-28-30*
6000		14-16-18-20	14-16-18-20	14-16-18-20	75000		24-26-28-30*	24-26-28-30*	24-26-28-30*
6250		14-16-18-20	14-16-18-20	14-16-18-20	80000		24-26-28-30*	24-26-28-30*	24-26-28-30*
6500		14-16-18-20	14-16-18-20	14-16-18-20	85000		24-26-28-30*	24-26-28-30*	24-26-28-30*
6750		14-16-18-20	14-16-18-20	14-16-18-20	90000		24-26-28-30*	24-26-28-30*	24-26-28-30*
7000		14-16-18-20	14-16-18-20	14-16-18-20	95000		24-26-28-30*	24-26-28-30*	24-26-28-30*
7250		16-18-20-22	16-18-20-22	16-18-20-22	100000		24-26-28-30*	24-26-28-30*	24-26-28-30*
7500		16-18-20-22	16-18-20-22	16-18-20-22					

*Note the 30 ft. refers to overall tube lengths.

HEI STANDARD

Laminar Film Condensation of a Flowing Vapor on a Horizontal Cylinder at Normal Gravity¹

V. E. DENNY

A. F. MILLS

Assistant Professor of Engineering,
 University of California,
 Los Angeles, Calif.

An analytical solution, based on the Nusselt assumptions, has been obtained for laminar film condensation of a flowing vapor on a horizontal cylinder. In so doing, a reference temperature for evaluating locally variable fluid properties is defined in the form $T_r = T_w + \alpha(T_s - T_w)$ and accounts for both the effects of fluid property variations and minor errors introduced by the assumptions in the analysis. Verification was obtained by comparison with exact numerical solutions based on a finite-difference analog to the conservation equations in boundary-layer form. In the analytical as well as the numerical developments, vapor drag was accounted for through an asymptotic solution of the vapor boundary layer under strong suction. It was found that, for angles up to 140 deg, there was less than a 2 percent discrepancy between the analytical predictions and the numerical results. As 180 deg is approached an increased discrepancy is expected due to a gross violation of the Nusselt assumptions. The values of the reference parameter α , which were previously derived for condensation on a vertical surface, were found to be appropriate for the horizontal cylinder as well.

Introduction

THE first analysis of film condensation on a horizontal cylinder was performed by Nusselt in 1916 [1].² Bromley [2] and Rohsenow [3] extended this analysis to account for the heat

¹This work was supported by the State of California through the University of California Statewide Water Resources Center on Grant No. 4-11-775-20537. Computer time for the numerical calculations was provided by the Campus Computing Network of the University of California, Los Angeles.

²Numbers in brackets designate References at end of paper.

Contributed by the Heat Transfer Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS for presentation at the ASME-AIChE Heat Transfer Conference, Minneapolis, Minn., August 3-6, 1969. Manuscript received by the Heat Transfer Division, January 1, 1969. Paper No. 69-HT-49.

capacity of the liquid condensate. Sparrow and Gregg [4] considered the boundary-layer form of the conservation equations governing the liquid film and used an approximate similarity transformation to obtain a solution. They thereby avoided the Nusselt assumptions of negligible effects due to liquid acceleration and energy convection. However the assumptions of negligible vapor drag and constant fluid properties remained. Shekrladze and Gomelaury [5] invoked the Nusselt assumptions to obtain an explicit analytical solution for the case of forced flow of a saturated vapor at zero gravity. Vapor drag was accounted for by the simple shear expression which results from the asymptotic solution of the vapor boundary layer for strong suction. For condensation in a gravitational field they stated that an explicit analytical solution was impossible. This last mentioned

Nomenclature

C_p = heat capacity [Btu/lb deg R]
 Fr = dimensionless Froude number
 $\frac{gR}{U_{\infty}^2}$
 g = gravitational acceleration
 32.174 ft/sec²
 h = heat transfer coefficient [Btu/sq ft sec deg R]
 $H = \lambda + \int_T C_p dT$ [Btu/lb]
 k = thermal conductivity [Btu/ft sec deg R]
 l_1, l_2 = metric coefficients [ft]
 $\dot{m}$ = rate of condensation [lb/sq ft sec]
 Pr = dimensionless Prandtl number
 $\frac{C_p \mu}{k}$
 P = pressure [lb/sq ft sec²]
 Re = dimensionless Reynolds number
 $\frac{U_{\infty} \rho R}{\mu}$
 R = tube radius [ft]
 T = absolute temperature [deg R]

U, V = vapor velocity components in x_1, x_2 plane [fps]
 u_1, u_2 = liquid velocity components in x_1, x_2 plane [fps]
 u, v = dimensionless liquid velocity components
 x_1, x_2 = boundary-layer coordinates [ft]
 x, y = dimensionless boundary-layer coordinates defined by
 $x = \frac{x_1}{R}$ and $y = \frac{x_2}{R}$
 α = dimensionless parameter which defines the reference temperature $T_r = T_w + \alpha(T_s - T_w)$
 γ = dimensionless parameter which establishes the spacing of the streamlines in the ξ_1, ξ_2 plane
 Γ = mass flow rate of condensate film [lb/ft sec]
 δ = dimensionless film thickness defined by $\delta = \frac{x_2, \delta}{R}$
 θ = dimensionless temperature $(T_r - T)/(T_s - T_w)$

ϕ = angle measured from the vertical axis
 λ = latent heat of vaporization [Btu/lb]
 μ = absolute viscosity [lb/ft sec]
 ν = kinematic viscosity [sq ft/sec]
 ξ_1, ξ_2 = dimensionless coordinates related to dx_1 and dx_2 by $dx_1 \equiv l_1 d\xi_1$ and $dx_2 \equiv l_2 d\xi_2$
 ρ = density [lb/ft³]
 ψ = stream function [lb/ft sec]

Subscripts

e = edge of vapor boundary layer
 0 = condition at $\phi = 0$
 r = reference state
 s = vapor-liquid interface
 v = vapor phase
 w = tube wall
 ∞ = oncoming vapor

Superscript

t = theoretically derived quantity

Overlines

$\sim, \approx$ = averages defined by equation (22)

situation is widely encountered in engineering practice and further investigation is warranted.

The objective of the present contribution is to demonstrate that a satisfactory explicit solution may be obtained for the problem left unsolved by Shchekladze and Gomelaui. The solution is obtained by averaging the two variable coefficients in the governing ordinary differential equation; a table of these average values as a function of angle is presented from which local values of the condensation rate may be calculated. The accuracy of this solution is evaluated by comparison with exact numerical solutions based on a finite-difference analog to the conservation equations in boundary-layer form. The complete partial differential equations are considered in the exact solutions and thus the nonsimilar character of the situation poses no problem. In addition, the finite-difference procedure allows the determination of appropriate reference temperatures to account for the property variation of real liquids. In the numerical solutions vapor drag is also accounted for by use of the asymptotic shear expression, thereby limiting the range of oncoming vapor velocities and condensation rates for which the conclusions are valid.

Theory

A schematic representation of the physical model and coordinate system is shown in Fig. 1. The dimensional coordinates are x_1 and x_2 , parallel and perpendicular to the tube surface, respectively. The corresponding dimensional liquid velocity components are u_1 and u_2 . The oncoming vapor flow is saturated at temperature T_s and has velocity $U_{s,\infty}$. The condenser tube wall is at temperature T_w ($T_w < T_s$) and is constant with angle ϕ . Following Nusselt [1] effects of fluid acceleration and energy convection are assumed negligible. The effect of temperature on the fluid properties is treated by means of the reference temperature concept. At the liquid surface there is the requirement of continuity of momentum flux between liquid and vapor phases. For a boundary-layer flow under strong suction, a simple asymptotic solution of the momentum conservation equation yields the required boundary condition [5, 6, 7]. In this limit the local shear is identically equal to the momentum given up by the condensing vapor ($mU_{s,\infty}$). The suction velocity is a function of angle ϕ resulting in an interaction with the inviscid vapor flow outside the boundary layer. If this interaction is assumed negligible then the potential flow solution is simply $U_{s,\infty} = 2U_{w,\infty}$

sin ϕ . Following [5], the effect of the pressure gradient around the cylinder periphery is assumed negligible and we have

$$\frac{d^2 u_1}{dx_2^2} = -\frac{g \sin \phi}{\nu} \left(1 - \frac{\rho_v}{\rho}\right) \quad (1)$$

$$\frac{d^2 T'}{dx_2^2} = 0 \quad (2)$$

$$T = T_w \text{ and } u_1 = 0 \text{ at } x_2 = 0 \quad (3)$$

$$T = T_s \text{ and } \mu \frac{du_1}{dx_2} = h(U_{s,\infty} - u_1) \text{ at } x_2 = x_{1,\delta} \quad (4)$$

Or, in dimensionless form

$$\frac{d^2 u}{dy^2} = -\text{Re Fr} \left(1 - \frac{\rho_v}{\rho}\right) \sin \phi = -C_0 \sin \phi \quad (5)$$

$$\frac{d^2 \theta}{dy^2} = 0 \quad (6)$$

$$\theta = 1 \text{ and } u = 0 \text{ at } y = 0 \quad (7)$$

$$\theta = 0 \text{ and } \frac{du}{dy} = h \text{ Re } (2 \sin \phi - u) \text{ at } y = \delta \quad (8)$$

where

$$u = \frac{u_1}{U_{w,\infty}} \quad \text{Re} = \frac{U_{s,\infty} R}{\nu}$$

$$\theta = \frac{T_s - T}{T_s - T_w} \quad \text{Fr} = \frac{g R}{U_{s,\infty}^2}$$

Solution of equations (5) and (6) subject to the boundary conditions (7) and (8) yields the velocity and temperature distributions

$$u = C_1 y - C_2 y^2 \quad (9)$$

$$\theta = \left(1 - \frac{y}{\delta}\right) \quad (10)$$

where

$$C_1 = C_0 \delta \sin \phi + \frac{\text{Re } h}{1 + \text{Re } m \delta} \left(2 \sin \phi - C_0 \sin \phi \frac{\delta^2}{2}\right) \quad (11)$$

$$C_2 = \frac{1}{2} C_0 \sin \phi \quad (12)$$

The local film thickness δ , as well as the local condensation rate $\dot{m}$ are obtained from the overall enthalpy and mass balances

$$\int_0^{x_1} k \frac{dT'}{dx_2} dx_1 = \int_0^{x_{1,\delta}} \rho u_1 [\lambda + C_p (T_s - T)] dx_2 \quad (13)$$

$$\dot{m} = \frac{d\Gamma}{dx_1} = \frac{d}{dx_1} \int_0^{x_{1,\delta}} \rho u_1 dx_2 \quad (14)$$

or in dimensionless form:

$$\int_0^x \frac{dx}{\delta} = \text{Re Pr} \int_0^\delta u \left(\frac{\lambda}{C_p \Delta T} + \theta\right) dy \quad (15)$$

$$\frac{\dot{m}}{\rho U_{s,\infty}} = \frac{d}{dx} \int_0^\delta u dy \quad (16)$$

Substitution of equations (9) and (10) into (15) and (16) followed by considerable algebra yields the enthalpy balance in the form

$$\int_0^x \frac{dx}{\delta} = a\delta + b\delta^2 \quad (17)$$

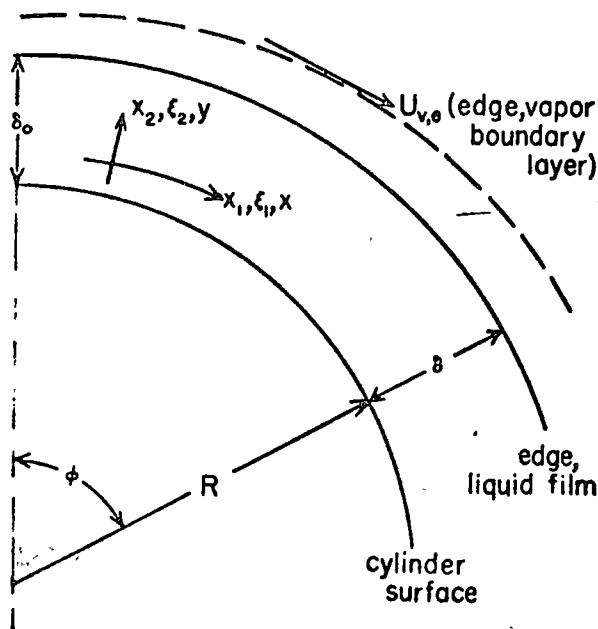


Fig. 1 Schematic diagram of the coordinate systems

$$a = \text{Re} \frac{\text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right)}{1 + \text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right)} \sin \phi = C_a \sin \phi \quad (18)$$

$$b = \frac{\text{Re}^2 \text{Fr}}{3} \left(1 - \frac{\rho_r}{\rho} \right) \text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right) \times \left(1 - \frac{3/4}{1 + \text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right)} \right) \sin \phi \quad (19)$$

$$= C_b \sin \phi$$

(The various coefficients of $C_p \Delta T$ which evolve naturally in the algebraic manipulations, $1/3$ and $2/3$, have been replaced by their average value 0.351 to effect a convenient simplification in the coefficients a and b .)

The integral equation (17) may be rewritten in the form:

$$dx = d(\sin \phi \delta^2) + d(b \delta^4) - \frac{a}{2} d(\delta^2) - \frac{b}{4} d(\delta^4)$$

or

$$dx = C_a \left[d(\sin \phi \delta^2) - \frac{\sin \phi}{2} d(\delta^2) \right] + C_b \left[d(\sin \phi \delta^4) - \frac{\sin \phi}{4} d(\delta^4) \right] \quad (20)$$

Equation (20) cannot be integrated exactly, but, applying the first mean value theorem of integral calculus we may extract

$$x = C_a \left[\sin \phi \delta^2 - \frac{\widetilde{\sin \phi}}{2} (\delta^2 - \delta_0^2) \right] + C_b \left[\sin \phi \delta^4 - \frac{\widetilde{\sin \phi}}{4} (\delta^4 - \delta_0^4) \right] \quad (21)$$

where

$$\widetilde{\sin \phi} = \frac{\int_{\delta_0^2}^{\delta^2} \sin \phi d(\delta^2)}{\delta^2 - \delta_0^2}; \quad \widetilde{\sin \phi} = \frac{\int_{\delta_0^4}^{\delta^4} \sin \phi d(\delta^4)}{\delta^4 - \delta_0^4} \quad (22)$$

Equation (21) may be solved for δ to yield:

$$\delta = \sqrt[3]{\frac{2x - C_a \delta_0^2 \widetilde{\sin \phi} - \frac{C_b \delta_0^4}{2} \widetilde{\sin \phi}}{C_a \left(\sin \phi - \frac{\widetilde{\sin \phi}}{2} \right) + \sqrt{C_a^2 \left(\sin \phi - \frac{\widetilde{\sin \phi}}{2} \right)^2 + 2C_b \left(\sin \phi - \frac{\widetilde{\sin \phi}}{4} \right) \left(2x - C_a \delta_0^2 \widetilde{\sin \phi} - \frac{C_b \delta_0^4}{2} \widetilde{\sin \phi} \right)}} \quad (23)$$

The local heat transfer coefficient h' is equal to $\frac{k}{R\delta}$ and is thus given by

$$h' = \frac{k}{R} \sqrt[3]{\frac{C_a \left(\sin \phi - \frac{\widetilde{\sin \phi}}{2} \right) + \sqrt{C_a^2 \left(\sin \phi - \frac{\widetilde{\sin \phi}}{2} \right)^2 + 2C_b \left(\sin \phi - \frac{\widetilde{\sin \phi}}{4} \right) \left(2x - C_a \delta_0^2 \widetilde{\sin \phi} - \frac{C_b \delta_0^4}{2} \widetilde{\sin \phi} \right)}}{2x - C_a \delta_0^2 \widetilde{\sin \phi} - \frac{C_b \delta_0^4}{2} \widetilde{\sin \phi}} \quad (24)$$

Initial Value of δ at $\phi = 0$. In order to evaluate h from equation (24) the film thickness at $\phi = 0$, δ_0 , must be known. Equation (20) may be rearranged in the form,

$$1 = \frac{C_a \sin \phi}{2} \frac{d\delta^2}{dx} + \frac{3C_b}{4} \frac{\sin \phi}{dx} \frac{d\delta^4}{dx} + (C_a \delta^2 + C_b \delta^4) \cos \phi \frac{d\phi}{dx} \quad (25)$$

$$\phi = \frac{x_1}{R} = x$$

$$\frac{d\phi}{dx} = 1$$

Therefore, as $x \rightarrow 0$, equation (25) reduces to:

$$1 = (C_a \delta_0^2 + C_b \delta_0^4)$$

or

$$C_b \delta_0^4 + C_a \delta_0^2 - 1 = 0 \quad (26)$$

Solving for δ_0 yields

$$\delta_0 = \sqrt[3]{\frac{2}{C_a + \sqrt{C_a^2 + 4C_b}}} \quad (27)$$

Total Condensation Rate. The condensation which occurs between $\phi = 0$ and $\phi = \phi$ is equal to 2Γ and may be obtained from equation (9) and the definition of Γ . After some algebra there results

$$2\Gamma = 2\rho R U_{\infty} \delta^2 \sin \phi \left[\text{Re Fr} \left(1 - \frac{\rho_r}{\rho} \right) \times \left(\frac{1}{3} - \frac{1/4}{1 + \text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right)} \right) + \frac{1/\delta^2}{1 + \text{Pr} \left(\frac{\lambda}{C_p \Delta T} + 0.351 \right)} \right] \quad (28)$$

The Averages $\widetilde{\sin \phi}$ and $\widetilde{\sin \phi}$. The utility of equation (24) is dependent on whether or not the averages $\widetilde{\sin \phi}$ and $\widetilde{\sin \phi}$ defined by equation (22) may be represented in a convenient manner.

Table 1 Values of the averages of $\sin \phi$ and $\sin \phi$ defined by equation (22)

ϕ	$\sin \phi$	$\sin \phi$	ϕ	$\sin \phi$	$\sin \phi$
1	.010908	.010908	91	.840406	.854332
2	.022905	.022905	92	.841903	.858960
3	.034656	.034657	93	.843452	.863306
4	.046341	.046344	94	.845241	.867604
5	.057995	.058000	95	.847169	.871664
6	.069628	.069636	96	.849235	.875517
7	.081244	.081256	97	.851436	.879250
8	.092842	.092860	98	.853770	.882769
9	.104424	.104450	99	.856235	.886102
10	.115988	.116023	100	.858831	.889246
11	.127533	.127580	101	.861558	.892199
12	.139057	.139118	102	.864416	.894957
13	.150560	.150637	103	.867406	.897518
14	.162039	.162135	104	.870528	.899887
15	.173492	.173610	105	.873781	.902035
16	.184919	.185062	106	.877165	.903985
17	.196317	.196488	107	.880680	.905724
18	.207685	.207888	108	.884326	.907251
19	.219027	.219259	109	.888104	.908562
20	.230334	.230601	110	.892016	.909653
21	.241591	.241911	111	.896063	.910521
22	.252821	.253188	112	.899346	.911162
23	.264013	.264430	113	.902865	.911573
24	.275163	.275637	114	.906621	.911751
25	.286272	.286805	115	.910614	.911691
26	.297337	.297935	116	.914846	.911390
27	.308355	.309023	117	.919317	.910845
28	.319327	.320069	118	.924028	.910050
29	.330249	.331071	119	.928981	.909003
30	.341120	.342021	120	.934177	.907699
31	.351939	.352936	121	.939618	.906134
32	.362703	.363795	122	.945306	.904304
33	.373412	.374604	123	.951249	.902204
34	.384062	.385361	124	.957458	.900830
35	.394653	.396064	125	.963935	.899177
36	.405183	.406712	126	.970680	.897240
37	.415650	.417302	127	.977695	.895015
38	.426052	.427833	128	.984981	.892496
39	.436388	.438304	129	.992548	.889679
40	.446655	.448712	130	.999396	.886558
41	.456853	.459057	131	.999999	.883127
42	.466980	.469336	132	.999999	.879382
43	.477033	.479547	133	.999999	.875315
44	.487011	.489690	134	.999999	.870929
45	.496913	.499762	135	.999999	.866214
46	.506735	.509761	136	.999999	.861178
47	.516479	.519686	137	.999999	.855821
48	.526141	.529555	138	.999999	.850142
49	.535719	.539306	139	.999999	.844149
50	.545212	.548998	140	.999999	.837832

Table (cont.)

51	.554618	.558609	141	.826326	.812453
52	.563936	.568136	142	.819949	.804197
53	.573164	.577579	143	.813258	.795548
54	.582300	.586934	144	.806244	.786497
55	.591342	.596202	145	.798899	.777034
56	.600289	.605379	146	.791214	.767150
57	.609140	.614464	147	.783180	.756835
58	.617892	.623455	148	.774787	.746077
59	.626543	.632350	149	.766024	.734865
60	.635093	.641147	150	.756880	.723186
61	.643539	.649841	151	.747341	.711028
62	.651880	.658410	152	.737394	.698377
63	.660114	.666932	153	.727024	.685219
64	.668240	.675319	154	.716215	.671536
65	.676255	.683599	155	.704948	.657314
66	.684158	.691769	156	.693205	.642533
67	.691948	.699828	157	.680963	.627175
68	.699623	.707773	158	.668198	.611219
69	.707180	.715604	159	.654884	.594642
70	.714619	.723317	160	.640991	.577420
71	.721938	.730910	161	.626485	.559525
72	.729135	.738382	162	.611328	.540929
73	.736209	.745730	163	.595476	.521599
74	.743157	.752953	164	.578880	.501498
75	.749979	.760018	165	.561482	.480588
76	.756672	.760703	166	.543216	.458824
77	.763234	.773846	167	.524003	.436154
78	.769665	.780545	168	.503750	.412521
79	.775963	.787107	169	.482342	.387859
80	.782125	.793531	170	.459642	.362091
81	.788150	.799814	171	.435477	.335126
82	.794036	.806941	172	.409627	.306855
83	.799782	.814418	173	.381803	.277147
84	.805336	.821791	174	.351616	.245837
85	.810847	.829191	175	.318511	.212713
86	.816162	.836935	176	.281659	.177491
87	.821330	.844124	177	.239697	.139773
88	.826349	.850655	178	.190070	.098944
89	.831218	.856427	179	.126576	.053930
90	.835934	.849637	180		

Table 1 presents values of $\sin \phi$ and $\sin \phi$ as a function of ϕ for the particular case of zero vapor drag; i.e., the unperturbed classical solution of Nusselt. The values of dimensionless film thickness used to construct the table are those presented by Abramowitz [8] and the averages were computed by simple trapezoidal quadrature using 0.5 deg intervals. On observing that the original solution presented by Nusselt [1] and quoted by Jakob [9] and Boelter, et al. [10], is in disagreement with Abramowitz, the latter results were checked, to six significant figures, by performing a 4th-order Runge-Kutta numerical integration of the original differential equation. It may be seen from the table that $\sin \phi$ and $\sin \phi$ are almost identical over most of the cylinder, a significant discrepancy occurring only as ϕ approaches 180 deg. That the $\sin \phi$ averages are relatively insensitive to whether they are taken with respect to δ^2 or δ^4 suggests that they should also be insensitive to perturbations of δ introduced by vapor drag. It is therefore proposed that the $\sin \phi$ averages presented in Table 1 should also be adequate for the vapor drag solutions. The validity of this proposition will be confirmed by comparison with exact numerical solutions.

Numerical Verification

A general computer program has been developed which solves the finite-difference analog of the conservation equations in boundary-layer form for laminar film condensation. In these equations the coordinates are considered in terms of metric coefficients h_1 and h_2 such that $h_1 d\xi_1$ and $h_2 d\xi_2$ represent the infinitesimal real distances dx_1 and dx_2 , respectively. The equations governing conservation of mass, momentum, and energy for steady laminar film condensation on a two-dimensional surface of arbitrary shape are, respectively, as follows:

$$\frac{\partial}{\partial \xi_1} (\rho u_1 h_2) + \frac{\partial}{\partial \xi_2} (\rho u_2 h_1) = 0. \quad (24)$$

$$\frac{\rho \mu_1}{h_1} \frac{\partial u_1}{\partial \xi_1} + \frac{\rho \mu_2}{h_2} \frac{\partial u_2}{\partial \xi_2} = \frac{1}{h_1 h_2} \frac{\partial}{\partial \xi_2} \left(\frac{h_1 \mu}{h_2} \frac{\partial u_1}{\partial \xi_2} \right) + \rho g \sin \phi - \frac{1}{h_1} \frac{\partial p}{\partial \xi_1} \quad (30)$$

$$\frac{\rho C_p u_1}{h_1} \frac{\partial T}{\partial \xi_1} + \frac{\rho C_p u_2}{h_2} \frac{\partial T}{\partial \xi_2} = \frac{1}{h_1 h_2} \frac{\partial}{\partial \xi_2} \left(\frac{h_1 k}{h_2} \frac{\partial T}{\partial \xi_2} \right) \quad (31)$$

Viscous dissipation and compressibility effects are omitted from the energy equation since low velocity liquid flow is under consideration. The boundary conditions are identical to those considered in the extended Nusselt theory and in the present notation are

$$T = T_w \text{ and } u_1 = u_2 = 0 \text{ at } \xi_2 = 0 \quad (32)$$

$$T = T_s \text{ and } \frac{\mu}{h_2} \frac{\partial u_1}{\partial \xi_2} = m(U_{\infty} - u_1) \text{ at } \xi_2 = \xi_{2,s} \quad (33)$$

The local thickness of the condensate film is determined from an overall enthalpy balance on a length of film between a reference location $\xi_{1,0}$ and the location in question, ξ_1

$$\int_0^{\delta} \rho u_1 h_2 d\xi_2 \Big|_{\xi_1} = \int_{\xi_{1,0}}^{\xi_1} \frac{k}{h_2} \frac{\partial T}{\partial \xi_2} h_2 d\xi_1 \Big|_{\xi_2=0} + \int_0^{\delta} \rho u_1 h_2 d\xi_2 \Big|_{\xi_{1,0}} \quad (34)$$

Modifying a technique developed by Patankar and Spalding [11], a transformation of independent variable is made wherein the cross-stream variable ξ_2 is defined in terms of the stream function as

$$\xi_2 = \left(\frac{\psi - \psi_w}{\psi_s - \psi_w} \right)^{1/\gamma} \quad (35)$$

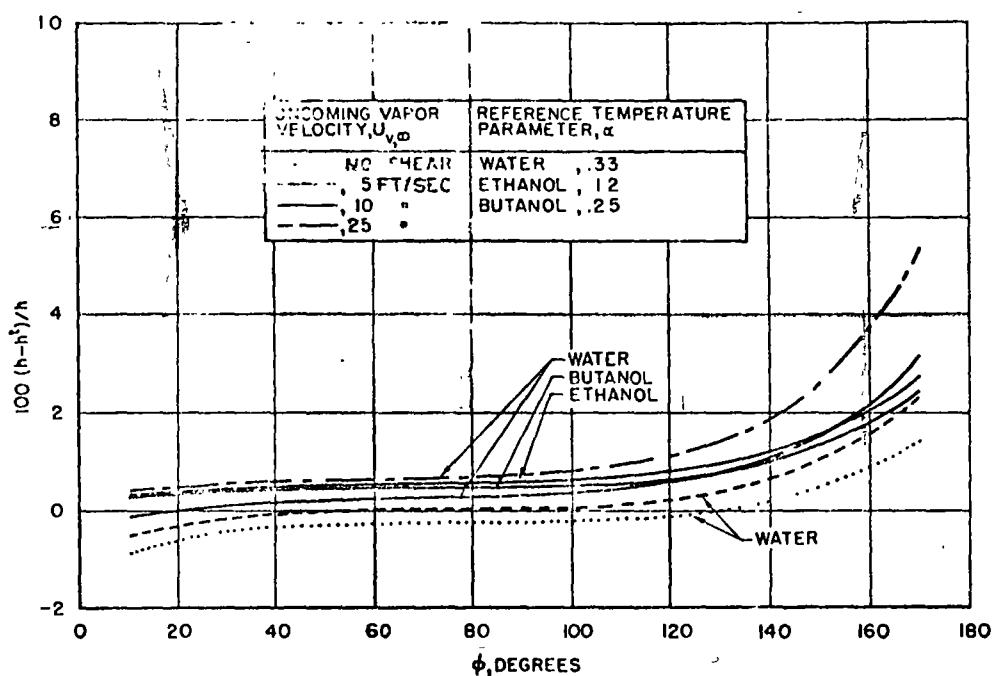


Fig. 2 Comparison of the theoretical local heat transfer coefficient h' , equation (24), with the numerical solution h . $T_s = 110$ deg F, $\Delta T = 40$ deg F, and $R = 0.125$ ft.

where a value of 2 for the constant γ proved to minimize the truncation errors in the computational procedure. Details of the transformed equations, their finite-difference analogs and the computer program are given in references [6,7]. The program has been shown to provide accurate solutions to a wide variety of laminar film condensation problems. Its application to the present problem is straightforward. The aforementioned references demonstrate the reliability of the numerical method by comparison with similar solutions and discuss such matters as machine times and computational errors. In the present calculations the step sizes and convergence criteria were chosen such that the numerical results would be in error by no greater than 0.5 percent.

Computations were performed for three fluids: water, ethanol and butanol. In reference [6] a reference temperature for evaluating locally variable fluid properties is defined in the form $T_r = T_w + \alpha(T_s - T_w)$ and accounts for both the effects of fluid properties and minor errors introduced by the Nusselt assumptions. The values of α derived in reference [6] for condensation on a vertical surface, namely, 0.33 for water, 0.12 for ethanol, and 0.25 for butanol were used for the present computations. Sources for the thermophysical property data are also listed in reference [6].

Results and Discussion

Comparison of Theory With Numerical Solutions. Fig. 2 shows a comparison of the theoretical local heat transfer coefficient h' from equation (24) with the numerical solution h , as a function of angle ϕ . In every case the discrepancy is less than 2 percent for ϕ less than 140 deg. The increased discrepancy as ϕ approaches 180 deg could be due to either inadequacies in the simplifying assumptions, or violation of the Nusselt assumptions. Fig. 3 shows a comparison of the inertial and gravitational terms in the momentum balance equation (1) obtained from the exact numerical solution for a typical case. It can be seen that the Nusselt assumption of negligible inertial effects is grossly violated as ϕ approaches 180 deg. Further investigation of the discrepancy between the theoretical and numerical solutions is thus of doubtful value. In conclusion it may be concluded that the reference temperature values derived for a vertical surface are adequate for the

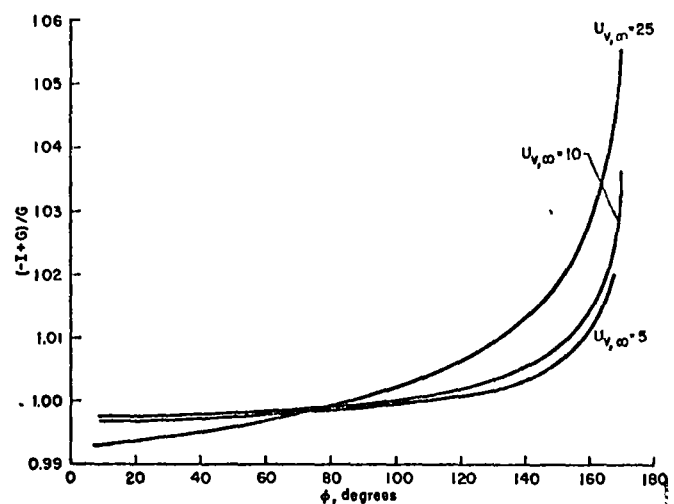


Fig. 3 Comparison of the relative magnitude of the inertial terms I , where

$$I = \int_0^{\xi_{1,2}} \rho(u_1 \partial u_1 / l_1 \partial \xi_1 + u_2 \partial u_1 / l_2 \partial \xi_2) d\xi_2$$

with the gravitational term G , where $G = \rho g \sin \phi$

horizontal cylinder as well. Values of the parameter α for a variety of fluids may be found in reference [6]. The absolute effect of vapor velocity on the local heat transfer coefficient is illustrated for water in Fig. 4.

Significance of the Results. It has been shown that the Nusselt assumptions and the proposed analytical solution are in excellent agreement for ϕ less than 140 deg. Restricting attention initially to the liquid film behavior, the foregoing result is important for two reasons. (i) In a typical situation 85 percent of the total condensation takes place when ϕ is less than 140 deg. (ii) Surface tension effects associated with liquid run-off cannot be ignored near the bottom of the tube. It is clear that analyses of more complex situations, such as the noncondensable gas problem, can now be justifiably and appreciably simplified by treating the liquid film on a constant property Nusselt basis.

It remains to evaluate the present results in relation to their

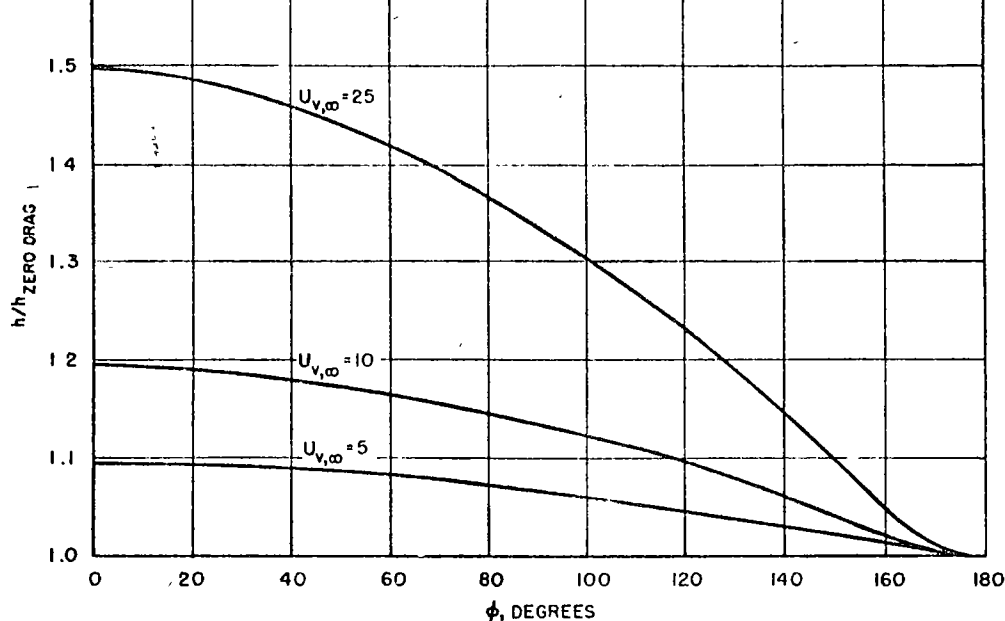


Fig. 4 Effect of oncoming vapor velocity on the local heat transfer coefficient for water. $T_s = 110$ deg F, $\Delta T = 40$ deg F, and $R = 0.125$ ft

utility in engineering practice. This requires the discussion of a number of issues.

1 Use of the asymptotic shear expression allows an analytical solution to be obtained for the Nusselt problem. The lower limit on the condensation rate for which the corresponding suction velocity is large enough to insure the validity of the analytical solution is discussed at length in reference [6]. For the range of variables considered in the present investigation the conclusion is that the error in the heat transfer introduced by

the asymptotic shear expression is less than 1 percent for $\frac{\dot{m}}{\rho_s U_{v,s}} \times$

$\sqrt{\frac{U_{v,s} x_1}{\nu_s}} > 2.0$. Precision with respect to this criterion is not of great importance since the various parameters involved vary over many orders of magnitude. Reference [6] presents numerical data for the suction parameter typical of engineering applications showing that the present results indeed have a wide range of utility.

2 If the vapor boundary layer separates, then the resulting reverse flow invalidates the shear expression and a boundary-layer analysis of the vapor flow becomes impossible. Shekrladze and Gomelauri [5] discuss this aspect in some detail and, invoking a criterion due to Prandtl, note that at sufficiently high condensation rates, separation will not take place. The problem for a separated vapor boundary layer is beyond the scope of the present contribution. For this reason attention has been focused on the prediction of local heat transfer rather than total condensation rate.

3 As mentioned in the theoretical development, the interaction of the ϕ -variation of the suction velocity with the inviscid vapor flow has been ignored. A solution procedure which may be used to account for the interaction is presented in reference [7]. It involves numerical evaluation of the trigonometric series appearing in the appropriate solution to La Place's equation. Sample calculations demonstrate that the error in the local heat transfer, due to ignoring the interaction, is in no case greater than 2 percent. It is suggested that inclusion of the interaction effect would only be warranted if a solution of vapor boundary-layer flow, more exact than the asymptotic form used here, were being attempted.

(iv) The effects of surface instabilities have been ignored.

The low film Reynolds numbers characteristic of condensation on horizontal tubes (in contrast to vertical tubes) insure that the usual criterion of $\frac{4\Gamma}{\mu} < 30$ for the absence of surface waves

will be met in most practical applications. The results of the present analysis are considered valid for situations where surface waves are not observed. The surface instability associated with liquid run-off from the bottom of the tube is discussed by Henderson and Marchello [12] where a correlation for the effect on total condensation rate is presented in terms of the Ohnesorge

number $Oh = \frac{\mu}{(\rho g R \sigma)^{1/2}}$.

The Zero Vapor Drag Solution. It has been pointed out that the numerical calculations of Abramowitz [8] indicate that the original solution of the zero vapor drag problem presented by Nusselt [1] and quoted by Jakob [9] and Boelter, et al. [10], is in error. The correct solution shows that the film thickness increases monotonically with ϕ and does not go through a minimum at $\phi \approx 5$ deg as found by Nusselt. This monotonic behavior is essential to the ease and success of the sin ϕ averaging employed in the present analysis. In addition, it is noted that an examination of the original governing ordinary differential equation shows directly that the film thickness must necessarily be a monotonic function of ϕ . It is surprising that this observation was not made earlier since the behavior of the film predicted by Nusselt's results contradicts physical intuition.

References

- 1 Nusselt, W., "Die Oberflächen-Kondensation des Wasserdampfes," *Zeitschrift des Vereins Deutscher Ingenieure*, Vol. 60, 1916, pp. 569-575.
- 2 Bromley, L. A., "Effect of Heat Capacity of Condensate in Condensing," *Industrial and Engineering Chemistry*, Vol. 44, 1952, pp. 2966-2969.
- 3 Rohsenow, W. M., "Heat Transfer and Temperature Distribution in Laminar-Film Condensation," *Trans. ASME*, Vol. 78, 1956, pp. 1615-1618.
- 4 Sparrow, E. M., and Gregg, J. L., "Laminar Condensation Heat Transfer on a Horizontal Cylinder," *JOURNAL OF HEAT TRANSFER*, *Trans. ASME, Series C*, Vol. 81, 1959, pp. 291-296.
- 5 Shekrladze, I. G., and Gomelauri, V. I., "Theoretical Study of Laminar Film Condensation of a Flowing Vapor," *International Journal of Heat and Mass Transfer*, Vol. 9, 1966, pp. 581-591.

6 Denny, V. E., and Mills, A. F., "Nonsimilar Solutions for Laminar Film Condensation on a Vertical Surface," to be published in the *International Journal of Heat and Mass Transfer*.

7 Denny, V. E., et al., "A General Computer Program for the Solution of the Equations Governing Laminar Film Condensation," Report No. 69-62, Department of Engineering, University of California, Los Angeles, 1969.

8 Abramowitz, M., "Tables of the Functions $\int_0^\phi \sin^{1/2} x dx$ and $\frac{4}{3} \sin^{-1/2} \phi \int_0^\phi \sin^{1/2} x dx$," *Journal of Research, National Bureau of Standards*, Vol. 47, pp. 288-290.

9 Jakob, M., *Heat Transfer*, Wiley, Vol. 1, New York, 1919, pp. 667-670.

10 Roelker, L. M. K., et al., "Heat Transfer Notes," University of California Press, Berkeley, Calif., 1948, pp. XV-9.

11 Patankar, S. V., and Spalding, D. B., "A Finite Difference Procedure for Solving the Equations of the Two-Dimensional Boundary Layer," *International Journal of Heat and Mass Transfer*, Vol. 10, 1967, pp. 1389-1411.

12 Henderson, C. L., and Marchello, J. M., "Role of Surface Tension and Tube Diameter in Film Condensation on Horizontal Tubes," *AIChE Journal*, Vol. 13, 1967, pp. 613-614.

LOUISIANA POWER & LIGHT COMPANY
STERLINGTON STEAM ELECTRIC STATION
1958 - 210 000 KW EXTENSION - UNIT NO.6

CONDENSER TESTS

General Information

Manufacturer	Foster Wheeler
Manufacturer's Number	
Condenser Serves	No. 6 Turbine
Object of tests	To determine performance

Surface Condenser

Description, Dimensions, etc.

Nominal size	125000 sq ft
Type of condenser	Top inlet
Number of passes	Two for each half
Tube data	
(a) Total number	16040
(b) In each half	8020
(c) Per pass	8020
(d) Length between tube sheets	29.77 ft.
(e) Outside diameter	1.0 in.
(f) Thickness	18 Bwg
(g) Material	Phosphorized admiralty
(h) Internal or bore area	0.00449 sq ft/tube, 35.68 sq ft/pass
(i) Number of inactive tubes	None
(j) How installed	Expanded at both ends
(k) Maker of tubes	Revere Copper and Brass.

Design Conditions

(a) Condenser heat load	1061000 K Btu per hour
(b) Quantity circulating water	104000 gpm
(c) Velocity thru tubes	6.5 ft/sec
(d) Inlet circulating water temp	90 F
(e) Back pressure with 85% clean tubes	3.13 in.

691

2

LOUISIANA POWER & LIGHT COMPANY
STERLINGTON STEAM ELECTRIC STATION
1958 - 210 000 KW EXTENSION - UNIT NO.6

HEAT BALANCE METHOD OF CALCULATING CONDENSER HEAT LOAD
200 000 KW TEST MARCH 8, 1960

a) Heat Input to Condenser Btu/hr:

Steam from Low Pressure Turbine	868,971 X 1038.45	=	902,382,935
Boiler Feed Pump Gland Return	18,000 X 58	=	1,044,000
Vent from LP-1 Heater	338 X 1134.2	=	383,360
Seal Line Vents	466 X 1401.2	=	652,959
Steam Seal Leakage	1,800 X 1401.2	=	2,522,160
Drains from Gland Steam Exhauster	2,367 X 103	=	243,801
Drains from LP-1 Heater	185,905 X 47.5	=	8,830,488
Total			916,059,703

b) Heat Leaving Condenser Btu/hr:

Condensate Flow	1,080,647 X 39.6	=	42,793,621

c) Condenser Heat Load Btu/hr:

Condenser Heat Load Equal C = a - b		=	873,266,082

LOUISIANA POWER & LIGHT COMPANY
STERLINGTON STEAM ELECTRIC STATION
1958 - 210 000 KW EXTENSION - UNIT NO.6

"ENERGY BALANCE" METHOD FOR CALCULATING CONDENSER HEAT LOAD

1 - Heat added to cycle:

- (a) By initial steam
- (b) By reheat steam
- (c) By enthalpy increase thru pumps
- (d) By boiler makeup, if any

2 - Heat leaving cycle not including that to cooling water:

- (a) Heat equivalent of internal or wheel kilowatts
- (b) An allowance for radiation

CALCULATIONS FOR TEST OF MARCH 8, 1960

$$1 - (a) = 1,244,000(1481.4 - 419.6) = 1,320,879,200 \text{ Btu/hr.}$$

$$1 - (b) = (1,137,748)(1525.6 - 1321.8) = 231,752,042 \text{ Btu/hr.}$$

$$\text{Total from boiler} = 1,552,752,242 \text{ Btu/hr.}$$

$$1 - (c) = \frac{1,244,120 \times 2.53 \times 2076}{778.3 \times 0.728} = 11,532,714 \text{ Btu/hr.}$$

Where:

- 1,244,120 = Feedwater Flow.
- 2.53 = Feet head per psi.
- 2076 = Feed pump discharge pressure, psia.
- 778.3 = Mechanical equivalent of heat.
- 0.728 = Feed pump efficiency, for this load (see note below).

$$1 - (d) = \text{None on this test}$$

$$2 - (a) = (199111 + 2875)(3412.75) = 689,327,722 \text{ Btu/hr where } 3412.75 = \text{heat equivalent of one kilowatt.}$$

$$2 - (b) = \text{Assumed cycle radiation loss taken as } 0.3\% \text{ of heat added to cycle by boiler} = 1,552,752,242 \times 0.003 = 4,658,257 \text{ Btu/hr.}$$

Condenser Load:

$$\begin{array}{r} 1,552,752,242 \\ 11,516,705 \\ - 4,658,257 \\ - 687,327,722 \\ \hline 870,282,968 \text{ Btu/hr} \end{array}$$

NOTE: Gpm per pump = $\frac{1.244.100}{2 \times 457.9} = 1360$

Where 457.9 = Pounds per hour/gpm for pump inlet pressure and temperature conditions during test. Find pump efficiency at 1360 gpm from characteristic curves for pumps.

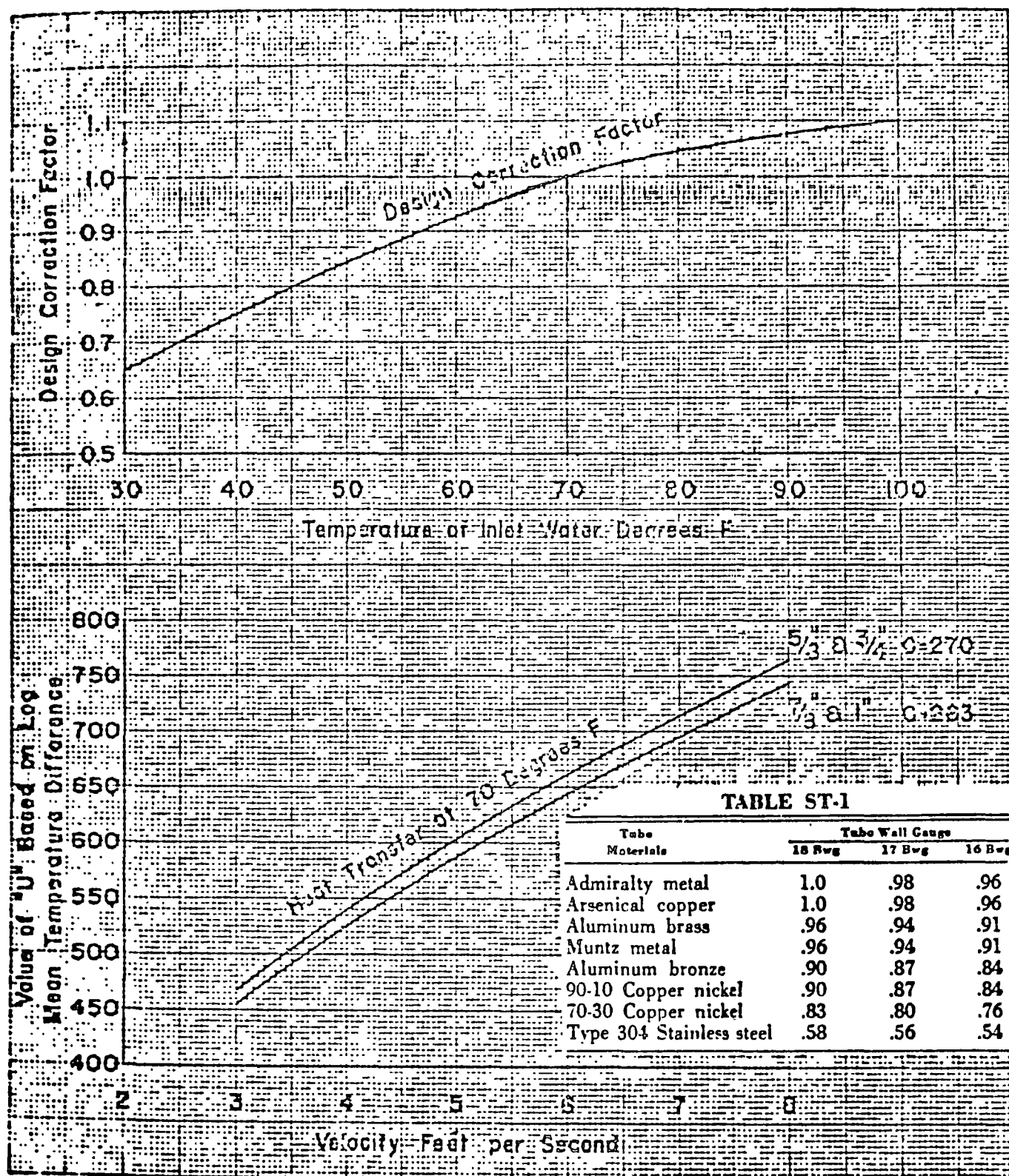


Fig. SF-2

Note 1: Heat transfer coefficient (U) based on $CV\sqrt{VEL}$ and on new, clean, bright and oxide free tubes. For the proper corrections to be used with various materials, gauges and loading, see paragraphs S-15 through S-18.

Note 2: Curves apply to condensers serving steam turbines.

Note 3: For condensers serving steam engines, use 65% of above values.

LOUISIANA POWER & LIGHT COMPANY
STERLINGTON STEAM ELECTRIC STATION
1958 - 210 000 KW EXTENSION - UNIT NO.6

-73-

6

CONDENSER TESTS

	Mar 25	Mar 8	May 4	April 2
1. Date of tests in 1960				
2. Unit load				
(a) Generation	235556	199111	176444	125778
(b) Electrical losses	2375	2043	1880	1528
(c) Mechanical or fixed losses	832	832	832	832
(d) Total wheel KW	238763	201986	179156	128138
3. Back pressure using basket type tips				
(a) In neck 24 in below turbine north end	1.089	0.867	1.437	1.274
(b) Ditto south end	1.165	0.937	1.445	1.279
(c) 30 in above tubes north end	1.045	0.817	1.414	1.256
(d) Ditto south end	1.057	0.813	1.438	1.280
4. Barometer at center line of turbine corrected to 32 F, Hg in.	29.99	30.11	29.81	30.03
5. Mercury column corrected to 32 F, Hg in.	28.85	29.12	28.27	28.69
6. Back pressure (Item 4 - Item 5)	1.14	0.99	1.54	1.34
7. Steam temperature entering condenser from average of Items 3a and 3b	82.71	75.91	90.41	86.59
8. Net heat added to cycle by boiler and reheater K Btu/hr	1841067.0	1552752.2	1393694.1	995730.2
9. Heat added to cycle by work of pumps K Btu/hr	13681.4	11532.7	10583.1	7973.6
10. Approximate radiation loss taken as 0.3 per cent of net heat added by boiler and reheater K Btu/hr	5523.2	4658.3	4181.1	2987.
11. Heat equivalent of wheel kilowatts K Btu/hr	814838.4	689327.7	611414.6	437303.0
12. Heat to circulating water equals (Item 8 + Item 9) minus (Item 10 + Item 11) K Btu/hr. This is condenser heat load by "Energy Balance"	1034386.8	870298.9	788681.5	563413.6
13. Steam to condenser				
(a) Lb/hr	1046374	868971	796380	563963
(b) Btu/lb	1032.6	1038.45	1045.2	1047.96
(c) K Btu/hr	1080485.8	902382.9	832376.4	591010.7
14. Other heat entering condenser K Btu/hr				
(a) From heater drips	12696.2	8830.5	9093.5	5271.6
(b) From gland condenser drips	269.3	243.8	273.1	231.4
(c) From BFP gland water return	1742.4	1044.0	1719.0	1582.2
(d) Vent from LPI heater	505.2	383.4	396.0	268.3
(e) Steam vent from seal line	653.9	652.9	651.7	651.1
(f) Steam from seals	2525.8	2522.2	2517.5	2515.1
(g) From makeup water	0	0	0	0
15. Total heat entering condenser K Btu/hr	1098878.6	916059.7	847027.2	601530.4
16. Condensate leaving condenser				
(a) Lbs/hr	1302144	1080647	964196	679388
(b) Temperature, F	78.7	71.6	88.7	84.
(c) Btu/lb	46.7	39.6	56.7	52.8
(d) K Btu/hr	60810.1	42793.6	54669.9	35871.7

Δt between exhaust steam and condenser heat well suit varied 4.3 F to 1.8 F

17. Condenser heat load from "Heat Balance" - K Btu/hr (Item 15 Minus Item 16d)	1038068.5	873266.1	792357.3	565658.7
18. Free air removed from condenser CFM	3.9	3.5	3.3	3.9
19. Number of inactive tubes	NONE	NONE	NONE	NONE
20. Condenser surface in use, sq ft	125000	125000	125000	125000
21. Actual heat transfer Btu/sq ft/hr State whether from "Energy Balance" or "Heat Balance"	HB 8305	HB 6986	HB 6338.9	HB 4525
22. Temperature of steam entering Condenser, F				
(a) From average of two Zimmerli gages 24 in below turbine joint	82.71	75.91	90.44	86.59
(b) From back pressure (Item 6)	83.06	78.73	92.56	88.12
23. Circulating water				
(a) Temperature entering, F	51.37	41.48	70.46	65.44
(b) Temperature leaving, F	69.80	57.44	84.84	82.83
(c) Temperature rise, F	18.43	15.96	14.38	17.39
(d) Initial temperature difference, F	31.34	34.43	19.98	21.15
(e) Terminal temperature difference, F	12.91	18.47	5.60	3.76
(f) Log mean temperature difference, F	20.78	25.62	11.31	10.07
(g) Average of entering and leaving Temperature, F	60.59	49.46	77.65	74.14
(h) Arithmetic mean temperature difference, F	22.12	26.45	12.79	12.45
(i) Specific heat	1.0	1.0	1.0	1.0
(j) Specific gravity	Actual weights of water used			
(k) Weight, M lbs/hr = $\frac{\text{condenser load}}{\text{temp rise} \times \text{sp. ht.}}$	56324.9	54707.4	55101.3	32527.8
(l) Weight/cu ft at entering temperature, lbs.	62.41	62.43	62.30	62.34
(m) Volume at entering temperature				
(1) Cu ft/sec	250.69	243.43	245.68	144.93
(2) GPM	112517	109259	110279	65049
24. Internal or bore area of tubes per pass, sq ft.	35.58	35.58	35.58	35.58
25. Velocity through tubes at inlet water temperature, ft/sec	7.05	6.84	6.91	4.07
26. Actual heat transfer coefficient, Btu/hr/sq ft/LMTD	399.6	272.67	560.47	449.38
27. Design heat transfer coefficient Btu/sq ft/hr/LMTD. From HEI curves, figure S-2 for 70F CW temp., clean tubes, and velocity during test. NOTE: For one inch 18 BWG clean tubes, and for velocities of 8.0 ft/sec or less, use 263 times square root of velocity in ft/sec.	698.1	687.92	691.11	530.80
28. Correct item 27 for actual CW inlet temperature, by use of HEI temperature correction curves, figure S-2	595.5	525.57	692.49	512.22
29. Expected heat transfer coefficient equals Item 28 times 0.85	506.17	446.73	588.62	435.39
30. Theoretical exhaust steam temp. from equations (2) and (3) or by equation (4) as shown on attached sheet (Methods for Calculating Condenser Back Pressure) based on item 29.	78.68	66.43	89.97	86.81

31.	Expected or theoretical back pressure corresponding to temperature found in item 30, Hg in NOTE: Compare this value with the average of 3a and 3b to determine whether performance is better or worse than expected.	0.989	0.654	1.421	1.285
32.	Apparent cleanliness factor				
	$\% = \frac{\text{Item 26}}{\text{Item 28}} \times 100$	67.1	51.9	80.9	87.73
33.	Pressure drop through tubes and water boxes				
	(a) Actual average differential Hg column, Hg inches underwater	17.31	17.77	17.11	6.82
	(b) Convert item 33a to feet water	18.14	18.64	17.92	7.14
	(c) Correct item 33b to 104000 gpm, feet water	15.50	16.89	15.94	18.25
34.	Guarantee with 90 F CW temp 85% clean tubes, and 104000 gpm, water feet	16.7	16.7	16.7	16.7

METHODS FOR CALCULATING CONDENSER BACK PRESSURE

A = Outside area of tubes, sq ft
 W = Weight of circulating water, lb/hr
 U = Heat transfer coefficient
 T_i = Temperature inlet circulating water, F
 T_o = Temperature outlet circulating water, F
 C_p = Specific heat circulating water
 T_s = Steam temperature entering condenser, F
 T_m = Log mean temperature difference, F
 $\Delta T = T_o - T_i$

H = Heat given up per lb steam times pounds of steam

$H = WC_p (T_o - T_i)$

Let Antilog_e = $\frac{\Delta T}{T_m} = P$

Equation (1)

$$\Delta T = \frac{H}{WC_p}$$

$$T_m = \frac{H}{UA}$$

$$\text{Log}_e \frac{T_m}{T_s - T_i} = \frac{\Delta T}{T_s - T_o}$$

Equation (2)

$$P = \text{Antilog}_e \frac{UA}{WC_p}$$

Equation (3)

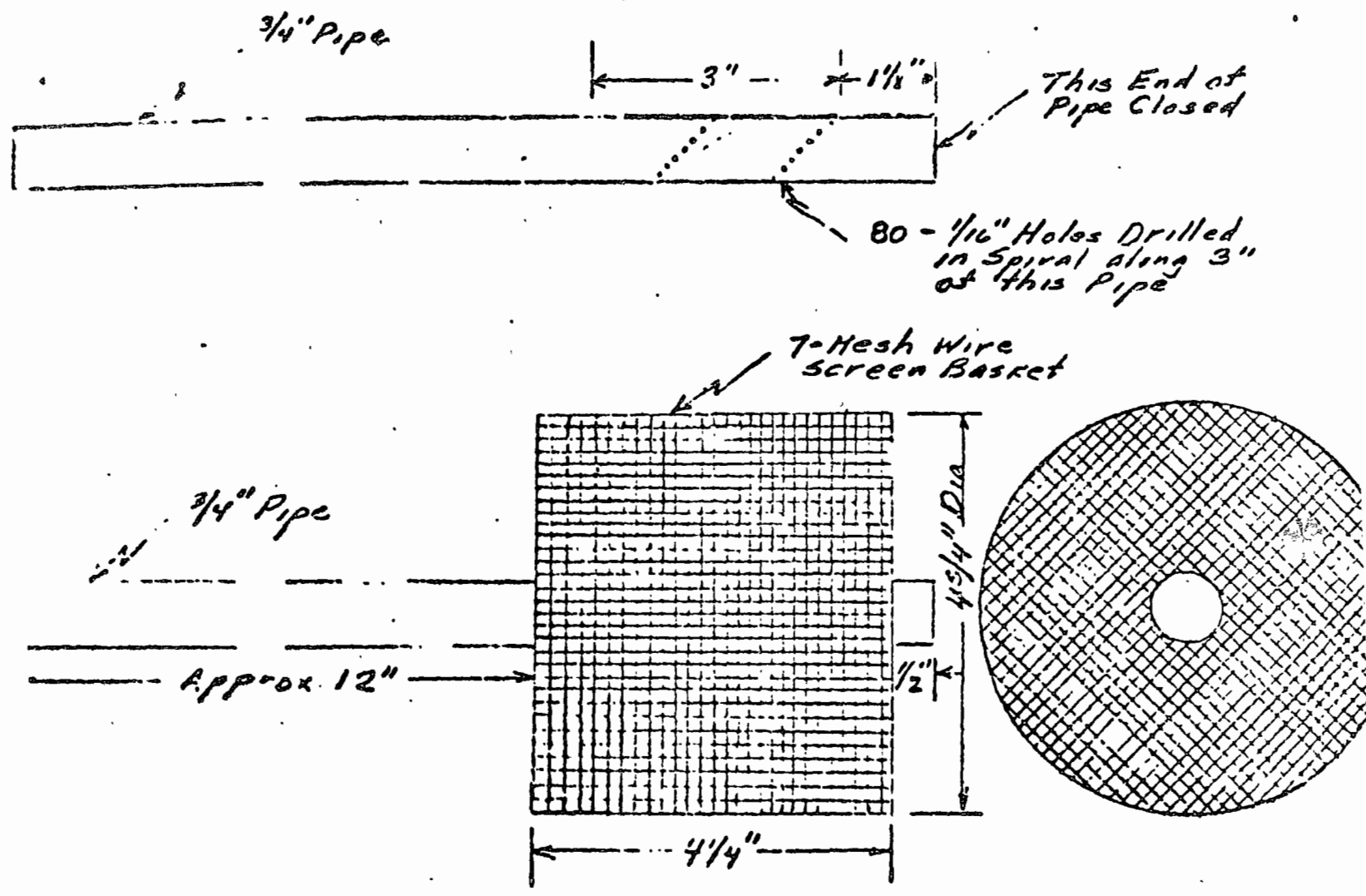
$$T_s = T_i + \frac{P \Delta T}{P - 1}$$

Equation (4)

$$T_s = T_i + \frac{\text{Condenser Load}}{WC_p \left(1 - \frac{1}{\text{Antilog}_e \frac{UA}{WC_p}} \right)}$$

Look up back pressure corresponding to T_s

LOUISIANA POWER AND LIGHT COMPANY
 NINE MILE POINT STEAM ELECTRIC STATION
 125000 KW EXTENSION - UNIT NO. 2



PRESSURE TIP FOR TURBINE EXHAUST

To be installed as near as practicable at plane of the turbine exhaust flange.

These pressure tips are relatively insensitive to the angle of the stream of flow



SURFACE CONDENSERS

SELECTION AND PERFORMANCE

PERFORMANCE

CONFIDENTIAL

INGERSOLL-RAND COMPANY
11 BROADWAY, NEW YORK

PERFORMANCE CALCULATIONS FOR ANY DESIGN CONDITIONS

D. PERFORMANCE CALCULATIONS FOR ANY CONDITIONS.

For performance calculations on a given condenser, or to calculate the required surface, circulating water quantity and velocity at any design point, the "K and R" method explained below may be used. This method is considered the simplest and most flexible of several alternate methods. All formulae are derived from fundamental equations of heat transfer for condensers and the standards of the Heat Exchange Institute.

In making calculations two points should be kept in mind, as explained in greater detail later.

1. At lower duties condenser performance must be modified because of the limitation of the air removal equipment.
2. The minimum terminal temperature difference (sometimes called approach) used in condenser calculations is 5.0 F. (Terminal TD is the difference between the inlet steam temperature and the outlet circulating water temperature. It is also equal to ITD - TR).

Symbols and abbreviations used—

- Tube constant depending on tube diameter and gauge. Values are listed in table 503.
- b - Tube material and gage correction factor (table 503)
- C - tube constant (varying with tube diameter) used in determining values of U in accordance with the HEI standards. (Table 503).
- f - Inlet water temperature correction factor. (Curve 23)
- K - variable factor calculated from other known factors.
- L - foot of total water travel through the tubes of all water passes. (Effective tube length x no. of passes).
- % clean - specified or assumed percent of clean tube heat transfer coefficient.
- R - ratio of temperature rise to initial temperature

difference. ($R = \frac{TR}{ITD}$). Also R has a definite

relationship to K ($R = 1 - \frac{K}{C}$)

- S - total sq ft of condenser surface (including the surface in the air cooler section).
- TR - temperature rise (deg F) of circulating water.
- TD - initial temperature difference between steam in condenser and inlet circulating water. (deg F).
- T_i - temperature of inlet circulating water (deg F).
- T_o - temperature of outlet circulating water (deg F).
- T_s - temperature of saturated steam corresponding to the design absolute pressure. (deg F). (Table 504).
- U - overall coefficient of heat transfer through the tubes. (Btu per hr per sq ft per deg. LMTD.)

CALCULATION OF REQUIRED SURFACE AND CIRCULATING WATER.

Before starting the actual calculations, the design conditions, as well as a specific tube size, tube length, number of water passes, and water velocity, must be either specified or assumed. See Ex. 1, Sheet 20.

1. Determine T_s (from steam tables) corresponding to the design absolute pressure. (Table 504).
2. Calculate ITD, from following formula:
 $ITD = T_s - T_i$
3. Determine water temperature correction factor from curve 23, page 18b.
4. Calculate the value of K by the following formula:

$$K = \frac{L \times a \times b \times f \times \% \text{ clean}}{\sqrt{v} \times 100}$$

Note: Use 65% clean for engine service.

5. Read the value of R corresponding to K, from table 506.
6. Determine TR, by following formula:
 $TR = R \times ITD$
7. CAUTION: ITD-TR from steps 4 and 5 above, must not be less than 5.0 F. If the terminal temperature difference comes out less than 5.0 deg. F, then reduce the TR by either increasing the water velocity, increasing the tube size, or decreasing the tube length until the resultant approach is equal to or greater than 5.0 F. If none of these factors can be changed, then the ITD must be increased by either increasing the design absolute pressure, or decreasing the design water temperature so that the new ITD will be at least equal to or greater than TR + 5 deg. F.
8. Calculate the gpm of circulating water required, by the following heat balance formula:

$$gpm = \frac{(\text{lb per hr of steam}) \times 950}{TR \times 500}$$

950—common assumed value for net Btu rejected to circulating water per lb of steam condensed for turbines. A value of 970 net BTU is often used for reheat turbines.

500—factor for conversion of gpm to lb per hour of water.

9. Find value of U by following formula:

$$U = C \sqrt{\frac{K}{S}} \times f \times b \times \frac{\% \text{ clean}}{100}$$

The standard value of U for 70° water and clean 18-BWG Admiralty tubes can also be read from curve No. 23, but must be corrected for other water temperatures, cleanliness factors, tube gages and materials as described above.

10. Calculate the sq ft of surface required, by—

$$S = K \times gpm \times 500$$



PERFORMANCE CALCULATION FOR ANY DESIGN CONDITIONS

11. Find the number of tubes required to make up the total surface.

$$\text{No. of tubes} = \frac{S}{(\text{tube length}) \times (\text{sq ft per linear ft of tube})}$$

Values for the sq ft per linear foot of tube for different tube sizes are listed in table 503.

CALCULATION OF THE EXPECTED PERFORMANCE OF A GIVEN CONDENSER.

Frequently it is necessary to calculate the absolute pressure that will be maintained by a given condenser having a specified or assumed surface and quantity of circulating water. In this case the method is similar, but the sequence of steps is different. (See Ex. 2, Sheet 20)

12. For the specific surface, tube size and tube length calculate the number of tubes as in step (11) above. (This can also be obtained from page 2500.40, Sheet 4.)

13. For the assumed quantity of water, calculate the water velocity by the following formula: (If the velocity is assumed instead of the gpm, then solve for the gpm using the same formula.)

$$V = \frac{\text{gpm}}{(\text{no. tubes per pass}) \times (\text{gpm per tube @ 1 fps})}$$

Values of the gpm per tube at 1.0 ft per sec are listed in table 503.

14. Determine water temperature correction factor from curve 23, page 18b.

15. Calculate the value of K, using the same formula as in (1) above.

16. Find R corresponding to K, from Table 506.

17. Find TR by the following heat balance formula:

$$TR = \frac{(\text{lb per hr of steam}) \times 950}{\text{gpm} \times 500}$$

950 is assumed for turbine service.

18. Find ITD, by $ITD = \frac{TR}{R}$

NOTE: Check the terminal TD. If ITD-TR is less than 5.0 F, then neglect the value of ITD calculated in step 16, and use the following formula: $ITD = TR + 5.0$.

19. Find T_s , by $T_s = ITD + T_i$.

20. From steam tables read the absolute pressure corresponding to the saturated steam temperature determined in step (16).

NOTE:—When the condenser design point is for circulating water temperatures below 60° F, it may be necessary to modify the condenser performance because of the limitation of the air removal equipment. If the design absolute pressure falls below the cutoff point read from Curve A fig. 24, a performance curve should be drawn, as described in Paragraph E and the design pressure taken from the performance curve.



SURFACE CONDENSERS

SELECTION AND PERFORMANCE

PERFORMANCE

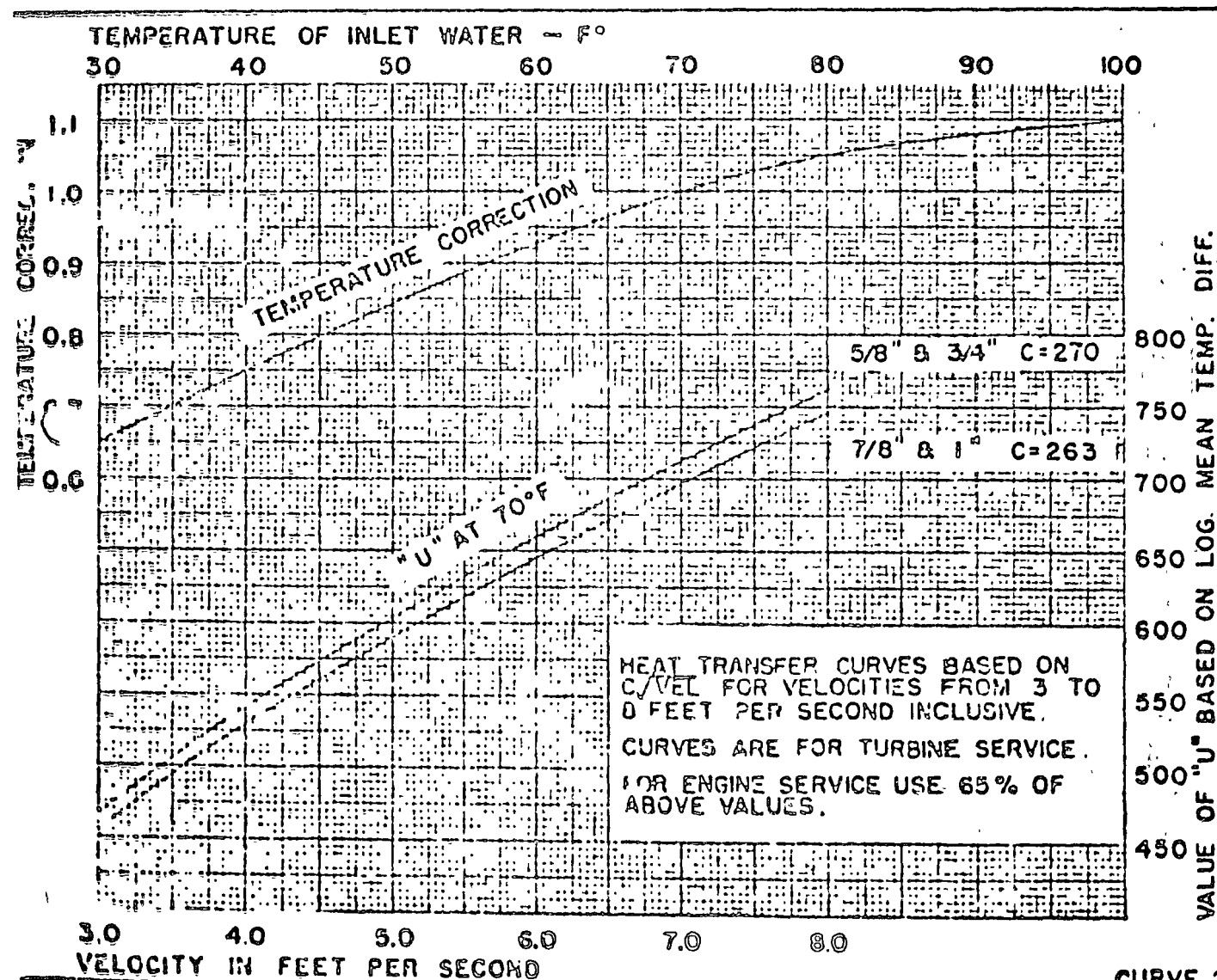
CONFIDENTIAL

INGERSOLL-RAND COMPANY
11 BROADWAY, NEW YORK

TUBE CONSTANTS

Table 303

Inch O.D.	Sq ft per linear ft. of tube	Values of Constant "C" in $U = \frac{C}{CV\sqrt{v}}$	GPM per tube at 1.0 ft. per sec. velocity		Value of "a" in formula $K = \frac{L \times a}{\sqrt{v}} \times b \times f \times \frac{\% \text{ clean}}{100}$		Values of "b" in formula $K = \frac{L \times a}{\sqrt{v}} \times b \times f \times \frac{\% \text{ clean}}{100}$		
			16 BWG	18 BWG	16 BWG	18 BWG	Tube Material	16 BWG	18 BWG
1/8	0.1636	270	0.60	0.68	0.1472	0.1299	Adm. & Aca. Cu.	.96	1.0
1/4	.1963	270	0.94	1.04	.1126	.1019	Al Brass & Muntz	.91	.96
3/8	.2291	263	1.36	1.48	.0887	.0815	Al. Bronze & 90: 10 Cu. Ni.	.84	.90
1/2	.2618	263	1.85	1.99	.0744	.0692	70: 30 Cu. Ni.	.76	.83



0.30
BACK OF
SHEET
1, 1904



SURFACE CONDENSERS

SELECTION AND PERFORMANCE

PERFORMANCE

CONFIDENTIAL

INGERSOLL-RAND COMPANY
11 BROADWAY, NEW YORK

14

Table 504

SATURATED STEAM TEMPERATURES

Absolute Pressure lb. Hg.	.40	.01	.02	.03	.04	.05	.06	.07	.08	.09	Absolute Pressure lb. Hg.
.2	34.56	35.78	36.96	38.09	39.18	40.23	41.23	42.22	43.17	44.08	.2
.3	46.96	45.83	46.67	47.48	48.28	49.05	49.80	50.53	51.25	51.96	.3
.4	52.64	53.31	53.98	54.62	55.25	55.88	56.48	57.08	57.66	58.24	.4
.5	58.80	59.35	59.90	60.43	60.96	61.48	62.00	62.49	62.99	63.47	.5
.6	63.96	64.43	64.90	65.35	65.81	66.26	66.70	67.13	67.56	67.99	.6
.7	68.40	68.82	69.23	69.63	70.03	70.43	70.81	71.20	71.58	71.96	.7
.8	72.33	72.70	73.06	73.42	73.78	74.13	74.48	74.83	75.17	75.51	.8
.9	75.85	76.18	76.51	76.83	77.15	77.47	77.79	78.11	78.42	78.73	.9
1.0	79.03	79.33	79.64	79.94	80.23	80.52	80.81	81.10	81.39	81.67	1.0
1.1	81.95	82.23	82.51	82.78	83.06	83.33	83.60	83.87	84.13	84.39	1.1
1.2	84.65	84.91	85.17	85.43	85.68	85.93	86.18	86.43	86.68	86.92	1.2
1.3	87.17	87.41	87.65	87.89	88.12	88.36	88.59	88.83	89.06	89.28	1.3
1.4	89.51	89.74	89.97	90.19	90.41	90.63	90.85	91.07	91.29	91.50	1.4
1.5	91.72	91.93	92.14	92.35	92.56	92.77	92.98	93.19	93.39	93.60	1.5
1.6	93.80	94.00	94.20	94.40	94.60	94.80	95.01	95.20	95.39	95.59	1.6
1.7	95.78	95.97	96.16	96.35	96.54	96.72	96.91	97.10	97.28	97.46	1.7
1.8	97.65	97.83	98.01	98.19	98.37	98.55	98.73	98.91	99.09	99.26	1.8
1.9	99.44	99.61	99.78	99.96	100.13	100.30	100.47	100.64	100.81	100.97	1.9
2.0	101.14	101.31	101.47	101.64	101.80	101.97	102.13	102.30	102.46	102.62	2.0
2.1	102.78	102.94	103.10	103.25	103.41	103.57	103.73	103.88	104.04	104.19	2.1
2.2	104.34	104.50	104.65	104.80	104.95	105.11	105.26	105.41	105.55	105.70	2.2
2.3	105.85	106.00	106.15	106.29	106.44	106.58	106.73	106.87	107.02	107.16	2.3
2.4	107.31	107.45	107.59	107.73	107.87	108.01	108.15	108.29	108.43	108.57	2.4
2.5	108.71	108.84	108.98	109.12	109.25	109.39	109.52	109.66	109.79	109.92	2.5
2.6	110.06	110.19	110.32	110.46	110.59	110.72	110.85	110.98	111.11	111.24	2.6
2.7	111.37	111.49	111.62	111.75	111.88	112.01	112.13	112.26	112.38	112.51	2.7
2.8	112.63	112.76	112.88	113.01	113.13	113.25	113.37	113.50	113.62	113.74	2.8
2.9	113.86	113.98	114.11	114.23	114.35	114.46	114.58	114.70	114.82	114.94	2.9
3.0	115.06	115.19	115.30	115.41	115.52	115.64	115.76	115.88	116.00	116.11	3.0
3.1	116.22	116.33	116.45	116.57	116.68	116.79	116.90	117.00	117.11	117.22	3.1
3.2	117.35	117.46	117.57	117.69	117.80	117.90	118.01	118.12	118.22	118.33	3.2
3.3	118.45	118.56	118.67	118.78	118.89	118.99	119.10	119.20	119.30	119.41	3.3
3.4	119.52	119.62	119.72	119.83	119.93	120.04	120.15	120.25	120.35	120.46	3.4
3.5	120.56	120.66	120.77	120.87	120.98	121.07	121.17	121.28	121.39	121.49	3.5
3.6	121.58	121.68	121.78	121.88	121.98	122.08	122.18	122.28	122.37	122.47	3.6
3.7	122.57	122.66	122.76	122.85	122.96	123.06	123.15	123.25	123.35	123.45	3.7
3.8	123.55	123.63	123.72	123.82	123.92	124.02	124.10	124.20	124.30	124.40	3.8
3.9	124.49	124.59	124.69	124.78	124.87	124.97	125.05	125.14	125.23	125.32	3.9
4.0	125.42	125.51	125.60	125.70	125.79	125.88	125.97	126.05	126.13	126.23	4.0
4.1	126.31	126.41	126.50	126.60	126.70	126.79	126.87	126.96	127.05	127.13	4.1
4.2	127.22	127.31	127.40	127.50	127.59	127.67	127.76	127.85	127.93	128.02	4.2
4.3	128.10	128.18	128.26	128.35	128.44	128.52	128.60	128.69	128.78	128.86	4.3
4.4	128.95	129.03	129.11	129.20	129.28	129.37	129.45	129.53	129.62	129.70	4.4
4.5	129.79	129.87	129.95	130.03	130.11	130.19	130.28	130.36	130.44	130.52	4.5
4.6	130.61	130.69	130.77	130.85	130.93	131.01	131.09	131.17	131.25	131.34	4.6
4.7	131.42	131.50	131.58	131.66	131.74	131.82	131.90	131.98	132.06	132.14	4.7
4.8	132.22	132.30	132.38	132.46	132.53	132.61	132.69	132.77	132.85	132.93	4.8
4.9	133.00	133.09	133.17	133.25	133.33	133.41	133.49	133.57	133.65	133.73	4.9

OPERACION Y CONTROL DE CONDENSADORES.-

El problema de operación y control de un condensador puede ser planteado tomando en consideración dos aspectos:

- a.- El aspecto económico directo, es decir, la temperatura del vapor de escape ó presión del condensador.
- b.- El aspecto económico indirecto que comprende la selección de los mejores métodos para corregir fallas como fugas de tubos, limpieza de estos, etc., para aumentar la disponibilidad del equipo.

Para el primer aspecto, es necesario, aumentar el interés y la habilidad del personal de la Planta para determinar el comportamiento de la unidad.- La habilidad para diagnosticar debe ser mejorada al punto que pueda determinarse si el consumo de calor específico no es el esperado y que parámetro de operación ó componente es el que está causando las pérdidas.

Se ha investigado que la falta de interés y habilidad se deben entre otras causas a la casi total ausencia de técnicas que puedan entenderse y utilizarse fácilmente.

Para tener una mejor idea de la importancia del aspecto económico, es conveniente recordar que por cada $^{\circ}\text{C}$ de disminución de temperatura en el vapor de escape se disminuye el consumo de calor en 0.25%. Esto significa que para una unidad de 150 000 KW con consumo.

específico de 2800 cal/Kw-hr trabajando con un factor de carga de 70%, con la sola economía obtenida al disminuir 1°C la temperatura del vapor de escape se pueden generar 6300 Kw-hr/día; si se considera que el costo es de \$0.10/Kw-hr se tendría una economía de \$630.00/día/unidad.

Una forma de determinar la alteración del consumo del calor específico es midiendo y comparando la presión en el condensador. La desviación de la presión de referencia puede deberse a las siguientes causas:

1.- Una reducción en la sección transversal del área de paso del agua de circulación lo cual producirá los siguientes efectos:

- a) Reducción en el flujo de agua de circulación.
- b) Aumento de la velocidad en los tubos, resultando pérdidas de carga de bombeo y pérdidas de fricción del flujo.

2.- Cambio en el coeficiente de transferencia de calor que resulta de los tubos sucios.

3.- Fugas de aire produciendo un aumento en la presión del condensador y fugas de agua de circulación tensión del condensado produciendo un aumento de conductividad en éste.

4.- Pérdidas causadas por la erosión en la bomba produciendo una disminución en el flujo.

5.- Taponeamiento de las rejillas de la succión produciendo un aumento de carga en la succión de la bomba.

Algunas de las maniobras que puede efectuar el operador para corregir esta anomalía son las siguientes:

- 1.- Arranque y operación de equipo adiciónamiento de extracción de aire.
- 2.- Infiltración del flujo de agua circulación a través del condensador.
- 3.- Arranque y operación de bombas de agua de circulación adicionales.
- 4.- Operación y limpieza de las rejillas.
- 5.- Arranque del equipo de exploración.

Sí la desviación no se corrige deberá informarse al supervisor para determinar el efecto de esta desviación en el consumo específico de calor y cuanto está costando.

Las referencias para determinar las desviaciones pueden obtenerse construyendo una familia de curvas como la mostrada en la figura 10, las cuales se determinan con datos del Instituto de Intercambio de Calor (HEI) y resolviendo la siguiente ecuación:

La nomenclatura y la obtención formal de esta ecuación se da en el apéndice I.

Las curvas deben construirse para cualquier condición de —

operación y presentarse al supervisor los resultados continuamente.

Una alternativa para presentar esta información es por medio de una computadora analógica que aproximaría las curvas de referencia de acuerdo con la siguiente ecuación:

$$P_s = f_1(t_1) - f_2(t_1) \cdot f(q)$$

en donde:

$f_1(t_1)$ es una función de t_1 en el punto 1 en las curvas de la Fig. 10.

$f_2(t_1)$ es una segunda función de t_1 en el punto 2 en las curvas de la figura 10.

f_q es una función de la carga térmica del condensador.

El paso siguiente es la determinación del parámetro de operación ó componente que está causando la desviación. Del análisis de la ecuación 1 se observa que, como el flujo de agua de enfriamiento normalmente es casi constante, la presión del condensador dependerá de lo siguiente:

- 1.- De la temperatura inicial (t_1) del agua de enfriamiento.
- 2.- De la carga térmica del condensador.
- 3.- Del coeficiente de transferencia de calor K , que a su vez vez depende de lo siguiente:
 - a) de la calidad del diseño del condensador.
 - b) De la limpieza de la superficie de transferencia de calor.
 - c) De la presión parcial del aire en la cámara de con

densación que depende de la relación entre la infiltración de aire y la capacidad de los inyectores.

Del análisis del coeficiente de transferencia de calor K para un condensador dado se tiene que este coeficiente está formado por las siguientes diferentes resistencias:

- 1.- Resistencia de contacto entre el vapor y la pared fría (k_1)
- 2.- Resistencia entre la película de condensado y el aire eventual (k_2)
- 3.- Resistencia a través de la pared metálica del tubo incluyendo una eventual capa de óxido ó de depósitos externos (k_3).
- 4.- Resistencia a través de la capa ó película interna de depósitos (k_4)
- 5.- Resistencia de convección en la película del agua de enfriamiento (k_5)

El coeficiente K se obtiene de los anteriores en la siguiente forma:

$$K = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4} + \frac{1}{k_5}}$$

k_1 tiene un valor bastante alto del orden de 20 000 a 100 000 $\text{cal/m}^2\text{°C/hr.}$

k_2 depende del arreglo físico de las haces de tubos; de la carga térmica, de la temperatura del agua de enfriamiento

y de la presión parcial del aire.

k3 Depende de la naturaleza del metal de los tubos y de su espesor.

k4 Está relacionado con la película ó capa interior de los tubos.

k5 Depende de la velocidad del agua en los tubos y no cambia cuando el flujo es constante.

Una vez determinadas las curvas de la fig. 10, cualquier desviación en la presión que no pueda ser corregida y las maniobras al alcance del operador se deberá ya sea a k2 ó a k4. Desafortunadamente no existen métodos sencillos de medición de estos parámetros por lo que será necesario efectuar pruebas para discriminar la causa y poder corregirla.

Un método sencillo que puede efectuarse con la unidad en operación para comprobar si las infiltraciones de aire son mayores que la capacidad de los eyectores, es la prueba del triángulo del aire que se muestra en la Fig. 11. En esta prueba se grafica la pérdida de vacío cuando se desconecta el eyector del condensador y enseguida la restauración del vacío cuando se vuelve a poner en servicio el eyector. Si el ángulo es mayor que el ángulo significa que las entradas de aire son mayores que la capacidad de los eyectores y serán las entradas de aire el factor que será necesario corregir. Por otra parte, si el ángulo es mayor que el ángulo, entonces no habrá duda que se trata de su capacidad interna de los tubos y se debe proceder a su limpieza para corregir la desviación.

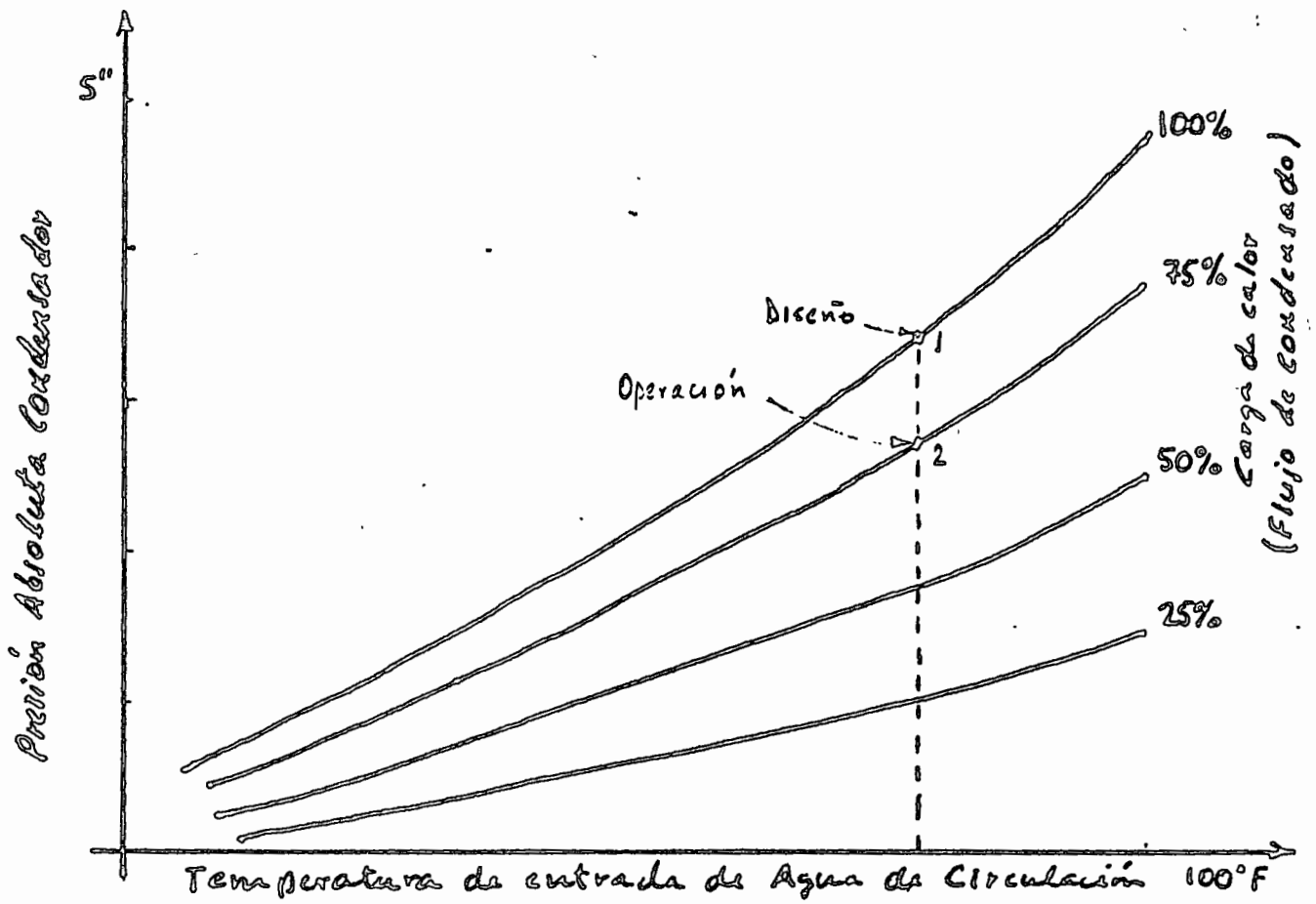


FIG. 10

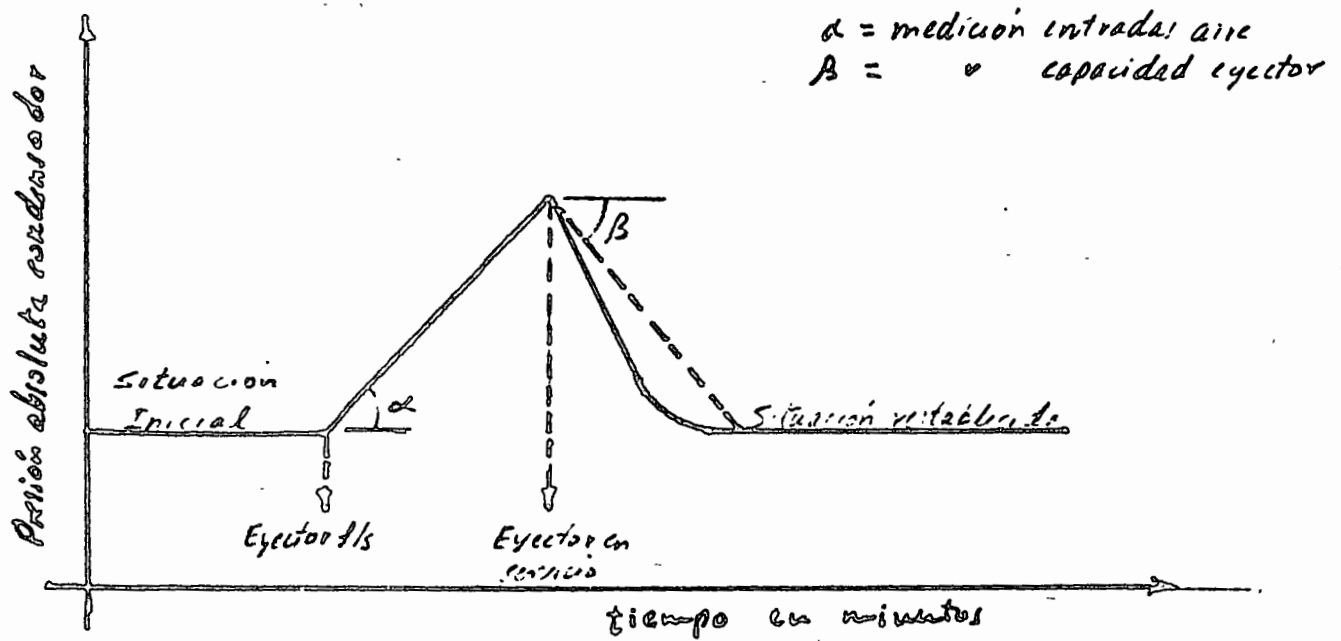


FIG. 11

MATERIALES MAS COMUNES EMPLEADOS EN LA FABRICACION DE CONDENSADORES.

1.- Cubierta, cuerpo ó carcasa.-

Se fabrica de acero especificación ASTM A-285 Grado C y ASTM A-70

2.- Cajas de agua y tapas. cajas de agua.-

Se fabrican ya sea de hierro o de acero dependiendo de las características del agua de circulación ó de enfriamiento.

El hierro que se emplea es el ASTM especificación A-48 clase 30 y - - ASTM 126 Grado A.

El acero que se utiliza es el ASTM especificación A-285 Grado C y - - ASTM A-70 calidad de bridas.

3.- Placas ó espejos de los tubos.-

Normalmente se fabrican de metal Muntz rolado ó de Admiralty, USN - - 46-B-6; Bronce naval rolado y ASTM especificación B-171.

4.- Placas soportes de los tubos.-

Se fabrican ya sea de hierro ó de acero.- El hierro es el tipo ASTM 126 Grado C y el acero el ASTM especificación A-285 Grado C y el ASTM A-70,

5.- Tubos.-

El material empleado para los tubos depende de las características del -- agua de circulación ó de enfriamiento. Para agua de buena calidad (de - pozo ó servicio municipal) se emplean tubos de metal Admiralty ó de cobre con arsénico, prefiriendo este último donde existen cantidades apreciables de polución orgánica en el agua que puedan causar la dezinificación del metal Admiralty.

En el caso de agua salada el empleo de tubos de bronce - aluminio da resultados satisfactorios en instalaciones estacionarias.

Cuando las condiciones del agua de circulación ó enfriamiento son bastante severas por su agresividad se requiere la instalación de tubos de aleación - - cobre-níquel.

En la Tabla III se dan las características ó composición de los métodos empleados en la fabricación de tuberías que quedan dentro del tipo ASTM especificación B-III.

COMPOSICION QUIMICA DE LOS MATERIALES DE LOS TUBOS.-

Aleacion	% (Porcentaje)										
	Cobre	Estano	Aluminio	Niquel	Plomo (max)	Hierro (max)	Zinc	Manganeso (max)	Antimonio	Aluminio	Fosforo
Metal Mista	59 a 63	-	-	-	0.3	0.07	resistente	-	-	-	-
- Admiralty:											
Tipo A	70 a 73	0.9 a 1.2	-	-	0.075	0.06	resistente	-	-	-	-
- B	70 a 73	0.9 a 1.2	-	-	0.075	0.06	✓	-	0.02 a 0.10	-	-
- C	70 a 73	0.9 a 1.2	-	-	0.075	0.06	✓	-	-	0.02 a 0.10	-
- D	70 a 73	0.9 a 1.2	-	-	0.075	0.06	✓	-	-	-	0.02 a 0.10
Bronce Rojo	64 (min)	-	-	-	0.075	0.06	✓	-	-	-	-
Bronce-Aluminio:											
Tipo B	76 (min)	-	1.75 a 2.50	-	0.075	0.06	✓	-	0.02 a 0.10	-	-
- C	76 (min)	-	1.75 a 2.50	-	0.075	0.06	✓	-	-	0.02 a 0.10	-
- D	76 (min)	-	1.75 a 2.50	-	0.075	0.06	✓	-	-	-	0.02 a 0.10
Bronce-Aluminio	resistente	-	5.0 a 6.5	-	0.10	0.10	-	-	0.35 (max)	-	-
70-30 Cupro-Niquel	✓	1.50 (max)	-	29 a 33	0.05	0.6	1.0 (max)	1.0	-	-	-
80-20 Cupro-Niquel											
Tipo A	resistente	1.0 (max)	-	19 a 23	0.05	0.6	1.0 (max)	1.0	-	-	-
- B	✓	1.0 (max)	-	19 a 23	0.05	0.6	3 a 6	1.0	-	-	-
90-10 Cupro-Niquel	✓	-	-	9 a 11	0.05	1.0	1.0 (max)	1.0	-	-	-
Cobre	99.9 (min)	-	-	-	-	-	-	-	-	-	0.04 max
Aluminio	99.4 (min)	-	-	-	-	-	-	-	0.15 a 0.50	-	-

I.- CONSIDERACIONES DE OPERACION.

- 1.- Se reconoce que el comportamiento de un condensador no puede ser pre-decido exactamente bajo cualquier condición posible de operación y --consecuentemente las curvas de comportamiento del condensador son da-das únicamente aproximadas exepcto para una condición específica "Pun-to de Diseño".
- 2.- Las condiciones comerciales de operación involucran variaciones meno--res incontrolables en el sistema de aire y capacidad de las bombas --de vacío, las cuales son despreciables bajo condiciones menos seve--ras pero es impractica la exacta predicción del comportamiento del --condensador en condiciones abajo de 5 grados F. de diferencia termi--nal de temperatura ó para cualquier condición de condensador con con--tenido de óxigeno menor de 0.03 c.c. por litro.
- 3.- Bajo ciertas condiciones y una construcción adecuada un contenido de óxigeno tan bajo como 0.01 c.c. por litro, puede obtenerse en la si--guiente forma:
 - a).- Con presiones del condensador no menores que la mostrada en la--gráfica de la figura 2.
 - b).- El aire y los gases no condensables removidos no deben exceder--el 25% de la capacidad del equipo extractor de aire. Para una tempe--ratura de entrada de agua de circulación menor de 50 grados F. el --equipo de aire deberá tener una capacidad mínima de 3 pies cúbicos --por minuto de aire libre a 5 pulgadas de mercurio de presión absolu--ta de succión y una temperatura de $7\frac{1}{2}$ grados F. abajo de la temperatura de saturación correspondiente a esta presión de succión.
 - c).- Las fugas de aire en el condensador abajo del nivel del conden--sador del pozo cal iente. *deben ser cero.*
 - d).- El total de agua entrando al condensador (~~deberá ser cero~~) a una --temperatura más baja de la entrada de vapor no deberá ser mayor del--5% del vapor que está siendo condensado.
 - e).- El método para muestra y medición de óxigeno deberá estar de --acuerdo con los estandar del Instituto de Intercambio de calor (HEI).
- 4.- Se reconoce también que velocidades del agua de enfriamiento en los--tubos del condensador menores de 3 pies por segundo no forman una re-sistencia de flujo en suficiente cantidad para forzar una cantidad --uniforme de agua atraves de todos los tubos y por lo tanto el compor--tamiento de un condensador en tales condiciones no puede predecirse.
- 5.- Cuando un condensador de superficie se usa en condiciones comercia--les de operación deberá considerarse el efecto del aire presente en--el vapor y la caída de presión necesaria para establecer el flujo a--traves del haz de tubos.

II.- TRANSFERENCIAS DE CALOR ESPECIFICAS.

- 1.- Con objeto de establecer las transferencias de calor específicas que son posibles de obtener bajo condiciones comerciales de operación y diseños eficientes de condensadores el Instituto de Intercambio de calor, ha realizado pruebas extensas para obtener valores que puedan representar máximos valores límites de diseño.
- 2.- Las curvas mostradas en la figura 3 deberán usarse en el diseño de condensadores de superficie. Estas curvas están basadas en tu bos nuevos, limpios, brillantes y libre de óxido.
- 3.- Las curvas de transferencia de calor de la figura 3 están basados a una temperatura de entrada de agua de circulación de 70 -- grados F. Para otras temperaturas de entrada de agua de circulación del coeficiente de transferencia de calor básico deberá mul tiplicarse por el correspondiente factor de corrección de temperatura.
- 4.- El coeficiente de transferencia de calor de la figura 3 se basa en tubos ^{metal} de la Almiralty calibre No. 18BWG. Para cualquier otro material y calibre, éstos coeficientes deben multiplicarse por -- el factor de corrección de la tabla 4.

III.- FACTOR DE LIMPIEZA.

Como se estableció anteriormente el coeficiente de transferencia de calor de la figura 3 están basados en los tubos nuevos, limpios, brillantes y libres de óxido; sin embargo debe de reconocerse que en condiciones reales de operación tanto el agua de -- circulación como las condiciones de vapor, producen películas re sistentes a la transferencia de calor sobre la superficies de los tubos las cuales tienen características relacionadas con el tipo de fluido y el material de los tubos.- Se reconoce tambien que *los tubos con ~~las~~ aleaciones sin cobre de tubos a bases de aleaciones de cobre,* ^{tiene menos resistencia a los depositos organicos que los} por lo tanto, las condiciones de agua y la pureza del vapor y sus efectos en el material de los tubos deberá considerarse al selec cionar un adecuado factor de limpieza asi como un coeficiente de diseño.

IV.- CURVAS DE COMPORTAMIENTO.

Teniendo establecido el correcto coeficiente de transferencia de calor para un condensador es posible graficar las curvas de comportamiento mostrando la presión absoluta para varias cargas ter micas del condensador y la temperatura de entrada de agua de circulación.

Se reconoce que abajas cargas las curvas de comportamiento deberán modificarse por la limitación del equipo de ^stracción de aire. La curva A de figura 4 muestra el límite superior al cual este efecto es el factor determinante, excepto cuando la presión absoluta es limitada por la diferencia terminal de temperatura ~~de~~ 5 grados F. La curva B muestra la mínima presión absoluta con cero de carga. En la construcción de curvas de comportamiento deberá llevarse una línea recta desde el punto del límite ^{superior} ~~inferior~~ al punto correspondiente a cero de carga para cada temperatura de entrada de agua de circulación.

V.- PERDIDAS HIDRAULICAS.

- 1.- Las perdidas hidráulicas (pérdidas de fricción) a través del condensador deberán ser también consideradas y las curvas de las figuras 5, 6 y 7 muestran las pérdidas por fricción que pueden esperarse en condensadores diseñados apropiadamente con tubos nuevos y limpios.
- 2.- Las figuras 5 y 6 indican las pérdidas de carga en pies por pies de recorrido a través de tubos nuevos y limpios.
- 3.- La figura 7 indica las pérdidas en las cajas de agua tomando en consideración las pérdidas de entrada y salida en los tubos.
- 4.- Las pérdidas por fricción total en condensadores se determinan sumando los valores de las curvas de la figura 5 ó 6 y la figura 7.

VI.- CALCULOS.

En los calculos de comportamiento de los condensadores, la diferencia de ~~la~~ temperatura ~~se mide~~ ~~logarítmica~~ ~~(media)~~ se expresa de la siguiente forma:

$$DMLT = \frac{t_2 - t_1}{\ln \frac{t_s - t_1}{t_s - t_2}}$$

EN DONDE:

t_s = Temperatura de entrada de vapor correspondiente a la presión de vapor seco saturado.

t_1 = Temperatura de entrada del agua de circulación.

t_2 = Temperatura de salida del agua de circulación.

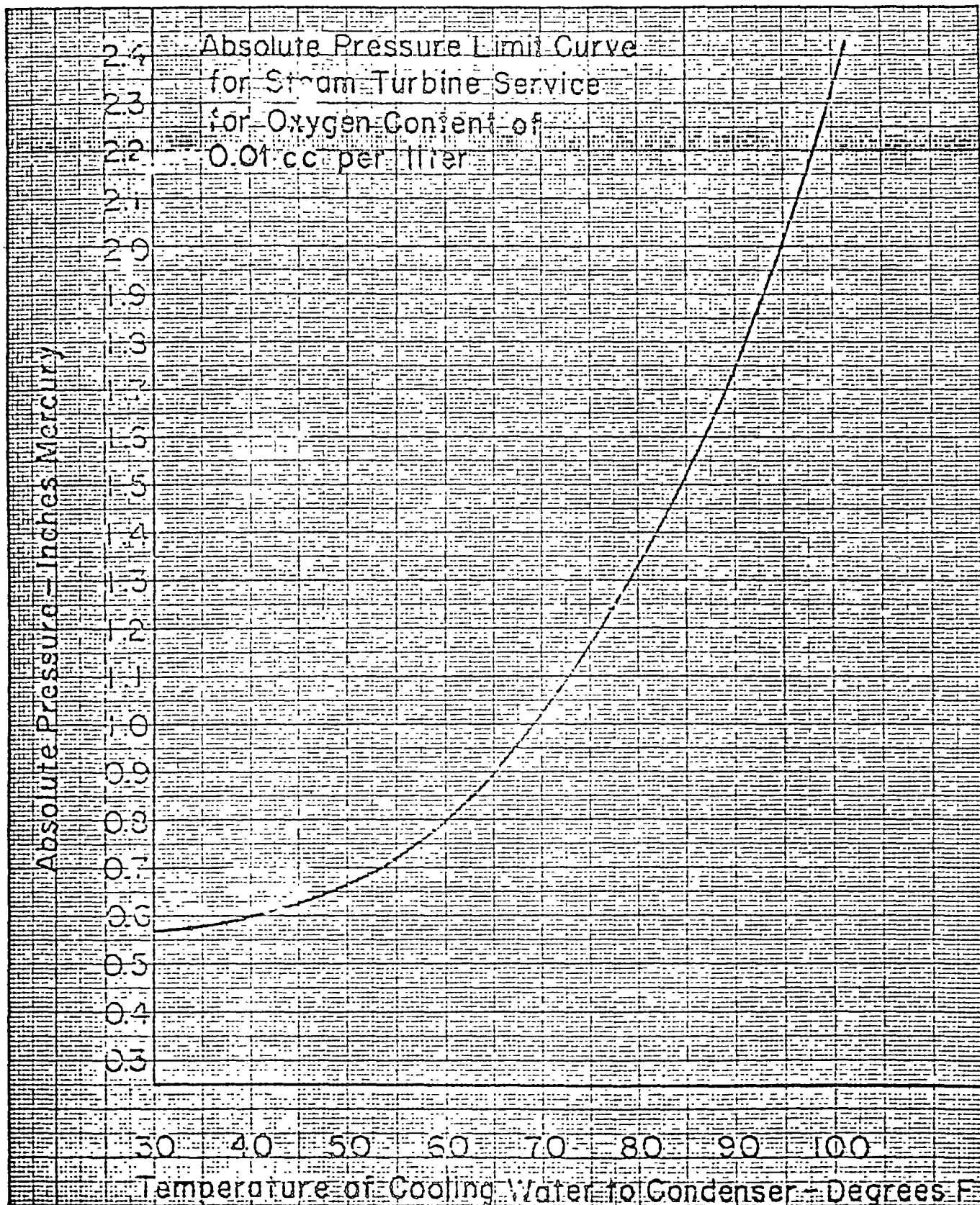


Fig. 2

STANDARDS OF THE HEAT EXCHANGE INSTITUTE

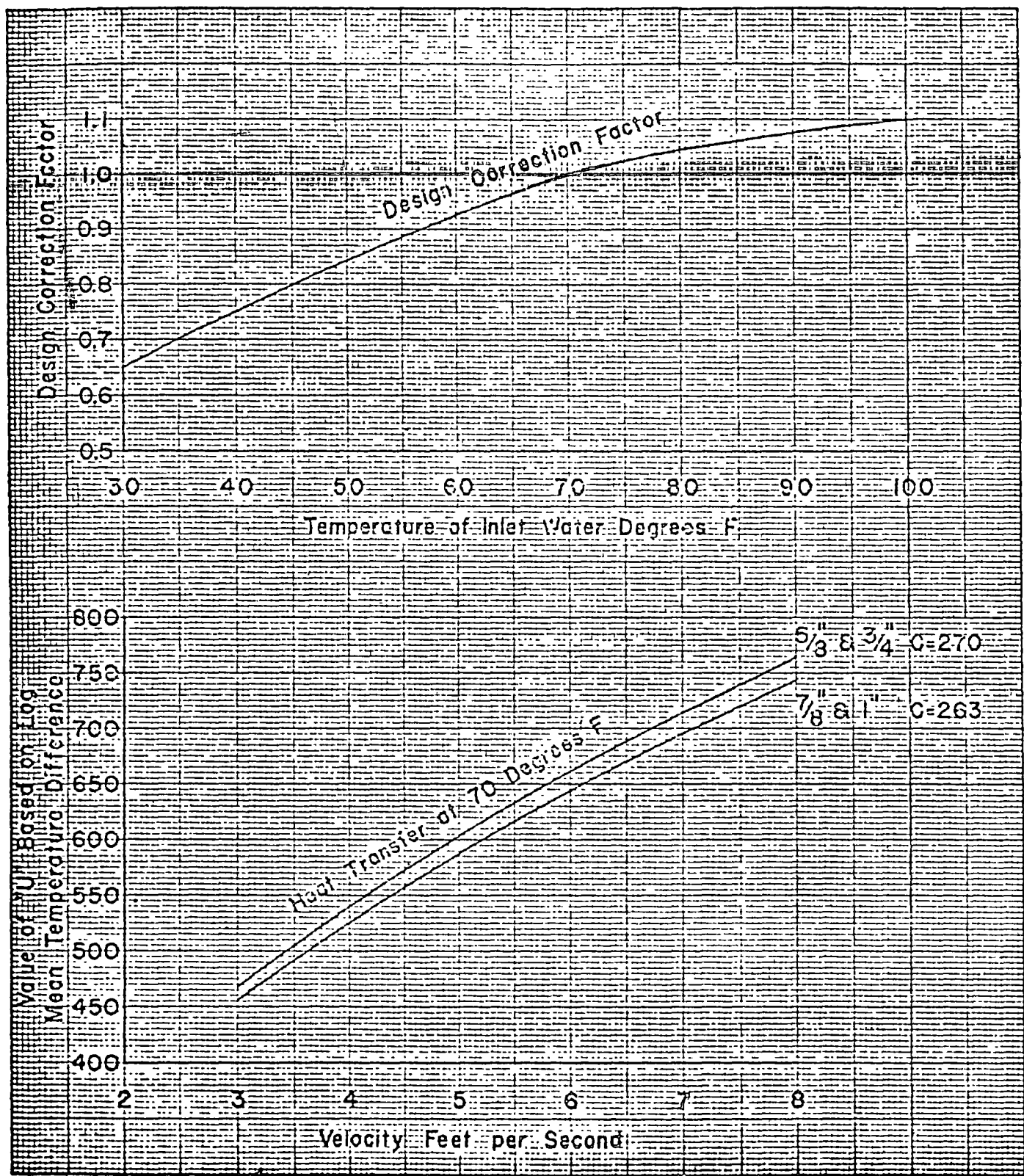


Fig. (SP-3)

Note 1: Heat transfer coefficient (U) based on $C \sqrt{VEL}$ and on new, clean, bright and oxide free tubes. For the proper corrections to be used with various materials, gauges and loading, see paragraphs S-15 through S-18.

Note 2: Curves apply to condensers serving steam turbines.

Note 3: For condensers serving steam engines, use 65% of above values.

TABLA IV

FALTORES DE CORRECCION DEL COEFICIENTE DE TRANSMISION
DE CALOR POR MATERIAL Y CALIBRE DE TUBOS

Material de los tubos	Calibre de la pared del tubo (BWG)						
	22	20	18	17	16	15	14
Metal Admiralty	—	—	1.0	.97	.93	.88	.82
Cobre - Arsenico	—	—	1.0	.97	.93	.88	.82
Aluminio *	—	—	1.0	.97	.93	.88	.8
~~Bronze~~ Aluminio-Latón	—	—	.96	.93	.89	.84	.78
Metal Muntz	—	—	.96	.93	.89	.84	.78
Bronze-Aluminio	—	—	.90	.87	.84	.79	.72
90-10 Cupro-Niquel	—	—	.90	.87	.84	.79	.72
70-30 Cupro-Niquel	—	—	.83	.80	.76	.71	.64
Acero inoxidable tipos 304 y 306	.69	.68	.65	—	.58	—	—

* ASTM especificación B-234, Aleaciones M-1A, GS-11A

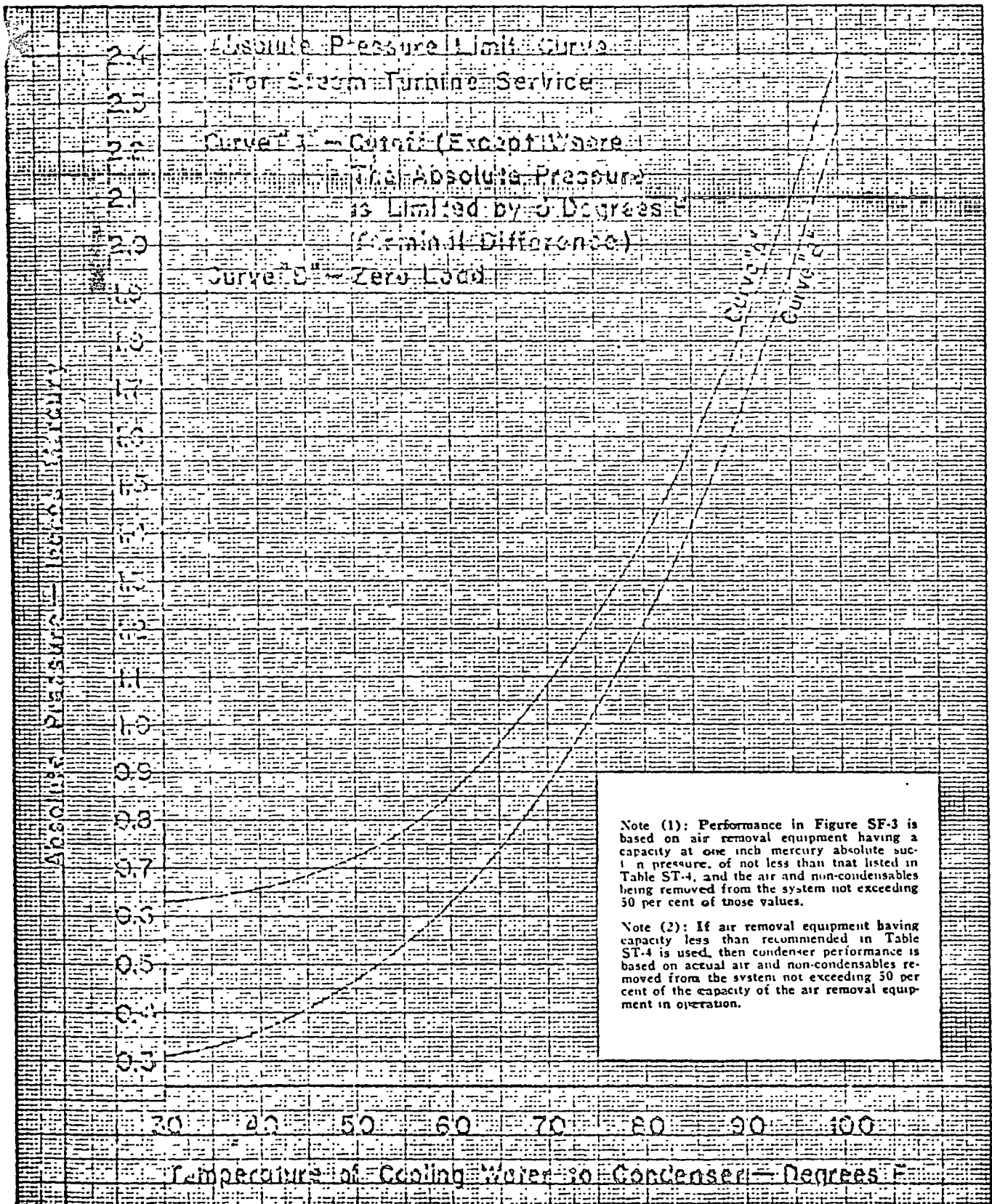


Fig. (SF-3) 4

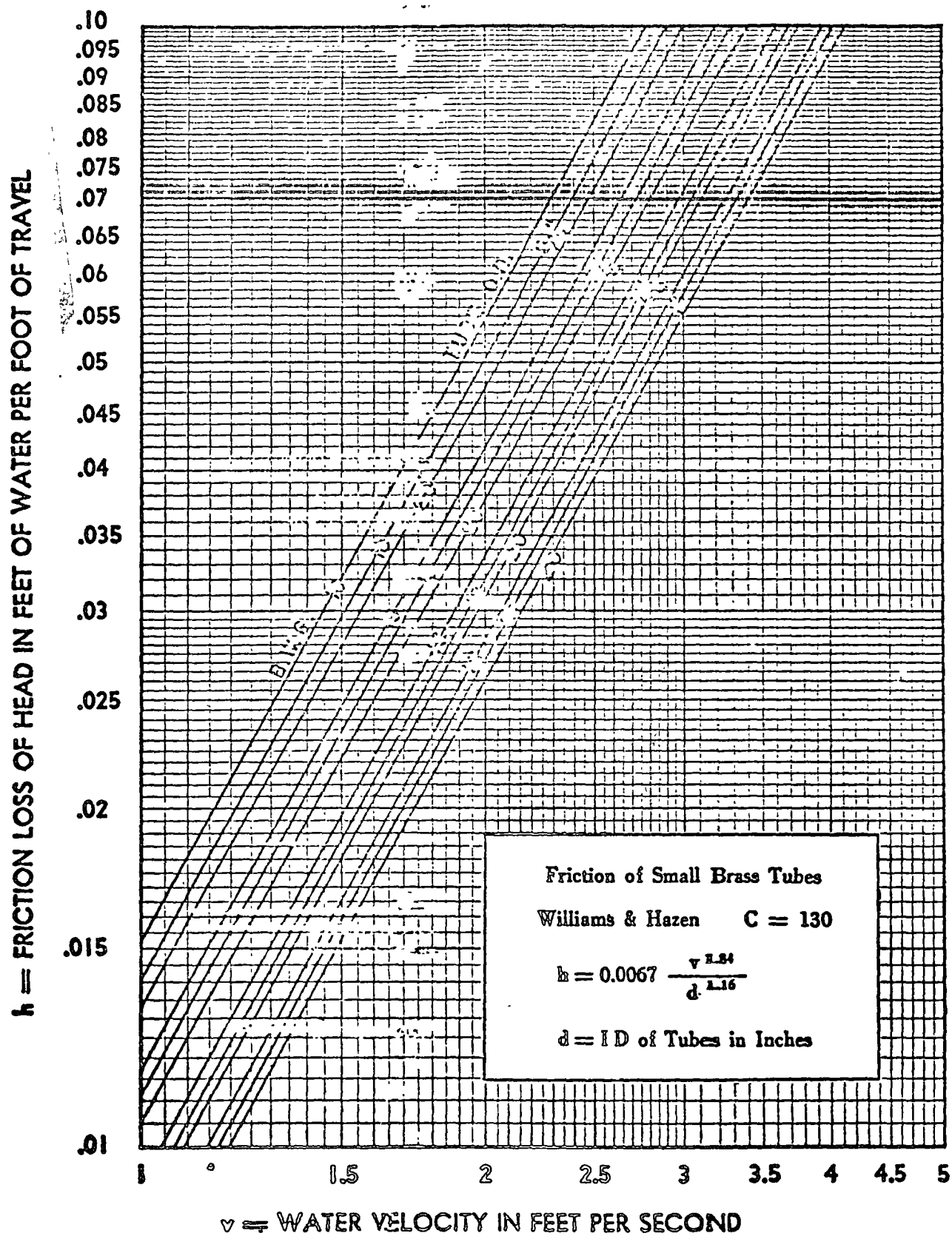


Fig. SF-5

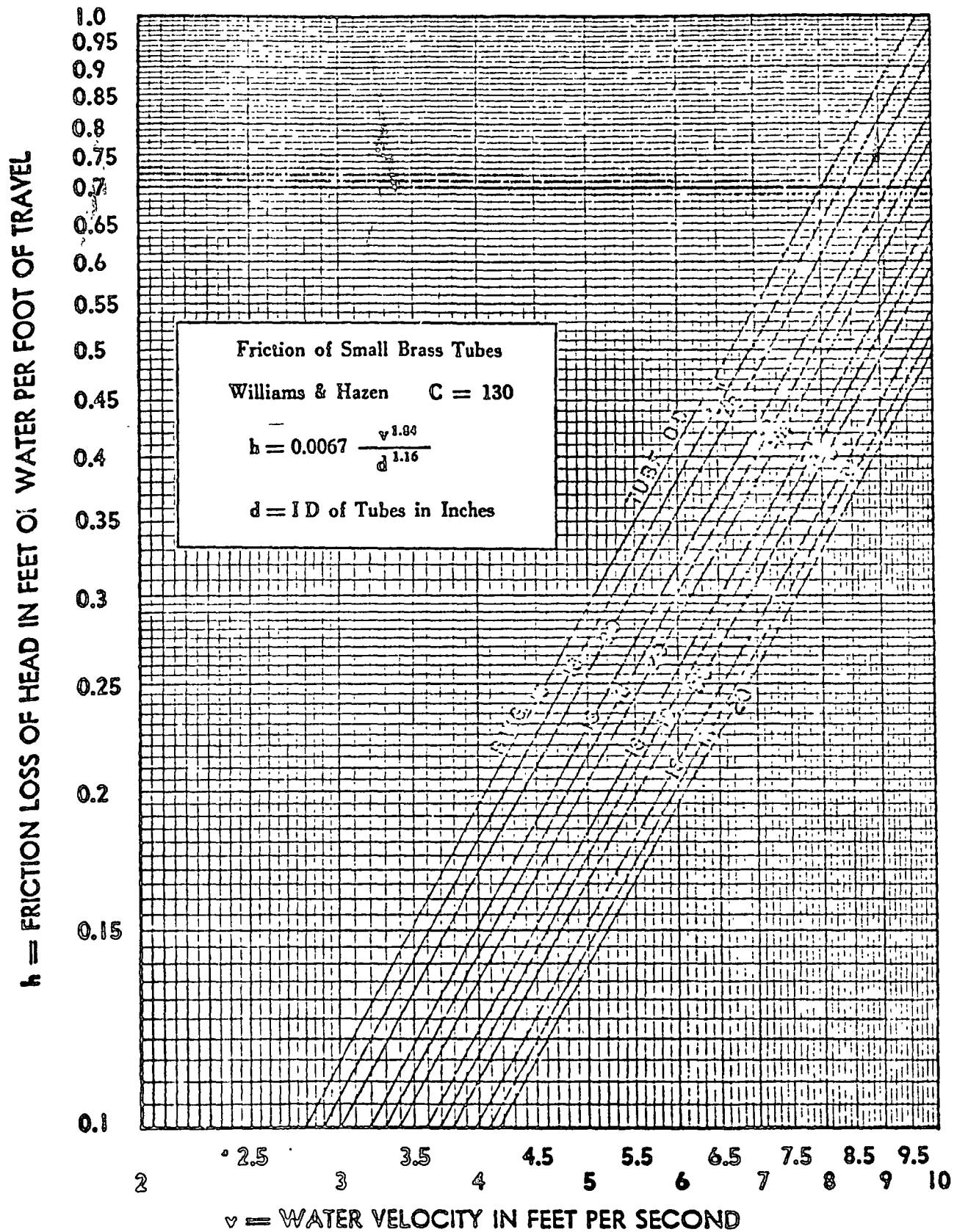
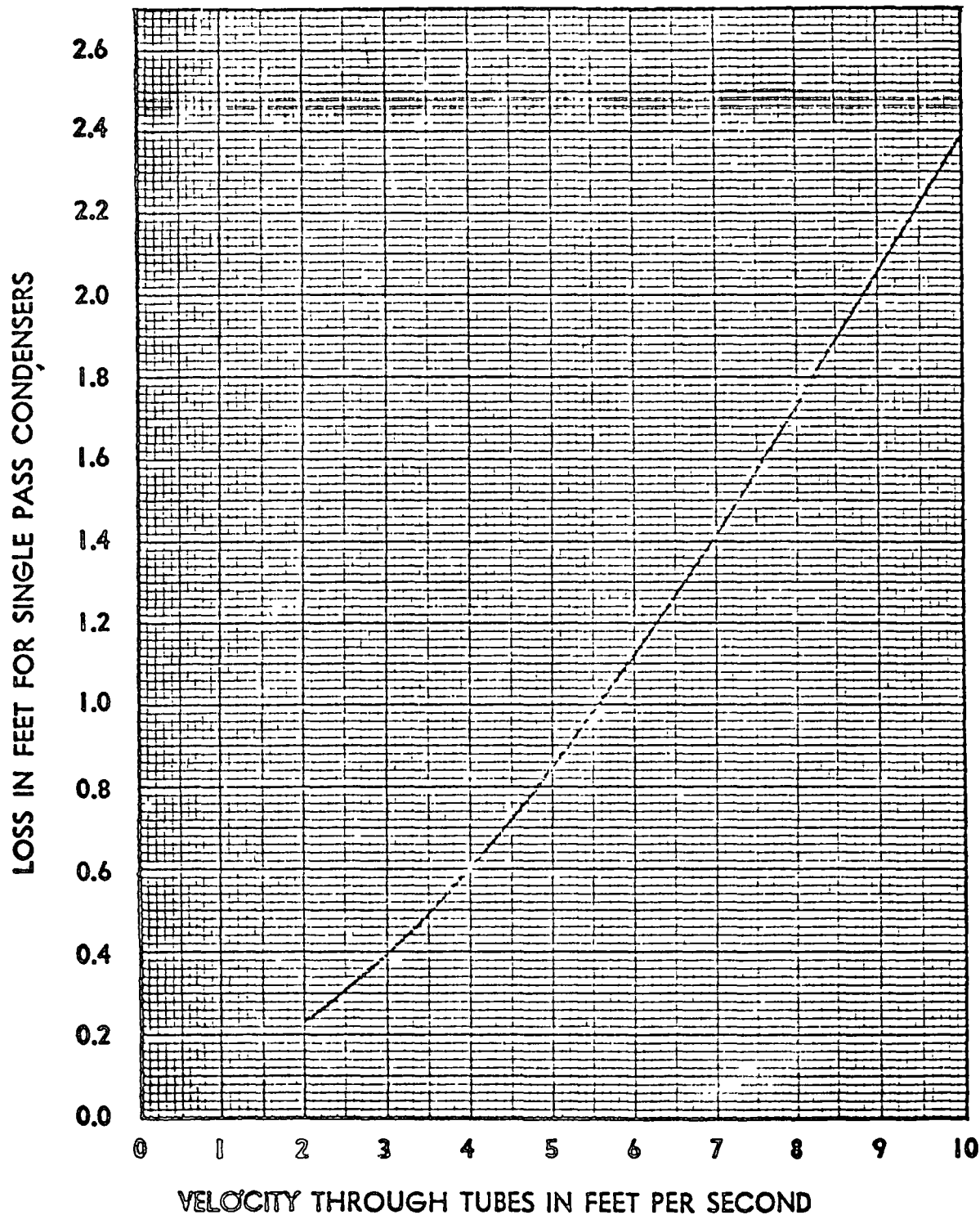


Fig. SF-6

PRESSURE LOSS IN WATER BOXES FOR SINGLE PASS CONDENSERS



Note: For multi-pass condensers the total pressure loss in the water boxes is obtained by multiplying the pressure loss from the curve for single pass condensers by the number of passes.

Fig. SF-7

PRUEBAS DE COMPORTAMIENTO

Los objetivos principales de estas pruebas son los siguientes :

- a) Comprobar las garantías ofrecidas por los fabricantes para su -
corrección ó aplicación de penas en su caso.
- b) Conocer lo que razonablemente puede esperarse del equipo.
- c) Tener referencias en el historial del equipo para pruebas sub-
secuentes y programar futuros mantenimientos.

I.- TIPOS DE PRUEBAS.-

Las determinaciones que se hacen del comportamiento de un con -
densador son principalmente las siguientes :

- 1.- La presión absoluta que puede ser mantenida a una carga tér -
mica dada del condensador y a un flujo y temperatura del agua
de circulación con un determinado factor de limpieza.
- 2.- La transferencia térmica de la superficie del condensador para
las condiciones indicadas en el punto anterior.
- 3.- El sub-enfriamiento del condensado.
- 4.- El porcentaje de Oxígeno disuelto en el condensado

II.- DESCRIPCION Y DEFINICION DE TERMINOS.-

1.- Carga del Condensador.- (q)

La rapidez a la cual el calor se transfiere al agua de enfria-
miento. Normalmente se considera la variable independiente en
cualquier prueba de condensador.

2.- Flujo de vapor.- (w_s)

La cantidad de vapor entrando al condensador del equipo prima-

3.- Presión del Condensador.- (p_s)

La presión absoluta del vapor en el punto donde este entra al condensador.

4.- Vacío en el condensador.- (p_v)

La diferencia entre la presión atmosférica y la presión del vapor donde esta entra al condensador.

5.- Temperatura de entrada de vapor.- (t_s)

La temperatura de saturación correspondiente a la presión del vapor entrando al condensador a menos que el vapor este sobrecalentado.

6.- Temperatura de condensado.- (t_c)

La temperatura de condensado saliendo del aparato de condensación. Donde el pozo caliente es parte integral del condensador, esa temperatura es la lectura del condensado tomado a la salida del cuerpo del condensador y antes de la entrada al almacenamiento del pozo caliente.

7.- Flujo de condensado.- (w_c)

Cantidad de condensado descargado del condensador. Esta puede incluir drenes si se descargan dentro del pozo caliente pero estas adiciones deben evitarse en lo posible.

8.- Entalpia del vapor de entrada.- (h_s)

La entalpia del vapor de entrada al condensador es del equipo primario normalmente obtenido de cálculos basados en el comportamiento del equipo primario.

9.- Entalpía de condensado.- (h_c)

La entalpía del líquido de las tablas de vapor para la temperatura del condensado saliendo del condensador.

10.- Temperatura de entrada del agua de enfriamiento.- (t_1)

La temperatura del agua de enfriamiento entrando al condensador.

11.- Temperatura de salida del agua de enfriamiento.- (t_2)

Temperatura del agua de enfriamiento saliendo del condensador.

12.- Elevación de temperatura.- (Δt_{2-1})

El aumento de temperatura del agua de enfriamiento desde la -- entrada a la salida del condensador.

13.- Calor específico del agua de enfriamiento.- (c_p)

Calor Específico del agua de enfriamiento de acuerdo con la -- temperatura media y con la salinidad medida durante la prueba.

14.- Diferencia inicial de temperatura.- (θ_1)

Diferencia entre la temperatura del vapor y la temperatura de -- entrada del agua de enfriamiento.

15.- Diferencia terminal.- (θ_2)

Diferencia entre la temperatura de vapor y la temperatura de sa-
lida del agua de enfriamiento.

16.- Diferencia media de temperatura.- (θ_m)

La diferencia media logarítmica de temperatura calculada entre -- el vapor y el agua de enfriamiento.

17.- Superficie de condensación.- (A)

El área externa total activa de los tubos del condensador inclu-
yendo los enfriadores de aire si se usan. El área de las placas

de los tubos no deben incluirse. El área activa excluye cualquier tubo que haya sido taponeado durante la prueba.

18.- Transferencia térmica total.- (U)

Es la cantidad de calor transferida por unidad de tiempo, por unidad de superficie por grado de diferencia media logarítmica. En condensadores de superficie esta cantidad es la medición — fundamental en el comportamiento de condensadores.

19.- Transferencia térmica total corregida.- (U')

La transferencia corregida para las condiciones contractuales ó condiciones promedio de prueba de velocidad y temperatura de agua y factor de limpieza.

20.- Transferencia de tubos nuevos.- (U_n)

Transferencia térmica de muestras de tubos nuevos, en la determinación de factor de limpieza.

21.- Transferencia de tubos viejos.- (U_o)

Transferencia térmica de muestras de tubos viejos ó usados, empleados en la determinación de factor de limpieza.

22.- Factor de limpieza.- (C_f)

Relación entre la transferencia térmica de tubos usados y tubos viejos bajo condiciones idénticas de operación.

23.- Flujo de agua de enfriamiento.- (W_w)

Cantidad de agua de enfriamiento pasando a través del condensador por unidad de tiempo.

24.- Densidad del agua de enfriamiento.- (d) .

Densidad del agua de enfriamiento para la temperatura y salinidad medida durante la prueba.

25.- Velocidad del agua.- (V_w)

Promedio de la velocidad del agua calculada a través de los tubos. Para condensadores de varios pasos con diferente número de tubos en los diferentes pasos el -- promedio de las velocidades promedios será el valor que deberá usarse.

26.- Presión del agua de entrada.- (p_1)

Presión estática del agua de enfriamiento cerca de la -- brida de entrada de la caja de agua.

27.- Presión de la salida del agua.- (p_2)

Presión estática del agua de enfriamiento donde es des - cargada toda la salida de la caja de agua.

28.- Carga de fricción del condensador.- (Δp_{1-2})

La pérdida de presión debido a la fricción en el circui- to de agua de enfriamiento entre los puntos donde el agua entra y sale del condensador. Este valor debera incluir la corrección de cualquier diferencia en elevación ó carga de velocidad entre los puntos de medición.

29.- Infiltraciones de aire.- (Q_a)

Es la cantidad de aire libre descargado por el equipo de extracción de aire. Por aire libre se quiere decir, el - aire a la presión atmosférica y temperatura ambiente.

30.- Temperatura del aire en la salida.- (t_a)

salida del cuerpo del condensador.

31.- Presión del aire de salida.- (p_a)

Es la presión de aire y mezcla de vapor en el punto de salida del cuerpo del condensador.

32.- Caída de presión del condensador.- (Δp_{s-a})

Es la pérdida de presión entre la entrada de vapor y el punto de salida de aire.

Además de los puntos anteriores, deberán considerarse algunos otros que corresponden a diferentes equipos que es necesario comprobar durante las pruebas de comportamiento.

III.- PRINCIPIOS GUIAS.-

- 1.- Se deberán tomar acuerdos sobre el límite de tiempo después de su instalación para efectuar las pruebas. La temperatura del agua de enfriamiento a la cual debe efectuarse la prueba, las desviaciones permisibles de flujo y correcciones para dicha desviación y los métodos de operación del condensador y equipo primario durante la prueba.
- 2.- Se recomienda que la prueba no se lleve a cabo hasta que las infiltraciones de aire se retengan a menor de 5 pcm.
- 3.- Las fugas de agua de enfriamiento no deberán ser mayores de los valores siguientes:

Para turbinas menores de	500 KW	1%
	500 a 1000 KW	.75%
	1000 a 5000 KW	.5%
	y arriba de 5000 KW	500 lbs/hr.

- 4.- El condensador deberá encontrarse en condiciones comerciales de operación durante la prueba.
- 5.- Se hará una corrida preliminar con el propósito de -- comprobar todos los instrumentos, entrenamiento de personal y efectuar ajustes menores.
- 6.- Cualquier condición cuya variación pueda afectar los resultados de la prueba deberá corregirse en lo posible y mantenerse constante durante la prueba.
- 7.- Se harán todos los esfuerzos para realizar la prueba - bajo las condiciones especificadas de velocidad del agua, carga, etc. con objeto de reducir al mínimo la aplica - ción de factores de corrección.
- 8.- La duración de la prueba normalmente se considera -- suficiente de 1 hora.
- 9.- Cuando se efectúa una prueba de una hora las temperaturas de agua de enfriamiento se tomarán cada 2 1/2 - minutos y las demás lecturas cada 5 minutos. Si la duración de la prueba es más de una hora la frecuencia de la lectura de la temperatura del agua se reducirá.
- 10.- Cada corrida de prueba se hará de acuerdo con un programa predeterminado.
- 11.- El resultado de las pruebas se reportarán calculadas directamente de las observaciones durante la prueba, las tolerancias aplicadas serán únicamente las establecidas con - tractuamente.

IV.- INSTRUMENTACION.

- 1.- La presión de entrada al condensador deberá medirse con manómetros de Mercurio ó por mutuo acuerdo con manómetros de presión absoluta. Como una regla empírica el número de manómetros usados deberá ser cuando menos 1 por cada 16 pies² de succión de entrada al condensador. La presión deberá medirse cerca ó a un lado adyacente al punto de conexión de la turbina y el condensador.
- 2.- La presión de salida deberá medirse con manómetros de Mercurio ó por mutuo acuerdo con manómetros de presión absoluta. Deberá usarse un manómetro por cada tubería de salida del condensador al equipo de extracción de aire.
- 3.- Para la medición de la presión atmosférica se usaran barómetros de cristal de Mercurio.
- 4.- Los barómetros y manómetros de Mercurio utilizados deberán corregirse por temperatura, longitud de la escala, meniscos ó capilaridad, gravedad y elevación si el manómetro y el barómetro no están a la misma altura.
- 5.- La presión de vapor auxiliares deberá medirse con manómetros tipo Burdón.
- 6.- Las temperaturas del agua de enfriamiento tanto de entrada como de salida deberán medirse con termómetros de cristal de Mercurio.
- 7.- La temperatura del aire de salida y condensado deberá medirse con termómetros de cristal de mercurio.

- termómetros de resistencia.
- 8.- La temperatura de vapor de auxiliares deberá medirse con --
termómetros de cristal de Mercurio ó termopares ó termómetros
de resistencia.
 - 9.- El flujo de condensado deberá medirse por medio de tanques y
escalas de pesos, volumétricas ó tubos Venturi, toberas ó pla -
cas de orificio según acuerdo entre las partes.
 - 10.- El nivel en el pozo caliente deberá observarse y anotarse al -
principio y al final de las pruebas.
 - 11.- La medición de vapor auxiliar puede efectuarse por medio de
tubos Venturi, toberas ó placas de orificio.
 - 12.- La medición del agua de repuesto se efectuará separadamente.
 - 13.- La cantidad del aire extraído del condensado se medirá por -
medio de gasómetros, toberas ó placas de orificio.
 - 14.- La medición de la caída de presión por fricción entre la en -
trada y salida de las cajas de agua se efectuará por medio de
manómetros diferenciales de Mercurio.
 - 15.- Las fugas de agua de enfriamiento se calcularán por medio de
la conductividad electrolítica, titulación con Nitrato de Plata
ó por peso directo.
 - 16.- El tiempo de iniciación y terminación de las pruebas se hará
por medio de relojes programadores, relojes sincronados, ó re -
lojes individuales de los observadores.
 - 17.- El oxígeno disuelto en el condensado se medirá en el lado de
succión de la bomba de condensado.

V.- CALCULOS.-

1.- La carga térmica del condensador (g) deberá calcularse en la siguiente forma:

a.- Cuando no hay repuestos ni drenes.

$$q = W_s (h_s - h_c)$$

b.- Para turbo-generadores con ciclo regenerativo:

$$W_s h_s = W_t (h_t - h_t) + W_s h_e + 0.003 p_f W_t -$$

$$- \frac{3413 P}{\text{Efic. Gen.} \times \text{Efic. Mec.}}$$

$$W_t = W_s + W_b ;$$

c.- Para turbo-generador con recalentamiento y ciclo regenerativo:

$$W_s h_s = W_t (h_t - h_f) + W_r (h_r - h''_r) + W_s h_e + 0.003 P W_t -$$

$$- \frac{3413 P}{\text{Efic. Gen} \times \text{Efic. Mec.}}$$

2.- Flujo de agua de enfriamiento.-

Puede calcularse por medio de un balance térmico igualando el calor cedido por el vapor al calor absorbido por el agua en la siguiente forma:

$$W_s (h_s - h_c) = W_w C_p (t_2 - t_1)$$

de donde:

$$W_w = \frac{W_s (h_s - h_c)}{C_p (t_2 - t_1)} = \frac{8}{C_p (t_2 - t_1)}$$

3.- Velocidad del agua a través de los tubos.-

Puede obtenerse dividiendo el flujo de agua entre el área de las secciones de los tubos en cada paso considerando el promedio de las secciones.

4.- Transferencia térmica total.-

La transferencia térmica total de la superficie del condensador se calculará de acuerdo con la siguiente ecuación que está basada en la diferencia media logarítmica de temperatura.

$$U = \frac{W_w C_p}{A} \ln \frac{\Theta_1}{\Theta_2}$$

5.- Corrección de la transferencia térmica para una velocidad, temperatura y factor de limpieza dados.

La transferencia térmica varía con una potencia fraccionaria de la velocidad del agua.

La corrección por temperatura deberá establecerse por acuerdo entre las partes.

Si ocurre ensuciamiento de tubos durante las pruebas, la transferencia puede corregirse a un factor de limpieza medio multiplicando por la relación del factor de limpieza medio al factor de limpieza de la prueba.

6.- Corrección de la Presión del Condensador para una velocidad del agua, temperatura del agua y factor de limpieza dados.-

Después de corregida la transferencia de calor, se determina la presión del condensador, cuya temperatura de saturación basada en la diferencia media logarítmica de temperatura es la siguiente:

$$T_s = T_1 = \frac{9}{W_w C_p (1 - \frac{1}{Ex})}$$

En donde :

$$X = U'A/W_w C_p$$

e = base logaritmos Neperianos = 2.718

La presión de saturación (P_s) correspondiente a ésta temperatura T_s se obtiene de las tablas de vapor y es la presión cometa de entrada al condensador.

7.- Diferencia media logarítmica de temperatura.

La diferencia media logarítmica de temperatura entre el -- vapor y el agua se obtiene de la siguiente fórmula:

En caso de acuerdo entre las partes puede usarse la diferencia media aritmética de temperaturas, debiendo consignar esto en los reportes de pruebas.

8.- Factor de limpieza.-

El factor de limpieza del condensador se calculará en la siguiente forma:

9.- Corrección del factor de limpieza por variación de -- velocidad.

Interpolando o extrapolando de los datos de prueba, es costumbre considerar que el factor de limpieza varía inversamente proporcional a alguna potencia de la velocidad del agua en los tubos en la siguiente forma:

Los subíndices 1 y 2 se refieren a los correspondientes valo-

- VI.- CARACTERISTICAS Y GARANTIAS CONDENSADOR (C.F.E.)
- VII.- FORMAS RECOMENDADAS POR ASME PARA REPORTAR RESULTADOS DE PRUEBAS.
- VIII.- PROCEDIMIENTO DE PRUEBAS EMPLEADO POR LA PACIFIC GAS - - & ELECTRIC CO. DE SAN FRANCISCO, CALIF.
- IX.- EJEMPLO DE CALCULO DE COMPORTAMIENTO DE UN CONDENSADOR (LOUSIANA POWER & LIGHT CO. UNIDAD DE 210 000 KW).

II.- C.- CARACTERISTICAS Y GARANTIAS CONDENSADOR.

No.	CONCEPTO	CARACTERISTICAS	GARANTIAS
1	Fabricante		
2	Tipo		
3	No de pasos		
4	Almacenamiento pozo caliente		
5	Superficie total		
6	No. de tubos		
7	Longitud tubos		
8	Diámetro tubos		
9	Espesor tubos		
10	Material tubos		
11	Sección/tubo		
12	Forma instalación tubos		
13	Fabricante tubos		
14	Flujo vapor		
15	Flujo no condensables		
16	Carga térmica		
17	Flujo agua circulación		
18	Velocidad agua circulación		
19	Temp. entrada agua circulación		
20	Procedencia ✓ ✓		
21	Contra-presión (vacío)		
22	Coeficiente transmisión de calor		

No.	CONCEPTO	CARACTERISTICAS	GARANTIAS
23	Factor de limpieza		
24	Caída presión (fricción)		
25	Sub-enfriamiento		
26	O ₂ disuelto en enfriamiento		
27			
28			
29			
30			

ASME POWER TEST CODES

Test Code for Steam-Condensing Apparatus

SECTION 8, REPORT OF RESULTS

162 The purpose of this table is to provide a convenient form for reporting results of tests. It is not intended to serve as a guide for determining the readings that should be taken. Some incidental readings which have no particular significance in appraising the apparatus, but which may be necessary for the computations, have been omitted. These data may be included, or other data added, if desired. On the other hand, many of the items will not apply to any one condenser and shall, therefore, be omitted. The table is divided under headings which make it convenient for selecting those groups of items which are applicable. Standardization of table forms is of great convenience to those who study or use reports. It is, therefore, recommended that this table form be used with as few modifications as possible.

General Information

- 1 Date of test.....
- 2 Condenser number (plant designation).....
 - (a) Manufacturer's serial number.....
- 3 Location
- 4 Owner
- 5 Manufacturer
- 6 Condenser serves prime mover. No.....size.....make, etc.
- 7 Object of test.....
- 8 Test conducted by.....

Surface Condenser

Description, Dimensions, etc.

- 9 Nominal size of unit.....sq ft
- 10 Type of condenser (horizontal or vertical).....
 - (a) Number of passes.....
 - (b) Special features
- 11 Distance from exhaust flange of prime mover to inlet flange of condenser.....ft
- 12 Area of steam inlet to condenser.....sq ft
 - (a) Diagram of steam inlet showing location of pressure connections.....
- 13 Elapsed time since condenser was placed in service months
- 14 Elapsed time since last cleaning.....days
 - (a) Method used to clean tubes.....
- 15 Number of tubes in each pass.....
 - (a) Active length of tubes between tube sheets.....ft
 - (b) Outside diameter of tubes.....in.
 - (c) Thickness of tube wall.....Bwg
 - (d) Material of tubes.....
 - (e) Number of inactive tubes, i.e., tubes removed or stoppered at time of test.....
 - (f) General condition of tube surfaces.....
 - (g) Kind of tube packing.....
- 16 Description or diagram of tube sheet layout.....
 - (a) Proportion of tube sheet area occupied by tubes.....per cent
- 17 Proportion of the volume of the shell, below the level of the top row of tubes, occupied by tubes per cent

- 18 Amount of heating surface provided for increasing the temperature of the condensate leaving the unit (state whether internal or external).....sq ft
 19 Amount of cooling surface in air cooler (state whether internal or external).....sq ft
 20 Source of cooling water supply.....
 (a) Kind of water (fresh, salt, or brackish).....
 21 Type of seal on atmospheric relief valve.....

Test Data and Results

- 22 Date and time of run.....
 23 Duration of run.....hr
 24 Condenser load 10⁶ Btu per hr
 25 Quantity of steam entering condenser from prime mover.....lb per hr
 (a) Steam entering condenser from other sources.....lb per hr
 (b) Cooling water leakage into steam space.....lb per hr
 26 Condenser pressure (i.e., pressure of steam at inlet).....in. Hg abs at 32 F
 (a) Saturation temperature corresponding to steam pressure (from steam tables).....F
 27 Enthalpy of steam entering condenser.....Btu per lb
 28 Barometric pressure (corrected for temperature and elevation).....in. Hg abs at 32 F
 (a) Temperature of barometer.....F
 (b) Elevation of barometer with reference to inlet pressure gages.....ft
 29 Pressure at each air removal point in condenser.....in. Hg abs at 32 F
 (a) Temperature of the air and vapor at each air removal point.....F
 (b) Average of the pressures at all air removal points.....in. Hg abs at 32 F
 (c) Average pressure drop through entire tube bank [Item 26-Item 29 (b)].....
in. Hg at 32 F
 30 Quantity of free air removed from condenser (state room temperature and pressure)....cfm
 31 Temperature of condensate leaving condenser.....F
 (a) Temperature leaving external heater supplied with steam from condenser.....F
 32 Quantity of condensate leaving condenser.....lb per hr
 33 Dissolved oxygen content of condensate.....cc per liter or ppm by wt
 34 Quantity of cooling water used.....cfs
 35 Temperature of cooling water entering condenser.....F
 (a) Temperature entering other passes of multipass condensers.....F
 36 Temperature of cooling water leaving condenser.....F
 37 Pressure of cooling water at inlet water box flange.....ft water abs
 38 Pressure of cooling water at outlet water box flange.....ft water abs
 39 Friction head of cooling water circuit in condenser (Item 37-Item 38).....ft water
 40 Salinity of cooling water.....per cent total salts by wt
 41 Condenser leakage per cent
 42 Overall heat transmittance (uncorrected).....Btu per hr per sq ft per F

Sample Tube Test Data

- 43 Locations of sample tubes (mark locations on tube sheet diagram).....
 44 Temperature of cooling water supply to sample tubes (state if water is obtained from the same source as for the entire condenser).....F
 45 Velocity of water through sample tubes.....fps
 46 Heat transmittances and cleanliness factors of sample tubes:

Group designation	A				B				Etc.			
Tube number	1	2	3	4	1	2	3	4	1	2	3	4
(a) U_n												
(b) U_o												
(c) C_r												

- (d) Average heat transmittance of new tubes.....Btu per hr per sq ft per F
 (e) Average heat transmittance of used tubes.....Btu per hr per sq ft per F
 47 Overall cleanliness factor [Item 46(e) ÷ Item 46(d)].....

Test and Guaranteed Values Adjusted for Comparison

48	Condenser load	10 ⁶ Btu per hr
49	Quantity of steam entering condenser (from all sources)	lb per hr
50	Steam condensed per unit area of condensing surface	lb per hr per sq ft
51	Quantity of cooling water passing through condenser	cfs
52	Average velocity of water through tubes	fps
53	Temperature of cooling water entering condenser	F
54	Over-all cleanliness factor	
55	Condenser pressure (i.e., pressure of steam entering):	
	(a) Guarantee	in. Hg abs at 32 F
	(b) Test	in. Hg abs at 32 F
56	Over-all heat transmittance:	
	(a) Guarantee	Btu per hr per sq ft per F
	(b) Test	Btu per hr per sq ft per F
57	Temperature of condensate leaving condensing apparatus:	
	(a) Guarantee	F
	(b) Test	F
58	Dissolved oxygen content of condensate:	
	(a) Guarantee	cc per liter or ppm by wt
	(b) Test	cc per liter or ppm by wt
59	Quantity of "free air" removed from condenser:	
	(a) Maximum allowable amount for which guarantees apply	cfm
	(b) Test	cfm
60	Friction head of cooling water circuit in condenser:	
	(a) Guarantee	ft water
	(b) Test	ft water
61	Power or steam consumption of auxiliaries:	
	(a) Guarantee	kw, ihp or lb steam per hr
	(b) Test	kw, ihp or lb steam per hr

Jet Condenser**Description, Dimension, etc.**

62	Elapsed time since condenser was placed in service	months
63	Type of condenser (low level or barometric)	
	(a) Special features	
64	Type of water distributor (jet or gravity)	
	(a) Number and type of injection nozzles	
65	Approximate volume of space available for condensation	cu ft
	(a) Area of steam inlet to condenser	sq ft
66	Distance from exhaust flange of prime mover to inlet flange of condenser	ft
67	Mechanical condition of apparatus	
68	Source of cooling water supply	
	(a) Kind of water (fresh, salt, brackish)	

Test Data and Results

69	Date and time of run	
70	Duration of run	hr
71	Condenser load	10 ⁶ Btu per hr
72	Quantity of steam entering condenser	lb per hr
73	Condenser pressure:	
	(a) Guarantee	in. Hg abs at 32 F
	(b) Test	in. Hg abs at 32 F
74	Enthalpy of the steam entering condenser	Btu per lb
75	Barometric pressure (corrected for temperature and elevation)	in. Hg abs at 32 F
	(a) Temperature of barometer	F
	(b) Elevation of barometer with reference to condenser pressure gages	ft
76	Quantity of cooling water:	
	(a) Guarantee	cfs
	(b) Test	cfs

- 77 Pressure of cooling water at injection nozzles:
 (a) Guarantee in. Hg abs at 32 F
 (b) Test in. Hg abs at 32 F
- 78 Differential pressure across injection nozzles [Item 77(b) — Item 73(b)]...in. Hg at 32 F
- 79 Temperature of cooling water entering condenser.....F
- 80 Temperature of cooling water and condensate mixture leaving.....F
 (a) Corresponding saturation pressure (from steam tables).....in. Hg abs at 32 F
 (b) Corresponding enthalpy of saturated liquid.....Btu per lb
- 81 Partial pressure of air in condenser:
 (a) Value on which guarantees are based in. Hg at 32 F
 (b) Test value [Item 73(b) — Item 80(a)].....in. Hg at 32 F
- 82 Quantity of air removed from condenser:
 (a) Guarantee cfm
 (b) Test cfm
- 83 Level of water in condenser or discharge pipe (referred to datum plane).....ft
- 84 Power or steam consumption of auxiliaries:
 (a) Guarantee kw, ihp or lb steam per hr
 (b) Test kw, ihp or lb steam per hr

Prime Mover Data for Computing Enthalpy of Steam Entering Condenser

- 85 Load kw or ihp
- 86 Steam pressure at throttle valve.....psia
- 87 Steam temperature at throttle valve.....F
 (a) Moisture content of steam (if in the wet vapor region).....per cent
 (b) Enthalpy of steam at throttle.....Btu per lb
- 88 Steam flow at throttle valve.....lb per hr
- 89 Quantity of steam reheated.....lb per hr
 (a) Pressure leaving turbine.....psia
 (b) Temperature leaving turbine.....F
 (c) Enthalpy leaving turbine.....Btu per lb
 (d) Pressure at reentrance psia
 (e) Temperature at reentrance.....F
 (f) Enthalpy at reentrance.....Btu per lb
- 90 Steam admitted at secondary admission points of mixed-pressure units.....lb per hr
 (a) Pressure psia
 (b) Temperature F
 (c) Moisture content per cent
- 91 Steam extracted at each extraction point.....lb per hr
 (a) Pressure psia
 (b) Temperature F
 (c) Moisture content per cent
 (d) Enthalpy Btu per lb
- 92 Quantity of steam discharged to condenser (same as Item 25).....lb per hr
- 93 Make-up feed added in condenser.....lb per hr
 (a) Feed added between condenser and final regenerative heater.....lb per hr
- 94 Feedwater leaving final regenerative heater:
 (a) Temperature F
 (b) Pressure (state whether nominal or measured).....psia
 (c) Enthalpy Btu per lb
- 95 Increase in enthalpy of the feedwater in final heater.....Btu per lb
- 96 Mechanical efficiency of prime mover.....per cent
- 97 Generator efficiency per cent
- 98 Enthalpy of steam entering condenser (same as Item 27).....Btu per lb

Auxiliary Equipment

Circulating, Removal, or Injection Pump

- 99 Pump No. (plant designation).....
 (a) Manufacturer's serial number.....

100	Manufacturer	
101	Type of pump (reciprocating, centrifugal, impeller, etc.)	
	(a) Number of stages	
	(b) Special features	
102	Rated capacity	cfs
103	Drive (motor, turbine or reciprocating engine)	
	(a) Rating	hp
	(b) Rated speed	rpm
104	Size of inlet and discharge openings (diameter)	in.
105	Mechanical condition of apparatus	
106	Quantity of water handled	cfs
	(a) Specific gravity if water is salty or brackish	
107	Inlet pressure (referred to center line of pump):	
	(a) Static pressure	ft water abs
	(b) Velocity pressure	ft water
	(c) Total pressure	ft water abs
108	Discharge pressure (referred to center line of pump):	
	(a) Static pressure	ft water abs
	(b) Velocity pressure	ft water
	(c) Total pressure	ft water abs
109	Head on pump [Item 108(c) - Item 107(c)]	ft water
110	Speed	rpm or strokes per min
111	Power or steam consumption of driving unit	kw, ihp or lb steam per hr
	(a) Operating pressures and temperatures	

Condensate Pump

112	Pump No. (plant designation)	
	(a) Manufacturer's serial number	
113	Manufacturer	
114	Type of pump (reciprocating, centrifugal, impeller, etc.)	
	(a) Number of stages	
	(b) Special features	
115	Rated capacity	lb per hr or cfs
116	Drive (motor, turbine, or reciprocating engine)	
	(a) Rating	hp
	(b) Rated speed	rpm
117	Size of inlet and discharge openings (diameter)	in.
118	Mechanical condition of apparatus	
119	Quantity of water handled	lb per hr or cfs
120	Height of water in hotwell above centerline of pump	ft
121	Discharge pressure (referred to centerline of pump):	
	(a) Static pressure	ft water abs
	(b) Velocity pressure	ft water
	(c) Total pressure	ft water abs
122	Head on pump	ft water
123	Speed	rpm or strokes per min
124	Power or steam consumption of driving unit	kw, ihp or lb steam per hr
	(a) Operating pressures and temperatures	

Steam-Jet Air Pump

125	Manufacturer	
	(a) Serial number	
126	Type (number of stages, etc.)	
	(a) Special features	
127	Rated capacity (free air)	cfm
	(a) Rated capacity is based on an inlet air and vapor pressure of	in. Hg abs at 32 F
128	Mechanical condition of apparatus	
129	Number of nozzles in each stage	
	(a) Size of nozzles (diameter)	in.

- 130 Amount of cooling surface:
 (a) Precooler sq ft
 (b) Intercooler sq ft
 (c) Aftercooler sq ft
 131 Source of water used in coolers (condensate or raw water).....
 132 Quantity of cooling water passing through air coolers.....cfs
 133 Temperature of cooling water (entering).....F
 134 Steam consumption of pump.....lb per hr
 (a) Initial steam pressure.....psia
 (b) Initial steam temperature F
 (c) Moisture content of steam.....per cent
 135 Inlet air and vapor pressure.....in. Hg abs at 32 F
 136 Inlet air and vapor temperature.....F
 137 Quantity of free air handled (state room temperature and pressure).....cfm

Hydraulic Air Pump

- 138 Manufacturer
 (a) Serial number
 139 Type
 (a) Size of rotating element (diameter).....in.
 (b) Special features
 140 Rated capacity of pump (free air).....cfm
 (a) Rated capacity is based on inlet and vapor pressure of.....in. Hg abs at 32 F
 141 Drive (motor, turbine, or reciprocating steam engine).....
 (a) Rated speed rpm
 142 Mechanical condition of apparatus.....
 143 Quantity of hurling water used.....cfs
 144 Static pressure of hurling water entering pump.....in. Hg abs at 32 F
 145 Temperature of hurling water entering pump.....F
 146 Inlet air and vapor pressure in. Hg abs at 32 F
 147 Inlet air and vapor temperature.....F
 148 Quantity of free air handled (state room temperature and pressure).....cfm
 149 Speed rpm
 150 Power or steam consumption of driving unit.....kw, ihp or lb steam per hr
 (a) Operating pressures and temperatures

Reciprocating Dry Vacuum Pump

- 151 Manufacturer
 (a) Serial number
 152 Type (horizontal or vertical).....
 (a) Number of stages.....
 (b) Size of cylinders (diameter and stroke) in.
 (c) Special features
 153 Rated capacity of pump (free air).....cfm
 (a) Rated capacity is based on an air and vapor pressure of.....in. Hg abs at 32 F
 154 Size of steam cylinders (diameter and stroke).....in.
 155 Mechanical condition of apparatus.....
 156 Inlet air and vapor pressure.....in. Hg abs at 32 F
 157 Inlet air and vapor temperature.....F
 158 Quantity of free air handled (state room temperature and pressure).....cfm
 159 Speed strokes per min
 160 Steam consumption of driving cylinder.....lb per hr
 (a) Operating pressures and temperatures

CONDENSADORES DE VAPOR DE SUPERFICIE.

NOMENCLATURA.

El diagrama mostrado es con el propósito de establecer una terminología standar, y el esquema es meramente ilustrativo para indicar los nombres de las partes.

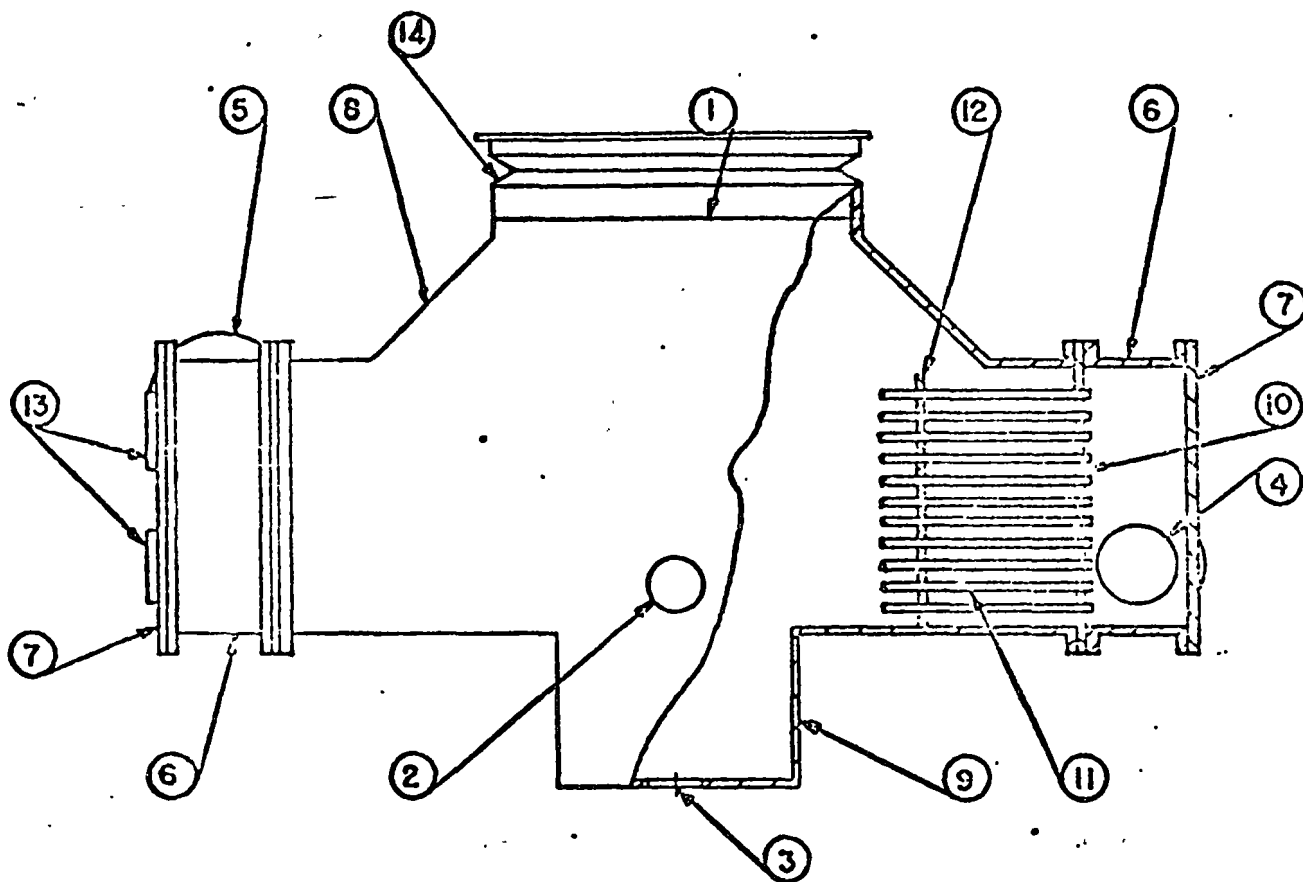


FIG. 1

- 1.- Entrada de vapor ó domo.
- 2.- Extracción de aire ó mezcla aire-vapor.
- 3.- Extracción ó salida de condensado.
- 4.- Entrada de agua de circulación (entrada agua fría)
- 5.- Salida agua de circulación.
- 6.- Cajas de agua.

b.

- 7.- Cubiertas ó tapas cajas de agua.
- 8.- Cubierta ó carcasa del condensador.
- 9.- Pozo caliente.
- 10.-Placas ó espejos de tubos.
- 11.-Tubos.
- 12.-Placas soporte de tubos.
- 13.-Registros de hombre.
- 14.-Junta de expansión.

berá, además, tener conexiones para presión y temperatura con objeto de realizar pruebas de acuerdo con Códigos.

2.-Extracción de aire.-

La extracción de aire y gases incondensables durante los arranques y operación normal se efectúa por medio de inyectores de aire ó bombas de vacío con objeto de evitar la formación de presiones parciales y el aumento de la presión en el condensador.- La extracción del aire y los gases incondensables se hace por medio de tuberías ó ductos colectores situados axialmente a lo largo total de los haces de tubos, instalados en tal forma que no promuevan disturbios del vapor, para éste objeto se instalan también mamparas con lo cual se evita la formación de bolsas de aire. Como durante los arranques es necesario evacuar rápidamente el aire, se utilizan extractores de mayor capacidad que los de operación normal; las capacidades de los evacuadores de aire de arranque y de operación normal se dan en las tablas I y II tomadas del Instituto de Intercambio de Calor (Heat Exchange Institute-HEI). Con objeto de facilitar la extracción del aire, los colectores se colocan en la parte inferior del condensador donde el agua está más fría; para la misma masa de aire se tendrá menor volumen manejado a mayor enfriamiento. En la Fig. 2 se muestra la disposición de un colector de aire instalado en el centro del condensador.

FUNCIONES DE LAS PARTES PRINCIPALES DE QUE CONSTA UN CONDENSADOR

En los condensadores de superficie, el vapor que sale de la turbina es condensado por contacto con la superficie exterior de los haces de tubos por el interior de los cuales circula agua de enfriamiento; el condensado es extraído por medio de una bomba que lo envía al sistema de calentamiento de agua de alimentación a través de calentadores cerrados y al deaerador de donde lo toma la bomba de agua de alimentación para enviarlo a la caldera y de aquí sale en forma de vapor a la turbina que después de efectuar su trabajo sale al condensador para repetirse el ciclo.

Mientras se condensa el vapor, el aire y los gases incondensables son extraídos continuamente del condensador por medio de inyectorres ó bombas de vacío, para evitar la formación de presiones parciales que aumenten la presión en el condensador.

Con esta breve introducción se iniciará a continuación la descripción de las funciones de cada una de las partes principales de que consta un condensador.

1.- Entrada de vapor ó domo.-

El objetivo principal del domo es el de recibir el vapor de la turbina y llevarlo hacia los haces de tubo para su condensación. Sus dimensiones quedan fijadas por el flujo ó masa de vapor, las pérdidas por fricción y por las características de la cimentación de la turbina. De

6.-Cajas de agua.-

El objetivo de las cajas de agua es el de recibir, dirigir y distribuir el agua de circulación ó de enfriamiento manteniendo al mínimo las pérdidas de presión y evitando disturbios en el flujo para lo cual se le da la forma y dimensiones adecuadas y complementándose con mamparas para cumplir sus objetivos ; adicionalmente se instalan venteos para expulsión del aire durante el llenado del sistema de agua de circulación en los arranques.

Normalmente las cajas de agua están divididas en tal forma que pueda seccionarse, mediante el cierre de válvulas, una parte para limpieza ó mantenimiento durante la operación de la unidad a carga reducida ó bien tener un sistema para retrolavado ; no deberán formarse cámaras de aire ni turbulencia en las esquinas para lo cual es necesario que tengan profundidad y forma adecuada.

7.-Cubiertas o tapas de agua.-

Son las cubiertas o tapas exteriores a las cajas de agua transversales a los tubos del condensador, van sujetas a las cajas con tornillos y se encuentran seccionadas para poderse retirar durante la operación a cargas reducidas para limpieza ó mantenimiento. Deberá proveerse de los medios adecuados para el manejo de éstas tapas.

3.-Extracción ó salida de condensado.-

La extracción de condensado se efectúa por medio de bombas cuya succión se conecta al fondo ó pozo caliente del condensador por medio de una tubería; normalmente se tienen para este objeto 2 bombas de 100% de capacidad con sus válvulas de aislamiento y filtros individuales.

4.-Entrada de agua de circulación (entrada agua fría).-

La entrada de agua de circulación ó entrada de agua fría se hace a un lado y por la parte inferior del condensador instalándose juntas de expansión para absorber las dilataciones y esfuerzos; las válvulas son del tipo mariposa y normalmente motorizadas por tratarse de diámetros grandes. Deberá tener conexiones para medición de presión y temperatura del agua.

5.-Salida agua de circulación.-

La salida del agua de circulación se hace por la parte superior del condensador; en los de simple paso la salida es por el lado opuesto a la entrada y en los de doble paso es por el mismo lado. También se instalan juntas de expansión para absorber dilataciones diferenciales térmicas y esfuerzos, y las válvulas son del tipo mariposa y normalmente motorizadas por tratarse de diámetros grandes, además de conexiones para medición de presión y temperatura.

normalmente no se instala en el cuerpo ó carcasa del condensador, si sirve para evitar que haya sobrepresiones en su interior por lo que es necesario mencionarla en este punto.

9.-Pozo caliente.-

El pozo caliente se encuentra en la parte del fondo del condensador y es donde se colecta el condensado y de donde lo succionan las bombas; normalmente se instalan uno ó varios indicadores y controladores de nivel para mantener y controlar el nivel del agua a un valor constante; normalmente se instala un filtro a la salida del pozo para evitar arrastre de materias extrañas al agua condensada.

El pozo es un depósito importante de agua teniendo volumen para almacenar agua a la máxima carga durante un período de 1 a 5 minutos. La velocidad de salida del agua se diseña para un máximo de 1.2 m/seg. considerando el NPHJ^S de las bombas de condensado.

10.-Placas o espejos de los tubos.-

Las placas ó espejos de los tubos son los extremos en donde van soportados y sujetos estos, se construyen normalmente de un material similar al de los tubos y de un espesor de 22.2 a 38.1 mm.

8.-Cubierta ó ~~carcaza~~ del condensador.-

La cubierta ó carcaza del condensador es una estructura totalmente construída de acero y suficientemente fuerte para soportar los tubos, todos los accesorios internos, la presión exterior producida por el vacío interior y para prevenir distorsiones durante el transporte.

A la cubierta ó carcaza del condensador se tienen varias conexiones, siendo éstas las siguientes principalmente:

- a.-Los drenes de condensado de los calentadores.
- b.-Los venteos de los calentadores.
- c.-La llegada del agua de repuesto del ciclo.
- d.-Purgas de algunas líneas de vapor.
- e.-Purga del condensador.
- f.-Dren de los sellos de vapor.
- g.-Deaereación de la succión de las bombas de extracción de condensado.
- h.-Drenes de los inyectores de aire.
- i.-Recirculación para asegurar flujo mínimo a inyectores durante los arranques.
- j.-Conexiones para instrumentos (indicadores de nivel, controladores de nivel, indicadores de presión, alarma alto nivel, etc.)
- k.-Deflectores a la llegada de drenes y purgas que pueden ser por medio de cabezales ó en forma individual.
- l.-Pozos para medición de temperatura.

Aún cuando la válvula de seguridad por disco de ruptura

- 0 -

registros de hombre tanto en las tapas de las cajas de agua como en la cubierta ó cuerpo del condensador; estos registros deben ir en lugares accesibles que no haya obstrucciones para el retiro de sus tapas y en ambos extremos y secciones de las tapas de agua así como en la cubierta del condensador.

14.-Junta de expansión.-

Debido a que durante los arranques, paro y operación de las turbinas se tienen dilataciones y contracciones, y el condensador se conecta directamente a la turbina, es necesario proporcionar un medio para que el condensador las absorba; en los condensadores grandes se emplea normalmente una junta de expansión cerca de la entrada del vapor y en condensadores pequeños se soportan estos sobre resortes.

11.-Tubos.-

Los tubos son el medio para el paso por su interior del agua de enfriamiento a una velocidad de unos 2.0 m/seg., estando limitada ésta agua por las pérdidas de carga (normalmente de 0.3 Kg/cm² para un solo paso), limitar la erosión de los tubos y el material de estos; el recorrido del agua puede ser de 1, 2 ó 3 pasos.

Los tubos empleados son de un diámetro de 15.9 a 25.4 mm. y longitudes variables hasta unos 20 m.; el lado de entrada de agua se mandrila acampanándolo con objeto de facilitar la entrada del agua y el lado de salida puede ser montado con empaques, soldados ó mandrilados.

El material de los tubos se selecciona de acuerdo a las características del agua y a la velocidad de ésta.

12.-Placas soporte de tubos.-

Debido a que se emplean tubos muy largos, es necesario soportarlos de tramo en tramo para evitar que se flexionen, -- pero permitiéndoles su dilatación y contracción; el material puede ser bronce para cojinetes y el espesor igual al diámetro de los tubos.- El espacio entre soportes se selecciona en tal forma de evitar vibraciones alejándose de la frecuencia natural de vibración de la máquina ó turbina y de la vibración causada por el flujo de vapor al haz de tubos.

13.-Registros de hombre.-

Con objeto de entrar al interior para inspección se instalan

TABLA I

CAPACIDADES DE BOMBAS DE VACÍO

Máximo Vapor Condensado en lbs/hora	pies ³ /min. a 70°F de aire seco libre	
	Turbinas	Maquinas
Hasta 25 000	3.0	6.0
De 25 001 a 50 000	4.0	8.0
De 50 001 a 100 000	5.0	10.0
De 100 001 a 250 000	7.5	15.0
De 250 001 a 500 000	10.0	—
Mas de 500 001	12.5	—

TABLA II

CAPACIDADES DE BOMBAS DE VACÍO DE ARRANQUE

Máximo Vapor Condensado en lbs/hora	pies ³ /min a 70°F de aire libre seco a 15" de Hg de Presion Absoluta
Hasta 75 000	150
De 75 000 a 250 000	300
De 250 001 a 600 000	600
Mas de 600 000	900

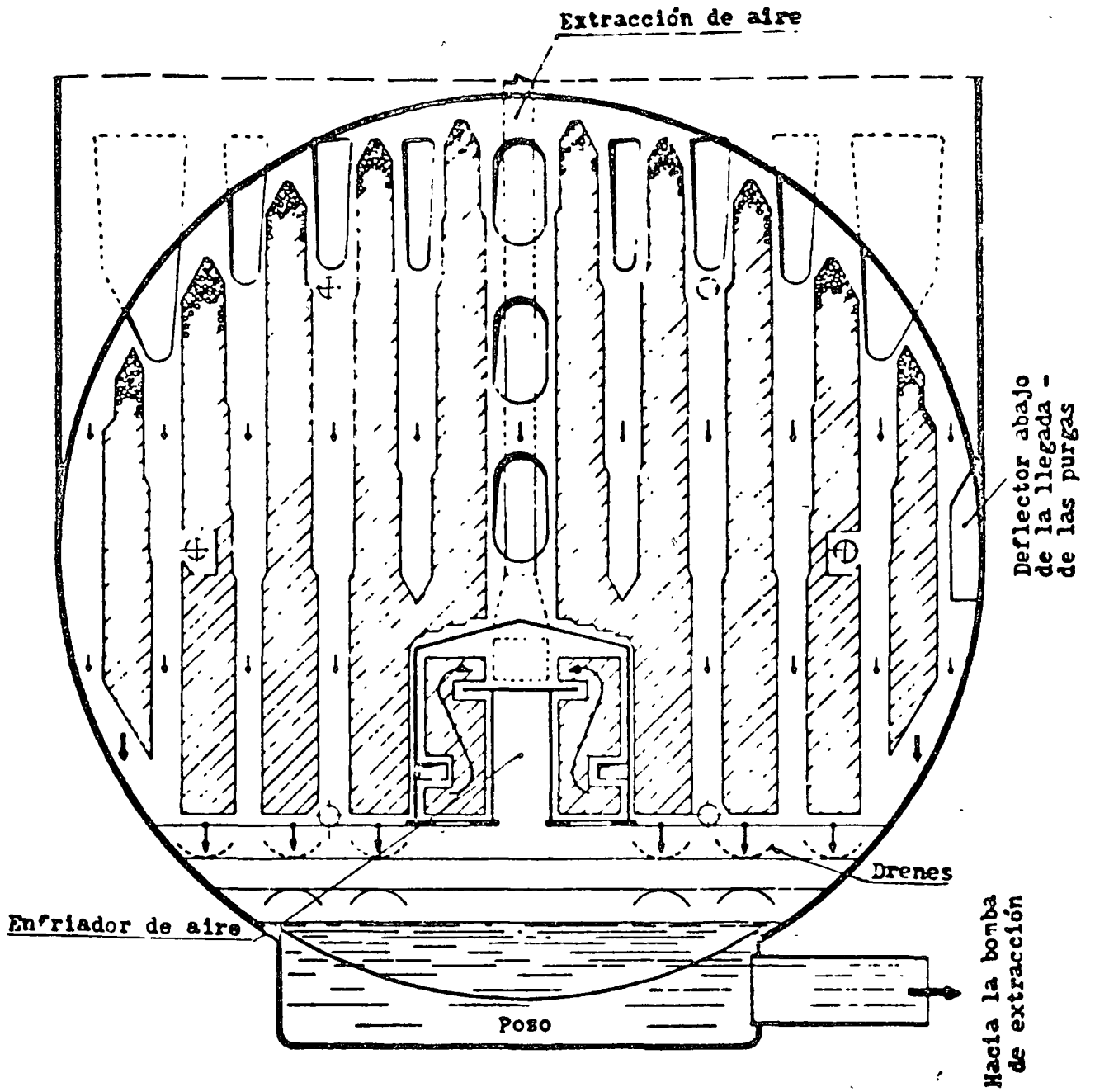
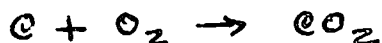


FIG. 1a

NOTAS SOBRE EL DISEÑO TERMICO DEL NUCLEO DE UN REACTOR NUCLEAR

Para comparar los órdenes de magnitud en la generación de calor, en combustibles convencionales y combustibles nucleares, examinemos los dos tipos de reacciones siguientes:

La reacción química de un átomo de carbón con dos átomos de oxígeno



libera 4 e.v en forma de calor

$$1 \text{ e.v} = 1.519 \times 10^{-22} \text{ Btu} = 4.44 \times 10^{-6} \text{ kw hr}$$

Una de las reacciones de fisión posibles en un reactor es:



y si en ella, hacemos un balance de masa entre los productos

y reactivos, vemos que ha "desaparecido" 0.208 a.m.u

(unidades atómicas de masa)

$$1 \text{ a.m.u} = 1.66 \times 10^{-24} \text{ gr}$$

y por la relación

$$\Delta E = \Delta m c^2$$

se deduce

$$1 \text{ a.m.u} = 9.31 \times 10^8 \text{ e.v.}$$

Por lo anterior, la energía liberada en esta reacción nuclear es de

$$0.2088 \times 9.31 \times 10^8 = 1.936 \times 10^8 \text{ e.v}$$

Para que se produzca una fisión, es necesario que un núcleo fisionable (tal como U^{235} , Pu^{239} o U^{233}) absorba un neutrón. Por lo tanto, la rapidez con que se producen las fisiones, depende de la población de neutrones (flujo de neutrones) y de un parámetro (sección recta macroscópica), que está relacionado con la probabilidad de que se produzca la absorción de neutrones.

$$\phi \text{ (flujo de neutrones) neutrones/cm}^2 \text{ seg}$$

$$\Sigma \text{ (sección macroscópica) } 1/\text{cm}$$

La rapidez de una reacción nuclear es

$$R = \Sigma \phi \text{ reacciones/cm}^3 \text{ seg}$$

En cada fisión se libera cierta cantidad de calor, que llamaremos G , cuyo orden de magnitud es la del ejemplo anterior, pero cuya determinación mas exacta involucra otras consideraciones. Por lo tanto, el calor que se genera por unidad de volumen y unidad de tiempo, en un medio donde esas reacciones son posibles, tal como un reactor nuclear, es

$$q''' = G \Sigma \phi \quad \dots\dots 1$$

ϕ es del orden de 10^6 neutrones/cm² seg para reactores de entrenamiento y de 10^{15} para reactores de potencia e investigación. Sin embargo, se puede decir que el nivel de ϕ no está limitado por razones nucleares, sino por razones térmicas.

La parte ocupada por el material fisiónable se denomina núcleo, y está constituido por elementos combustibles. Estos elementos los podemos considerar como una "fuente de calor", semejante a una resistencia eléctrica. Ellos son de dimensiones laterales pequeñas y demasiado largos y están formados por el combustible propiamente dicho (tal como uranio metálico o una aleación del mismo) y una camisa o cladding que impide que salgan los productos de fisión.

A diferencia de una resistencia eléctrica, en donde el calor generado por unidad de volumen es constante, en un elemento combustible esto no sucede, pues él depende de ϕ como hemos visto y ϕ varía (suponiendo un núcleo cilíndrico) con Z y r .

Sin embargo, dadas las características geométricas de los elementos combustibles, para uno de ellos sólo es significativa su dependencia . Esta dependencia se puede expresar - aproximadamente como sigue

$$\phi = \frac{H^2}{2} \frac{d^2 \phi}{dz^2} \quad \dots \quad 2$$

esto se muestra en la siguiente figura:

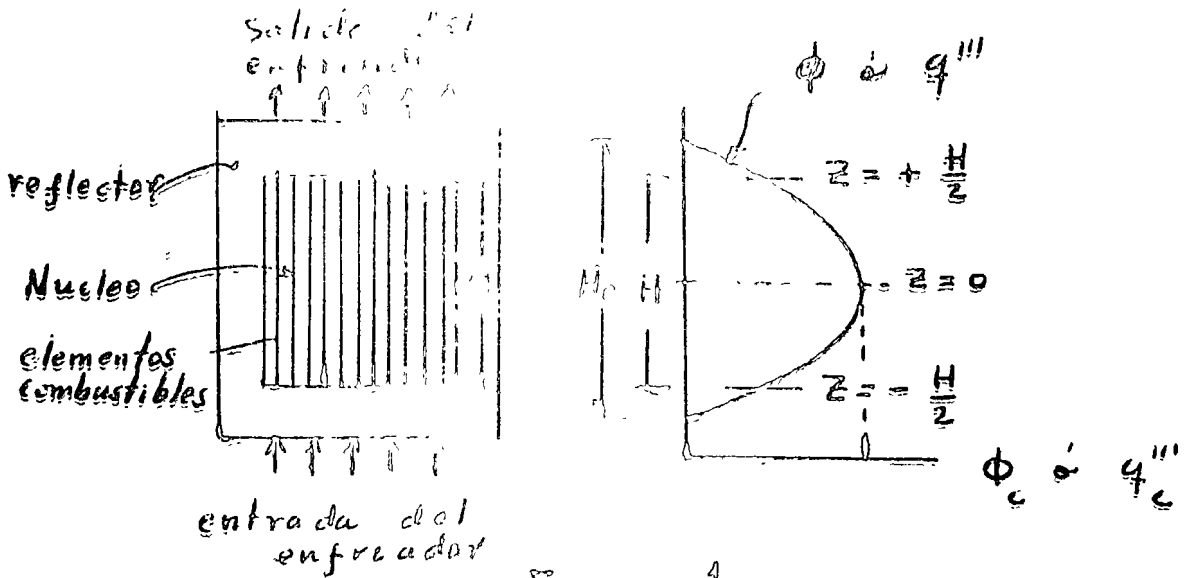


Figura 3

donde:

H es la altura del núcleo

$\pm H/2$ son los puntos donde ϕ es cero

ϕ_c es el valor máximo de ϕ el cual se produce en $z = 0$

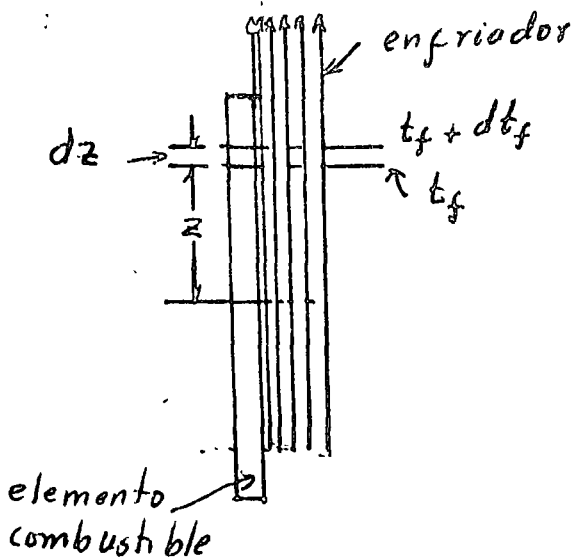
Por la ecuación 1, vemos que el calor generado por unidad de volumen y unidad de tiempo, tendría la misma forma que la de ϕ , esto es:

$$q''' = \frac{P}{H} \cos \frac{\pi z}{H} \quad \dots \quad 3$$

q_c''' es el valor máximo de q''' el cual se produce en $z = 0$

El calor generado en el combustible será removido por el en-

friador que fluye paralelo al mismo, como se muestra en la figura 2



Si consideramos un balance de calor en un elemento del elemento combustible, como se muestra en la figura 2, tendremos

$$\dot{m} c_p dt_f = q''' A_c dz \quad \dots 4$$

en donde:

Figura 2

$\dot{m}$ flujo de masa del fluido enfriador

c_p calor específico del fluido enfriador

dt_f elevación de la temperatura del fluido enfriador entre z y $z + dz$

A_c sección recta de un elemento combustible

Si sustituimos la ecuación 3 en la 4 e integramos entre

$z = -H/2$ y z tenemos

$$\dot{m} c_p \int_{t_{f1}}^{t_f} dt_f = q'''_c A_c \int_{-H/2}^z \cos \frac{\pi z}{H_c} dz$$

donde t_{f1} y t_f son las temperaturas del enfriador en

$z = -H/2$ y z . Por lo tanto

$$t_f = t_{f_1} + \frac{q_c''' A_c H_e}{\pi C_p \dot{m}} \left(\sin \frac{\pi z}{H_e} + \sin \frac{\pi H}{2 H_e} \right) \dots\dots 5$$

La temperatura de salida del "canal" se obtiene si hacemos

$z = H/2$, esto es:

$$t_{f_2} = t_{f_1} + \frac{2 q_c''' A_c H_e}{\pi C_p \dot{m}} \sin \frac{\pi H}{2 H_e} \dots\dots 6$$

Es de interés conocer la temperatura exterior del cladding

t_c . Para ello tenemos en cuenta que en el punto z , el calor transferido a través de $\mathcal{C} dz$, donde $\mathcal{C}$ es el perímetro del elemento combustible, está dado por la ecuación

$$h \mathcal{C} dz (t_c - t_f) = q_c''' A_c dz$$

donde h es el coeficiente convectivo el cual supondremos constante. Bajo esta suposición vemos que q_c''' es proporcional a $t_c - t_f$, por lo que podemos escribir

$$t_c - t_f = (t_c - t_f)_c \cos \frac{\pi z}{H_e} \dots\dots 7$$

pero en el centro del combustible tenemos:

$$h \mathcal{C} (t_c - t_f)_c = q_c''' A_c \dots\dots 8$$

Combinando las ecuaciones 5, 7 y 8 tenemos:

$$t_c = t_{f_1} + q_c''' A_c \left[\frac{H_e}{\pi C_p \dot{m}} \left(\sin \frac{\pi z}{H_e} + \sin \frac{\pi H}{2 H_e} \right) + \frac{1}{h \mathcal{C}} \cos \frac{\pi z}{H_e} \right] \dots\dots 9$$

z_c es positivo y por lo tanto localizado arriba del punto medio del núcleo.

Procediendo en forma análoga podemos obtener z_s y z_m , puntos donde se producen los valores máximos de t_s (temperatura de la superficie del combustible) y t_m (temperatura del centro del combustible) respectivamente. Por ejemplo la expresión de t_m es:

$$t_m = t_f + \frac{q_c''' A_c H_e}{\pi C_p \dot{m}} \left(\sin \frac{\pi z}{H_e} + \sin \frac{\pi H}{2 H_e} \right) + \frac{q_c''' R^2}{4 k_f} \cos \frac{\pi z}{H_e} + \frac{q_c''' R^2}{2} \left[\frac{1}{k_c} \ln \frac{R+c}{R} + \frac{1}{h(R+c)} \right] \cos \frac{\pi z}{H_e} \dots 11$$

$$z_m = \frac{H_e}{\dot{m}} \tan^{-1} \frac{H_e}{C_p \dot{m} \left\{ \frac{1}{4 k_f} + \frac{1}{2} \left[\frac{1}{k_c} \ln \frac{R+c}{R} + \frac{1}{h(R+c)} \right] \right\}} \dots 12$$

donde k_f y k_c son las conductividades térmicas de combustible y cladding respectivamente.

La dependencia de t_f , t_c , t_s y t_m se muestran en la figura 4

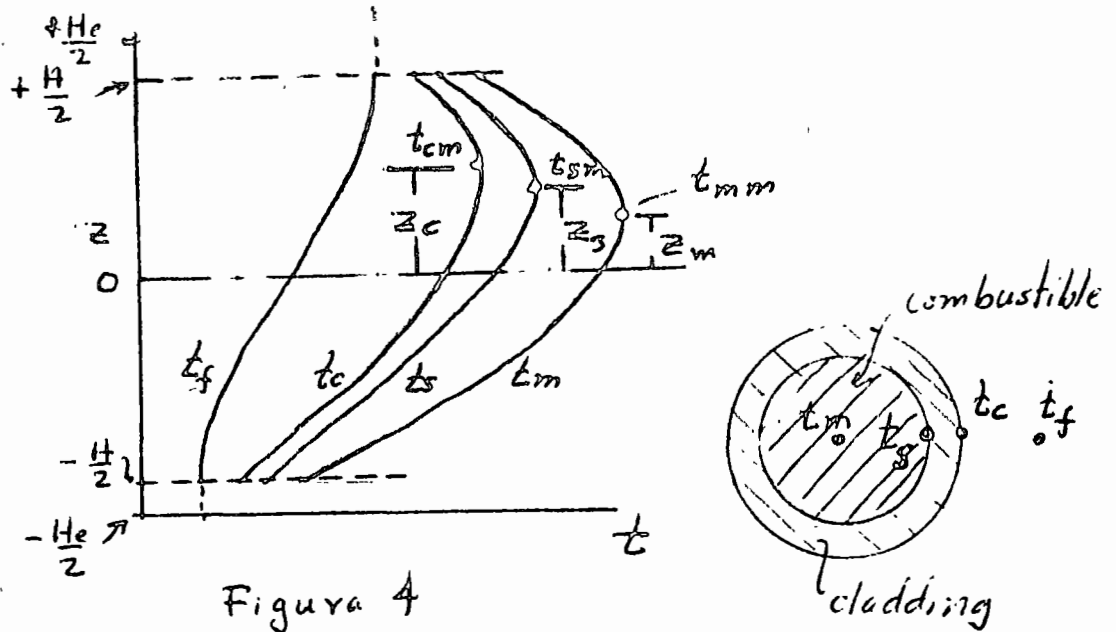


Figura 4

La ecuación 11, puede utilizarse para determinar la rapidez con que se extrae el calor en el núcleo de un reactor y es equivalente a la ecuación familiar

$$Q = UA \Delta t \quad \text{-----} \quad 13$$

Q rapidez de remoción de calor

U coeficiente de transferencia de calor total

A área a través de la cual el calor se transfiere

Δt diferencia de temperatura entre el generador de calor y el enfriador

En nuestro caso vemos que q_c''' depende de k_f, k_c, h

A (área lateral del combustible) y $t_m - t_f$.

Analicemos cada uno de estos parámetros.

k_f depende del tipo de combustible y de su temperatu-

ra. Existen suficientes datos para los diferentes combustibles usados a las temperaturas que se emplean.

k_c su valor debe ser grande, exigiéndose además que el cladding tenga bajo poder de absorción de neutrones, resista altas temperaturas y resista la acción corrosiva del fluido enfriador. Los materiales que se usan actualmente como cladding son acero inoxidable, aluminio, zirconio y magnesio. El espesor del cladding que suministra la máxima transferencia de calor es

$$e_o = R + \frac{k_c}{h}$$

h Depende de las características físicas del enfriador y de sus condiciones de operación, como es su velocidad. Existen diferentes correlaciones para determinar h , según el tipo de enfriador y la geometría del núcleo. Para un núcleo cilíndrico con elementos cilíndricos, la correlación que se recomienda es:

$$Nu = C Re^{0.8} Pr^{1/3} \dots 14$$

donde

$$Nu = \text{Número de Nusselt} = h D_e / k$$

$$Re = \text{Numero de Reynolds} = D_e \bar{V} \rho / \mu$$

$$Pr = \text{Número de Prandtl} = c_p \mu / k$$

De diámetro equivalente

C es una constante que depende del arreglo de los elementos y toma los siguientes valores

$$C = 0.042 \frac{S}{D} - 0.024$$

para un arreglo cuadrado con

$$1.1 \leq \frac{S}{D} \leq 1.3$$

$$C = 0.026 \frac{S}{D} - 0.006$$

para un arreglo triangular con

$$1.1 \leq \frac{S}{D} \leq 1.5$$

S es la distancia entre los centros de los elementos combustibles

D el diámetro de los elementos.

A Mas que el área lateral de cada elemento combustible, importa el área total que arrojan todos ellos, pues mientras más grande sea ella mas calor se puede extraer del núcleo. Esta área total depende en gran parte del tamaño del núcleo. El tamaño de los

núcleos de los reactores son variables, existiendo algunos de miles de pies cúbicos y otros de decenas. Es obvio que se debe desarrollar gran ingenio para obtener una área total de transferencia grande en núcleos pequeños.

Por ejemplo el reactor de ~~c~~ria enfriado por sodio denominado Enrico Fermi, tiene un volumen de 11.65 pies cúbicos y produce 268 Mw(t), por lo que su densidad de potencia es de 23 Mw(t) por pie cúbico; su área de transferencia es 1378 pies cuadrados y su sección recta total por donde fluye el enfriador es de 1.99 pies cuadrados, llevando una velocidad de 31.2 pies por segundo. En cambio los reactores enfriados por gas tienen una densidad de potencia del orden de .010 Mw(t) por pie cúbico, por lo que un reactor de este tipo, con la potencia del Enrico Fermi tendría un volumen de 2680 pies cúbicos

$t_m - t_f$ debe ser grande. Para ello existen dos posibilidades.

La primera es reducir t_f , pero ello ocasiona que disminuya la eficiencia del ciclo de potencia que genera la energía eléctrica, pues el enfriador del núcleo cede el calor que remueve de éste, para generar vapor a través de un generador de vapor.

(Reactores de agua presurizados). Por lo tanto no conviene disminuir t_f

La segunda es aumentar t_m . Sin embargo este aumento está limitado por razones metalúrgicas y de "burnup". Por ejemplo, una razón metalúrgica que impide elevar la temperatura del U metálico arriba de $1235^{\circ}F$, es que esta temperatura cambia al metal de la fase α a la β con incremento de su volumen; sin embargo existe una razón de "burnup" para que la temperatura de este combustible no se eleve a mas de $750^{\circ}F$ y ella consiste en que a esta temperatura disminuye la resistencia del combustible, permitiendo que los gases de fisión se difundan y formen cavidades microscópicas dentro del mismo combustible; a medida que se aumenta el consumo de U^{235} ("burnup") estas cavidades aumentan pudiendo ocasionar fallas en el combustible.

Con el uso de UO_2 y UC se pueden tener temperaturas de $4500^{\circ}F$ para el combustible, pasando a ser el punto de fusión del cladding, el determinante para definir el valor máximo de t_m .

En resumen, aquí como en otros campos de la ingeniería hay conflicto, en este caso el conflicto radica entre los requisitos nucleares, metalúrgicos, estructurales, termodinámicos y de transferencia de calor.

Pasemos a otro problema, si examinamos la ecuación 6 notamos que t_{f_2} es función de q_c''' , el cual varia de un elemento combustible a otro de acuerdo con su situación en el núcleo. Para un núcleo cilíndrico q_c''' es función de r y una aproximación adecuada es la siguiente

$$q_c''' = q_{c_0}''' J_0 \left(\frac{2.405}{R_e} r \right) \dots\dots 15$$

donde q_{c_0}''' es el valor de q_c''' en el centro del núcleo. Ver figura 5

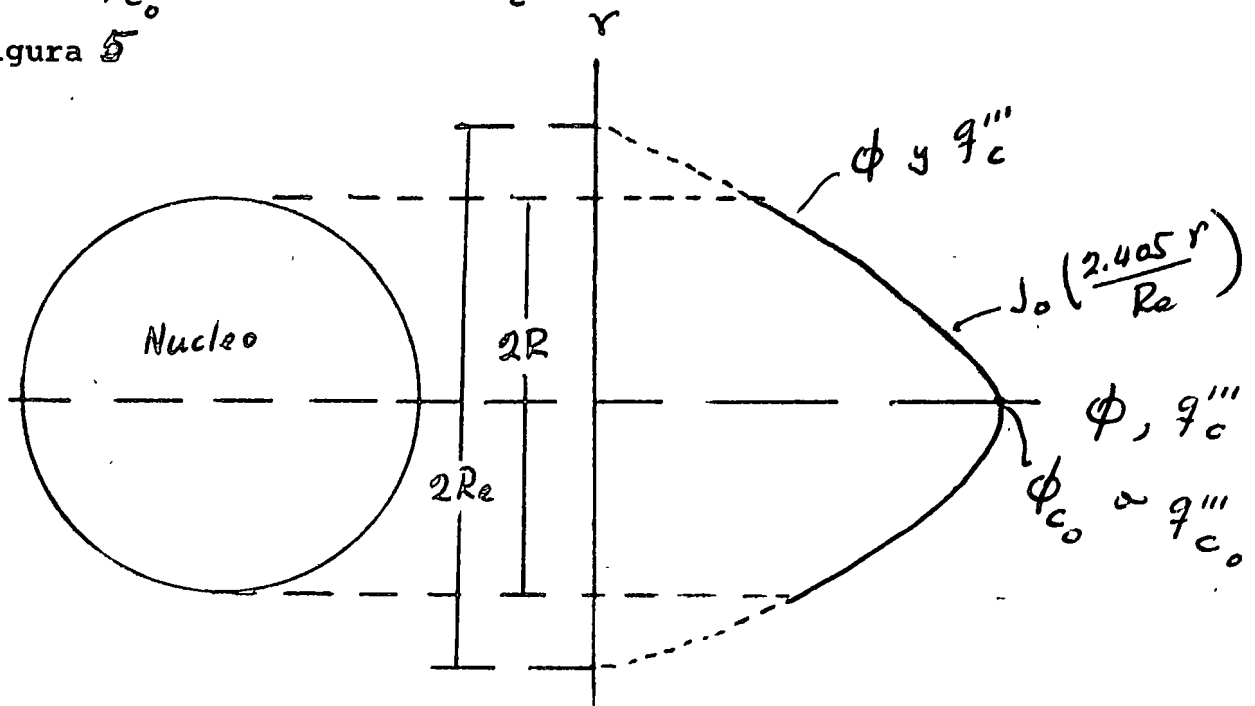


Figura 5.

Esta situación ocasiona que, si $\dot{m}$ y h son constantes en todo el núcleo, el canal central genere alrededor del 10 % mas de calor que el canal alejado $0.25 R$ del centro.

Bajo estas condiciones y considerando t_f , igual para todos los canales, el valor máximo de t_{mm} se producirá en el elemento central del núcleo pudiendo excederse, este elemento, del límite de seguridad, mientras que los elementos periféricos

pueden tener valores máximos de t_{mm} , muy por abajo de aquel límite. Sumado a lo anterior, también tendremos que t_{f2} disminuirá de canal en canal con el radio.

Para remediar lo anterior los núcleos se diseñan de tal suerte que exista una reducción gradual, del centro a la periferie, del flujo del enfriador. Esta reducción se logra aumentando la caída de presión o disminuyendo la sección recta de los canales.

La reducción gradual de $\dot{m}$ puede tener dos objetivos igualar t_{f2} para todos los canales o igualar el valor máximo de t_{mm} para todos los elementos combustibles.

Para el primer objetivo se hace

$$\dot{m} = \dot{m}_0 \int_0 \left(\frac{2.405 r}{Re} \right) \quad \text{--- 16}$$

donde $\dot{m}_0$ y $\dot{m}$ son los flujos del enfriador del canal central y a una distancia r respectivamente. Con las ecuaciones 15 y 16 se tiene

$$\frac{q_c'''}{\dot{m}} = cte \quad \text{--- 17}$$

y con la ecuación 6 se ve que t_{f2} es constante para todos los canales. Sin embargo, esta distribución del flujo del refrigerante hace que t_c disminuya con el radio (lo mismo sucederá con t_m). Esto se debe a que h , según la ecuación 14, depende de $Re^{0.8}$, por lo que podemos escribir

$$h = \frac{\dot{m}^{0.8}}{a} \quad \text{--- 18}$$

donde a es una constante; teniendo en cuenta las relaciones 17 y 18 en la ecuación 9 tenemos:

$$t_c = t_f + \frac{q_c''' A_c}{\dot{m}} \left[\frac{H_c}{\pi c_p} \left(\sin \frac{\pi z}{H_c} + \sin \frac{\pi H}{2H_c} \right) + \frac{a \dot{m}^{0.2}}{c} \cos \frac{\pi z}{H_c} \right] \dots 19$$

lo que nos muestra que todo permanece constante, a excepción del segundo término del paréntesis rectangular y es éste término, el que ocasiona que t_c disminuya con r , aun cuando mas lentamente que en ausencia de reducción gradual de $\dot{m}$

Para lograr el segundo objetivo, esto es para que el máximo de t_m sea el mismo en todos los elementos combustibles, se procede a definir este máximo en el elemento central con ayuda de las ecuaciones 11 y 12. Una vez definido este máximo, se sigue un procedimiento de tanteos, para cada conjunto de elementos colocados en la misma posición respecto al centro, como sigue

- i) Se supone una $\dot{m}^0$
- ii) Se calcula h con la ecuación 18
- iii) Se calcula Z_m con la ecuación 12
- iv) Se calcula t_{mm} con la ecuación 11 en la cual q_c''' se ha substituido por $q_{c0}''' \int_0 \left(\frac{2.405 r}{R_e} \right)$

Si no coincide con el valor fijado se supone otra

También siguiendo un procedimiento análogo, se puede uniformizar la calidad del vapor a la salida de los diferentes canales en el núcleo de un reactor de ebullición.

En seguida nos referiremos a los coeficientes que se usan, en el diseño del núcleo térmico de un reactor, con objeto de hacer seguro su funcionamiento.

Cuando se construye una planta nuclear, obviamente no resulta exactamente con las especificaciones de diseño. Por ejemplo las dimensiones de los elementos y espesor del cladding; el enriquecimiento del combustible; la caída de presión y sección recta de los canales; el flujo de neutrones etc., tendrán variaciones, pequeñas si se quiere, de las especificadas en el diseño.

Estas desviaciones ocurrirán en ambos sentidos de los valores especificados y sólo nos interesan aquellas desviaciones que ocasionen que las temperaturas (del cladding y combustible principalmente) excedan su valor nominal. Los factores de diseño relacionados con estas desviaciones se denominan factores "hot-Spot"

El significado de ellos lo analizaremos a través de un ejemplo.

Supongamos que un núcleo ha sido diseñado para que todos los canales tengan una sección recta nominal de $A_n = 2 \text{ cm}^2$ y que consideramos aceptable una desviación de 0.015 cm^2

Como primer paso, determinemos la probabilidad de que A , sección recta de un canal cualquiera, no exceda el valor A_h donde

$$A'_h = 1.015 A_n$$

Para esto ajustamos la distribución de las medidas de las a una distribución normal o Gaussiana. Para ello medimos secciones rectas y calculamos la desviación standard S con la expresión

$$S = \left[\frac{\sum (A_i - \frac{\sum A_i}{n})^2}{n-1} \right]^{0.5}$$

$\frac{\sum A_i}{n}$ no es necesariamente igual a A aún cuando sus valores son muy próximos si la muestra estadística es grande.

Supongamos para nuestro caso que

$$S = 0.01 \text{ cm}^2$$

La distribución normal está dada por

$$g(A) = \frac{1}{S\sqrt{2\pi}} e^{-\frac{1}{2} \left(\frac{A - A_n}{S} \right)^2} \dots \dots 21$$

y cuya representación gráfica se muestra en la figura 6

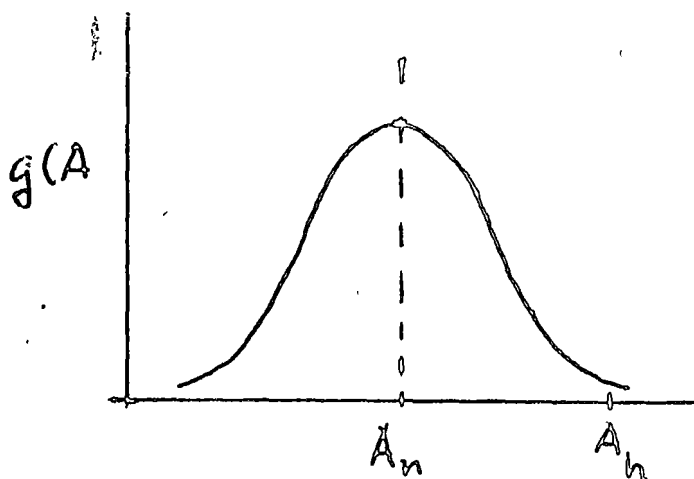


Fig 6

La probabilidad que A exceda a A_h está dada por

$$P(A > A_h) = \int_{A_h}^{\infty} g(A) dA$$

Si definimos la desviación standard adimensional y los factores adimensionales f y R como:

$$\sigma = \frac{S}{A_n}, \quad f = \frac{A_h}{A_n} \quad \text{y} \quad R = \frac{A}{A_n}$$

se puede demostrar que

$$P(A > A_h) = P(R > f)$$

El factor f usualmente se iguala a una función lineal de la desviación standard σ , como

$$f = 1 + B\sigma$$

donde B es una constante, frecuente^{mente}, pero no ^enecesariamente, entera.

Para nuestro caso

$$f = 1.015$$

$$\sigma = \frac{.01}{2} = .005$$

$$B = \frac{f-1}{\sigma} = 3$$

B es el número de desviaciones standard σ y $P(A > A_h)$ es la probabilidad de que A sea mayor que A_h . Su representación geométrica se muestra en la figura 7

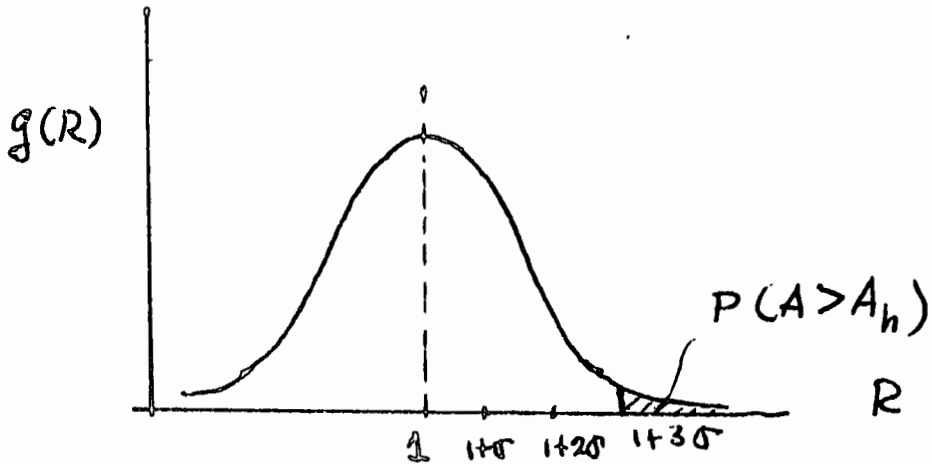


Figura 7

Los valores de $P(A > A_h)$ se encuentran tabulados en la tabla siguiente:

No. de Desviaciones Std. B	Probabilidad $P(A > A_h)$	Falla probable en 1000	% de Límite de confiabilidad
0	.500000	500.000	50.0000
0.5	.308538	308.538	69.1462
1.0	.158665	158.655	84.1345

3.0	.001350	1.35	99.8650

Para nuestro ejemplo, el número de desviaciones standard es 3 por lo que la probabilidad según la tabla es .00135 lo que significa que hay posiblemente 1.35 A'' s en cada 1000 que exceden de $A_h = 1.015 A_n$, lo que corresponde a un límite de confiabilidad de 99.865% equivalente a 3 desviaciones standard.

El coeficiente

$$f = \frac{A_n}{A_h} = 1.015 \quad \text{(para nuestro caso) pues se trata de reducción}$$

representa un subfactor hot-spot de la sección recta de los canales, el cual origina dos factores hot-spot, como veremos mas adelante.

En la fórmula de Darcy podemos ver que si la sección recta

A se reduce, permaneciendo todo constante, a excepción de la velocidad, entonces la velocidad y por lo tanto $\dot{m}$ también se reducen y como q_c'' permanece constante tenemos que debido a la reducción de A se pueden ocasionar las dos elevaciones de temperaturas siguientes:

- 1) De la ecuación 5 y suponiendo que para una Z dada se produce una reducción en área, vemos que $\Delta t_c(z) = t_f - t_{f,i}$,

crece y por lo tanto

$$(t_f - t_{f,i})_h > (t_f - t_{f,i})_n$$

$$\Delta t_{c,h}(z) > \Delta t_{c,n}(z) \quad \dots \dots \dots 25$$

Donde la h y n indican hot-spot y nominal respectivamente.

Como $t_{f,}$ no se altera, entonces

$$t_{f,h}(z) > t_{f,n}(z)$$

Para calcular el factor hot-spot de la temperatura del enfriador a partir de subfactor hot-spot de las secciones rectas de los canales, procedemos como sigue. De la fórmula de Darcy

$$\Delta p = f \frac{H f V^2}{D_e 2 g_c} \quad \dots 27$$

f = factor de fricción

H = altura del canal

D_e = diámetro equivalente

V = velocidad del enfriador,

vemos que los parámetros afectados por la desviación en el área del flujo son V , D_e y f , los demás permanecen constantes. Por lo tanto

$$\frac{f V^2}{D_e} = cte$$

y comparando el caso nominal con el hot-spot tenemos

$$\frac{V_h}{V_n} = \left(\frac{f_n}{f_h} \right)^{0.5} \left(\frac{D_{e,h}}{D_{e,n}} \right)^{0.5} \quad \dots 28$$

Se sabe que para flujo turbulento y tubos circulares

$$f \propto (D_e V)^{-0.2}$$

Tomando esta aproximación tenemos

$$\frac{V_h}{V_n} = \left(\frac{D_{e,h}}{D_{e,n}} \right)^{0.5} \left(\frac{D_{e,h} V_h}{D_{e,n} V_n} \right)^{0.1}$$

$$\text{ó} \quad \frac{V_h}{V_n} = \left(\frac{D_{e,h}}{D_{e,n}} \right)^{\frac{2}{3}} \quad \text{--- 29}$$

Como $D_e = \frac{4A}{P}$ y como P no cambia entonces

$$D_e \propto A$$

$$\text{y} \quad \frac{V_h}{V_n} = \left(\frac{A_h}{A_n} \right)^{\frac{2}{3}} \quad \text{--- 30}$$

Si en la ecuación 5 sustituimos $\dot{m} = \rho A V$ vemos que para una z dada y bajo estas condiciones, todo es constante a excepción de t_f , A y V . Por lo tanto

$$(t_f - t_{f,i}) A V = cte$$

$$\text{ó} \quad \Delta t_c(z) A V = cte$$

y comparando el caso nominal con el hot-spot tenemos:

$$\frac{\Delta t_{c,h}(z)}{\Delta t_{c,n}(z)} = \frac{A_n V_n}{A_h V_h} = \left(\frac{A_n}{A_h} \right)^{\frac{5}{3}} \quad \text{--- 31}$$

Para nuestro caso el factor hot-spot para la elevación de la temperatura del enfriador, debido al subfactor hot-spot de

la desviación de la sección recta de los canales, es

$$\frac{\Delta t_{c,h}}{\Delta t_{c,n}} = f_c = (1.015)^{1.667} \dots 32$$

- 2) Hemos visto que la reducción de la sección recta de los canales, ocasiona disminución en $\dot{m}$ y V . Veamos ahora como afecta esto la temperatura del cladding.

La ecuación 9 puede escribirse en la siguiente forma

$$t_c - t_f = \frac{q_c''' A_c}{h C} \cos \frac{\pi z}{H_e}$$

Para una z dada los parámetros afectados por la disminución de $\dot{m}$ son t_c , t_f y h . Por lo que podemos escribir

$$(t_c - t_f) h = \text{cte}$$

$$\Delta t_f(z) h = \text{cte}$$

y por lo tanto

$$\frac{\Delta t_{f,h}}{\Delta t_{f,n}} = \frac{h_n}{h_h}$$

De la ecuación 14 y de la definición de Nu podemos escribir

$$h \propto \frac{V^{0.8} D_e^{0.8}}{D_e} = \frac{V^{0.8}}{D_e^{0.2}}$$

Substituyendo en la expresión anterior y teniendo en cuenta que $D \propto A$ y la ecuación 30, tenemos:

$$\begin{aligned} \frac{\Delta t_{f,h}}{\Delta t_{f,n}} &= \left(\frac{V_n}{V_h} \right)^{0,8} \left(\frac{D_{e,h}}{D_{e,n}} \right)^{0,2} \\ &= \left(\frac{D_{e,n}}{D_{e,h}} \right)^{\frac{1,6}{3}} \left(\frac{D_{e,h}}{D_{e,n}} \right)^{0,2} \\ &= \left(\frac{D_{e,n}}{D_{e,h}} \right)^{\frac{1}{3}} = \left(\frac{A_n}{A_h} \right)^{\frac{1}{3}} \end{aligned}$$

Por lo tanto el factor hot-spot para la elevación de la temperatura a través de la película del cladding f_f , debido al subfactor hot-spot de la desviación de la sección recta de los canales es

$$\frac{\Delta t_{f,h}}{\Delta t_{f,n}} = f_f = (1.015)^{\frac{1}{3}}$$

Vemos que t_c incrementa debido a dos factores hot-spot, ocasionados por la misma incertidumbre física.