THE NEW HEAT TRANSFER

VOLUME 3

EQUIPMENT DESIGN AND ANALYSIS

the ventuno press

NEW ENGINEERING SERIES

Adiutori	The New Heat Transfer	Vol 1	1974
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EQUIPMENT DESIGN AND ANALYSIS

eugene f. adiutori
stability consultants

the ventuno press
Box 40321
Cincinnati, Ohio 45240

PREFACE

This third and final volume of TNHT has a highly specific purpose: to describe and illustrate the solution of equipment design/analysis problems using the concepts/methods of the new heat flow described in Volume 1. Toward this end, the book includes a number of equipment design/analysis problems and their new way solutions. The problems are practical and fundamental--practical in that they are problems encountered every day by equipment designers/analysts, fundamental in that they serve to illustrate the design/analysis of a broad spectrum of heat flow equipment.

It is desirable but not necessary to have read Volume 1 before reading Volume 3. The only prerequisite for reading this book is an understanding of the following simple ideas on which the new way solutions are based:

> If at all possible, problems should be solved with the primary variables SEPARATED. (Farewell to the ratio $q/\Delta T$ (alias h) which makes it IDENTICALLY IMPOSSIBLE to separate q and ΔT.)
>
> "Heat" does not disappear.
>
> The total temperature difference is the sum of the individual temperature differences.

Chapter 1 presents a summary of the "new engineering" and illustrates that the new heat flow is merely one branch of an altogether new engineering science which is just over the horizon. This new engineering bears little resemblance to today's engineering for two reasons: because it abandons many of the old parameters such as electrical resistance, fluid friction factor, heat transfer coefficient, elastic modulus, etc; because it is so much simpler and more logical. The remainder of the book deals specifically with the new heat flow. Chapter 2 describes how to translate old heat transfer correlations to the language of the new heat flow. Chapter 3 deals with equipment design/analysis problems involving the simplest possible thermal behavior--proportional. This is the only type of thermal behavior which can be dealt with effectively using the old heat transfer, and it will be noticed that there is little practical difference between the new and the old way solution of problems of this type. The reason for the small difference is that problems involving only proportional behavior are so simple that it makes little practical difference whether or not the solutions are based on separated variables.

Chapters 4 through 6 deal with equipment design/analysis problems involving nonlinear thermal behavior. Problems of this type are solved quite simply using the new heat flow, yet they are virtually impossible to solve using the old heat transfer. The reason for the large difference is that nonlinear behavior is virtually impossible to deal with UNLESS one separates the variables--ie unless one separates q and ΔT by abandoning h. In particular, it should be noted that the new way solution of the nonlinear problems in Chs 4 through 6 is EXACTLY THE SAME as the new way solution of the simple, proportional problems in Ch 3!!!

This book completes The New Heat Transfer, but it is not the end--it is merely the end of the beginning--the beginning of the new heat flow--the beginning of the new engineering--the beginning of an exciting and rewarding period for all those who love science.

5/29/76

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FOR SCIENCE

NOMENCLATURE

The nomenclature in this book is somewhat different from that of the old engineering--not for the sake of being different, but for the sake of being better. Better from an engineering standpoint.

The symbols in this book denote both the parameters AND their dimensions. For example, the symbol "q" denotes the heat flux in B/hr ft^2, and the expression

$$q = 147000 \qquad (1)$$

indicates that the heat flux is 147000 B/hr ft^2. We do NOT write q = 147000 B/hr.ft^2 because the dimensions would be redundant.

The symbol → indicates that there is numerical identity but not necessarily dimensional identity. For example, the symbol ΔT stands for the temperature difference in degrees F, and the relation

$$q \rightarrow 3.74(\Delta T)^{.75} \qquad (2)$$

states that the heat flux in B/hr ft^2 is numerically equal to 3.74 times the temperature difference in degrees F raised to the .75 power. (In the language of the old engineering, relation 2 would be written with an equals sign and the equation would be called a "dimensional equation", indicating that the pure constant 3.74 magically possesses whatever dimensions are required in order to make the equation dimensionally consistent. As we discussed in Book 2, "dimensional consistency" is a contrived, unnecessary invention and we abandon it in the new engineering.) Except in Chapter 1, the equals sign denotes both numerical and dimensional identity.

The symbol { } is a shorthand way of writing "is a function of". For example, q{H,ΔT} is to be read "q is a function of H and ΔT" and is written in place of the unspecified function.

The symbols used are:

a	a constant
A	Area, ft^2
b	a constant
c	a constant
C	specific heat, B/# F
D	diameter, ft
E	potential difference, volts
f	friction factor
F	constant of proportionality between q and dT/dx; used ONLY for materials which behave in such a way that q is proportional to dT/dx
F	factor used in heat exchanger design to "correct" the log mean temperature difference based on counterflow
g	gravity constant
G	mass flow rate, #/hr ft^2
h	heat transfer coefficient, B/hr ft^2 F
H	enthalpy, B/#
I	electric current, amps
k	thermal conductivity, B/hr ft F
L	length, ft
LMTD	log mean temperature difference
M	constant of proportionality between q and ΔT; used ONLY when thermal behavior is such that q is proportional to ΔT
M	ratio of slot mass flow rate to mainstream mass flow rate
NTU	Number of Transfer Units
Nu	Nusselt number
P	pressure, psia
P	power, watts
Pr	Prandtl number
Q	heat flow rate, B/hr
r	radius, ft
R	electrical resistance, ohms
Re	Reynolds number
s	slot height, ft
St	Stanton number
t	thickness, ft
T	temperature, F
TDF	Thermal Driving Force
V	velocity, ft/hr
V	voltage
U	overall heat transfer coefficient, B/hr ft^2 F
W	flow rate, #/hr
W'	flow rate per unit slot length, #/hr ft
x	distance, ft
α	a dimensional constant
β	temperature coefficient of volume expansion, 1/F
Δ	difference between values
ε	radiative emissivity
ε	temperature coefficient of electrical resistivity, 1/F
ε	heat exchanger effectiveness
ε	strain
η	film cooling effectiveness
μ	viscosity, #/hr ft
ρ	density, $\#/ft^3$
ρ	electrical resistivity, ohms per foot cube
σ	stress, $\#/ft^2$
σ	Stefan-Boltzmann constant

<u>Subscripts</u>

A	fluid A
B	fluid B
b	boiling interface
bl	boiling liquid
bo	botherm (a hypothetical fluid)
c	cold fluid
c	coolant
c	condensing interface
cond	conduction

conv	convection
f	fouling
g	gas or mainstream
h	hot fluid
hi	high value
i	interface
i	inside diameter
i	ith
in	into the interface
in	inlet end of node or exchanger
ins	insulation
L	liquid
L	left side of node
lam	laminar
lo	low value
max	maximum
mid	midpoint of node
min	minimum
n	number
n	nth
o	outside diameter
o	reference value
OA	overall
out	out of interface
out	outlet end of node or exchanger
radn	radiation
s	shell side
s	single equivalent
s	slot
sat	saturated
sh	superheat
ss	steady-state
t	total
t	tube side
trans	transmittance
V	vapor
w	wall
xs	cross section

CHAPTER 1 UNDERSTANDING ENGINEERING PHENOMENA

BOOK 3

This final volume of The New Heat Transfer describes how to design and analyze engineering equipment using the new engineering science presented in Book 1. The new engineering is conceptually very simple, and this simplicity makes possible a major improvement in equipment design/analysis because it permits us to abandon many of the old engineering parameters without adding any new ones. Although this may seem anomalous, the explanation is quite simple--we abandon only the old parameters which are unnecessary. And since they are unnecessary, there is not the slightest need to replace them with anything. In short, the design and analysis of equipment using the new engineering requires NO new parameters--it requires merely that the unnecessary old parameters be abandoned and that the remaining parameters be used in a new way.

For example, in the old electrical engineering, the first thing we learn is how to design and analyze simple electrical circuits using the concepts of potential, current, and resistance--ie using the parameters volts, amps, and ohms. But in the new engineering, we recognize that the concept of resistance is altogether superfluous and therefore we abandon resistance in general and electrical resistance in particular. In the new electrical engineering, the first thing we learn is how to design and analyze simple electrical circuits using ONLY volts and amps. In other words, we abandon the unnecessary old parameter of electrical resistance and we use the old parameters volts and amps in a new way.

WHY abandon many of the old engineering parameters simply because they are unnecessary? WHY revolutionize the method of equipment design/analysis simply because the old method is based on parameters which are unnecessary? Because engineering is a science and the proper goal of every science is simplicity. Unnecessary parameters do not simplify--they only complicate and confuse--and that is sufficient reason to reject anything in science. At best, the unnecessary old parameters force us to think and to solve design/analysis problems in a roundabout way. At worst, they make it impossible to think clearly or to solve certain design/analysis problems of practical importance. (Electrical resistance is an "at best" example; thermal resistance is an "at worst" example.)

The book begins with an in-depth discussion of the old concept of resistance because it is the most widely used unnecessary concept of the old engineering. The resistance concept is used in dealing with all types of flow phenomena and in designing all types of flow equipment. The resistance concept results in numerous resistance parameters--static electrical resistance, dynamic electrical resistance, thermal resistance, hydraulic resistance, diffusion resistance, etc. Since electrical resistance is the most widely used resistance parameter, our discussion of resistance centers mainly on electrical resistance and the design/analysis of simple electrical circuits. However, it is important to note that the purpose of this discussion is to bring about the realization that the resistance concept is GENERALLY unnecessary/undesirable and that ALL resistance parameters should be abandoned.

A considerable portion of this chapter is devoted to a discussion of design and analysis in the new engineering in general. This general discussion is intended to: emphasize that the new heat flow is merely one branch of the new engineering; present the new heat flow within the proper framework of the new engineering; show that the new heat flow fits logically into the framework of the new engineering; make this book a useful pilot for similar books on the other branches of engineering.

CONCEPTS

The first requirement of any engineering science is that it must be based on simple concepts which enable us to understand and to describe engineering phenomena. Without this understanding, it would be altogether impossible to design or analyze engineering equipment. For example, 150 years ago, it was impossible to design or analyze even a simple electrical circuit because no one knew how volts and amps were related. Then, in 1827, Ohm made the important discovery that the potential difference across wires was generally proportional to the electric current through them, at least for most common materials. Within the framework of the old engineering and its concept of resistance, this EMPIRICALLY OBSERVED proportionality suggested the so-called "Ohm's Law"

$$E = I R \qquad (1)$$

volts = amps x ohms

Ohm's Law and its electrical resistance parameter made it possible to understand and to quantitatively describe the behavior of the proportional electrical components which were important in Ohm's time. (Nonlinear electrical components did not arrive on the scene until much later.) This new understanding made it possible for the first time to effectively design and analyze proportional electric circuits, and this was a giant step forward for science in general and for engineering in particular.

When Ohm published eq 1, it was vigorously attacked by his contemporaries in spite of the fact that it was an empirical result and therefore was not open to question. Equation 1 was not the result of theory/supposition/guesswork--it was the result of exhaustive experiment and careful analysis which demonstrated that volts and amps were generally proportional. Given this empirically observed proportionality, the next problem was how to describe it in a simple and useful way. Ohm CHOSE to describe the proportionality between volts and amps in terms of the resistance concept. That is why Ohm's Law takes the particular form of eq 1 and why the design and analysis of proportional electric circuits has been based on electrical "resistance" for 150 years.

But suppose we ignore Ohm's choice of the resistance concept. Suppose we go back 150 years in order to independently answer the question which confronted Ohm:

> What is the simplest and most useful way to describe the empirical observation that volts and amps are proportional?

The correct answer to this question is to simply write

$$E \propto I \tag{2}$$

Relation 2 could be used as the fundamental relation of the new electrical engineering IF electrical components were generally proportional (as Ohm believed). Is rel 2 sufficient to replace Ohm's Law and its contrived parameter, electric resistance? YES!! EVERY design/analysis problem that can be solved with eq 1 and electric resistance can be solved just as readily (and in fact more readily) using rel 2.

But there is a problem with both relations 1 and 2. The problem is that neither relation is generally applicable because all electrical components do NOT behave in a proportional way. And how do we handle this problem in the old engineering? We handle it by setting up two sub-sciences--one to deal with proportional components, the other to deal with nonlinear components. First we learn about volts and amps and how to deal with them using Ohm's Law and the electric resistance parameter. And then when we have mastered

this method of dealing with volts and amps, we learn that it is not generally applicable and that it must be used ONLY when dealing with proportional components. If we wish to deal with nonlinear components, we must learn an altogether different method of dealing with volts and amps. And this different method is so different that it has NOTHING TO DO with Ohm's Law--NOTHING TO DO with "Ohm's Law resistance". This different method is based on dealing directly with volts and amps and with a different type of "resistance".

How do we handle this problem in the new engineering? We recognize that rel 2 is not a fundamental relation--it is merely a phenomenological description of the behavior of certain electrical components. The fundamental relation/concept of the new electrical engineering describes the behavior of ALL electrical components, and this relation is

$$E = f\{I\} \tag{3}$$

$$\text{volts} = f\{\text{amps}\}$$

Relation 3 states that the potential difference across an electrical component is a function of the current through the component. It is the SINGLE relation/concept which enables us to deal with ALL electrical components IN THE SAME SIMPLE WAY--ie it is the single basis for designing/analyzing circuits involving BOTH proportional components AND nonlinear components.

This singularity of concept and method is one of the principal advantages of the new engineering. Problems involving nonlinear components are no different than problems involving only proportional components. When one has learned to solve proportional design/analysis problems using the new engineering, he has also learned to solve nonlinear design/analysis problems because the solutions are EXACTLY THE SAME. On the other hand, the opposite is true in the old engineering, and one must learn to solve nonlinear problems in a way which little resembles the method of solving proportional problems.

The resistance concept has been part of the old engineering for many decades, and the result of this long usage is that we no longer question whether or not it is desirable to deal with flow phenomena in terms of "resistance"--ie we treat resistance as though it were fundamental rather than contrived. The contrived nature of "resistance" can best be appreciated by considering the question:

What EXACTLY is the thing we call resistance?

The correct and revealing answer to this question is that "resistance" is merely the name we assign to certain constants and/or variables which are determined by the functionality between driving force and flow rate. For example,

> Resistance is the name we sometimes give to the constant of proportionality between driving force and flow rate.
>
> Resistance is the name we sometimes give to the ratio of driving force to flow rate.
>
> Resistance is the name we sometimes give to the first derivative of driving force with respect to flow rate.

"Static electrical resistance" is an example of the first type. "Thermal resistance" is an example of the second type. "Dynamic electrical resistance" is an example of the third type.

And now let us question whether or not it is desirable to deal with flow phenomena in terms of "resistance". What do we gain by inventing and utilizing "resistance"?

The correct answer is that we gain NOTHING by inventing and utilizing "resistance". ANY problem that can be solved with resistance can be solved just as well without it. Then why do we retain and utilize the many resistance parameters of the old engineering? Because it has been done that way for many decades--and long precedent has a way of making unnecessary things seem necessary and desirable. What is the harm in inventing and utilizing resistance parameters? The harm is that since these parameters are unnecessary, they serve only to complicate and confuse the solution of every problem they touch. They force us to think and to design and to analyze in terms of parameters of only related importance rather than in terms of parameters of direct, first order importance. They add altogether unnecessary dimensions to every problem. And in the case of the second type of resistance, this added dimension introduces so much artificial complication and confusion that it becomes virtually (if not identically) impossible to think clearly or to solve certain design/analysis problems dealing with highly nonlinear phenomena such as boiling.

In the old engineering, we recognize that 2-dimensional problems are simpler to solve than 3-dimensional problems. For that reason, we often strive to find unimportant dimensions which can be neglected in the solution of the problem. But conceptually, we do just the opposite in the old engineering! We invent unnecessary parameters and the end result is that 2-parameter problems often become 3-parameter problems! Recall that Ohm was confronted with the problem of how to describe the proportionality between volts and amps--a problem involving only two parameters. How did Ohm solve this 2-parameter problem? He solved it in terms of 3 parameters--and in order to do so, it was of course necessary to invent a parameter--the parameter of "electrical resistance". And because of this parameter invented 150 years ago, the design and analysis of proportional electrical components and circuits has ever since been based on THREE parameters when in fact only TWO are necessary!!!

The new engineering lacks many of the key concepts and parameters of the old engineering, and therefore we must deal with engineering phenomena and equipment in an altogether new way. For example, the new engineering has no resistance concept and therefore we must think about flow phenomena in terms of the driving force and the flow rate ONLY--we must describe flow component behavior in terms of the driving force and the flow rate ONLY--and we must design/analyze flow equipment in terms of the driving force and the flow rate ONLY. And now let us consider precisely how this is done in the new engineering. Let us use the new engineering to solve some simple problems involving the design/analysis of flow equipment. Since all flow phenomena are quite similar, it will not be necessary to consider all the various types of flow equipment. However, it should be born in mind that the problems are intended to illustrate the new way to design/analyze flow equipment IN GENERAL, as well as to demonstrate that the concept of resistance is GENERALLY unnecessary and that ALL resistance parameters should be abandoned.

DESIGNING/ANALYZING SIMPLE ELECTRIC CIRCUITS

In the old engineering, we describe the behavior of proportional electrical components in terms of their "static electric resistance" which is usually abbreviated to the single word "resistance". For example, we might say

> This electrical component behaves in such a way that its resistance is 2 ohms.

But what EXACTLY does this statement mean? It means exactly the following:

> This electrical component behaves in such a way that E = 2I.

In other words, the information that the "resistance" is 2 ohms is of NO VALUE unless we can translate this

information into the equivalent and more intelligible form, E = 2I. (Throughout this book, the symbols indicate the dimensions as well as the parameters. For example, E indicates electric potential difference in volts, I indicates electrical current in amps, R indicates electrical resistance in ohms. Thus the equation E = 2I means that the potential difference in volts is equal to twice the current in amps.) And this brings up the questions

> What is the particular advantage in stating that R = 2 when we actually mean that E = 2I? Why do we design/analyze proportional electric circuits in terms of the three parameters R, E, and I when the two parameters E and I are sufficient?

The correct answer is that there is NO advantage in stating R = 2 when we mean E = 2I. There is NO good reason to use three parameters when only two are sufficient. Quite simply, the invention and utilization of the parameter R serve NO USEFUL PURPOSE.

Everyone reading this book knows how to use the concept of resistance to solve simple electrical circuits which contain only "Ohm's Law resistors". One might therefore suppose that the particular advantage of "resistance" is that it somehow simplifies the solution of proportional electrical circuits (often inappropriately called "linear electrical circuits" in the old engineering). But the truth of the matter is that it is even simpler to solve proportional electrical circuits withOUT using the concept of resistance--withOUT using "Ohm's Law resistance". To prove this, let us solve some simple design/analysis problems dealing with electrical circuits. First we will solve them the old way using Ohm's Law and the concept of "resistance", then we will solve them the new way using relation 3 and the new concept of "behavior".

PROBLEM 1: What is the current in the circuit shown in Figure 1?

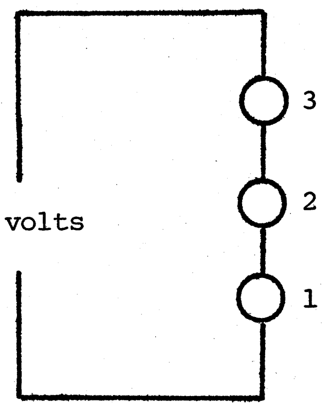


FIGURE 1

Old way analysis based on Ohm's Law

Given: The electrical components in Fig 1 are "Ohm's Law resistors"--ie they behave in a proportional way. Their behavior is described by $R_1 = 3.0$, $R_2 = 4.5$, and $R_3 = 6.2$.

Solution: From resistance theory, we know that resistances in series are additive and therefore that components 1, 2, and 3 are equivalent to a single component whose resistance R_s is given by

$$R_s = 3.0 + 4.5 + 6.2 = 13.7 \quad (4)$$

From this equivalent resistance and "Ohm's Law", we know that the current in the circuit is given by

$$I = \frac{E}{R_s} = \frac{100}{13.7} = 7.30 \quad (5)$$

Problem 1 cont.

New way analysis based on relation 3

Given: The behavior of the electrical components in Fig 1 is described by $E_1 = 3.0I$, $E_2 = 4.5I$, $E_3 = 6.2I$.

Solution: From potential theory, we know that the voltage drops across series-connected components are additive and therefore we may write

$$E_1 + E_2 + E_3 = 100 \quad (6)$$

Substituting the given component behavior into eq 6 gives

$$3.0I + 4.5I + 6.2I = 100 \quad (7)$$

and therefore $I = 7.30$.

PROBLEM 2: What power is dissipated in component 3 of Figure 2?

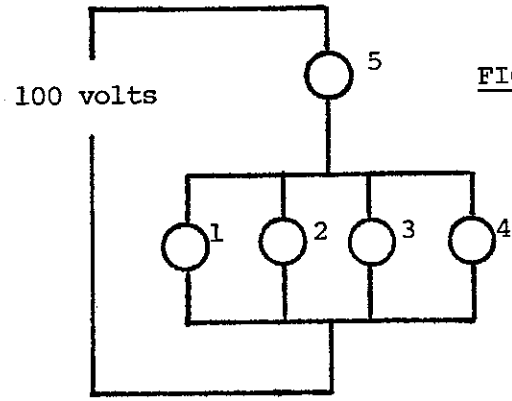


FIGURE 2

Old way analysis based on Ohm's Law

Given The electrical components in Fig 2 are "Ohm's Law resistors"--ie they behave in a proportional way. Their behavior is described by $R_1 = 2.4$, $R_2 = 4.4$, $R_3 = 2.5$, $R_4 = 4.7$, $R_5 = 1.9$.

Solution In order to determine the power dissipated in component 3, we must first solve for the current which passes through it. Using the resistance concept, we calculate this current by noting that resistors in parallel are equivalent to a single resistor whose resistance is given by the reciprocal of the sum of the reciprocals of the individual resistances. Expressed mathematically, this means that components 1, 2, 3, and 4 are equivalent to a single resistor whose resistance is given by

$$R_s = \frac{1}{1/2.4 + 1/4.4 + 1/2.5 + 1/4.7} \qquad (8)$$

$$R_s = 0.796$$

Problem 2 cont.

Since this equivalent resistor is in series with component 5, the total resistance of the circuit is given by

$$R = .796 + 1.9 = 2.696 \qquad (9)$$

Using Ohm's Law, this result tells us that overall current is

$$I = \frac{100}{2.696} = 37.1 = I_s = I_5 \qquad (10)$$

and therefore that the voltage drop across component 3 is given by

$$E_3 = I_s R_s = 37.1\ (.796) = 29.5 \qquad (11)$$

Therefore the current through component 3 is (from Ohm's Law)

$$I_3 = E_3/R_3 = 29.5/2.5 = 11.8 \qquad (12)$$

Therefore the power dissipated in component 3 is, from resistance theory,

$$P = I^2R = 11.8^2\ (2.5) = 348 \qquad (13)$$

Problem 2 cont.

New way analysis based on relation 3

Given The behavior of the electrical components in Fig 2 is described by $E_1 = 2.4I$, $E_2 = 4.4I$, $E_3 = 2.5I$, $E_4 = 4.7I$, $E_5 = 1.9I$.

Solution In order to determine the power dissipated in component 3, we must determine its voltage drop and current. We determine these by noting that components 1 to 4 are equivalent to a single component described by

$$I_s = I_1 + I_2 + I_3 + I_4 \quad (14)$$

$$E_s = E_1 = E_2 = E_3 = E_4 \quad (15)$$

Substituting the given information into eq 14 and using eq 15 gives

$$I_s = .417E_s + .227E_s + .400E_s + .213E_s \quad (16)$$

Since the equivalent component is in series with component 5, we may write

$$E_s + E_5 = 100 \quad (17)$$

$$\therefore \quad E_s + 1.9I_s = 100 \quad (18)$$

The solution of eqs 16 and 18 tells us that E_s and therefore E_3 are equal to 29.5. Since E_3 is 29.5, the given information tells us that $I_3 = 29.5/2.5 = 11.8$ and

Problem 2 cont.

therefore, from potential theory, the power dissipated in component 3 is given by

$$P_3 = E_3I_3 = 29.5 \times 11.8 = 348 \quad (19)$$

PROBLEM 3: What is the current in the circuit shown in Figure 3?

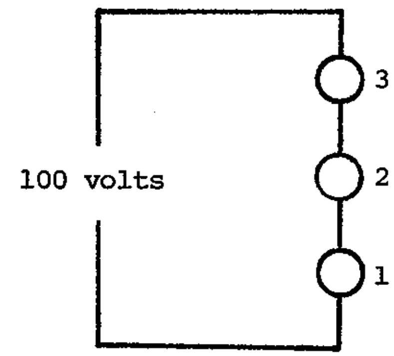


FIGURE 3

Old way analysis based on Ohm's Law

Given Components 1 and 2 in Fig 3 are "Ohm's Law resistors" and their behavior is described by $R_1 = 3.0$, $R_2 = 6.2$. Component 3 is NOT an Ohm's Law resistor. Its behavior is described by the nonlinear relation $E_3 = 5.1(I)^{.5}$.

Solution It is IDENTICALLY IMPOSSIBLE to solve this problem using Ohm's Law. Because of the nonlinear behavior of component 3, its Ohm's Law resistance is not defined. In order to solve this problem using the old engineering, we would first have to learn the "other" method of solving such problems.

Problem 3 cont.

This "other" method of the old electrical engineering is used only when dealing with electrical components which behave in a nonlinear way. We will not go into this other method except to say that it is based on volts, amps, and the type of resistance usually called "dynamic electrical resistance". (This other type of resistance is, by definition, the first derivative of E with respect to I. It is dynamic in the sense that it describes how E responds to changes in I.)

New way analysis based on relation 3

Given The behavior of the components in Fig 3 is described by $E_1 = 3.0I$, $E_2 = 6.2I$, and $E_3 = 5.1(I)^{.5}$.

Solution The new way solution of this nonlinear problem is EXACTLY THE SAME as the new way solution of Problem 1 which was a proportional problem. Since the components are series-connected, we may write

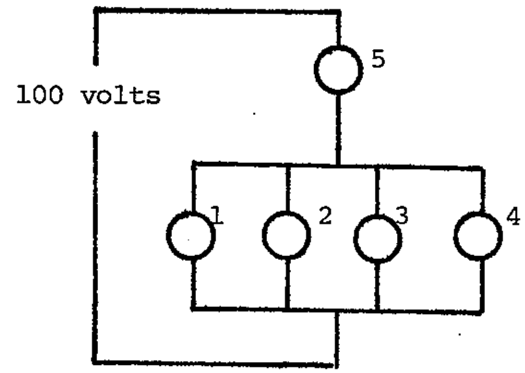
$$E_1 + E_2 + E_3 = 100 \qquad (20)$$

Substituting the given component behavior into eq 20 gives

$$3.0I + 6.2I + 5.1(I)^{.5} = 100 \qquad (21)$$

The solution of eq 21 gives $I = 9.19$.

PROBLEM 4: What power is dissipated in component 3 of Figure 4?



Old way analysis based on Ohm's Law

Given Components 1, 2, 4, and 5 are Ohm's Law resistors. Their behavior is described by $R_1 = 2.4$, $R_2 = 4.4$, $R_4 = 4.7$, $R_5 = 1.9$. Component 3 is NOT an Ohm's Law resistor. Its behavior is described by $E_3 = 3.2(I)^{.5}$.

Solution It is IDENTICALLY IMPOSSIBLE to solve this problem using Ohm's Law because of the nonlinear behavior of component 3.

New way analysis based on relation 3

Given The behavior of the components in Fig 4 is described by $E_1 = 2.4I$, $E_2 = 4.4\ I$, $E_3 = 3.2(I)^{.5}$, $E_4 = 4.7I$, $E_5 = 1.9I$.

Solution In order to determine the power dissipated in component 3, we must determine its voltage drop and current. We determine these by noting that components 1 to 4 are

Problem 4 cont.

equivalent to a single component described by

$$I_s = I_1 + I_2 + I_3 + I_4 \quad (22)$$

$$E_s = E_1 = E_2 = E_3 = E_4 \quad (23)$$

Substituting the given information into eq 22 and using eq 23 gives

$$I_s = .417E_s + .227E_s + .0977E_s{}^2 + .213E_s \quad (24)$$

Since this single component is in series with component 5, we may write

$$E_s + E_5 = 100 \quad (25)$$

$$\therefore\ E_s + 1.9I_s = 100 \quad (26)$$

The solution of eqs 24 and 26 tells us that E_s and also E_3 are equal to 17.2. Since E_3 is 17.2, the given information tells us that

$$I_3 = (E_3/3.2)^2 = (17.2/3.2)^2 = 28.9 \quad (27)$$

and therefore the component 3 power is given by

$$P_3 = E_3I_3 = 17.2 \times 28.9 = 497 \quad (28)$$

THE SIGNIFICANCE OF PROBLEMS 1 AND 2

Problems 1 and 2 demonstrate that

A. The old engineering parameter "electrical resistance" is not necessary for the solution of proportional design/analysis problems.

B. The new way of solving proportional design/analysis problems is even simpler than the old way.

We can reach these same conclusions in a more general way by noting that the old way of solving proportional design/analysis problems is based on the following:

1. The voltage drops across series-connected components are additive--ie

$$E_{total} = E_1 + E_2 + E_3 + \text{etc} \quad (29)$$

2. The currents through parallel-connected components are additive--ie

$$I_{total} = I_1 + I_2 + I_3 + \text{etc} \quad (30)$$

3. The power dissipated in an electrical component is given by the product of the voltage drop and the current--ie

$$P = EI \quad (31)$$

4. The proportionality constant between voltage drop and current is defined to be the "resistance", R.

5. The resistance of series-connected components is given by the sum of the resistances of the individual

components--ie

$$R_{total} = R_1 + R_2 + R_3 + \text{etc} \tag{32}$$

6. The resistance of parallel-connected components is given by the reciprocal of the sum of the reciprocal of the resistances of the individual components--ie

$$R_{total} = \frac{1}{1/R_1 + 1/R_2 + 1/R_3 + \text{etc}} \tag{33}$$

7. The power dissipated in an electrical component is given by the product of the current squared and the resistance--ie

$$P = I^2R \tag{34}$$

On the other hand, the new way of solving proportional design/analysis problems (and also nonlinear problems) is based on ONLY ITEMS 1 TO 3! ANY problem that can be solved with items 1 to 7 can be solved just as readily with only items 1 to 3. The "information" in items 4 to 7 is superfluous--redundant--unnecessary. It becomes necessary only if we decide to invent and utilize the parameter "electrical resistance". In the new engineering, we do not invent resistance and that is why we have no use for the information in items 4 to 7. And now let us prove in the most rigorous way that items 4 to 7 are unnecessary.

Item 4 is obviously unnecessary because it is merely a matter of definition--of arbitrarily assigning a name to a pure constant which has no need of a name.

The proof that item 5 is unnecessary is based on proving that item 5 is IDENTICALLY THE SAME as item 1. In other

words, item 5 is unnecessary because it is merely a disguised form of item 1. Note that, since R is the proportionality constant between E and I, we may substitute E/I for R in eq 32 and so obtain

$$\frac{E_{total}}{I_{total}} = \frac{E_1}{I_1} + \frac{E_2}{I_2} + \frac{E_3}{I_3} + \text{etc} \tag{35}$$

Since the components are in series, we may write

$$I_{total} = I_1 = I_2 = I_3 = \text{etc} \tag{36}$$

Combining eqs 35 and 36 gives

$$E_{total} = E_1 + E_2 + E_3 + \text{etc} \tag{37}$$

proving that item 5 is merely a disguised form of item 1 and therefore is unnecessary.

The proof that item 6 is unnecessary is based on proving that item 6 is IDENTICALLY THE SAME as item 2. Since R is the proportionality constant between E and I, we may rewrite eq 33 in the form

$$\frac{E_{total}}{I_{total}} = \frac{1}{I_1/E_1 + I_2/E_2 + I_3/E_3 + \text{etc}} \tag{38}$$

Since the resistances are parallel-connected, we may write

$$E_{total} = E_1 = E_2 = E_3 = \text{etc} \tag{39}$$

Combining eqs 38 and 39 gives

$$I_{total} = I_1 + I_2 + I_3 + \text{etc} \qquad (40)$$

proving that item 6 is merely a disguised form of item 2 and therefore is unnecessary.

It is obvious that item 7 is merely a disguised form of item 3 and therefore is unnecessary.

In summary, Problems 1 and 2 illustrate how to solve proportional design/analysis problems using the new method based on relation 3 and its behavioral concept rather than Ohm's Law and its resistance concept. The problems demonstrate that the parameter "electric resistance" is unnecessary and therefore undesirable in the solution of proportional design/analysis problems. This same conclusion is reached in a very general way by demonstrating that items 5 to 7 are redundant. The old method of solving proportional design/analysis problems requires an understanding of twice as much information as the new method (6 items instead of 3), and for that reason it is twice as difficult to learn how to use the old method. In a very real sense, the old method requires us to learn TWO languages--the language of volts and amps and the language of resistance. The new method is better and simpler because it requires us to learn only ONE language--the language of volts and amps.

THE SIGNIFICANCE OF PROBLEMS 3 and 4

Problems 3 and 4 demonstrate that

A. The old method based on "Ohm's Law" and "resistance" can NOT cope with problems involving nonlinear behavior. In the old engineering, ANOTHER METHOD must be learned if one wishes to deal with nonlinear components (such as vacuum tubes and transistors).

B. The new method based on relation 3 and "behavior" CAN cope with nonlinear behavior. In fact, the new way solution of problems involving nonlinear behavior is EXACTLY THE SAME as the new way solution of problems involving only proportional behavior. The new way solution of problems is ALTOGETHER UNAFFECTED by the type of behavior involved.

Problems 3 and 4 illustrate the principal advantage of the new method based on relation 3 and the concept of "behavior". The principal advantage is the fact that the new method replaces TWO old methods--the Ohm's Law method which can be used ONLY when dealing with proportional components and the "other" method which must be used when dealing with nonlinear components.

Notice that the new way solution of Problem 3 is exactly the same as the new way solution of Problem 1, in spite of the fact that Problem 3 involves nonlinear behavior whereas Problem 1 involves only proportional behavior. (And similarly for Problems 4 and 2.) The point is that when one has learned to solve proportional problems using relation 3 and "behavior", he has also learned to solve nonlinear problems because the new way solution is UNIVERSAL--it is applicable to ALL TYPES of component behavior. And, as shown by Problems 1 to 4, this new way solution is so simple it is readily learned and applied to problems involving nonlinear components, even if one has never learned the "other" method of the old engineering.

In Ohm's time, only proportional components were of practical importance and therefore "Ohm's Law" and "resistance" were adequate. But in the twentieth century, nonlinear electrical components possess tremendous practical importance and that is why we must abandon "Ohm's Law" and the concept of "resistance" and replace them with relation 3 and the concept of "behavior".

THERMAL RESISTANCE

The old heat transfer is based on the concept of resistance, and the thermal resistance parameter is properly defined in the following way:

> Thermal resistance is the name given to the ratio of temperature difference ΔT to heat flux q--ie thermal resistance is another name for the reciprocal of the "heat transfer coefficient" h or U.

In the old engineering, electrical resistance and thermal resistance are viewed as analogous parameters, and thermal resistance is often introduced with the aid of the so-called "electrical analogy". But in the new engineering, we recognize that there is little analogy between these two parameters. They only SEEM analogous in the old engineering because resistance parameters are dealt with as though they were fundamental parameters of Nature when in fact they are contrived parameters of the intellect. In the new engineering, we are consciously aware of the contrived nature of resistance parameters and it is obvious that electrical resistance and thermal resistance are NOT analogous because they are contrived in completely different ways.

Electrical resistance is the name given to a constant, thermal resistance is the name given to a variable. How can a constant be considered analogous to a variable? Electrical resistance is applicable to only ONE type of behavior, thermal resistance is applicable to ALL types of behavior. Where is the analogy in that?

Because thermal resistance is a VARIABLE (rather than a constant of proportionality), it is defined for ALL types of thermal behavior and can be used in the solution of ALL thermal design/analysis problems. At first glance, the universality of "thermal resistance" may seem to be an advantage, but it is in fact a decided DISadvantage. The difficulty is that analyses based on "thermal resistance" are MATHEMATICALLY ABSURD if the problem involves thermal behavior which is not strictly proportional.

As we saw in Book 1, the thermal resistance parameter ($\Delta T/q$ or $1/h$) makes it IDENTICALLY IMPOSSIBLE to separate the variables we most want to separate, namely q and ΔT. The impossibility of separating the variables makes relatively little difference if we are dealing with strictly proportional behavior. But if we are dealing with nonlinear thermal behavior, it makes a world of difference! If we do not separate q and ΔT, we greatly increase the difficulty of solving problems involving nonlinear thermal behavior--and we make the correct solution of certain practical problems so difficult as to be virtually impossible.

When we are dealing with proportional behavior, the ratio of driving force to flow rate is identically the same as the proportionality constant between driving force and flow rate. For that reason, thermal resistance SEEMS analogous to electrical resistance whenever we are dealing with proportional thermal behavior. But when we are dealing with nonlinear behavior, the ratio of driving force to flow rate oftentimes BEARS NO RESEMBLANCE to the proportionality constant between driving force and flow rate. In this latter case, there is no analogy at all between thermal resistance and electrical resistance--the former is defined and thus exists, the latter is not defined and thus does not exist. How can something which DOES exist be analogous to something which does NOT exist?

In the new engineering, the behavior concept is the basis for dealing with ALL flow phenomena including the flow of electricity, the flow of heat, the flow of fluids, the flow of solids, etc. The behavior concept is used in the same way in all branches of the new engineering, and this explains the close analogies one finds throughout the new engineering. For example, the fundamental relation of the new heat flow is

$$q = f\{TDF\} \tag{41}$$

and this relation is closely analogous to relation 3, the fundamental relation of the new electrical engineering. (The analogy may be more apparent if we note that relation 41 can also be written in the form

$$TDF = f\{q\} \tag{42}$$

where of course the f refers to a different function than that in relation 41.) Relation 41 is to be read "the heat flux q is some function of the Thermal Driving Force", and similarly for relation 42.

Relation 41 (or 42) is the fundamental, generic relation of the new heat flow and is the basis for the specific relations

$$q_{conv} = f\{\Delta T\} \tag{43}$$

$$q_{trans} = f\{dT/dx\} \tag{44}$$

$$q_{radn} = f\{T\} \tag{45}$$

Relations 43 through 45 are based on the new concept of behavior and they replace the old relations

$$q_{conv} = h\ \Delta T \tag{46}$$

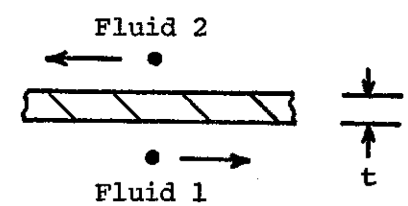
$$q_{cond} = k(dT/dx) \tag{47}$$

$$q_{radn} = \varepsilon\sigma T^4 \tag{48}$$

based on the old concept of resistance. Now let us use the old relations and the new relations to solve some simple design/analysis problems in heat flow.

PROBLEM 5: What is the heat flux through the wall in Figure 5?

FIGURE 5



Old way analysis based on thermal resistance

Given The thermal resistance at the wall/fluid interfaces is described by the equation

$$Nu = .023\ Re^{.8}\ Pr^{.4} \tag{49}$$

Problem 5 cont.

Translating rel 49 into real world parameters gives

$$h = .023(k/D)(DG/\mu)^{.8}(C\mu/k)^{.4} \qquad (50)$$

$$h = .023\,\frac{k^{.6}G^{.8}C^{.4}}{D^{.2}\mu^{.4}} \qquad (51)$$

For the conditions in the Fig 5 system, we are given that eq 51 indicates $h_1 = 720$, $h_2 = 950$.

The wall in Fig 5 is described by $t = .01$, $k = 15$.

The fluid temperatures are $T_1 = 350$, $T_2 = 273$.

Solution The old way solution is based on noting that the overall heat transfer coefficient (ie the inverse of the overall thermal resistance) is given by

$$U = \frac{1}{1/h_1 + t/k + 1/h_2} \qquad (52)$$

$$\therefore U = \frac{1}{1/720 + .01/15 + 1/950} = 322 \qquad (53)$$

$$q = U\,\Delta T \qquad (54)$$

$$\therefore q = 322(350 - 273) = 24{,}800 \qquad (55)$$

Problem 5 cont.

New way analysis based on thermal behavior

Given The thermal behavior at the wall/fluid interfaces is described by the equation

$$q = .023\,\frac{k^{.6}G^{.8}C^{.4}}{D^{.2}\mu^{.4}}\,\Delta T \qquad (56)$$

(Note that eq 56 is the new way translation of eq 51 and that it was obtained by replacing h with $q/\Delta T$ and then separating q and ΔT.) For the conditions in the Fig 5 system, we are given that eq 56 indicates $q_1 = 720\ \Delta T_1$ and $q_2 = 950\ \Delta T_2$.

The wall in Fig 5 is described by $t = .01$, $q_w = 15\ \Delta T_w/t$.

The fluid temperatures are $T_1 = 350$, $T_2 = 273$.

Solution The new way solution is based on noting that

$$\Delta T_t = \Delta T_1 + \Delta T_w + \Delta T_2 \qquad (57)$$

Substituting the given information into eq 57 and noting that $q_1 = q_w = q_2$ gives

$$350 - 273 = q/720 + .01q/15 + q/950 \qquad (58)$$

$$\therefore q = 24{,}800$$

PROBLEM 6: What is the heat flux through the wall in Figure 5, page 1-27?

Old way analysis based on thermal resistance

Given: The thermal resistance of the interface between fluid 1 and the wall is described by eq 51. The thermal resistance of the interface between fluid 2 and the wall is described by

$$Nu = .012(Gr\ Pr)^{.333} \qquad (59)$$

Translating eq 59 into real world parameters gives

$$h = .012(\rho^2 g\beta\Delta TCk^2/\mu)^{.333} \qquad (60)$$

For the conditions in Fig 5, we are given that eq 51 indicates that $h_1 = 100$ and eq 60 indicates that

$$h_2 = .22\ (\Delta T_2)^{.333} \qquad (61)$$

The wall in Fig 5 is described by $t = .20$, $k = 0.35$.

The fluid temperatures are $T_1 = 225$, $T_2 = 110$.

Solution: The old way solution is based on writing eq 52 and then combining it with the given information to obtain

$$U = \frac{1}{1/100 + .20/.35 + 1/h_2} \qquad (62)$$

Problem 6 cont.

We cannot solve eq 62 for U in a direct manner because h_2 is a function of ΔT_2 and we do not know the value of ΔT_2. The indirect method generally used to solve problems of this sort is:

a) Arbitrarily pick a value for ΔT_2.

b) Use this arbitrary value of ΔT_2 and eq 61 to solve for the corresponding value of h_2.

c) Use this value of h_2 to solve for a first estimate of U.

d) Note that if we picked the correct value for ΔT_2, the q defined by

$$q = h_2\ \Delta T_2 \qquad (63)$$

will be equal to the q defined by

$$q = U\ \Delta T_{total} \qquad (64)$$

If these two values of q are not equal, then we did not pick the correct value for ΔT_2 and we must repeat steps a through d until the q from eq 63 equals the q from eq 64.

The reader is encouraged to obtain the numerical answer to this simple problem using the above method of the old heat transfer and to compare his solution to the new way solution below.

Problem 6 cont.

New way analysis based on thermal behavior

Given The thermal behavior of the interface between fluid 1 and the wall is described by eq 56. The thermal behavior of the interface between fluid 2 and the wall is described by

$$q = .012(\rho^2 g\beta C k^2/\mu)^{.333}\ \Delta T^{1.333} \quad (65)$$

(Eq 65 is the result of translating eq 60 from the old way to the new way.)

For the conditions in Fig 5, we are given that eq 56 indicates that $q_1 = 100\ \Delta T_1$ and eq 65 indicates that

$$q_2 = .22(\Delta T_2)^{1.333} \quad (66)$$

The wall in Fig 5 is described by $t = .20$, $q_w = .35\ \Delta T/t$.

The fluid temperatures are $T_1 = 225$, $T_2 = 110$.

Solution The new way solution is based on noting that

$$\Delta T_t = \Delta T_1 + \Delta T_w + \Delta T_2 \quad (67)$$

Substituting the given information into eq 67 and noting that $q_1 = q_w = q_2$ gives

$$225 - 110 = q/100 + .20q/.35 + (q/.22)^{.75} \quad (68)$$

Problem 6 cont.

The solution of eq 68 gives

$$q = 69.3 \quad (69)$$

(Note that the new way solution of Problem 6 is exactly the same as the new way solution of Problem 5, even though the former involves nonlinear behavior and the latter involves only proportional behavior. Note also that the Problem 5 solution leads to a proportional equation (eq 58) which is solved for q whereas the Problem 6 solution leads to a nonlinear equation (eq 68) which is solved for q.)

PROBLEM 7: What is the heat flux through the wall in Figure 5, page 1-27?

Old way analysis based on thermal resistance

Given The thermal resistance of the interface between fluid 1 and the wall is described by eq 51. At the Fig 5 conditions, we are given that eq 51 indicates $h_1 = 1050$.

The thermal resistance of the interface between fluid 2 and the wall is highly nonlinear and cannot be described by a simple, analytical function. It is described graphically in Fig 6, page 1-34, for the conditions in the Fig 5 system.

The wall in Fig 5 is described by $t = .013$, $k = 12$.

The fluid temperatures are $T_1 = 500$, $T_2 = 340$.

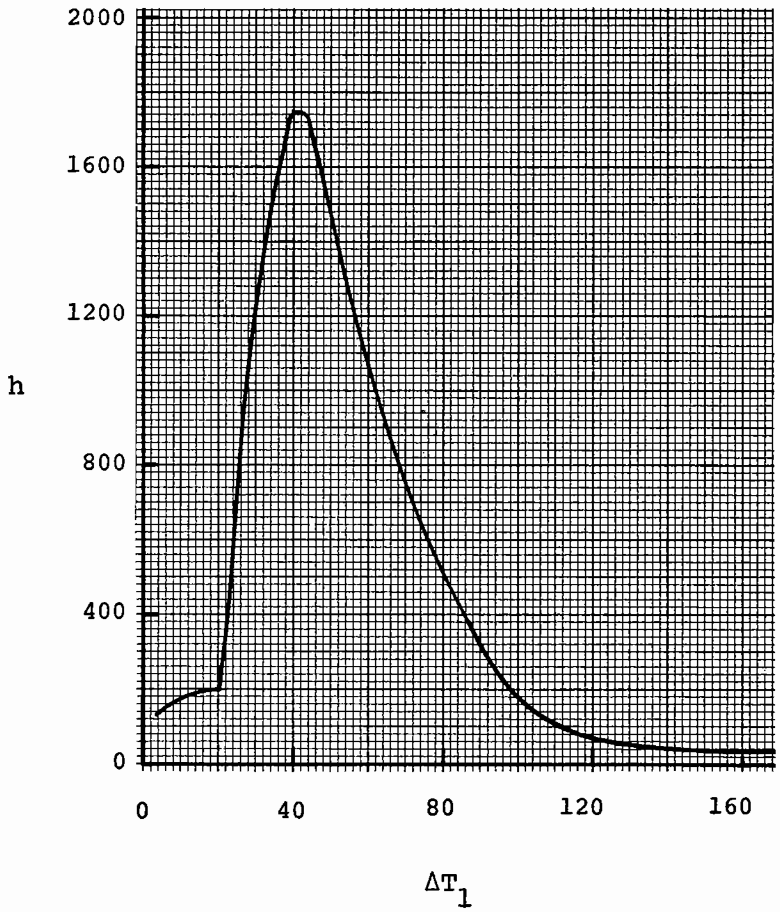


FIGURE 6

The Thermal Resistance $h\{\Delta T\}$

of the Fluid 1 Interface in Problem 7

Problem 7 cont.

Solution The old way solution of this problem is so difficult that it has never been solved, as evidenced by the fact that textbooks on the old heat transfer generally present an incorrect solution to the problem. The correct solution of this problem must be regarded as outside the "knowledge envelope" of the old heat transfer.

The reader is encouraged to attempt the old way solution of this problem before going on to the new way solution. This attempt would seem to be the best way to convince the reader that the old way solution of this problem is virtually impossible--and to convince him that this problem has not been solved with the old heat transfer.

New way analysis based on thermal behavior

Given The thermal behavior of the interface between fluid 1 and the wall is described by eq 56. At the Fig 5 conditions, we are given that eq 56 indicates $q_1 = 1050\ \Delta T_1$.

The thermal behavior of the interface between fluid 2 and the wall is highly nonlinear and cannot be described by a simple, analytical function. It is described graphically in Fig 7, page 1-36, for the conditions in the Fig 5 system.

The wall in Fig 5 is described by $t = .013$, $q_w = 12\ \Delta T_w/t$.

The fluid temperatures are $T_1 = 500$, $T_2 = 340$.

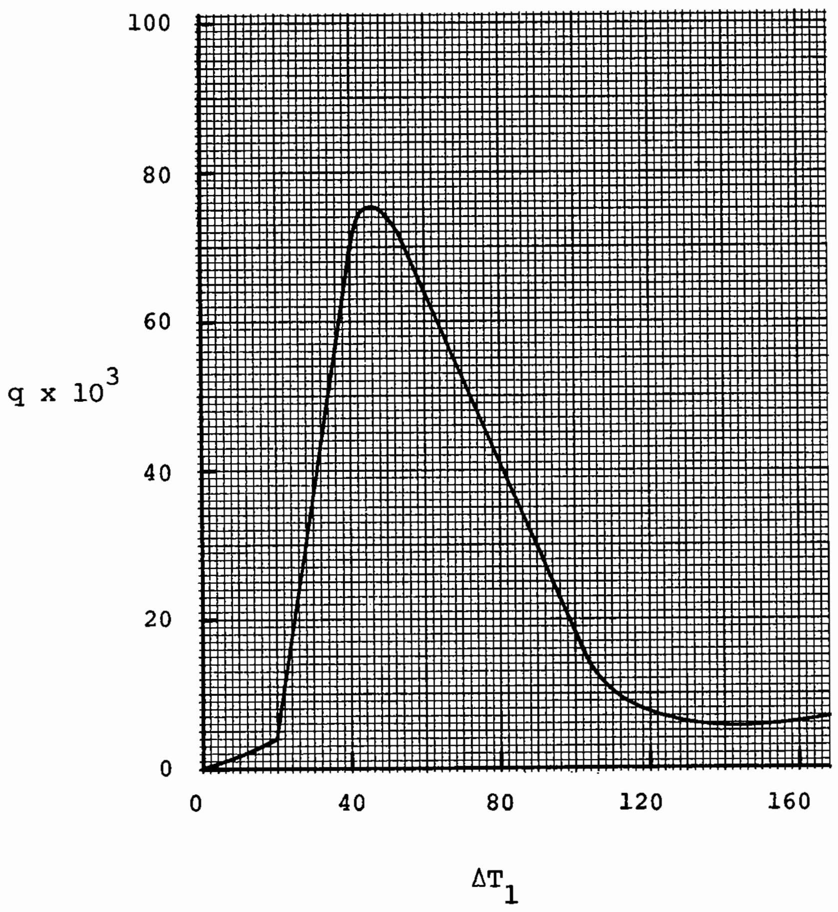


FIGURE 7

The Thermal Behavior q{ΔT} of the Fluid 1 Interface in Problem 7

Problem 7 cont.

Solution The new way solution is based on noting that

$$\Delta T_t = \Delta T_1 + \Delta T_w + \Delta T_2 \quad (70)$$

Substituting the given information into eq 70 and noting that $q_1 = q_w = q_2$ gives

$$500 - 340 = q/1050 + .013q/12 + \Delta T_2 \quad (71)$$

$$\therefore\ q = 78{,}600 - 491\Delta T_2 \quad (72)$$

Because the thermal behavior of the fluid 2/wall interface is not given in analytical form, we cannot reduce eq 72 to an equation with a single unknown (as we were able to do with eqs 58 and 68). In this case, the solution for q is based on noting that eq 72 and the curve in Fig 7 are both of the form $q\{\Delta T_2\}$ and that the q described by eq 72 must equal the q described by the curve in Fig 7. In other words, the solution for q involves the solution of the two simultaneous relations described by eq 72 and Fig 7. Since one of these relations is graphical, the solution must be graphical. This means we must plot eq 72 against the Fig 7 curve and the intersection(s) will define q and ΔT_2 at the operating point of the equipment. The simultaneous solution of eq 72 and the Fig 7 curve is shown in Fig 8, page 1-38. As shown in Fig 8, there are three possible solutions for q and ΔT_2:

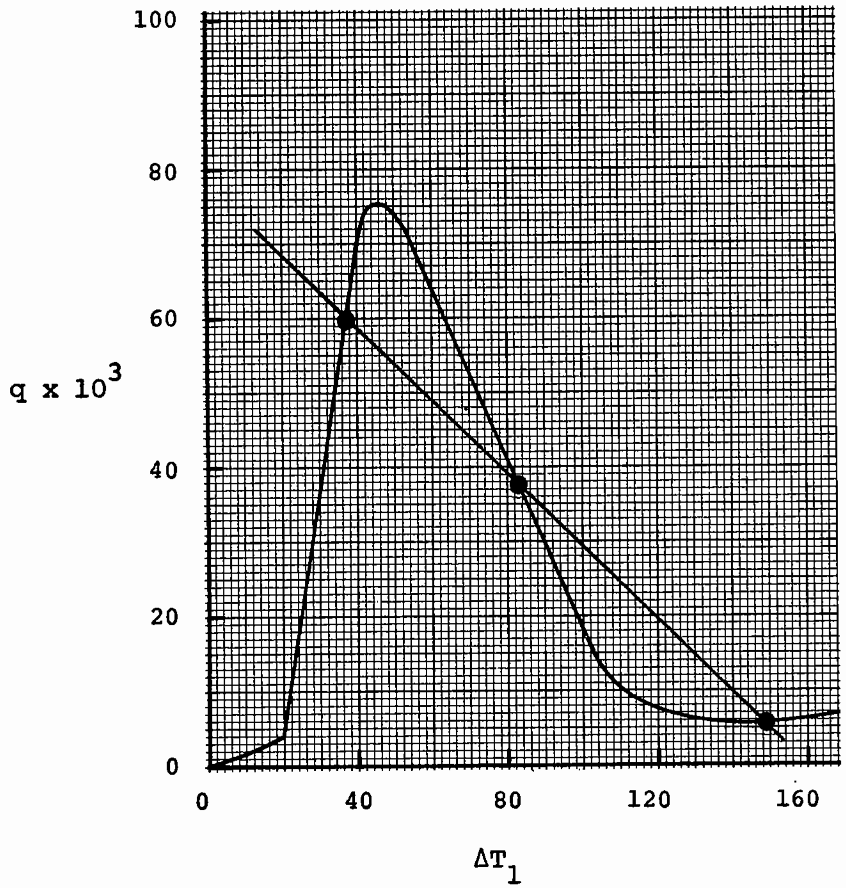


FIGURE 8

Graphical Solution for
the Heat Flux in Problem 7

Problem 7 cont.

Possible values of q and ΔT_2
(from Figure 8)

$q = 60{,}000$ at $\Delta T_2 = 37$

$q = 38{,}000$ at $\Delta T_2 = 83$

$q = 12{,}000$ at $\Delta T_2 = 148$

Since the operating point at q = 38,000 is thermally unstable, the answer to Problem 7 is that q is either 60,000 or 12,000 and we do not have sufficient information to determine which of these two values is the correct one. (Recall from Book 1 that the criterion for thermal stability is

$$(dq_{in}/dT_i)_{ss} < (dq_{out}/dT_i) \qquad (73)$$

With regard to Fig 8, eq 73 states that stable intersections are those where the slope of the eq 72 curve is more negative than the slope of the curve from Fig 7. Note that eq 72 actually describes q_{in}-- ie actually describes the heat flow into the fluid 2/wall interface and that the curve from Fig 7 actually describes the heat flow out of this interface. Inspection of Fig 8 shows that this criterion is not satisfied at the middle intersection and is satisfied at the two outside intersections.) In order to determine whether the equipment were actually operating at q = 60,000 or 12,000, we would need to know the prior operating history of the equipment.

THE SIGNIFICANCE OF PROBLEMS 5, 6, AND 7

Problems 5, 6, and 7 demonstrate how to use the new heat flow to determine local heat flux. Although the problems do not deal with the overall behavior of heat flow equipment, it should be noted that the problems specifically relate to equipment design/analysis in that the determination of local heat flux is oftentimes the key step in designing/analyzing heat flow equipment. Once the local heat flux is determined, it is integrated over the surface of the equipment and this integral establishes the overall thermal behavior of the equipment.

Problem 5 is an extremely simple problem because it involves only behavior which is strictly proportional. Due to this simple behavior, the difficulty in solving the problem is little affected by whether or not we separate the variables q and ΔT. For this reason, both the old way and the new way solutions are in fact quite simple. But notice that the old way solution is not a direct solution. The problem statement directed us to solve for q, but instead of solving for q, we first solved for U! And why did we solve for U? We solved for U so that we could then multiply it by ΔT_t in order to obtain what we were after in the first place, namely q. Note that this is EXACTLY what we did in eqs 52 to 54. And the point is that if one is consciously aware that U is nothing more than a shorthand way of writing $q/\Delta T_t$, then it will be obvious that this indirect, old way solution is mathematically and conceptually absurd.

On the other hand, notice that the new way solution of Problem 5 is a direct solution. The problem statement directed us to solve for q and so we solved for q. Note also that the new way solution was not possible until we translated the given information from the language of the old heat transfer to the language of the new heat flow--from the language of thermal resistance to the language of thermal behavior. This

involved nothing more than replacing h with $q/\Delta T$, k with $qt/\Delta T$, and separating the variables q and ΔT. (The substitution of $q/\Delta T$ for h is always valid, but the substitution of $qt/\Delta T$ for k is valid only when we are given that "k" is a constant.) To be entirely consistent, we should also have translated the k in eq 56 to its new way equivalent. However, let us overlook this inconsistency for now and come back to it in a later chapter.

Problem 6 involves moderately nonlinear thermal behavior at the fluid 2/wall interface. Because of this more or less complicated behavior, it makes a considerable difference whether or not the solution is based on separating the variables q and ΔT. The difference is that the solution is much more cumbersome and indirect if these variables are not separated as in the old way solution. The reader can best verify this by using the old way to obtain a numerical solution to Problem 6 and then comparing his solution with the new way solution which of course is simple and direct. Notice that the solution of Problem 6 is the same as the solution of Problem 5 except that we must solve a nonlinear equation (eq 68) rather than a proportional equation (eq 58). In other words, the new way solution of problems involving nonlinear behavior is really no different or more complicated than the solution of problems involving only proportional behavior.

It should be noted that Problem 6 is a highly practical problem which is often confronted in heat flow engineering. As most readers probably recognized, it is a problem dealing with free convection from an insulated surface. The old way solution is as described in Zemansky's "Heat and Thermodynamics", 4th ed., 1957, McGraw-Hill, cited in Book 1, page 2-21.

Problem 7 involves highly nonlinear behavior such as one might encounter in dealing with a boiling liquid. In spite of the fact that highly nonlinear thermal behavior has had great practical significance for well over 100 years, problems of this type have NEVER been solved with the old heat transfer. Of course, texts on the old heat transfer do not state that "problems of this type have not yet been solved". Instead, they either avoid this type of problem altogether or, more often, they present an incorrect solution to it. For example, Rohsenow (1) states:

> With condensing vapor as the heat source on one side of a wall, any point on the entire curve, Fig. 2, can be reached under stable conditions. (Rohsenow's Fig. 2 is titled "Typical pool boiling curve" and closely resembles Fig 7 on page 1-36 of this book.)

The new way solution of Problem 7 shows that Rohsenow's statement is NOT true because the Problem 7 hardware is similar to a pool boiler "with a condensing vapor as the heat source on one side of a wall" and it obviously can NOT reach "any point on the entire curve (of Fig 7) under stable conditions". In other words, Rohsenow's statement represents an incorrect solution to Problem 7, and this incorrect solution is widely accepted in the old heat transfer. Rohsenow's statement is actually based on the old heat transfer view that, if one has a thermal heat source (such as a condensing vapor or a flowing liquid), the heat flux is uniquely determined by the temperature of the thermal heat source. The Problem 7 solution demonstrates that this old view is incorrect because the heat flux in Problem 7 is NOT uniquely determined by the temperature of the thermal heat source.

In the new heat flow, we begin to actually deal with nonlinear thermal behavior, and this means that we require a simple and quantitative method of describing nonlinear behavior. There are really only two good ways of describing engineering behavior--analytical and graphical. If the behavior is quite simple, such as the behavior described by eq 72, then we can describe it either analytically (as in eq 72) or graphically (as

in the curve of eq 72 shown on Fig 8, page 1-38). But if the behavior is complex, such as the nonlinear behavior described in Fig 7, then it may not be possible to describe the behavior with a simple, analytical correlation and we would have to settle for a simple, graphical correlation.

The old heat transfer has little need for graphical correlations because it is unable to cope with highly nonlinear behavior and largely avoids it. This is generally accomplished by inventing more or less linear "regimes" to replace nonlinear phenomena and/or by simply ignoring problems which can result from highly nonlinear behavior, such as the problem of instability. But in the new heat flow, we have the capability and the desire to deal with highly nonlinear behavior and therefore we will often need and use graphical correlations in order to design and analyze equipment in which highly nonlinear phenomena take place.

Problem 7 illustrates the manner in which graphical correlations are used in equipment design/analysis. In the old heat transfer, graphical correlations such as that in Fig 7 would be regarded as empirical and therefore unscientific. But in the new heat flow, we recognize that graphical correlations are just as scientific as analytical correlations and moreover that graphical correlations are more useful when dealing with highly nonlinear behavior.

Problems 5, 6, and 7 illustrate that the improvement one obtains from the new heat flow is largely determined by the complexity of the phenomena one works with. Those who work with problems involving only strictly proportional phenomena (such as Problem 5) will find little improvement in the new heat flow, since the old heat transfer does provide a simple, effective method of solving such problems. Those who work with problems involving moderately nonlinear behavior (such as Problem 6) will find a considerable improvement in the new heat flow since it provides a much simpler, more effective way of solving such problems. And those who work with problems involving highly nonlinear phenomena (such as Problem 7) will find a remarkable improvement in the new heat flow since it allows them to solve practical problems never before solved with the old heat transfer.

HYDRAULIC RESISTANCE

There are several different forms of hydraulic resistance, the principal of which is the so-called "friction factor", usually given the symbol "f". But what EXACTLY is the friction factor?

> "Friction factor" is the name we give to the ratio ($\Delta P/V^2$) multiplied by the ratio ($gD/L\rho$) multiplied by a constant which is sometimes 2 and sometimes 0.5. In other words, friction factor is a resistance parameter found only in the intellect.

Is the friction factor necessary? No. Is the friction factor desirable? No. Is the friction factor part of the new engineering? No. The friction factor is a resistance parameter and there are NO resistance parameters in the new engineering. The friction factor is a parameter found only in the intellect, and the new engineering deals only with parameters found in Nature.

The friction factor f is the hydraulic analog of the reciprocal of the heat transfer coefficient--ie f closely resembles 1/h--and all the arguments against h hold equally against f. The hydraulic analog of a statement made earlier about h is the following:

> When one becomes consciously aware that f is nothing more than a shorthand way of writing the group ratio ($2\Delta PgD/V^2L\rho$), then it becomes obvious that f is neither necessary nor desirable and that f should be abandoned because it brings nothing but complexity and confusion to every problem it touches.

We are not going to solve the hydraulic analogs of Problems 5, 6, and 7, but let us consider them for a moment.

The analog of Problem 5 would be a problem in laminar flow, since such problems are readily solved both with the friction factor and without it. However, it should be noted that it makes absolutely no sense to deal with laminar flow problems using the friction factor. For example, in the old fluid flow, one finds expressions of the form

$$f_{lam} = \frac{64}{Re} \tag{74}$$

and eq 74 enables us to determine the ΔP for a given flow rate (and conversely) from the expression

$$\Delta P = \frac{f\rho V^2 L}{2gD} \tag{75}$$

In the new fluid flow, we have no use for f or Re and so the new way translation of eq 74 is

$$\frac{2\Delta PgD}{V^2L\rho} = \frac{64\ \mu}{DV\rho} \tag{76}$$

$$\therefore\ \Delta P_{lam} = \frac{32\mu L}{gD^2}\ V \tag{77}$$

Equation 77 is the new way replacement for BOTH eqs 74 and 75. Equation 77 is very easy to use and to understand. Equation 77 obviously states that, in laminar flow, ΔP is proportional to V and independent of ρ. Now look at eq 75. Doesn't it seem to say something altogether different? Doesn't it seem to say that ΔP is NOT proportional to V and that ΔP DEPENDS on ρ? Does it make any sense to use eq 75 when it seems to say something altogether different from what it actually says? Eqs 74 and 75 actually say the same thing that eq 77 says. But eqs 74 and 75 say it in the complicated, confusing language of the old fluid flow whereas eq 77 says it in the simple, straightforward language of the new fluid flow--in the language of separated variables. The old way solution of laminar flow problems is based on eqs 74 and 75, the new way solution is based on eq 77. The old way solution makes no sense at all. The new way solution makes a great deal of sense because it is simple and direct.

The analog of Problem 6 would probably be a problem based on the so-called Moody plot (2), a plot of friction factor vs Reynolds' Number--ie a plot of f vs Re. The problem would serve to point out that a cumbersome, trial-and-error solution is required whenever we are given ΔP and we must solve for the flow rate. (Almost without exception, textbook problems illustrate how to find ΔP when the flow rate is given, probably because this type of problem can be solved directly even with the ineffective Moody plot. The thermal analog of this error of omission is the fact that textbook problems on free convection heat transfer generally illustrate how to find q when ΔT is given, probably because this type of problem can be solved directly even with the ineffective Nu vs GrPr plots of the old heat transfer, whereas the inverse problem must be solved by trial-and-error if one must use Nu vs GrPr plots.) And it would demonstrate that if we translated the Moody plot to the language of the new fluid flow--ie if we transformed the graph of f vs Re which is really a graph of $(2\Delta PgD/V^2L\rho)$ vs $(DV\rho/\mu)$ to the new way form

$$(2\Delta PgD^3\rho/L\mu^2) \text{ vs } (DV\rho/\mu)$$

then this new way graph would eliminate trial-and-error solutions because it would enable us to solve directly for flow rate given ΔP and for ΔP given flow rate.

The hydraulic analog of Problem 7 would deal with highly nonlinear flow behavior such as that encountered in two phase flow. In two phase flow, the $\Delta P\{W\}$ behavior often closely resembles the $q\{\Delta T\}$ behavior described by Fig 7. The new way solution of the hydraulic analog problem would be very similar to the solution of Problem 7. And the point to be made by the problem would be that problems of this type are virtually impossible to solve if the solution is based on f and quite simple to solve if the solution is based on abandoning f and using the new concept of behavior.

OTHER RESISTANCE PARAMETERS

The old engineering contains a great many resistance/conductance parameters which are generally called "coefficients" or "factors". They are ALL abandoned in the new engineering because the new engineering has NO USE for resistance parameters. The new engineering abandons ALL parameters of the intellect and deals EXCLUSIVELY with parameters of Nature.

MODULI

In the field of stress/strain, the fundamental concept is that of the modulus, and we have the "modulus of elasticity" and the "modulus of plasticity". But what EXACTLY is a "modulus"--for example, what EXACTLY is the "modulus of elasticity"?

> "Modulus of elasticity" is the name we give to the proportionality constant between stress and strain in the region where the stress is essentially

proportional to the strain--ie in the "elastic" region.

Similarly, "modulus of plasticity" is the name we give to the derivative of stress with respect to strain and is generally used only in the region where the stress is NOT proportional to the strain--ie only in the "plastic" region. And the important thing to notice is that, although stress/strain is not a flow phenomenon, the modulus of elasticity is formulated in EXACTLY THE SAME WAY as Ohm's Law resistance and the modulus of plasticity is formulated in EXACTLY THE SAME WAY as dynamic electric resistance. For this reason, the arguments against electric resistance parameters apply equally to these modulus parameters.

Are modulus parameters necessary? No. Are modulus parameters desirable? No. Are modulus parameters found in the new engineering? No. What replaces the modulus concept in the new engineering? The concept of behavior--the SAME concept we use to deal with flow phenomena in the new engineering! In dealing with flow phenomena, the behavior concept is contained in expressions such as

$$\text{Driving force} = f\{\text{flow rate}\} \qquad (78)$$

$$\text{Flow rate} = f\{\text{driving force}\} \qquad (79)$$

$$E = f\{I\} \qquad (80)$$

$$q = f\{\Delta T\} \qquad (81)$$

$$\Delta P = f\{W\} \qquad (82)$$

Similarly, in dealing with stress/strain phenomena,

the stress/strain behavior concept is contained in the expression

$$\sigma = f\{\varepsilon\} \qquad (83)$$

and eq 83 is in fact the fundamental relation used to deal with stress and strain in the new engineering.

The new concept of stress/strain behavior does not improve our ability to deal with elastic behavior, since the old stress/strain does provide a simple, effective method of dealing with elastic behavior. But this new concept will bring about a marked improvement in our ability to deal with plastic behavior, and that is the highly practical reason why the old modulus concept will soon be abandoned in favor of the new stress/strain behavior concept.

THE NEW ENGINEERING

The new engineering is founded on the concept of "behavior", and this new concept is an important part of every branch of the new engineering. The behavior concept eliminates the need to invent the "parameters of the intellect" which are the bases of the old engineering, and the result of this elimination is that only parameters of Nature are used in the new engineering. The behavior concept asks "How is the current related to the voltage drop? How is the convective heat flow related to the ΔT? How is the fluid flow related to the ΔP? How is the stress related to the strain?" And by insisting that we answer in terms of Natural parameters, the behavior concept ensures that the primary variables will remain separate and thus that equipment design/analysis problems will be solved in the simplest and therefore most reliable way.

The behavior concept so permeates the new engineering that it brings a remarkable unity and simplicity to the science of engineering, making possible a new engineering education so broad and so simple that it would be deemed identically impossible from the viewpoint of the old engineering.

CONCLUSIONS

The behavior concept of the new engineering will replace the resistance/conductance/coefficient/factor/modulus concepts of the old engineering and will bring about the abandonment of "parameters of the intellect" such as electrical resistance, heat transfer coefficients, fluid flow friction factors, stress/strain moduli, etc. which result from the old concepts. In this Chapter, we have been concerned with engineering in general and we have used the behavior concept to solve a number of simple, fundamental problems dealing with equipment design/analysis. These problems illustrated the application of the behavior concept and demonstrated that this new concept simplifies the solution of all design/analysis problems and makes possible the solution of problems never solved with the old engineering. In the remaining Chapters, we will focus our attention on heat flow engineering in particular and will use the behavior concept to solve design/analysis problems dealing with heat flow equipment.

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CHAPTER 2 TRANSLATING AND IMPROVING OLD WAY CORRELATIONS

INTRODUCTION

If a new component or system fails to perform as intended, the failure is generally attributed to the designer or the analyst and the equipment is said to be "underdesigned" or "overdesigned". But in spite of this common practice, it must be recognized that equipment performance is not uniquely determined by the designer and the analyst. Equipment designers and analysts base their work on correlations generated by researchers and developers. These correlations are intended to accurately describe real world behavior and thus to make it possible to design and analyze equipment with confidence and accuracy. But when the correlations do not accurately describe real world behavior, then the equipment will not perform as expected even if the design and analysis are perfect--ie even if the design and analysis represent the perfect interpretation of the correlations.

Whether or not the correlations of the old heat transfer closely resemble real world behavior, they can all be readily translated to the language of the new heat flow. But it must be emphasized that this translation does not improve the accuracy of the old correlations--it merely puts them in the form necessary for the new heat flow. (Of course, as soon as the old heat transfer dies out, the language of the new heat flow will be universal and no translation will be necessary. Note that this translation is not an extra step required by the new heat flow--it is an extra step required by the old heat transfer because it is based on unseparated variables and "parameters of the intellect".)

The translation of old way correlations is merely a stopgap measure, and these translated correlations are intended to be used only until they can be replaced (or verified) by altogether new correlations obtained within the framework of the new heat flow, as described in Book 2. These altogether new correlations will be better and more useful than the correlations of the old heat transfer--better in that they will more accurately describe real world behavior, more useful in that they will be in a form more suited to the design and analysis of engineering equipment.

In this Chapter, our main concern is with obtaining new way correlations by translating old way parameters and correlations. These translated correlations will be the bases for the equipment design and analysis problems we shall cope with in the remainder of this book. But we will also look beyond the end of this book toward the altogether new correlations which are coming, and we will briefly discuss how and why these altogether new correlations will differ from those obtained by translating the old heat transfer.

TRANSLATING OLD WAY PARAMETERS AND CORRELATIONS

As we have discussed before, it is a simple matter to translate correlations from the language of the old heat transfer to the language of the new heat flow. This translation requires two steps:

1. Replace old way parameters with their new way counterparts. For example, replace h with $q/\Delta T$, Re with DG/μ, etc.

2. Separate q and ΔT--ie rearrange the equation so that q is on one side and ΔT is on the other.

To this point, we have more or less glossed over the details involved in translating k, and now let us correct this error of omission.

The old heat transfer treats thermal conductance (thermal

transmittance) as though it were IN FACT a proportional phenomenon--as though q were strictly proportional to dT/dx. In the new heat flow, we consider that thermal transmittance may or may not be a proportional phenomenon, and the question of functionality requires a serious answer.

The fundamental relation of the old thermal conductance is

$$q = -k \frac{dT}{dx} \quad (1)$$

where k is in fact a shorthand way of writing "the ratio of q to dT/dx" and is determined by experiment. In the experiment, a heat flux q is passed through the material of interest and the resultant temperature gradient, dT/dx, is measured. k is obtained from the data by noting that

$$k = -q \frac{dx}{dT} \quad (2)$$

The old heat transfer assumption that q is strictly proportional to dT/dx is equivalent to assuming that qdx/dT and k are independent of q, and thus the experiment need be (and often is) performed at only one value of q.

The fundamental relation of the new thermal transmittance is

$$q \rightarrow f\{dT/dx\} \quad (3)$$

which states simply that q is some function of dT/dx. (As we discussed in Book 2, the new heat flow abandons the old requirement that correlations be written as dimensional identities, and it should be noted that the dimensions on the left side of relation 2 do NOT equal the dimensions on the right side. That is the reason we have abandoned the old = and replaced it with the new →. However, there is little practical difference between the two symbols. Those readers who prefer may substitute = for →, but should bear in mind that dimensional identity is not preserved. In the old heat transfer, dimensional identity is obtained by assigning dimensions to what are in fact pure constants. Since this seems a pointless task, we avoid it in the new heat flow.)

In the new heat flow, the functionality in relation 3 is of course determined by experiment, just as the k in relation 1 is determined by experiment. In the new experiment, various levels of heat flux q are passed through the material of interest and the resultant values of the temperature gradient, dT/dx, are measured. No assumption is made about the functionality between q and dT/dx. Instead, the functionality is determined by examining the data. Of course, the possibility exists that the data may indicate a proportional relationship between q and dT/dx, but the point is that we determine this by experiment and NOT by assumption--and we allow whatever functionality is indicated by the data rather than arbitrarily insisting that the functionality is proportional.

Throughout this book, we are merely translating the old heat transfer correlations (with the exception of Figs 4 and 6 in Ch 6)--ie we are PRETENDING that the old heat transfer accurately describes real world behavior. With regard to thermal transmittance, this means that we will PRETEND that the old view of proportionality accurately applies, and this pretense allows us to write relation 3 in the form

$$q \propto - dT/dx \qquad (4)$$

Relation 4 states that q is proportional to dT/dx and therefore that the unspecified functionality in rel 3 is described by a constant times dT/dx. This constant of proportionality is a pure constant--it has no dimensions, no name, no physical significance. Since we are PRETENDING in this book that rel 4 accurately describes the real world behavior of ALL materials, we will oftentimes wish to refer to this supposed constant of proportionality. For convenience, let us assign the symbol "F" to this constant of proportionality and thus we write rel 4 in the form

$$q \rightarrow -F \frac{dT}{dx} \qquad (5)$$

IN THIS BOOK, we PRETEND that rel 5 accurately describes the real world behavior of all materials, but it must be recognized that rel 3 is the fundamental relation for thermal transmittance in the new heat flow. Rel 5 is merely a particular form of relation 3, and this particular form is suggested by the old heat transfer which is based on the view that it is universally applicable. We are using this form in this book because we are translating the old heat transfer correlations, and they state without exception that "conduction" is a proportional phenomenon. But it must be remembered that the new heat flow will include other particular forms of rel 3 when/if it is empirically discovered that some materials do NOT exhibit thermal transmittance behavior which is proportional.

Note that k and F are really quite different. k has a conceptual basis, F is nothing more than a pure constant. k is fundamental, universal, permanent in the old heat transfer, F is incidental, limited, temporary in the new heat flow. The only similarity between k and F is that they are numerically equal WHEN F IS DEFINED--ie when the transmittance behavior of a material is in fact such that it exhibits proportional behavior. In the new heat flow, we will seldom deal with F by itself because we are usually concerned with the functional relationships which relate q to the thermal driving force. For that reason, when we translate k, we will usually be translating it into an expression which relates q to dT/dx (or to $\Delta T/t$ if we are given that k is independent of T). For example,

Old way	New way
k	$q \to -F(dT/dx)$
k = 9.3	$q \to 9.3\ \Delta T/t$
k = 7.4(1 + .002T)	$q \to -7.4(1 + .002T)(dT/dx)$

The only time we will use F by itself is when we are translating some of the dimensionless parameters of the old heat transfer. For example, Pr becomes $C\mu/F$ when translated to the language of the new heat flow. (Note that Pr is actually $C\mu dT/qdx$ which simplifies to $C\mu/F$ only because of the old view that q is proportional to dT/dx and thus that qdx/dT is a constant.)

Note that Pr is dimensionless whereas the dimensions in $C\mu/F$ do not cancel. If this seems illogical, recall that we are not concerned about "dimensional consistency" in the new heat flow. Moreover, even in the old heat transfer, dimensional consistency is only a sometime thing. For example, the old heat transfer contains many correlations like

$$h = .19(\Delta T)^{1/3} \qquad (6)$$

which obviously is NOT dimensionally consistent. The point is that dimensional consistency is neither necessary nor Natural and so we abandon it altogether in the new heat flow. Expressions like $C\mu/F$ are taken to mean merely "the numerical value obtained by multiplying the numerical value of C times the numerical value of μ divided by the numerical value of F where

C, μ, and F are all based on the same system of dimensions", just as the ΔT in rel 6 is taken to mean merely the numerical value of ΔT without its dimensions.

Conceptually, μ and k are very similar, and it may seem anomalous to retain μ and reject k. The reason we do not reject μ, which is in fact a shorthand way of writing "the ratio of shear stress to velocity gradient", is because μ does not prevent us from separating the primary heat flow variables, q and ΔT. But if we were dealing with problems in which the primary variables were shear stress and velocity gradient, we would certainly reject μ in order that we might separate them.

In the old heat transfer, fluid properties are very important because of the widespread use of generalized correlations which relate heat transfer coefficients to fluid properties. In the new heat flow, fluid properties are not nearly so important because generalized correlations are largely replaced by specific correlations, each of which refers to a particular fluid and is based on temperature and pressure rather than fluid properties such as μ and k. To illustrate, the old generalized correlation

$$Nu = a\ Re^{b}\ Pr^{c} \qquad (7)$$

is of the form

$$h = a\ D^{d}\ G^{e}\ \mu^{f}\ C^{g}\ k^{h} \qquad (8)$$

which, for a particular fluid, can be written in the specific form

$$h \to a\ D^{d}\ G^{e}\ f\{T\} \qquad (9)$$

Relation 9 translates to the expression

$$q \rightarrow a\ D^d\ G^e\ f\{T\}\ \Delta T \qquad (10)$$

and specific correlations like rel 10 will increasingly be used in the new heat flow. Relation 10 is a new way, specific correlation which contains no fluid properties and which we have obtained by translating the old way generalized correlation. (See Book 2, Ch 9 for specific correlations for water and air in the form of rel 10.) When altogether new specific correlations are obtained within the framework of the new heat flow, they will be of the form

$$q \rightarrow f_1\{D\}\ f_2\{G\}\ f_3\{T\}\ f_4\{\Delta T\} \qquad (11)$$

and it may very well be found that the functionality predicted by rel 7 little resembles real world behavior.

Now that we have discussed the details of translating k, we are ready to prepare an old-way-to-new-way dictionary for heat flow parameters and correlations. Table 1 is such a dictionary (see page 2-9). Those correlations which are not listed in Table 1 can be translated by analogy.

IMPROVING OLD WAY CORRELATIONS

With regard to the generation of correlations to describe engineering phenomena, the methods of the old heat transfer have two serious shortcomings:

1. They lead to correlations which do not closely resemble real world behavior and of course this makes it impossible to obtain truly optimum equipment designs.

TABLE 1

TRANSLATING OLD WAY PARAMETERS AND CORRELATIONS

Old way	New way
q	q
T	T
ΔT	ΔT
$\frac{dT}{dx}$	$\frac{dT}{dx}$
$q = h\ \Delta T$	$q \rightarrow f\{\Delta T\}$
h	$q/\Delta T$
$h = 114$	$q \rightarrow 114\ \Delta T$
$h = .19(\Delta T)^{1/3}$	$q \rightarrow .19(\Delta T)^{4/3}$
U	$q/\Delta T_{total}$
$U = \frac{1}{1/h_1 + 1/h_2}$	$\Delta T_{total} = \Delta T_1 + \Delta T_2$
$U = 384$	$q \rightarrow 384\ \Delta T_{total}$
$q = -k\frac{dT}{dx}$	$q \rightarrow f\{dT/dx\}$ Note: In translating the old heat transfer, this simplifies to $q \rightarrow -F\frac{dT}{dx}$
$k = 9.3$	$q \rightarrow 9.3\ \Delta T/t$
$k = 7.4(1 + .002T)$	$q \rightarrow -7.4(1 + .002T)(dT/dx)$
Nu	$qD/\Delta TF$

TABLE 1 cont.

TRANSLATING OLD WAY PARAMETERS AND CORRELATIONS

Old way	New way
St	$q/\Delta TCG$
Gr	$D^3\rho^2 g\beta\Delta T/\mu^2$
Pr	$C\mu/F$
$Nu = .023\ Re^{.8}Pr^{.4}$	$q \rightarrow .023\ \frac{F^{.6}G^{.8}C^{.4}}{D^{.2}\mu^{.4}}\ \Delta T$
$Nu = .13(GrPr)^{1/3}$	$q \rightarrow .13(\rho^2 F^2 g\beta C/\mu)^{1/3}\ \Delta T^{4/3}$
$Nu = 5 + .025(RePr)^{.8}$	$q \rightarrow (5\ \frac{F}{D} + .025\ \frac{F^{.2}G^{.8}C^{.8}}{D^{.2}})\ \Delta T$
$Nu = .332\ Re^{1/2}Pr^{1/3}$	$q \rightarrow .332\ \frac{F^{2/3}G^{1/2}C^{1/3}}{D^{1/2}\mu^{1/6}}$
$h = .41\ \frac{k}{x}\ (GrPr)^{1/4}$	$q \rightarrow .41(\rho^2 g\beta CF^3/x)^{1/4}\ \Delta T^{5/4}$
$St = .023\ Re^{-.2}Pr^{-.6}$	$q \rightarrow .023\ \frac{F^{.6}G^{.8}C^{.4}}{D^{.2}\mu^{.4}}\ \Delta T$
fouling factor = .0012	$q_{foul} \rightarrow 833\ \Delta T_{foul}$

2. They do not take into account the fact that correlations are intended to be used in the design and analysis of equipment. The end result of this error of omission is that correlations are oftentimes presented in a form which is difficult to use, even though the correlation developer had it within his power to make the correlation quite simple to use.

If a correlation is to be truly useful for equipment design/analysis, it must closely resemble real world behavior--ie it must describe how the important parameters affect each other. As obvious as this seems, it is simply not recognized by the old heat transfer. The proof of this is that the methods of the old heat transfer place NO EMPHASIS on demonstrating that the parametric functionality described by the correlation agrees with the functionality described by the experiment. In the old heat transfer, all the emphasis is placed on demonstrating that the overall prediction of the correlation more or less agrees with the values measured in the experiment. The difficulty with this is that it is readily possible to obtain overall "agreement" even though there is little resemblance between correlation functionality and real world functionality.

For example, recall the film cooling experiment by Hartnett, Birkebak, and Eckert (1) which we discussed in Book 1, Ch 6 and again in Book 2, Ch 10. In this experiment, the researchers dealt with η, x, G_g, G_c, and s, and they correlated their data with

$$\eta = 16.9(G_c s/xG_g)^{.8} \qquad (12)$$

and with a graphical correlation in the form $\eta\{G_c s/xG_g\}$. With regard to their graphical correlation, the authors demonstrate that it "represent(s) the reported values (of several researchers) with a deviation of ±40 per cent". They also state that their graphical correlation can be used to obtain "a reasonable estimate of the effectiveness for practical applications". Thus the researchers show that their graphical correlation more or less agrees with the overall behavior of film cooling--at least it more or less agrees with the values REPORTED by several investigators.

If the old heat transfer recognized the importance of obtaining agreement between correlation functionality and experiment functionality, then researchers would generally take some pains to demonstrate this agreement. For example, the above researchers would have demonstrated that their data in fact indicated

$$\eta \propto G_c^{.8} \quad (13)$$

in accordance with their rel 12. But did they make this demonstration? No, they did not. It was not necessary--not in the old heat transfer. Moreover, it would have been IMPOSSIBLE to make such a demonstration because they had performed their experiment in such a way that their data contained NO INFORMATION about the functionality between η and G_c! The authors would have demonstrated that their data in fact indicated

$$\eta \propto s^{.8} \quad (14)$$

in accordance with their rel 12. Did they make this demonstration? No, they did not. It would have been impossible to make such a demonstration because their data contained NO INFORMATION about the functionality between η and s. The authors would have demonstrated that their data in fact indicated

$$\eta \propto G_g^{-.8} \quad (15)$$

in accordance with their rel 12. Did they make this demonstration? No, they did not. It would have been impossible to make such a demonstration because their data contained NO INFORMATION about the functionality between η and G_g. The authors would have demonstrated that their data in fact indicated

$$\eta \propto x^{-.8} \quad (16)$$

in accordance with their rel 12. Did they make this demonstration? They tried to, but unfortunately they found that rel 16 DID NOT AGREE with their data. Did they then reject rel 12 because their $\eta\{x\}$ data did not agree with it? No, it was not necessary to reject rel 12 simply because it did not agree with the data. The methodology of the old heat transfer has the cure for this malady. The cure is called "regimes". And what exactly is a regime? A regime is a range of applicability which is arbitrarily chosen so as to be sufficiently narrow that correlations and data will SEEM to agree even though they violently DISagree. And so the authors availed themselves of this cure and invented the "away from slot" regime and restricted their rel 12 to this regime which is sufficiently narrow to conceal the violent disagreement between rel 12 and the data. (The violent disagreement is indicated by the fact that, at large values of $G_c s/xG_g$, rel 12 indicates that η aproaches infinity whereas the data indicates that η approaches one!!!) But if we ignore the regimes invented by the authors and ask simply what $\eta\{x\}$ functionality is indicated by ALL the authors' published data, the answer is

$$\eta \propto \frac{1}{1 + ax} \quad (17)$$

which obviously bears little resemblance to the $\eta\{x\}$ functionality in rel 12, particularly at small values of x.

And what is the point of this discussion? It is to demonstrate that the methods of the old heat transfer and the old engineering place NO EMPHASIS on the importance of obtaining correlations which accurately describe the parametric functionality of the real world. Is the experiment by Hartnett, Birkebak, and Eckert an isolated example? Or does it truly reflect the general application of the methods of the old heat transfer? One has only to leaf through any journal on the old heat transfer to be convinced that it truly reflects the general application. Virtually without exception, no effort is ever made in the old heat transfer to demonstrate agreement between correlation functionality and experiment functionality.

What difference does it make whether or not correlations accurately describe parametric functionality? It makes the difference between equipment design based on engineering science and equipment design based on trial-and-failure. It makes the difference between optimum design and poor design. In short, it makes the difference between science and art.

Why will the altogether new correlations of the new engineering accurately describe parametric functionality? Because, as described in Book 2, the R & D methodology of the new heat flow is altogether different than that of the old heat transfer. The difference is that the new R & D methodology deemphasizes speculation and emphasizes real world behavior, and this difference makes it NECESSARY and SIMPLE to determine parametric functionality by observing and correlating real world behavior rather than by speculation.

If the ref 1 experiment were reported within the framework of the new engineering, the authors would report that they could make NO STATEMENT about the functionality between η and s, or between η and G_c, or between η and G_g because they had failed to investigate the effect of s, G_c, and G_g on η. They would have to report that

they had investigated only the functionality between η and x, and only at one level of s, G_c, and G_g. And of course there would be some question as to whether this restricted experiment was of sufficient scope to permit publication of the results.

In summary, the R & D methods of the new engineering lead to correlations which accurately describe real world behavior, and that is why correlations obtained within the framework of the new engineering will represent a marked improvement over their old engineering counterparts.

MAKING CORRELATIONS SIMPLER TO USE

Engineering correlations are intended to be used in the design and analysis of equipment, and it is self-evident that they should be as simple as possible to use. Yet, in the old heat transfer, very little effort is expended in making correlations simple to use in the design and analysis of equipment. What often happens is that the researcher generates the correlation in a manner which is convenient for himself rather than in a manner which would be convenient for the equipment designer/analyst.

For example, in the old heat transfer, many heat transfer coefficient correlations are based on bulk fluid properties such as C, μ, k in spite of the fact that these are not the parameters of immediate concern to designers/analysts. Temperature T and pressure P ARE parameters of immediate concern to designers/analysts, and since C, μ, k are determined by T and P, it would be possible to write C, μ, k functions for a given fluid in terms of T, P functions. For example, the old expression

$$Nu = .023\ Re^{.8}\ Pr^{.4} \tag{18}$$

is of the form

$$h = .023\ G^{.8}\ D^{-.2}\ f\{C,\mu,k\} \tag{19}$$

Since C, μ, k are defined by T and P, it would be possible to write rel 19 in the form

$$h \rightarrow .023\ G^{.8}\ D^{-.2}\ f\{T,P\} \tag{20}$$

for any fluid. Moreover, since C, μ, k are only weakly dependent on P, we could closely approximate rel 20 with a relation of the form

$$h \rightarrow .023\ G^{.8}\ D^{-.2}\ f\{T\} \tag{21}$$

$$q \rightarrow .023\ G^{.8}\ D^{-.2}\ f\{T\}\ \Delta T \tag{22}$$

Relations 21 (old way) and 22 (new way) would be very simple and convenient to use. (Book 2, Ch 9 presents water and air correlations in the form of rel 22, and describes how to convert rel 19 to the form of rels 21 and 22.) Note that rels 21 and 22 IN NO WAY depend on fluid properties. This means that, if we were writing a computer program for the design/analysis of water/air heat exchangers, rels 21 and 22 would enable us to omit the six subroutines normally required to generate C, μ, and k for water and for air.

Another example is the dimensionless group x/Ms (which is really xG_g/G_cs) often used in the old heat transfer to correlate film cooling data. This group strongly suggests to the designer/analyst that x, G_g, G_c, and s strongly influence film cooling behavior. Of these parameters, the designer of the film cooling system normally has control over the values of G_c and s. The

design problem normally involves obtaining the optimum design, and in this case, optimum usually means use the minimum possible coolant. Thus the design problem reduces to

> What are the optimum values of G_c and s--ie what values of G_c and s will give the required amount of cooling with the minimum amount of coolant?

To determine these values, we must somehow relate the cooling flow W_c to the group x/Ms. This requires that we note the following:

$$M = G_c/G_g \tag{23}$$

$$G_c = W_c/\text{slot area} = W_c/sL = W_c'/s \tag{24}$$

$$\therefore\ x/Ms = xG_g/G_cs = xG_g/W_c' \tag{25}$$

Relation 25 tells us that expressions of the form $\eta\{x/Ms\}$ are really of the form $\eta\{xG_g/W_c'\}$ and that the value of x/Ms IN NO WAY depends on the value of M or s!!! Therefore correlations of the form $\eta\{x/Ms\}$ actually state that film cooling behavior IN NO WAY depends on G_c or s--in spite of the fact that they strongly suggest that film cooling behavior importantly depends on G_c and s!!! And this tells us that correlations of the form $\eta\{x/Ms\}$ should always be written in the form $\eta\{x/G_g/W_c'\}$ in order that they will seem to say what they actually do say, and thus will be as simple as possible to use.

The point of this discussion is to illustrate that the difficulty involved in applying a correlation to the design and analysis of equipment is largely controlled by the developer of the correlation. In the new heat flow, a conscious effort is made to obtain correlations which are as simple as possible to use. This will be accomplished by requiring that the researchers who develop correlations be familiar with what is involved in designing and analyzing engineering equipment.

CONCLUSIONS

For the remainder of this book, we will PRETEND that the correlations of the old heat transfer accurately describe real world behavior and we will simply translate them into the language of the new heat flow. But it should be born in mind that the R & D methods of the old heat transfer oftentimes result in correlations which little resemble real world behavior, and it should be expected that correlations obtained within the framework of the new heat flow will be quite different from those obtained by translating the old correlations.

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1. J. P. Hartnett, R. C. Birkebak, and E. R. G. Eckert, Velocity distributions, temperature distributions, effectiveness and heat transfer for air injected through a tangential slot into a turbulent boundary layer, ASME Journal of Heat Transfer, pp 293-306, August 1961

CHAPTER 3 THE THERMAL DESIGN/ANALYSIS OF SIMPLE SHELL AND TUBE HEAT EXCHANGERS

INTRODUCTION

This chapter deals with the thermal design and analysis of "simple" shell and tube heat exchangers using the new heat flow and its free form concept of thermal behavior. A "simple" heat exchanger is taken to mean one in which the heat flow fluids, interfaces, and walls exhibit the simple thermal behavior described by

a. $q_{interfaces} \propto \Delta T_{interfaces}$

b. $q_{wall} \propto \Delta T_{wall}$

c. Proportionality in a and b is independent of location

d. Wall thickness is uniform

e. C_{fluid} is independent of temperature.

This type of behavior is so simple that even the old heat transfer can deal with it effectively, and the problems in this chapter were correctly solved long ago using the old heat transfer. The reason for this old way effectiveness is that the thermal behavior described above is PROPORTIONAL and this is the ONLY type of behavior with which the old heat transfer can deal effectively because the old heat transfer is itself based on a proportional concept. On the other hand, the new heat flow deals effectively with ALL forms of thermal behavior because it is based on a free form concept which assumes whatever functionality is described by the phenomenon being considered.

The free form concept of the new heat flow is contained in the general and simple expression

$$q \rightarrow f\{\underline{T}\text{hermal } \underline{D}\text{riving } \underline{F}\text{orce}\} \quad (1)$$

In this chapter, we deal EXCLUSIVELY with proportional thermal behavior and therefore this general, free form expression takes the particular forms

$$q \propto TDF \quad (2)$$

$$q \rightarrow \text{constant} \cdot TDF \quad (3)$$

$$q_{conv} \rightarrow \text{constant} \cdot \Delta T_{interface} \quad (4)$$

$$q_{tran} \rightarrow \text{constant} \cdot \frac{dT}{dx} \quad (5)$$

Since these particular forms of the new heat flow closely resemble the general forms of the old heat transfer, we should expect (and will find) many close parallels between the new and the old ways of solving the design/analysis problems which are the subject of this chapter. However, it must be emphasized that relations 2 through 5 are merely particular forms which we are using in this chapter ONLY BECAUSE this chapter deals with thermal behavior which is strictly proportional.

In this chapter, we solve problems which illustrate the thermal design/analysis of simple heat exchangers using the new heat flow. We begin with problems involving a double pipe heat exchanger and then we go on to discuss more complex geometries. We concentrate on those problem areas which are handled differently in the new heat flow, and we just briefly discuss those problem areas which are handled in essentially the same way in both the new heat flow and the old heat transfer.

THE OVERALL THERMAL PERFORMANCE OF "SIMPLE" DOUBLE PIPE HEAT EXCHANGERS

In the old heat transfer, the overall thermal performance of "simple" double pipe heat exchangers is given by

$$Q = U\ A\ \Delta T_{LM} \quad (6)$$

where Q is the total heat flow rate from one fluid stream to the other, A is the total heat flow area in the exchanger, and ΔT_{LM} is the log mean temperature difference. Relation 6 is obtained by integrating the old heat transfer expression

$$dQ = U\ \Delta T_t\ dA \quad (7)$$

In the new heat flow, the overall thermal performance of "simple" double pipe heat exchangers is given by

$$Q = q_{LM}\ A \quad (8)$$

where q_{LM} refers to the local heat flux evaluated at the log mean temperature difference--ie

$$q_{LM} = q\{\Delta T_t = \Delta T_{LM}\} \quad (9)$$

Equation 8 is obtained by integrating the new way expression

$$dQ = q\ dA \quad (10)$$

(We are not going to bother with the details of integrating eqs 7 and 10 to obtain eqs 6 and 8.

For those who are interested, the integration of eq 7 to obtain eq 6 is described in most texts on the old heat transfer, and the integration of eq 10 to obtain eq 8 is quite similar.) Notice that eq 8 can be interpreted as a statement that the total heat flow equals the average heat flux times the heat flow area, and in this case the average heat flux happens to be the heat flux which would result from $\Delta T_t = \Delta T_{LM}$.

Equation 8 is in fact the new way translation of eq 6. Both equations apply to the same problems. Both are useful ONLY when the heat flow fluids, interfaces, and walls exhibit the simple thermal behavior described on page 3-1. We will have considerable use for eq 8 in this chapter which deals with proportional thermal behavior--and no use for eq 8 outside this chapter where we will generally deal with "nonproportional" thermal behavior.

DESIGN/ANALYSIS PROBLEMS INVOLVING "SIMPLE" DOUBLE PIPE HEAT EXCHANGERS

Now let us solve some more or less typical design/analysis problems involving "simple" double pipe heat exchangers where "simple" has the meaning described on page 3-1. Rather than solve the problems using both the old heat transfer and the new heat flow, let us forget about the old heat transfer and consider only the new way solutions. In other words, let us forget about the solutions based on eq 6 and consider only the solutions based on eq 8.

PROBLEM 1 A double pipe heat exchanger is to be designed to heat Fluid A to a temperature of 360 F. Given the design specs below, how long must the exchanger be?

Given The inner pipe of the exchanger is described by

$$D_i = .06 \qquad D_o = .07$$

The outer pipe of the exchanger is described by

$$D_i = .09 \qquad D_o = .10$$

The fluid streams are to be countercurrent. Fluid A is on the tube side, Fluid B is on the shell side. The fluid streams are described by

	Fluid A	Fluid B
W	1200	1600
T_{in}	200	400
C	0.60	0.95

The thermal behavior of the stream/wall interfaces is described by

$$q_A \rightarrow 480\ \Delta T_A \qquad (11)$$

$$q_B \rightarrow 620\ \Delta T_B \qquad (12)$$

The thermal behavior of the wall material is described by

$$q_w \rightarrow -13\ \frac{dT}{dx} \qquad (13)$$

Problem 1 cont.

Given In cylindrical coordinates, rel 13 is

$$q_w \rightarrow 13 \frac{\Delta T_w}{r \ln(r_o/r_i)} \qquad (14)$$

$$q_{w,i} \rightarrow 13 \frac{\Delta T_w}{r_i \ln(r_o/r_i)} \qquad (15)$$

where $q_{w,i}$ is the heat flux at the inner wall. Rel 15 gives, for the inner pipe,

$$q_{w,i} \rightarrow 2811\ \Delta T_w \qquad (16)$$

Solution Calculate Q, the total heat flow, by making a heat balance on Fluid A:

$$Q = W_A C_A \Delta T_A = 1200(0.60)(360 - 200) \qquad (17)$$

$$\therefore Q = 115{,}200$$

Calculate $T_{B,out}$ by making a heat balance on Fluid B:

$$T_{B,out} = T_{B,in} - \frac{Q}{W_B C_B} \qquad (18)$$

$$\therefore T_{B,out} = 400 - \frac{115{,}200}{1600(.95)} = 324$$

Calculate the relationship between q and ΔT_t in the exchanger--ie calculate $q\{\Delta T_t\}$:

Problem 1 cont.

Solution At any point within the exchanger, the total temperature difference ΔT_t is given by

$$\Delta T_t = T_A - T_B = \Delta T_A + \Delta T_w + \Delta T_B \qquad (19)$$

where ΔT_A refers to the temperature difference across the wall/Fluid A interface, etc. Combining eq 19 with the thermal behavior described by eqs 11, 12, and 16 gives

$$\Delta T_t \rightarrow \frac{q_A}{480} + \frac{q_{w,i}}{2811} + \frac{q_B}{620} \qquad (20)$$

Since the diameter of the A interface is .06 and that of the B interface is .07, and since the wall heat flux in eq 20 is evaluated at the inner diameter, we may write

$$q_A = q_{w,i} = \frac{.07}{.06} q_B = q_i \qquad (21)$$

Combining eqs 20 and 21 gives

$$\Delta T_t \rightarrow \frac{q_i}{480} + \frac{q_i}{2811} + .857 \frac{q_i}{620} \qquad (22)$$

$$\therefore q_i \rightarrow 262\ \Delta T_t \qquad (23)$$

Calculate the log mean temperature difference, ΔT_{LM}:

Problem 1 cont.

Solution From the given information and the result of eq 18, the temperature differences at the ends of the exchanger are (324 - 200) and (400 - 360). Therefore the log mean temperature difference is given by

$$\Delta T_{LM} = \frac{124 - 40}{\ln(124/40)} = 74.2 \qquad (24)$$

Calculate $q_{i,LM}$--ie calculate $q_i\{\Delta T_t = \Delta T_{LM}\}$:

$$q_{i,LM} \rightarrow 262 \times 74.2 = 19{,}440 \qquad (25)$$

Calculate the heat flow area in the required exchanger based on the inner diameter of the inner pipe, and calculate the length of the required exchanger:

$$Q = q_{i,LM} A_i \qquad (26)$$

$$\therefore A_i = 115{,}200/19440 = 5.93 \qquad (27)$$

$$A_i = \pi D_i L \qquad (28)$$

$$\therefore L = 5.93/.06\pi = 31.5 \qquad (29)$$

Answer In order to heat Fluid A to a temperature of 360 F, the subject heat exchanger must be 31.5 feet long.

PROBLEM 2 Describe the overall thermal behavior of the heat exchanger in Problem 1--ie describe how the total heat flow Q is related to the overall thermal driving force TDF_{OA}--ie determine $Q\{TDF_{OA}\}$.

Solution The overall thermal driving force in Problem 1 is given by

$$TDF_{OA} = T_{B,in} - T_{A,in} \qquad (30)$$

Let us call this temperature difference ΔT_{OA}. In Problem 1, we found that

$$Q\{\Delta T_{OA}=200\} = 115{,}200 \qquad (31)$$

Since we know that the heat exchanger in Problem 1 behaves in a proportional way, eq 31 tells us that

$$Q \rightarrow \frac{115{,}200}{200} \Delta T_{OA} = 576 \ \Delta T_{OA} \qquad (32)$$

Answer The overall thermal behavior of the heat exchanger in Problem 1 is described by

$$Q \rightarrow 576 \ \Delta T_{OA} \qquad (33)$$

where ΔT_{OA} is the temperature difference between the two incoming streams. The overall thermal behavior is also shown graphically in Figure 1, below.

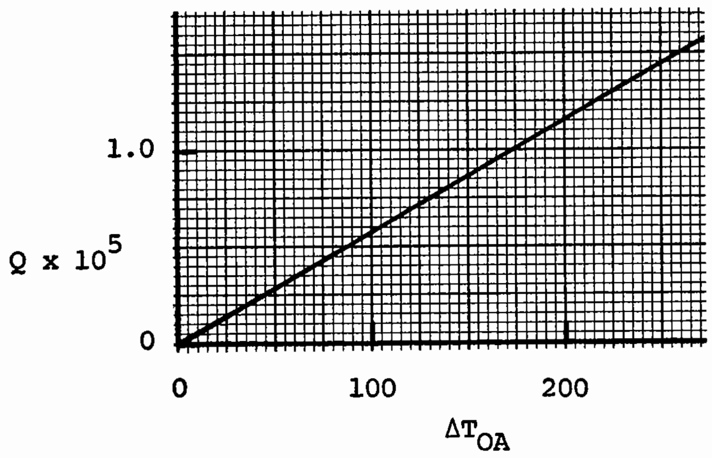

FIGURE 1

Overall Thermal Behavior of the Problem 1 Heat Exchanger

DISCUSSION OF PROBLEMS 1 AND 2

The new way solution of Problem 1 is conceptually simpler and mathematically more straightforward than the old way solution would have been. Notice that we had no use for the old concept of the "heat transfer coefficient", and that we calculated q in a direct way rather than the old roundabout way of first calculating the ratio $q/\Delta T_t$ (otherwise known as "U").

Problem 2 introduces the subject of the overall thermal behavior of equipment, a subject seldom (if ever) mentioned in texts on the old heat transfer. It is a subject of first order importance in the new heat flow, and one which we will frequently take up in the remainder of this book. Its importance results from the fact that it reveals a great deal about the static and dynamic performance of the equipment, as suggested by Problems 3, 4, and 5.

PROBLEM 3 Suppose the heat exchanger in Problem 1 were installed and operating as intended. What changes would result if the inlet temperature of Fluid A increased to 250 F and remained there?

Solution If the inlet temperature of Fluid A increased to 250 F, then the temperature difference between the incoming streams would be given by

$$\Delta T_{OA} = 400 - 250 = 150 \tag{34}$$

Relation 33 indicates that this value of ΔT_{OA} would result in a total heat flow of

$$Q \rightarrow 576\ \Delta T_{OA} = 576(150) = 86{,}400 \tag{35}$$

This value of heat flow would result in fluid outlet temperatures given by

$$T_{A,out} = 250 + \frac{86{,}400}{1200(.60)} = 370 \tag{36}$$

$$T_{B,out} = 400 - \frac{86{,}400}{1600(.95)} = 343 \tag{37}$$

Answer If the Fluid A inlet temperature increased to 250 F, the heat flow in the exchanger would decrease to 86,400 B/hr, the outlet temperature of Fluid A would increase to 370 F, and the outlet temperature of Fluid B would increase to 343 F.

PROBLEM 4 Suppose the heat exchanger in Problem 1 were installed and operating as intended. Describe the functionality between the outlet and inlet temperatures of Fluid A--ie determine $T_{A,out}\{T_{A,in}\}$. Repeat for Fluid B.

Solution The inlet and outlet temperatures of Fluid A are related by

$$T_{A,out} = T_{A,in} + \frac{Q}{W_A\ C_A} \tag{38}$$

Combining eqs 38 and 33 gives

$$T_{A,out} = T_{A,in} + \frac{576(T_{B,in} - T_{A,in})}{1200(.60)} \tag{39}$$

$$\therefore\ T_{A,out} = .20\ T_{A,in} + .80\ T_{B,in} \tag{40}$$

Repeating the above for Fluid B gives

$$T_{B,out} = .62\ T_{B,in} + .38\ T_{A,in} \tag{41}$$

Answer In the Problem 1 heat exchanger, the functionality between the inlet and outlet temperatures of Fluid A is described by eq 40. For Fluid B, this functionality is described by eq 41.

PROBLEM 5 Suppose the heat exchanger in Problem 1 were installed and operating as intended. Determine the following thermal characteristics of the exchanger:

$$\frac{dQ}{d\Delta T_{OA}} \qquad \frac{dQ}{dT_{A,in}} \qquad \frac{dQ}{dT_{B,in}}$$

$$\frac{dT_{A,out}}{dT_{A,in}} \qquad \frac{dT_{A,out}}{dT_{B,in}}$$

Solution Differentiation of rel 33 gives

$$\frac{dQ}{d\Delta T_{OA}} \rightarrow 576 \tag{42}$$

Recall that rel 33 is actually

$$Q \rightarrow 576(T_{B,in} - T_{A,in}) \tag{43}$$

$$\therefore\ \frac{dQ}{dT_{A,in}} \rightarrow -576 \quad \text{and} \tag{44}$$

$$\frac{dQ}{dT_{B,in}} \rightarrow 576 \tag{45}$$

Differentiation of eq 40 gives

$$\frac{dT_{A,out}}{dT_{A,in}} = .20 \quad \text{and} \tag{46}$$

Problem 5 cont.

Solution

$$\frac{dT_{A,out}}{dT_{B,in}} = .80 \qquad (47)$$

Answer If the heat exchanger in Problem 1 were installed and operating as intended, it would exhibit the thermal characteristics described by rels 42, 44, 45, 46, and 47.

DISCUSSION OF PROBLEMS 3, 4, AND 5

Problems 3, 4, and 5 illustrate the importance and usefulness of the new way concept of "overall thermal behavior". This simple concept is contained in the generic relation

$$Q \rightarrow f\{TDF_{OA}\} \qquad (48)$$

(Notice that rel 48 is the logical outgrowth of and closely resembles the fundamental relation of the new heat flow,

$$q \rightarrow f\{TDF\} \qquad (49)$$

where q and TDF refer to local values.) When dealing with "simple" heat exchangers, rel 48 takes the general form

$$Q \rightarrow constant(T_{h,in} - T_{c,in}) \qquad (50)$$

where the temperatures refer to the hot and cold incoming fluid streams. For example, the overall thermal behavior of the heat exchanger in Problem 1 is described by the relation

$$Q \rightarrow 576(T_{B,in} - T_{A,in}) \qquad (51)$$

The overall thermal behavior is a concise description of the steady-state thermal behavior of heat exchangers, and it is much more useful than its old heat transfer counterpart. Recall that in the old heat transfer, the steady-state heat transfer behavior of a "simple" heat exchanger is usually described in terms of its "UA" value. For example, the UA value of the Problem 1 heat exchanger is (from eqs 23 and 27)

$$UA = 262(5.93) = 1554 \text{ B/hr F} \qquad (52)$$

and thus the steady-state heat transfer behavior is described by

$$Q = 1554\ \Delta T_{LM} \qquad (53)$$

Notice that rel 51 is much more useful and informative than its old heat transfer counterpart, eq 53. For example, given that the incoming stream temperatures are 425 F and 227 F, rel 51 tells us that the total heat flow in the exchanger is 114,000 B/hr. On the other hand, given this same information, it is IDENTICALLY IMPOSSIBLE to determine Q from eq 53!

The overall thermal behavior describes how the equipment responds to changes in the steady-state values of operating parameters, and the derivatives in Problem 5 are a useful way to describe the response characteristics. It should be emphasized that the derivatives in Problem 5 are "steady-state derivatives"--ie they are defined in a steady-state sense. For example,

$$\frac{dQ}{d\Delta T_{OA}} = \frac{\Delta Q_{ss}}{\Delta\Delta T_{OA,ss}} \quad \text{as } \Delta\Delta T_{OA,ss} \to 0 \qquad (54)$$

where the subscript ss stands for steady-state. The derivatives in Problem 5 actually describe the "dynamic, steady-state behavior" of the Problem 1 heat exchanger. This terminology may seem more rational if one notes that the information

$$\frac{dQ}{dT_{A,in}} \to -576 \quad \text{and} \quad \frac{dT_{A,out}}{dT_{A,in}} = .20$$

tell us that a 4F increase in $T_{A,in}$ would result in

$$\Delta Q = \frac{dQ}{dT_{A,in}} (\Delta T_{A,in}) = -576(4) = -2300 \qquad (55)$$

$$\Delta T_{A,out} = .20(4) = .80 \qquad (56)$$

In other words, the equipment response to a 4F increase in $T_{A,in}$ would be a decrease of 2300 B/hr in the heat flow and an increase of 0.8F in $T_{A,out}$.

The concept of overall thermal behavior is not within the "knowledge envelope" of the old heat transfer, and this is readily verified by reference to the many texts on the old heat transfer. It is not surprising that overall thermal behavior is not part of the old heat transfer, since it simply does not fit in with the old heat transfer coefficient concept. Notice that if we were to force rel 50 into the heat transfer coefficient mold, we would write

$$Q = U^* A(T_{h,in} - T_{c,in}) \qquad (57)$$

Based on eq 57, the definition of U^* would be

$$U^* = \frac{Q}{A(T_{h,in} - T_{c,in})}$$

and this definition is altogether different than the definition of U in the old heat transfer. Thus, if the concept of overall thermal behavior were forced into the old heat transfer coefficient mold, the result would be two altogether different types of heat transfer coefficients, and this would obviously be unsatisfactory. On the other hand, as noted above, the concept of overall thermal behavior is a logical result in the new heat flow.

It should be noted that the effectiveness-NTU method of the old heat transfer leads to the equation

$$Q = \varepsilon\,(WC)_{min}\,(T_{h,in} - T_{c,in}) \qquad (58)$$

and that this equation is of the same form as rel 50. Notice, however, that there are no heat transfer coefficients in eq 58 and therefore that eq 58 is also part of the new heat flow. (We will use eq 58 in one of the later problems in this chapter.)

In summary, Problems 3, 4, and 5 describe the concept of "overall thermal behavior" and illustrate its usefulness and application. This concept is useful even when dealing with "simple" heat exchangers, but its real importance lies in improving our ability to deal with "nonsimple" heat exchangers--as we shall see in the later chapters of this book.

PROBLEM 6 Suppose that the design specs in Problem 1 had included a design allowance for a fouling deposit on the inner surface of the inner pipe, and that the thermal behavior of this deposit was described by

$$q_f \rightarrow 1200\ \Delta T_f \tag{58}$$

What would have been the required length of the Problem 1 heat exchanger?

Solution The fouling causes rel 20 to become

$$\Delta T_t \rightarrow \frac{q_A}{480} + \frac{q_f}{1200} + \frac{q_{w,i}}{2811} + \frac{q_B}{620} \tag{59}$$

Since the fouling deposit is on the inner surface of the inner pipe, eq 21 becomes

$$q_A = q_f = q_{w,i} = \frac{.07}{.06} q_B = q_i \tag{60}$$

Combining rels 59 and 60 gives

$$\Delta T_t \rightarrow \frac{q_i}{480} + \frac{q_i}{2811} + \frac{.857 q_i}{620} + \frac{q_i}{1200} \tag{61}$$

$$\therefore\ q_i \rightarrow 215\ \Delta T_t \tag{62}$$

The design value of the log mean temperature difference is unaffected by the fouling. Therefore eq 24 is applicable and $q_{i,LM}$ is given by

$$q_{i,LM} \rightarrow 215(74.2) = 16000 \tag{63}$$

Problem 6 cont.

The design heat flow rate is unaffected by the fouling and is 115,200 B/hr as in Problem 1 (see eq 17). Therefore A_i is given by

$$A_i = \frac{Q}{q_{i,LM}} = \frac{115,200}{16,000} = 7.20 \tag{64}$$

$$\therefore\ L = \frac{7.2}{.06\pi} = 38.2 \tag{65}$$

Answer If the design specs in Problem 1 had included a design allowance for a fouling deposit described by rel 58, the required length of the heat exchanger would have been 38.2 ft.

DISCUSSION OF PROBLEM 6

In the old heat transfer, fouling deposits are usually dealt with in terms of "fouling factors"--ie in terms of thermal resistance--ie in terms of the value of 1/h. In the new heat flow, fouling deposits are dealt with in terms of their thermal behavior--ie in terms of $q_f\{\Delta T_f\}$. Problem 6 illustrates the new way of dealing with fouling deposits in terms of their thermal behavior.

PROBLEM 7 If the heat exchanger in Problem 6 were installed and operating as intended, what values of Q and $T_{A,out}$ would be expected in the unfouled condition?

Solution The solution of the problem is based on noting that

$$Q = q_{i,LM} A_i \tag{66}$$

From Problem 1, rel 23, q_i in the unfouled condition is given by

$$q_i \rightarrow 262\ \Delta T_t \tag{67}$$

$$\therefore\ q_{i,LM} \rightarrow 262\ \Delta T_{LM} \tag{68}$$

Combining rels 64, 66, and 68, we obtain

$$Q \rightarrow 1886\ \Delta T_{LM} \tag{69}$$

The value of ΔT_{LM} is determined by the fluid inlet and outlet temperatures. We are given the inlet temperatures and we establish the outlet temperature of Fluid B in terms of the outlet temperature of Fluid A by noting that

$$Q = W_A\ C_A\ (T_{A,out} - T_{A,in}) \tag{70}$$

$$\therefore\ Q = 1200(.60)(T_{A,out} - 200) \tag{71}$$

Problem 7 cont.

Solution A heat balance on Fluid B gives

$$T_{B,out} = T_{B,in} - \frac{Q}{W_B\ C_B} \tag{72}$$

Combining eqs 71 and 72 and recalling that W_B is 1600 and that C_B is 0.95 gives

$$T_{B,out} = 495 - .474\ T_{A,out} \tag{73}$$

Using eq 73 and recalling that $T_{A,in}$ is 200 and $T_{B,in}$ is 400, we obtain

$$\Delta T_{LM} = \frac{(295-.474T_{A,out})-(400-T_{A,out})}{\ln \dfrac{(295-.474T_{A,out})}{(400-T_{A,out})}} \tag{74}$$

Combining rels 69 and 71, we obtain

$$\Delta T_{LM} = .3818(T_{A,out} - 200) \tag{75}$$

Combining rels 74 and 75, we obtain

$$.3818(T_{A,out} - 200) =$$

$$\frac{(295-.474T_{A,out})-(400-T_{A,out})}{\ln \dfrac{(295-.474T_{A,out})}{(400-T_{A,out})}} \tag{76}$$

Problem 7 cont.

Solution The solution of eq 76 gives

$$T_{A,out} = 370 \tag{77}$$

and therefore eq 71 indicates that

$$Q = 1200(.60)(370 - 200) = 122{,}400 \tag{78}$$

Answer If the heat exchanger in Problem 6 were installed and operating as intended, the expected heat flow would be 122,400 B/hr and the expected outlet temperature of Fluid A would be 370 F, in the unfouled condition.

DISCUSSION OF PROBLEM 7

Problem 7 is the first problem in which we have had to determine an outlet temperature by working with the log mean temperature difference. (In the preceding problems, we were able to determine the unknown temperatures by making heat balances.) As indicated by eq 76, the solution of this type problem leads to an equation which must be solved iteratively if the solution is based on eq 66. The old heat transfer avoids the iteration by using the effectiveness-NTU method. As shown in Problem 8, this method is readily translated to and applied in the new heat flow.

PROBLEM 8 Translate the effectiveness-NTU method of the old heat transfer into the language of the new heat flow. Use the results to solve Problem 7.

Solution The effectiveness-NTU method of the old heat transfer is based on dealing with the ratio Q/Q_{max} where Q refers to the actual heat transfer rate and Q_{max} refers to the maximum possible heat transfer rate that could take place between the two streams--ie Q_{max} refers to the heat transfer rate that would result if the heat exchanger area were infinite. This ratio is given the name "effectiveness" and the symbol ε. The usefulness of the effectiveness concept is based on noting that

$$Q = \varepsilon\, Q_{max} \tag{79}$$

$$Q_{max} = WC_{min}(T_h - T_c)_{in} \tag{80}$$

$$\therefore\ Q = \varepsilon\, WC_{min}(T_h - T_c)_{in} \tag{81}$$

where WC_{min} refers to the stream which has the smaller value of WC and the subscript "in" refers to the inlet condition. The value of the effectiveness is determined by the two parameter groups

$$AU/WC_{min} \qquad WC_{min}/WC$$

where the WC without subscript refers to the fluid stream which has the larger value of WC. Both groups are dimensionless. The first group is generally named the

Problem 8 cont.

Solution "number of heat transfer units" and is abbreviated NTU. The second group is not usually given a name. For a double pipe heat exchanger, the functionality between the dimensionless groups ε, NTU, and WC_{min}/WC is given by

$$\varepsilon_{\uparrow\downarrow} = \frac{1-\exp[-NTU(1-WC_{min}/WC)]}{1-(WC_{min}/WC)\exp[-NTU(1-WC_{min}/WC)]} \qquad (82)$$

$$\varepsilon_{\uparrow\uparrow} = \frac{1-\exp[-NTU(1+WC_{min}/WC)]}{1+WC_{min}/WC} \qquad (83)$$

where the subscript ↑↓ denotes a counterflow heat exchanger and the subscript ↑↑ denotes parallel flow.

In the new heat flow, we will of course obtain essentially the same results, but the language and emphasis are somewhat different. We do not assign names to parameter groups and so we abandon the names "effectiveness" and "number of heat transfer units" and also their shorthand counterparts, ε and NTU. We do not have the old U--but since we are dealing with proportional thermal behavior, we do have the proportionality constant between q and ΔT_t. Let us assign this constant the symbol "M". The translation of eqs 79 through 83 is as follows. Equation 79 is unnecessary (since it is really the defining equation for ε which we no longer have). Equation 80 is the same in both the old and the new ways. Equation 81 becomes

Problem 8 cont.

Solution $$Q = (Q/Q_{max})(WC_{min})(T_h - T_c)_{in} \qquad (84)$$

Equations 82 and 83 become

$$Q_{\uparrow\downarrow} \to \frac{1-\exp[-AM/WC_{min}(1-WC_{min}/WC)]}{1-(WC_{min}/WC)\exp[\text{as above}]}(WC_{min})(T_h - T_c)_{in} \qquad (85)$$

$$Q_{\uparrow\uparrow} \to \frac{1-\exp[-AM/WC_{min}(1+WC_{min}/WC)]}{1+WC_{min}/WC}(WC_{min})(T_h - T_c)_{in} \qquad (86)$$

Notice that rels 85 and 86 describe the overall thermal behavior of the equipment--ie they are in the form

$$Q \to f\{TDF_{OA}\} \qquad (87)$$

which, for "simple" exchangers, gives

$$Q \to \text{constant } (T_h - T_c)_{in} \qquad (88)$$

Relation 85 is used to solve Problem 7 by recalling that $A = 7.20$, $M = 262$, $WC_{min} = 1200(.60) = 720$, $WC = 1600(.95) = 1520$, $T_{h,in} = 400$, $T_{c,in} = 200$. Substituting these values into rel 85 gives

$$Q = 122{,}400 \qquad (89)$$

$$\therefore T_{A,out} = 200 + \frac{122{,}400}{720} = 370 \qquad (90)$$

Problem 8 cont.

Answer The translation of the old effectiveness-NTU method results in relations 84, 85, and 86. Relation 84 is the general relation of the Q/Q_{max} method, 85 and 86 are particular relations which apply only to double pipe heat exchangers. It should be emphasized that both the old way and the new way translation apply ONLY to "simple" heat exchangers (see page 3-1).

DISCUSSION OF PROBLEM 8

In Problem 8, we translated the old effectiveness-NTU method to the new way and then applied it to the solution of a problem we had already solved using the q_{LM} method. The particular advantage of the "Q/Q_{max}" method described in Problem 8 is that it avoids the iterative solution which is required in the q_{LM} method whenever the fluid outlet temperatures cannot be determined from a heat balance (as was the case in Problem 7).

As illustrated in Problem 8, the Q/Q_{max} method is readily applied to the analysis of double pipe heat exchangers and results in an expression which describes the overall thermal behavior of the equipment.

PROBLEM 9 The heat exchanger in Problem 1 was designed for counterflow operation but, due to faulty communication, it was installed as a parallel flow exchanger. What heat flow rate and Fluid A outlet temperature should have resulted from parallel flow operation?

Solution The solution of this problem can be based on either the q_{LM} method of Problem 1 or the Q/Q_{max} method of Problem 8. Since neither outlet temperature is given, the q_{LM} method would require an iterative solution and so we will use the Q/Q_{max} method. Recalling from Problem 1 that $A = 5.93$, $M = 262$, $WC_{min} = 720$, $WC = 1520$, $T_{h,in} = 400$, $T_{c,in} = 200$, the solution of rel 86 gives

$$Q \rightarrow .6504(720)(200) = 93{,}700 \qquad (91)$$

$$\therefore T_{A,out} = 200 + \frac{93{,}700}{720} = 330 \qquad (92)$$

Answer Parallel flow operation of the Problem 1 heat exchanger should have resulted in a heat flow rate of 93,700 B/hr and a Fluid A outlet temperature of 330 F. (The counter-flow values were 115,200 B/hr and 360 F).

PROBLEM 10 A double pipe heat exchanger is to be added to a process system in order to recover the process heat added to a stream of water. Given the design specs below, perform a "simple" analysis to determine approximately how long the exchanger must be in order to recover 70% of the process heat. Comment on the accuracy/inaccuracy of the result.

Given The water enters the process system at 200F and exits at 400F. (With regard to the exchanger design, this means that the hot stream enters at 400F and the cold stream enters at 200F.)

The dimensions of the inner and outer pipes of the exchanger are to be

$D_i = .06 \qquad D_o = .07$

$D_i = .09 \qquad D_o = .10$

The exchanger is to be installed with the streams running countercurrent.

The water flow rate is 1000 lbs/hr. The water properties are listed on page 9-7 of Book 2.

The thermal behavior of the wall/water interfaces is described by

$$q_i \rightarrow .023 \frac{G^{.8}\ F^{.6}\ C^{.4}}{D^{.2}\ \mu^{.4}} \Delta T \qquad (93)$$

(Notice that rel 93 is the new way translation of the old expression

$$Nu = .023\ Re^{.8}\ Pr^{.4} \qquad (94)$$

Problem 10 cont.

Given The translation details are described in Chapter 2.)

The thermal behavior of the wall material is described by

$$q_w \rightarrow 13 \frac{dT}{dx} \qquad (95)$$

As in Problem 1, rel 95 leads to

$$q_{w,i} \rightarrow 2811\ \Delta T_w \qquad (96)$$

Basis of Solution This heat exchanger is not "simple" because the specific heat of the water and the thermal behavior of the wall/water interfaces (rel 93) are affected by temperature and therefore are not constant throughout the exchanger. However, we do know that water properties are well-behaved functions of temperature. Therefore, let us perform the analysis using the water properties which correspond to the average of the hot and cold inlet temperatures. Then, in order to appraise the accuracy of this result, let us repeat the analysis using the water properties which correspond to the hot inlet temperature and also the water properties which correspond to the cold inlet temperature. These repeat analyses will firmly establish the accuracy of the result since they will place upper and lower limits on the exchanger length required.

Problem 10 cont.

Solution The average of the hot and cold inlet temperatures is

$$T_{avg,in} = \frac{400 + 200}{2} = 300 \qquad (97)$$

At this temperature, the water properties are (from Bk 2, pg 9-7)

$F = .395$

$C = 1.026$

$\mu = .45$

Calculate G_i and G_o:

$$G_i = \frac{1000}{(\pi/4)(.06)^2} = 354{,}000 \qquad (98)$$

$$G_o = \frac{1000}{(\pi/4)(.09^2 - .07^2)} = 398{,}000 \qquad (99)$$

where G_i is the mass flow rate in the inner pipe and G_o is the mass flow rate in the outer pipe.

Calculate $q_i\{\Delta T_i\}$ and $q_o\{\Delta T_o\}$ from rel 93:

$$q_i \rightarrow .023 \frac{(354000)^{.8}(.395)^{.6}(1.026)^{.4}}{(.06)^{.2}(.45)^{.4}} \Delta T_i$$

$$q_i \rightarrow 884\ \Delta T_i \qquad (100)$$

Problem 10 cont.

Solution $$q_o \rightarrow .023 \frac{(398000)^{.8}(.395)^{.6}(1.026)^{.4}}{(.02)^{.2}(.45)^{.4}} \Delta T_o$$

$$q_o \rightarrow 1209\ \Delta T_o \qquad (101)$$

where $q_i\{\Delta T_i\}$ describes the thermal behavior of the inner wall/water interface and $q_o\{\Delta T_o\}$ describes the thermal behavior of the outer wall/water interface.

Calculate $q_i\{\Delta T_t\}$:

$$\Delta T_t = \Delta T_i + \Delta T_w + \Delta T_o \qquad (102)$$

Combining rels 96, 100, and 102 gives

$$\Delta T_t \rightarrow \frac{q_i}{884} + \frac{q_{w,i}}{2811} + \frac{q_o}{1209} \qquad (103)$$

From a consideration of the pipe diameters, we can write

$$q_i = q_{w,i} = \frac{.07}{.06} q_o \qquad (104)$$

Combining rels 103 and 104 gives

$$q_i \rightarrow 455\ \Delta T_t \qquad (105)$$

Relation 105 describes the relationship between the heat flux at the inner wall

Problem 10 cont.

Solution and the total temperature difference, and of course both refer to local values.

Calculate the outlet temperatures and ΔT_{LM} by noting that we are to recover 70% of the heat added in the process and therefore

$$T_{h,out} = T_{h,in} - .70(T_{h,in} - T_{c,in}) \qquad (106)$$

$$\therefore T_{h,out} = 400 - .70(400 - 200) = 260 \qquad (107)$$

$$T_{c,out} = 200 + .70(400 - 200) = 340 \qquad (108)$$

$$\therefore \Delta T_{LM} = 60 \qquad (109)$$

(Note that ΔT_t has the same local value throughout the exchanger because WC is the same value on both sides of the inner pipe and because the heat capacity is independent of temperature in this "simple" analysis.)

Calculate q_{LM} from rels 105 and 109:

$$q_{LM,i} \rightarrow 455(60) = 27300 \qquad (110)$$

Calculate Q by noting that

$$Q = W_h C_h (T_{h,in} - T_{h,out}) = 144000 \qquad (111)$$

Calculate A_i and L by noting that

Problem 10 cont.

Solution

$$Q = q_{LM,i} A_i \qquad (112)$$

$$\therefore A_i = \frac{144000}{27300} = 5.27 \qquad (113)$$

$$\therefore L = \frac{5.27}{\pi(.06)} = 28.0 \qquad (114)$$

Evaluating the water properties at 400F and repeating the above, we obtain

$$q_i \rightarrow 999 \ \Delta T_i \qquad (115)$$

$$q_o \rightarrow 1367 \ \Delta T_o \qquad (116)$$

$$q_{LM} = 30240 \qquad (117)$$

$$Q = 149400 \qquad (118)$$

$$L = 26.2 \qquad (119)$$

Evaluating the water properties at 200F and repeating the above, we obtain

$$q_i \rightarrow 715 \ \Delta T_i \qquad (120)$$

$$q_o \rightarrow 978 \ \Delta T_o \qquad (121)$$

$$q_{LM} = 22800 \qquad (122)$$

Problem 10 cont.

Solution $Q = 140600$ (123)

$L = 32.7$ (124)

Answer The "simple" analysis of this real world problem indicates that the double pipe heat exchanger should be about 30 feet long in order to recover 70% of the process heat added to the stream of water. The analysis further indicates that this length is within about 10% of the actual length required for 70% recovery. This accuracy seems acceptable since the inaccuracy in the interface correlations probably exceeds 10%.

DISCUSSION OF PROBLEM 10

Problem 10 illustrates the application of "simple" analyses to real world design/analysis problems. The "simple" criteria on page 3-1 NEVER apply to real world systems, and the question the designer/analyst must decide is "Are the deviations from the "simple" criteria serious enough to compromise the "simple" answer and thus require a "nonsimple" analysis?" In Problem 10, we answered this question by repeating the analysis with the water properties evaluated at the extreme temperatures, and this told us that the accuracy of the "simple" analysis was acceptable. This same method can be used whenever one is dealing with thermal behavior which is strictly proportional, as in Problem 10.

PROBLEM 11 Repeat Problem 10 using rel 125 to describe the thermal behavior of the wall/water interfaces.

Given The thermal behavior of the wall/water interfaces is described by

$$q \rightarrow .023 \frac{G^{.8}}{D^{.2}} (a + bT + cT^2)\ \Delta T \tag{125}$$

where $a = .2068$, $b = .002631$, $c = -2.218 \times 10^{-6}$.

With the single exception that rel 125 replaces rel 93, the design specs of Problem 10 apply equally to this problem.

Solution Relation 125 indicates that, at 300F, the thermal behavior of the wall/water interfaces is described by

$$q_i \rightarrow 884\ \Delta T_i \tag{126}$$

$$q_o \rightarrow 1209\ \Delta T_o \tag{127}$$

Analysis similar to that in Problem 10 leads to a heat exchanger length of 28.0 ft, demonstrating that rel 125 and rel 93 give exactly the same result. (Note that rels 126 and 127 are the same as 100 and 101)

At 400 F, relation 125 indicates that the thermal behavior of the wall/water interfaces is described by

$$q_i \rightarrow 1004\ \Delta T_i \tag{128}$$

Problem 11 cont.

Solution $q_o \rightarrow 1373\ \Delta T_o$ (129)

Analysis based on rels 128 and 129 leads to a heat exchanger length of 26.1 ft (compared to the Problem 10 result of 26.2 ft). (Note that the proportionality constants in rels 128 and 129 agree within a fraction of a per cent with the constants in rels 115 and 116.)

At 200 F, relation 125 indicates that the thermal behavior of the wall/water interfaces is described by

$$q_i \rightarrow 715\ \Delta T_i \tag{130}$$

$$q_o \rightarrow 978\ \Delta T_o \tag{131}$$

Analysis based on rels 130 and 131 leads to a heat exchanger length of 32.7 ft, exactly the same result we obtained in Problem 10. (Note that rels 130 and 131 are the same as 120 and 121.)

Answer The analysis and the result were little affected by the fact that the thermal behavior of the wall/water interfaces was based on rel 125 rather than rel 93. Based on rel 125, a heat exchanger length of 30 ft is required in order to recover approximately 70% of the process heat added to the stream of water.

DISCUSSION OF PROBLEM 11

Relation 125 is a highly specific relation which applies ONLY to water. We derived rel 125 in Bk 2, Ch 9 by ASSUMING that the old way expression

$$Nu = .023\ Re^{.8}\ Pr^{.4} \tag{132}$$

accurately applies to real world behavior and by noting that eq 132 is of the form

$$q \rightarrow .023\ \frac{G^{.8}}{D^{.2}}\ f\{T\}\ \Delta T \tag{133}$$

where f{T} is some unspecified function of T. We determined f{T} for water by noting that

$$f\{T\} \rightarrow \frac{F^{.6}\ C^{.4}}{\mu^{.4}} \tag{134}$$

and evaluating the right side of rel 134 using a water properties table. In other words, rel 125 is an approximation of rel 132 and was in fact derived from rel 132. As shown by problems 10 and 11, rel 125 is a very close approximation of rel 93, the new way translation of rel 132.

The particular advantage of rel 125 is that it is much more convenient to use than rel 132 or rel 93 because rel 125 IN NO WAY depends on water properties! If this seems to be only a small advantage, imagine that we wished to determine the thermal behavior of the wall/water interfaces in Problem 11 at 387F. If we use rel 93 or 132, we must go to a reference book which contains the properties of water, we may have to convert from centipoises to lbs/hr ft, and we will certainly have to interpolate between the entries in the tables. On the other hand, rel 125 tells us WITHOUT REFERENCE TO ANY OTHER MATERIAL that, at 387F, the thermal behavior of the inner wall/water interface in Problem 11 is described by

$$q_i \rightarrow 991\ \Delta T_i \qquad (136)$$

Notice also that it is much easier to program rel 125 than rel 93. Three subroutines would be required in order to program the thermal behavior of wall/water interfaces based on rel 93--subroutines would be required in order to generate F{T}, C{T}, and μ{T}. On the other hand, the thermal behavior of wall/water interfaces based on rel 125 can be programmed much more readily since three constants take the place of the three subroutines. (The .023 in rel 125 should be combined with a, b, c to limit the number of constants to three. Also, rel 133 should be written

$$q \rightarrow \frac{G^{.8}}{D^{.2}}\ f\{T\}\ \Delta T \qquad (137)$$

The .023 in rels 125 and 133 was retained merely to clarify the development of the f{T} method of expressing thermal behavior correlations.)

The assumption that eq 132 accurately describes real world behavior makes it possible to determine the temperature function in rel 137 for all fluids whose physical properties are known. For some time to come, temperature functions for various fluids will be based on this assumption (or one like it involving another generalized correlation). However, this assumption will gradually be replaced by experimental determination of the temperature functions, and these experiments may very well show that the real world temperature functions little resemble those suggested by the functionality in eq 132.

THE THERMAL DESIGN/ANALYSIS OF "SIMPLE" SHELL AND TUBE HEAT EXCHANGERS

The thermal design/analysis of "simple" (see page 3-1) shell and tube heat exchangers is quite similar to the thermal design/analysis of "simple" double pipe heat exchangers. In both types of equipment, the fundamental relation is

$$q \rightarrow f\{\text{Thermal Driving Force}\} \qquad (138)$$

which, because of the "simple" criteria, takes the particular form

$$q \rightarrow \text{constant}(\Delta T_t) \qquad (139)$$

Also, in both types of equipment, the total heat flow is obtained by integrating the expression

$$dQ = q\ dA \qquad (140)$$

When dealing with shell and tube heat exchangers in the old heat transfer, the designer/analyst can avoid the actual integration by utilizing graphs which describe the results of integration. These graphs are presented in most old heat transfer texts and handbooks and take several forms. These same graphs are used in the new heat flow when dealing with "simple" shell and tube heat exchangers, but the terminology and the emphasis is somewhat different. In this section, we will translate two of the more common graphical forms into the terminology of the new heat flow.

In the old heat transfer, one method of avoiding the actual integration is by using the so-called "LMTD correction factor plots". This "correction factor" is usually assigned the symbol "F" and is viewed as a factor which "corrects" the LMTD (Log Mean Temperature Difference). F is useful in that the product of F and LMTD gives the true mean temperature difference and allows Q to be determined without integration from the expression

$$Q = U\ A\ F\ \Delta T_{LM} \tag{141}$$

The correction factor plots are presented in the form F vs P at constant values of R (also called Z). P is often referred to as the "temperature efficiency" and R is often referred to as the "hourly heat capacity ratio". F, P, and R are defined by

$$F = \frac{\text{true mean temperature difference}}{\text{log mean temperature difference}} \tag{142}$$

$$P = \frac{T_{t,out} - T_{t,in}}{T_{s,in} - T_{t,in}} \tag{143}$$

$$R = \frac{(WC)_t}{(WC)_s} \tag{144}$$

where the subscript t refers to tube side and s refers to shell side. The correction factor plots of the old heat transfer translate directly into the new heat flow except that the parameters are described in a different way. In the new heat flow, we do not refer to the "temperature efficiency", or to the "LMTD correction factor", or to the "hourly heat capacity ratio", nor do we use the shorthand symbols F, P, R, or Z. In the new heat flow, we note that

$$Q = q_{avg}\ A \tag{145}$$

which states the obvious fact that the total heat flow rate is given by the product of the average heat flux and the heat exchanger area. Recall that, for a "simple" double pipe heat exchanger,

$$Q = q_{LM}\ A \tag{146}$$

and note from 145 and 146 that, for this case, q_{avg} is equal to q_{LM}. However, for "simple" shell and tube heat exchangers in general, q_{avg} is NOT equal to q_{LM} and thus eq 146 is NOT generally applicable. Nonetheless, we can determine the total heat flow rate for "simple" shell and tube heat exchangers without integrating eq 140 by noting that

$$Q = \frac{q_{avg}}{q_{LM}}\ q_{LM}\ A \tag{147}$$

and by presenting q_{avg}/q_{LM} graphically. In the new heat flow, the graphs are presented in the form q_{avg}/q_{LM} vs $(T_{t,out} - T_{t,in})/(T_{s,in} - T_{t,in})$ at constant values of $(WC)_t/(WC)_s$, and the curves and the coordinates are the same as those of the old "LMTD correction factor" graphs. Eq 147 is the new way translation of eq 141.

Another popular way of presenting the integration results in the old heat transfer is in the form of so-called "effectiveness-NTU plots" where ε is plotted vs NTU at constant values of $(WC)_{min}/(WC)_{max}$. (We discussed the effectiveness-NTU method on page 3-23.) This method allows the total heat flow rate to be determined without integration from the expression

$$Q = \varepsilon\ (WC)_{min}(T_{h,in} - T_{c,in}) \tag{148}$$

In the new heat flow, these same graphs are presented in the form Q/Q_{max} vs AM/WC_{min} at constant values of

WC_{min}/WC_{max}, and the curves and the coordinates are the same as those of the old "effectiveness-NTU" graphs. The Q/Q_{max} values obtained from these graphs are used with the expression

$$Q = \frac{Q}{Q_{max}} (WC)_{min}(T_{h,in} - T_{c,in}) \qquad (149)$$

in order to obtain the value of Q without integrating eq 140.

In summary, the translation of the old F and ε methods of avoiding integration requires merely that the old labels on the graphs be replaced by the new labels described above, and that eqs 147 and 149 replace the old equations 141 and 148.

Since the old way graphs are available in most texts on the old heat transfer, we are not going to repeat them here merely to update the labels. The next two problems illustrate the application of eqs 147 and 149, and are based on the assumption that the reader has access to an old heat transfer text which contains F and ε graphs.

PROBLEM 12 Given the information below, determine the heat flow area required in order to heat Fluid A from 200F to 260F.

Given The heat exchanger is one shell pass, two tube pass. The shell pass is cocurrent with the incoming tube pass. Fluid A is tube side, Fluid B is shell side. The fluids are described by

$W_A = 2250 \qquad C_A = 0.68$

$W_B = 1910 \qquad C_B = 1.00$

$T_{B,in} = 300$

The local thermal behavior within the heat exchanger is described by

$$q_i \rightarrow 140\ \Delta T_t \qquad (150)$$

Solution A heat balance on Fluid A gives

$$Q = 2250(.68)(260 - 200) = 91800 \qquad (151)$$

$$\therefore\ T_{B,out} = 300 - \frac{91800}{1910(1.00)} = 249 \qquad (152)$$

Calculate ΔT_{LM} and q_{LM}:

$$\Delta T_{LM} = \frac{49 - 40}{\ln(49/40)} = 44.3 \qquad (153)$$

$$\therefore\ q_{LM} \rightarrow 140(44.3) = 6200 \qquad (154)$$

Problem 12 cont.

Solution (Note that ΔT_{LM} is based on counterflow of the fluid streams.)

Calculate heat flow area from

$$Q = \frac{q_{avg}}{q_{LM}} q_{LM} A \tag{155}$$

by obtaining q_{avg}/q_{LM} from graph. (See old heat transfer text such as Kreith or Holman and find "F correction-factor plot" for one shell pass, even number of tube passes.) The value of q_{avg}/q_{LM} is obtained from the graph by noting that

$$\frac{WC_t}{WC_s} = \frac{2250(.68)}{1910(1.00)} = .80 \tag{156}$$

$$\frac{T_{t,out} - T_{t,in}}{T_{s,in} - T_{t,in}} = \frac{60}{100} = .60 \tag{157}$$

Locating the above values on the graph, obtain

$$\frac{q_{avg}}{q_{LM}} = 0.71 \tag{158}$$

$$\therefore A = \frac{91800}{6200(.71)} = 20.9 \tag{159}$$

Answer The heat flow area required to heat Fluid A from 200F to 260F is 20.9 ft^2.

PROBLEM 13 Suppose that the heat exchanger in Problem 12 becomes fouled, and that the thermal behavior of the fouling deposit is described by

$$q_{f,i} \rightarrow 650\ \Delta T_f \tag{160}$$

What total heat flow rate and Fluid A outlet temperature would be expected?

Solution The total temperature difference locally is

$$\Delta T_t = (\Delta T_i + \Delta T_w + \Delta T_o) + \Delta T_f \tag{161}$$

From rel 150, we know that

$$q_i \rightarrow 140(\Delta T_i + \Delta T_w + \Delta T_o) \tag{162}$$

Combining 160, 161, and 162, we obtain

$$q_i \rightarrow 115\ \Delta T_t \tag{163}$$

Determine Q from

$$Q = \frac{Q}{Q_{max}} (WC)_{min}(T_{h,in} - T_{c,in}) \tag{164}$$

by obtaining Q/Q_{max} from graph. (See old heat transfer text such as Kreith or Holman and find "effectiveness-NTU" plot for one shell, two tube passes.) The value of Q/Q_{max} is obtained from the graph

Solution by noting that

$$\frac{WC_{min}}{WC_{max}} = .80 \qquad \frac{AM}{WC_{min}} = 1.57$$

At the above values, the graph indicates that Q/Q_{max} is 0.56. Substituting this value into eq 164, we obtain

$$Q = .56(2250)(.68)(300 - 200) = 85700 \qquad (165)$$

$$\therefore T_{A,out} = 200 + \frac{85700}{2250(.68)} = 256 \qquad (166)$$

Answer At the stated conditions, the expected heat flow rate would be 85700 B/hr and the expected Fluid A outlet temperature would be 256F.

DISCUSSION OF PROBLEMS 12 AND 13

Problems 12 and 13 illustrate the solution of "simple" shell and tube design/analysis problems using the new heat flow. The problems demonstrate that the new way solution is the same as the old way solution except that we think in a different way, we view the parameters in a different light, and we assign the parameters their "natural" names rather than inventing artificial names for them like "number of heat transfer units" or "temperature efficiency" or "effectiveness". The problems also demonstrate that the translation of the old "LMTD correction factor plots" and the "effectiveness plots" requires merely that the parameters be assigned their new heat flow names.

CONCLUSIONS

This chapter deals EXCLUSIVELY with thermal behavior which is STRICTLY PROPORTIONAL. The analyses in this chapter do NOT accurately describe the behavior of real world equipment because real world equipment is NOT accurately described by the "simple" criteria given on page 3-1. However, this type of analysis has considerable practical importance because the "simple" criteria greatly simplify the design/analysis problem, and oftentimes these criteria describe real world behavior with "engineering accuracy"--which means that if we base the design/analysis on the "simple" criteria and if we include a safety/ignorance factor, then the end result will probably be satisfactory. In each case, the magnitude of the safety/ignorance factor must be decided by the designer/analyst on the basis of experience or analysis.

The obvious purpose of the problems in this chapter is to illustrate the solution of "simple" heat exchanger design/analysis problems using the new heat flow. But these problems have a much more important purpose, and that is to help the reader forget about heat transfer coefficients and begin to think and analyze in terms of thermal behavior--in terms of the local thermal behavior $q\{\Delta T_t\}$ and the equipment overall thermal behavior in the form $Q\{\Delta T_{OA}\}$.

The "simple" criteria make the problems in this chapter so simple as to be no problem at all, and for that reason the concept of thermal behavior may seem less than necessary. But beyond this chapter, we are going to abandon the "simple" criteria and the results which are based on them. Beyond this chapter, we are going to deal with problems involving "nonproportional" thermal behavior--problems which are NOT accurately described by

$$q \rightarrow \text{constant}(\Delta T_t) \qquad (167)$$

$$Q \rightarrow \text{constant}(\Delta T_{OA}) \quad (168)$$

and which must be described in a more general way by

$$q \rightarrow f\{\Delta T_t\} \quad (169)$$

$$Q \rightarrow f\{\Delta T_{OA}\} \quad (170)$$

Beyond this chapter, we will deal with problems which are accurately described as "problems" since many of them are so difficult they have never been solved within the framework of the old heat transfer. And we will find that "thermal behavior" is the key to the solution of these "problems".

CHAPTER 4 THE THERMAL DESIGN/ANALYSIS OF "NONSIMPLE" HEAT FLOW EQUIPMENT SUCH AS POOL BOILERS

INTRODUCTION

In the old heat transfer, equipment design/analysis is based on heat transfer coefficients. In the new heat flow, equipment design/analysis is based on thermal behavior. In Chapter 3, we dealt with equipment design/analysis problems which involved only strictly proportional thermal behavior, but the heat flow phenomena of the real world exhibit vastly different types of thermal behavior ranging from essentially proportional to highly nonlinear. The correlations of the old heat transfer indicate that one phase, forced convection heat flow at small to moderate temperature differences oftentimes exhibits essentially proportional thermal behavior and therefore is quite accurately described analytically by rel 1 and graphically by Fig 1:

$$q \rightarrow \text{constant}(\Delta T) \quad (1)$$

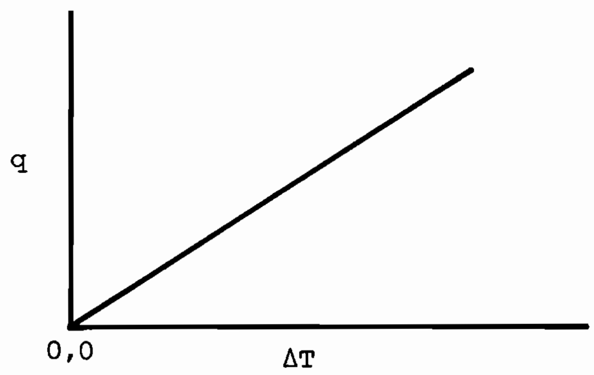

FIGURE 1

Proportional Thermal Behavior

One phase, free convection heat flow is moderately nonlinear, and the correlations of the old heat transfer indicate that the thermal behavior of this type of heat flow is described analytically by rel 2 and graphically by Fig 2:

$$q \to \text{constant}(\Delta T)^n \qquad n \simeq 1.3 \qquad (2)$$

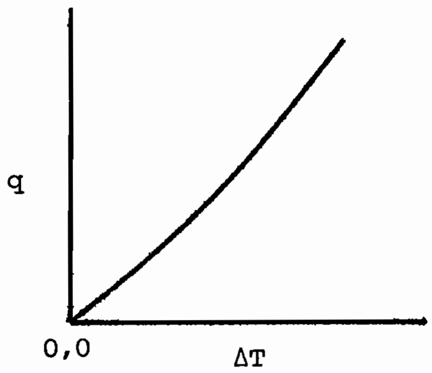

FIGURE 2

Moderately Nonlinear Thermal Behavior

Heat flow to a boiling liquid exhibits perhaps the most highly nonlinear form of thermal behavior as evidenced by the so-called "pool boiling curve" shown in Fig 3:

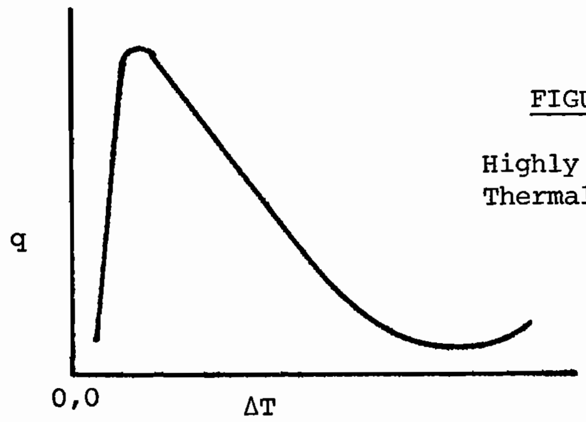

FIGURE 3

Highly Nonlinear Thermal Behavior

Notice that the behavior in Fig 3 is so highly nonlinear that it can not be accurately described by a simple analytical expression such as rel 1 or 2. For that reason, the pool boiling curve is normally described graphically and no attempt is made to analytically describe its nonlinear character. (The regime oriented correlations of the old heat transfer fragment the pool boiling curve and thereby avoid rather than describe its nonlinear character.)

In this chapter, we leave behind the simple criteria of Chapter 3 which caused us to consider only strictly proportional thermal behavior. We go to the opposite end of the spectrum and consider equipment design/analysis problems which involve highly nonlinear thermal behavior. In order that the problems might be simple enough to completely solve by hand, they deal with the simplest type of hardware in which highly nonlinear thermal behavior occurs--a vented pool boiler. However, it is important to note two things about this choice:

1. Pool boilers have considerable real world application and therefore the problems in fact deal with hardware of practical importance.

2. The thermal behavior of a vented pool boiler closely resembles the thermal behavior of a differential element of a forced convection boiler. Since an understanding of local behavior is prerequisite to an understanding of overall equipment behavior, the problems in this chapter are an excellent preparation for an understanding of the overall behavior of forced convection boilers, a subject which we will take up in Chapter 6.

It should also be noted that, although the problems continually refer to the boiling process, nothing in the treatment is peculiar to boiling, and the problems actually illustrate how to deal with highly nonlinear thermal behavior in a general way.

DESCRIBING HIGHLY NONLINEAR THERMAL BEHAVIOR

In the new heat flow, we wish to actually deal with highly nonlinear thermal behavior such as that shown in Fig 3. This can only be done if we have some way of describing such behavior. Inspection of Fig 3 indicates that no truly simple analytical expression can accurately describe the curve in the figure, and so we really have only two choices:

1. Describe highly nonlinear thermal behavior with complicated analytical expressions.

2. Describe highly nonlinear thermal behavior with graphical expressions and do not bother to translate them into analytical expressions.

The particular drawback of the first choice is that the resultant mathematics is so cumbersome that even simple design/analysis problems become quite difficult to solve, and there is the danger that we will become so preoccupied with mathematical problems that we will lose sight of the real, physical problem. The particular drawback of the second choice is that graphical expressions require graphical solutions, and of course we would prefer analytical solutions to design/analysis problems.

Rather than debate the relative merits of the above two choices, let us agree that the first choice is better in an aesthetic sense and that the second choice is better in a practical sense. And since we are more concerned with the practical aspect, let us agree on the second choice. Picking the second choice does not mean we are giving up anything. We are merely agreeing to solve this type of design/analysis problem graphically, but these graphical solutions also illustrate the analytical solution of the same problems because every graphical solution we perform could be translated into the corresponding analytical solution. We will not bother with the translation because, when dealing with highly nonlinear behavior, the graphical solution is much easier to obtain and to understand. In other words, the analytical solutions are implicit in the graphical solutions we perform, and we do not bother to make the analytical solutions explicit because that would not add to our understanding of the problems.

In summary, in the new heat flow, we describe highly nonlinear thermal behavior with graphical expressions, and we use graphical analyses to solve equipment design/analysis problems involving highly nonlinear thermal behavior.

THE GRAPHICAL ANALYSIS OF PROBLEMS INVOLVING PROPORTIONAL THERMAL BEHAVIOR

In order to prepare for the graphical analysis of problems involving nonlinear thermal behavior, and also to illustrate the parallel between analytical solutions and graphical solutions, let us solve some problems involving proportional thermal behavior. First we will state and solve the problems analytically, then we will state and solve the same problems graphically.

Problem 1, Analytical statement/solution:

Given the information below, determine the overall thermal behavior, $q\{\Delta T_t\}$, of the system in Fig 4:

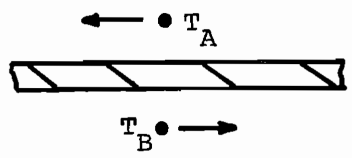

FIGURE 4

Problem 1, Analytical cont.

Given The thermal behavior of the wall/fluid interfaces and of the wall in Fig 4 are described by

$$q_A \rightarrow 370\ \Delta T_A \qquad (3)$$

$$q_B \rightarrow 260\ \Delta T_B \qquad (4)$$

$$q_w \rightarrow 980\ \Delta T_w \qquad (5)$$

where ΔT_A refers to the temperature difference across the wall/Fluid A interface, etc.

The wall in Fig 4 is a flat plate.

Sol'n Inspection of Fig 4 shows that

$$\Delta T_t = \Delta T_A + \Delta T_w + \Delta T_B \qquad (6)$$

Since the wall in Fig 4 is a flat plate, it follows that

$$q_A = q_w = q_B = q \qquad (7)$$

Substituting 3, 4, and 5 into 6 gives

$$\Delta T_t \rightarrow \frac{q_A}{370} + \frac{q_w}{980} + \frac{q_B}{260} \qquad (8)$$

Combining 7 and 8 gives

Problem 1, Analytical cont.

Sol'n $q \rightarrow 132\ \Delta T_t$ (9)

Ans'r The overall thermal behavior of the system in Fig 4 is described by rel 9.

Problem 1, Graphical statement/solution:

Given the information below, determine the overall thermal behavior, $q\{\Delta T_t\}$, of the system in Fig 4.

Given The wall in Fig 4 is a flat plate.

The thermal behavior of the wall/fluid interfaces and of the wall in Fig 4 are described by Fig 5:

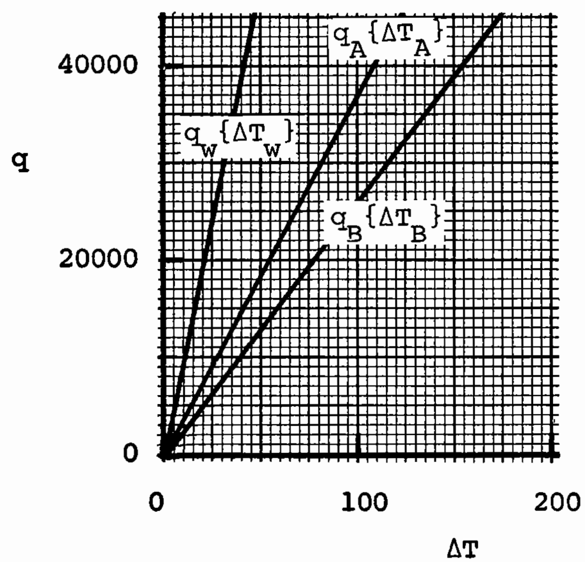

FIGURE 5

Thermal Behavior of Problem 1 Components

Problem 1, Graphical cont.

Sol'n Inspection of Fig 4 shows that

$$\Delta T_t = \Delta T_A + \Delta T_w + \Delta T_B \quad (10)$$

Since the wall in Fig 4 is a flat plate, it follows that

$$q_A = q_w = q_B = q \quad (11)$$

With regard to Fig 5, eqs 10 and 11 tell us that, at any value of q, the total ΔT is the sum of the individual ΔT's. Referring to Fig 5 at q = 40,000, it can be seen that

$\Delta T_A = 108$ $\Delta T_w = 41$ $\Delta T_B = 154$

and therefore $\Delta T_t = 303$, thus locating a point on the $q\{\Delta T_t\}$ function. Repeating the above graphically at several other values of q, we obtain the $q\{\Delta T_t\}$ points shown in Fig 6, pg 4-9.

The $q\{\Delta T_t\}$ function is obtained by connecting the $q\{\Delta T_t\}$ points on Fig 6. The result of connecting these points is shown on Fig 7.

Ans'r The overall thermal behavior of the system in Fig 4 is shown graphically in Fig 7, pg 4-9.

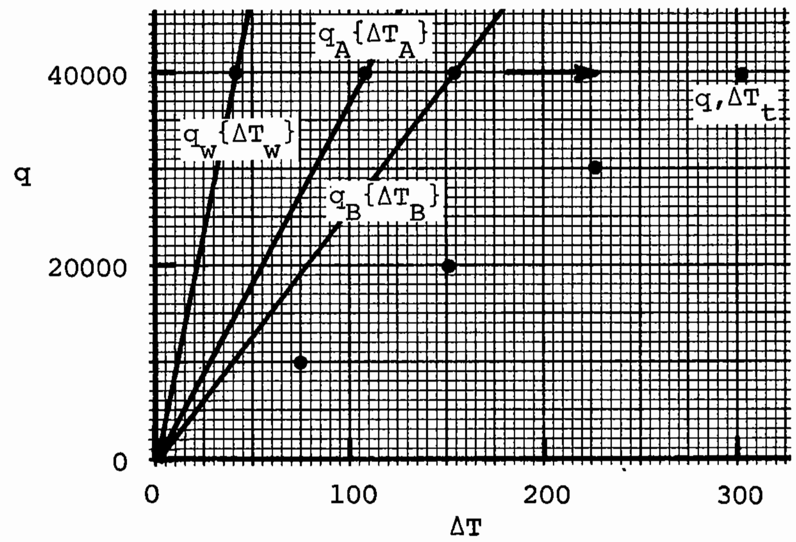

FIGURE 6 Graphical Determination of $q, \Delta T_t$ Points, Problem 1

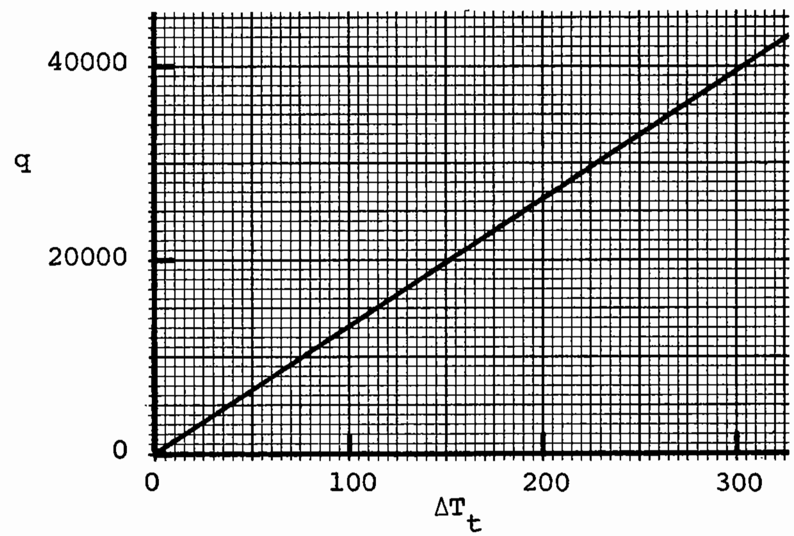

FIGURE 7 Overall Thermal Behavior of System in Problem 1

DISCUSSION OF PROBLEM 1

Problem 1 illustrates the graphical determination of overall thermal behavior based on graphical descriptions of the interface and wall thermal behavior. It also demonstrates that analytical solutions and graphical solutions are fundamentally the same.

Notice what a simple task it is to solve Problem 1 graphically. Moreover, the simplicity of the graphical solution is not the result of the proportional behavior in Problem 1--it is the result of the fact that graphical solutions are inherently simple, and we will find that the graphical solution of problems involving highly nonlinear behavior is no more difficult than the graphical solution of Problem 1.

Notice also that the analytical analysis and the graphical analysis are in fact the same analysis. Both are based on the observations that the total temperature difference is the sum of the individual temperature differences and that the heat flux is the same value at the interfaces and through the wall. Both give the same result, the only difference being that one result is expressed in the language of equations and the other is expressed in the language of curves on a graph.

INTRODUCTION TO PROBLEM 2

In the new heat flow, we will oftentimes wish to determine the value of the heat flux and/or appraise the thermal stability at a particular value of the total temperature difference. Such analyses are conveniently performed by determining expressions for the heat flux into and the heat flux out of an interface. The value of the heat flux is determined from the requirement that the heat flux into the interface must equal the heat flux out of the interface--ie the heat

flux must be continuous at the interface--ie eq 12 must be satisfied at the interface:

$$q_{in} = q_{out} \qquad (12)$$

where the subscripts in and out stand for into and out of the interface. The thermal stability is appraised by differentiating both sides of eq 12 with respect to the wall temperature at the interface and noting that thermal stability requires that the criterion

$$\frac{dq_{in}}{dT_{w,i}} < \frac{dq_{out}}{dT_{w,i}} \qquad (13)$$

be satisfied. Since we will usually be working with equipment in which the sink temperature is independent of the heat flux, we can replace criterion 13 with the more convenient form of criterion 14:

$$\frac{dq_{in}}{d\Delta T_i} < \frac{dq_{out}}{d\Delta T_i} \qquad (14)$$

where ΔT_i refers to the temperature difference across the interface.

In Problem 2, we perform the analysis described above, first analytically and then graphically. The purpose is the same as that of Problem 1--to illustrate the graphical solution of such problems and to demonstrate that there is no real difference between analytical analysis and graphical analysis--ie they are equally rigorous and equally "scientific".

Problem 2, Analytical statement/solution:

Determine the heat flux and appraise the thermal stability of the system described in Problem 1 given that the overall temperature difference is 205F. Perform the analysis at the wall/Fluid B interface in the manner suggested by eq 12 and criterion 14.

Sol'n From Problem 1, it is apparent that q_{out} at the wall/Fluid B interface is given by

$$q_{out} = q_B \rightarrow 260\ \Delta T_B \tag{15}$$

From inspection of Fig 4, it is apparent that q_{in} is determined by the value of $(\Delta T_A + \Delta T_W)$. From the information in Problem 1,

$$(\Delta T_A + \Delta T_W) \rightarrow \frac{q_{in}}{980} + \frac{q_{in}}{370} \tag{16}$$

$$\therefore\ q_{in} \rightarrow 269(\Delta T_A + \Delta T_W) \tag{17}$$

In the problem statement, we are given that

$$\Delta T_t = 205 = \Delta T_A + \Delta T_W + \Delta T_B \tag{18}$$

$$\therefore\ (\Delta T_A + \Delta T_W) = 205 - \Delta T_B \tag{19}$$

Combining 17 and 19 gives

$$q_{in} \rightarrow 269(205 - \Delta T_B) \tag{20}$$

Problem 2, Analytical cont.

Sol'n As suggested by eq 12, we equate q_{in} from eq 20 and q_{out} from eq 15 and obtain

$$269(205 - \Delta T_B) = 260\ \Delta T_B \tag{21}$$

$$\therefore\ \Delta T_B = 104 \tag{22}$$

$$\therefore\ q = 260(104) = 27000 \tag{23}$$

Differentiating 15 and 20, we obtain

$$\frac{d\dot{q}_{out}}{d\Delta T_B} \rightarrow 260 \tag{24}$$

$$\frac{dq_{in}}{d\Delta T_B} \rightarrow -269 \tag{25}$$

Therefore the thermal stability criterion is satisfied since, as required by criterion 14,

$$-269 < 260 \tag{26}$$

Ans'r At an overall temperature difference of 205F, the heat flux in the Problem 1 system is 27000 B/hr ft^2, and the system is thermally stable.

Problem 2, Graphical statement/solution:

Graphically determine the heat flux and appraise the thermal stability of the Problem 1 system at an overall temperature difference of 205F.

Sol'n The graphical solution is based on plotting $q_{in}\{\Delta T_B\}$ and $q_{out}\{\Delta T_B\}$ on the same graph. These functions are obtained either from Fig 5 or from rels 15 and 20. The graphical solution is as shown in Fig 8 below.

Ans'r The intersection in Fig 8 tells us that, at a total temperature difference of 205F, the heat flux in the Problem 1 system is 27000 B/hr ft^2 (since this is the value at which $q_{in} = q_{out}$) and that the system is thermally stable (since the slope of the q_{in} curve is more negative than the slope of the q_{out} curve, as required by crit 14).

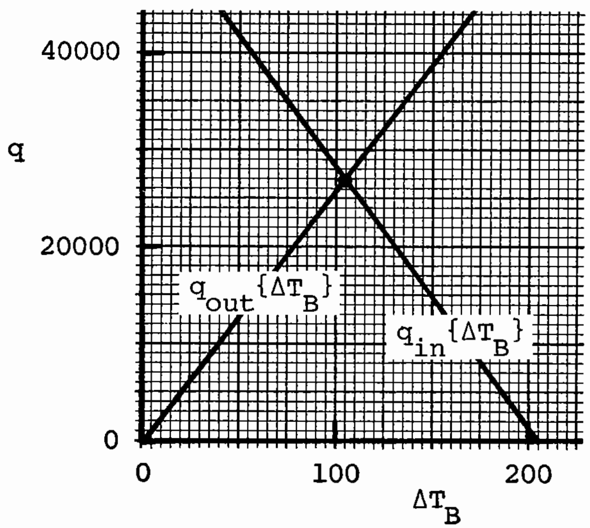

FIGURE 8 Graphical Solution of Problem 2

INTRODUCTION TO PROBLEM 3

Because of the proportional thermal behavior in Problems 1 and 2, the analytical solutions were simple and straightforward, and there was no need for the graphical solutions other than for illustration. On the other hand, Problem 3 deals with highly nonlinear behavior and there is a real need for a graphical solution because an analytical solution could hardly be described as simple. As shown by Problem 3, the graphical solution of problems involving highly nonlinear behavior is quite simple--in fact, it is no more difficult than the graphical solution of Problems 1 and 2. If this seems unlikely, notice that the graphical solution of Problem 3 is EXACTLY THE SAME as the graphical solution of Problems 1 and 2!

Problem 3 is the last problem which is primarily intended to illustrate graphical analysis. The later problems in this chapter are primarily concerned with equipment design/analysis, and the fact that we use graphical analysis is only incidental.

Problem 3, Graphical statement/solution:

Suppose that the thermal behavior of the wall/Fluid B interface is as described in Fig 9, page 4-17, and that the other characteristics of the system are as described in Problem 1. Determine the heat flux and appraise the thermal stability of the system at an overall temperature difference of 205F. Also determine the overall thermal behavior and discuss any hysteresis effects.

Sol'n The heat flux at 205F is determined in exactly the same way as in Problem 2. We plot q_{in} and q_{out} on the same graph and the intersection of these two functions defines the heat flux, since

Problem 3, Graphical cont.

Sol'n at the intersection, $q_{in} = q_{out}$ as required by continuity. We perform the analysis at the nonlinear interface between the wall and Fluid B. Thus q_{out} is $q_{out}\{\Delta T_B\}$ and we must determine $q_{in}\{\Delta T_B\}$. The first function is described by the curve in Fig 9, the second function was determined in Problem 2 and is described by rel 20. (The second function could also have been determined graphically by noting that

$$q_{in}\{\Delta T_B\} = q_{in}\{\Delta T_t - \Delta T_B\} \tag{27}$$

and graphically subtracting $q\{\Delta T_B\}$ of Fig 5 from $q\{\Delta T_t\}$ of Fig 7.) Plotting these two functions on the same graph results in Fig 10, page 4-17.

The three intersections in Fig 10 tell us that the heat flux could be any of three values:

$$q = 42000 \text{ or } 33000 \text{ or } 3000$$

Inspection of Fig 10 and the thermal stability criterion that

$$\frac{dq_{in}}{d\Delta T_B} < \frac{dq_{out}}{d\Delta T_B} \tag{28}$$

indicate that the system would be thermally stable at heat flux values of 42000 and 3000 B/hr ft^2, and thermally unstable at 33000. Fig 10 also indicates that the result of the thermal instability is that the system tends to move away from the intersection at 33000 and towards either of the other two intersections.

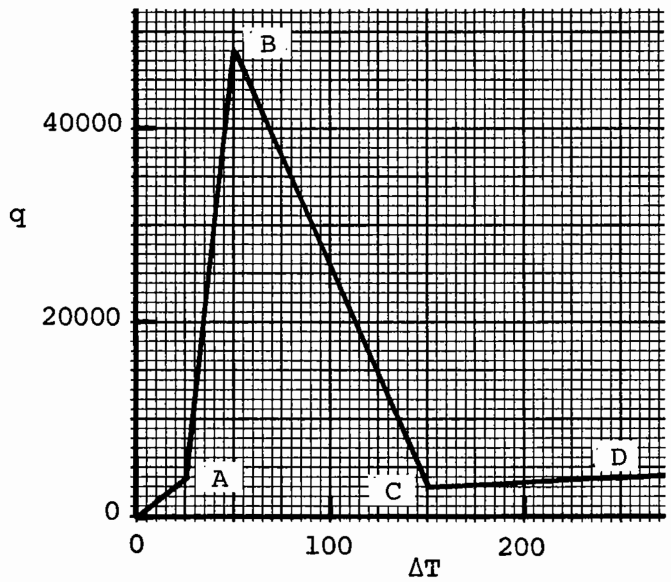

FIGURE 9 Thermal Behavior of Fluid B Interface, Problem 3

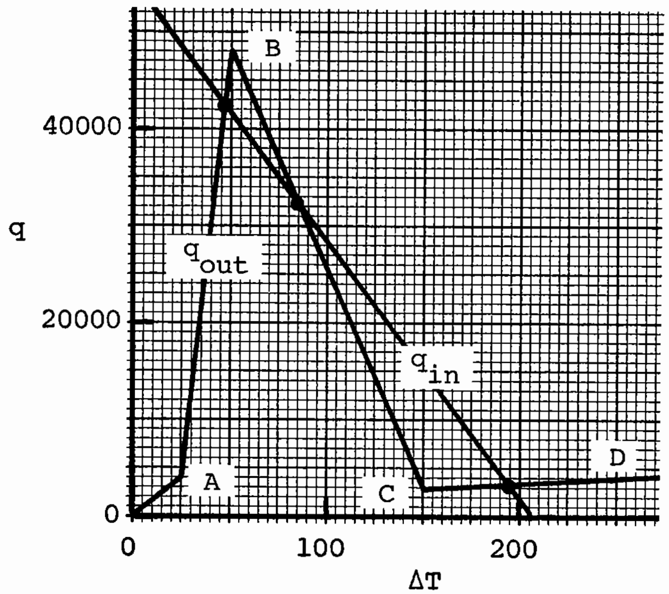

FIGURE 10 Solution of Problem 3

Problem 3, Graphical cont.

Sol'n at the intersection, $q_{in} = q_{out}$ as required by continuity. We perform the analysis at the nonlinear interface between the wall and Fluid B. Thus q_{out} is $q_{out}\{\Delta T_B\}$ and we must determine $q_{in}\{\Delta T_B\}$. The first function is described by the curve in Fig 9, the second function was determined in Problem 2 and is described by rel 20. (The second function could also have been determined graphically by noting that

$$q_{in}\{\Delta T_B\} = q_{in}\{\Delta T_t - \Delta T_B\} \tag{27}$$

and graphically subtracting $q\{\Delta T_B\}$ of Fig 5 from $q\{\Delta T_t\}$ of Fig 7.) Plotting these two functions on the same graph results in Fig 10, page 4-17.

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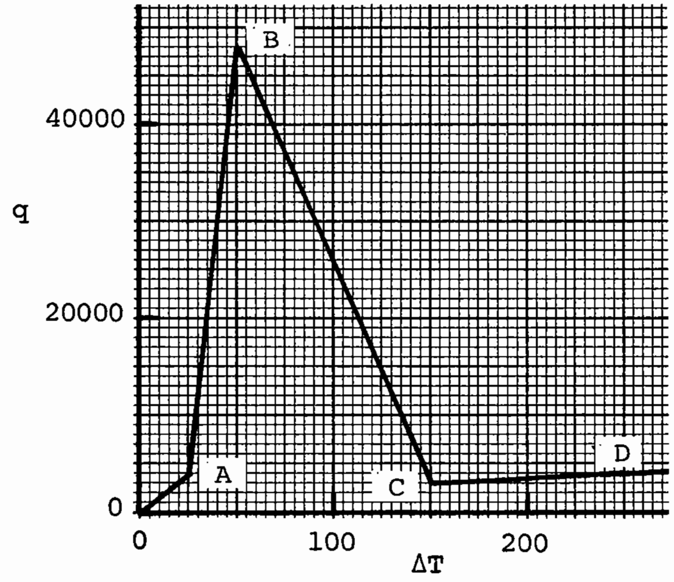

FIGURE 9 Thermal Behavior of Fluid B Interface, Problem 3

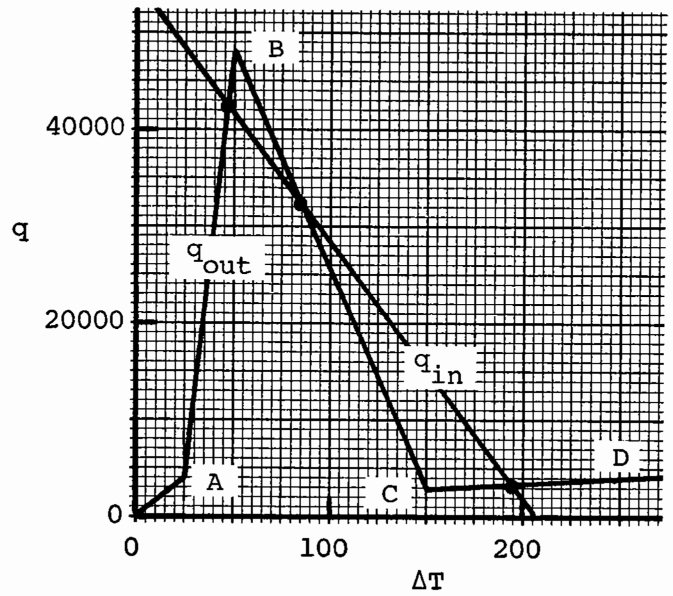

FIGURE 10 Solution of Problem 3

Problem 3, Graphical cont.

Sol'n We therefore conclude that the steady-state heat flux is either 42000 or 3000 B/hr ft^2 and we note that the given information is not sufficient to uniquely determine the result. (In order to determine whether the equipment would be at 42000 or 3000 B/hr ft^2, we would need to be given a description of the prior operating history.)

The overall thermal behavior is determined in exactly the same way as in Problem 1. At various values of q, we add up the individual ΔT's and this gives us $q\{\Delta T_t\}$ coordinates. For example, from Figs 5 and 9, we could generate the following table:

q	ΔT_A	ΔT_w	ΔT_B	ΔT_t
10000	27	10	30	67
10000	27	10	133	170
20000	54	20	35	109
20000	54	20	111	185
30000	81	31	40	152
30000	81	31	89	201

The graphical description of the overall thermal behavior is obtained by plotting the $q\{\Delta T_t\}$ coordinates from a table such as the above, and Fig 11, pg 4-19 is the result. Figure 11 tells us that there is pronounced hysteresis in the overall thermal behavior of the equipment, and that the hysteresis is as described by Fig 12. (Figure 12 is obtained by inspecting Fig 11 to determine the result of a monotonic increase and a monotonic decrease in ΔT_t.) Figure 12

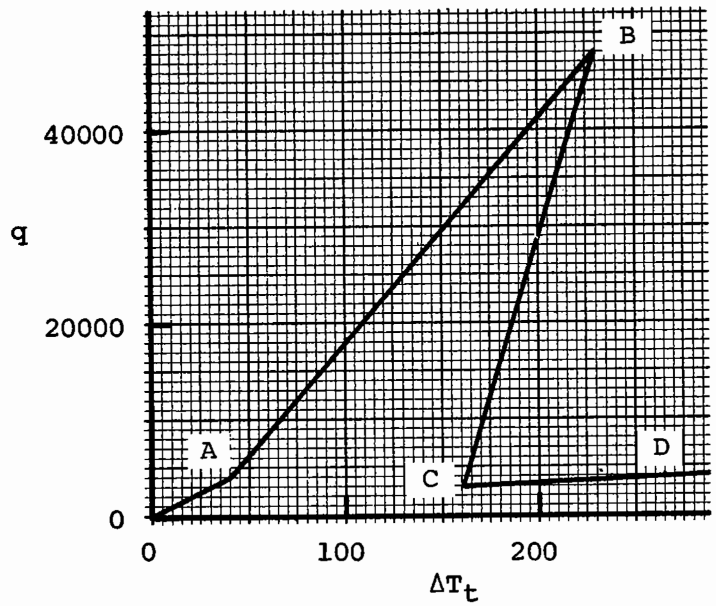

FIGURE 11 Overall Thermal Behavior of System in Problem 3

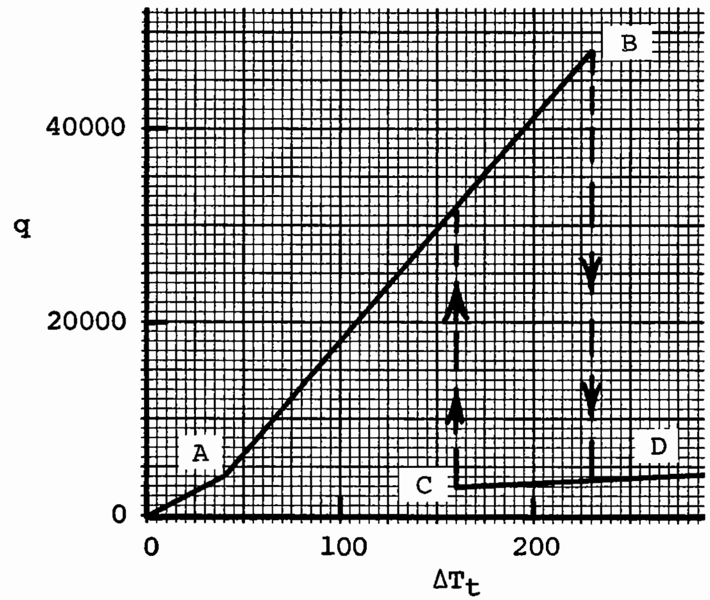

FIGURE 12 Hysteresis Behavior of System in Problem 3

Problem 3, Graphical cont.

Sol'n indicates that the equipment will operate satisfactorily along lines AB and CD of Figs 9 through 12, but that it will "refuse" to operate along line BC--ie there is no value of ΔT_t which will result in stable operation of the system anywhere on line BC. (Also notice by inspection of Fig 10 that the slope of the BC line is more negative than the slope of the q_{in} line--ie the thermal stability criterion is not satisfied anywhere on line BC. Therefore the system will not operate stably anywhere on line BC.)

Ans'r At an overall temperature difference of 205F, the heat flux in the subject system would be either 42000 or 3000 B/hr ft^2 and the system would be thermally stable. The overall behavior of the subject system is described by Fig 11, and the hysteresis behavior is described by Fig 12. The system is thermally unstable at all points along line BC of Fig 9. The result of this thermal instability is pronounced thermal hysteresis at overall temperature differences of 160 to 230F.

POOL BOILERS

Now let us turn our attention from graphical analysis to the solution of design/analysis problems involving highly nonlinear thermal behavior. A good place to begin would be with the simplest type of hardware in which highly nonlinear thermal behavior normally occurs. And this criterion leads us directly to vented, flat-plate pool boilers. The simplicity of this type of hardware is the result of the following:

1. There is no net flow of the boiling liquid. Therefore we do not have to cope with the flow rate or its effects or with the vapor quality.

2. The temperature of the boiling pool is not affected by the heat flux. (The vent maintains the pressure above the boiling liquid at essentially one atmosphere, thus fixing the temperature of the boiling pool.)

3. The flat boiler plate means that we can, with reasonable accuracy, ignore end effects and assume that the heat flux is uniform over the entire boiler plate--ie we can assume that

$$\int q\,dA = q\,A \tag{29}$$

$$\therefore Q = q\,A \tag{30}$$

Although the problems deal with pool boilers, they closely relate to forced convection boilers in that a differential element of a forced convection boiler behaves in much the same manner as a vented pool boiler. To illustrate, suppose we are considering a very short length of a forced convection boiler tube. If we vary the ΔT_t (ie the overall thermal driving force) within the short length and maintain it constant outside the short length, the variation will have no effect on the flow rate through the element or on the pressure and vapor quality within the element. And this is exactly the situation which prevails in a vented pool boiler! Thus we have every reason to expect that the thermal behavior of a short element of a forced convection boiler tube closely resembles the thermal behavior of a vented pool boiler.

A more subtle point is the fact that the pool boiler problems closely relate not only to forced convection boilers in particular, but also to forced convection heat exchangers in general. In the problems, it will be noticed that we keep referring to the boiling process, but there is nothing in the problems which is peculiar to boiling. The problems deal with highly nonlinear thermal behavior, and whether this behavior is the result of boiling heat flow or some other type of heat flow makes not the slightest difference. Thus the pool boiler problems actually relate to the behavior of a differential element of a forced convection heat exchanger in the most general sense.

INTRODUCTION TO PROBLEM 4

The pool boiler in Problem 4 closely resembles the pool boiler used by Berenson (1) to obtain the data discussed in Bk 1, Ch 7. This particular design was chosen primarily because ref 1 contains digital data which relates the heat flux to the temperature difference across the boiling interface--ie it contains a digital description of the thermal behavior of the boiling interface. And this is PRECISELY the information we require for design/analysis in the new heat flow. This choice also permits us to compare the new understanding of pool boiler behavior with the old understanding.

Since the thermal behavior of a boiling interface is quite complex, we will not try to describe its behavior analytically. We will simply plot the boiling interface $q\{\Delta T\}$ coordinates given by Berenson, and we will use this graphical description to solve design/analysis problems dealing with boilers which closely resemble the one used by Berenson. Since the behavior of the boiling interface is described graphically, we will solve the problems using graphical analysis.

In spite of the fact that the following problems deal with highly nonlinear behavior, the analyses are quite simple and straightforward. The problems describe the new way solution of equipment design/analysis problems which are virtually impossible to solve with "heat transfer coefficients"--and which in fact have never been solved within the framework of the old heat transfer.

PROBLEM 4

Given the boiler design information below, describe the overall thermal behavior of the equipment--ie describe the functionality between the total heat flow rate and the overall temperature difference--ie describe the function $Q\{\Delta T_t\}$.

Given The pool boiler design is shown schematically in Fig 13:

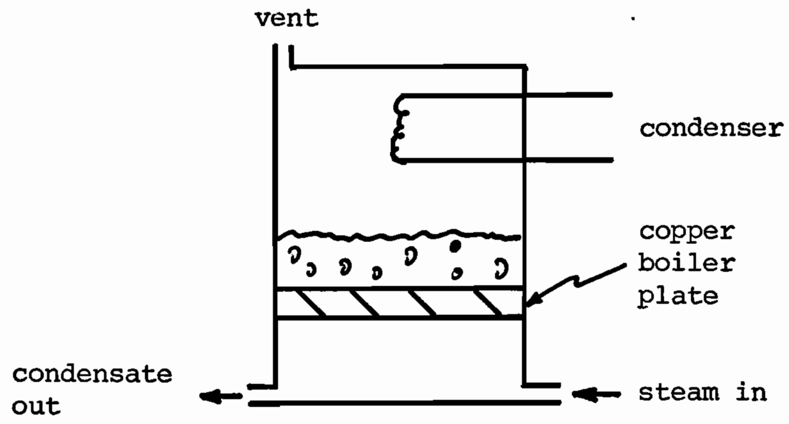

FIGURE 13

The boiler plate diameter is 2.0" and the thickness is 2.3". The lower surface of the boiler plate is well finned. The thermal behavior of the plate material is described by

$$q_w \rightarrow 220 \frac{\Delta T_w}{t} \tag{31}$$

$$\therefore q_w \rightarrow 1148 \, \Delta T_w \tag{32}$$

Problem 4 cont.

Given The boiling liquid is n-pentane. The thermal behavior of the boiling interface is described by the curve in Fig 14, pg 4-25. (The data points in Fig 14 are those reported by Berenson. In the region where no data points were reported, the curve is drawn arbitrarily.)

The heat source is steam condensing on the finned lower surface of the boiler plate. The thermal behavior of the condensing interface is described by

$$q_c \rightarrow 10000\ \Delta T_c \quad (33)$$

based on the unfinned area. (The subscript c refers to the condensing interface.)

Sol'n As discussed on page 4-21, the overall thermal behavior of the equipment is obtained by noting that

$$Q = q\,A \quad (34)$$

for the type of boiler we are analyzing. In eq 34, q is really $q\{\Delta T_t\}$, and we determine this function using the method of Problems 1 and 3-- by noting that $q\{\Delta T_t\}$ is also $q\{\Delta T_c + \Delta T_w + \Delta T_b\}$ and by adding the individual ΔT's at various values of q. (The subscript b refers to the boiling interface.) In this way, we transform the $q\{\Delta T_b\}$ points of Fig 14 to the $q\{\Delta T_t\}$ points of Fig 15. The details of this simple transformation are presented in Table 1, pg 4-26.

Figure 15 is transformed to Fig 16 by noting from eq 34 that

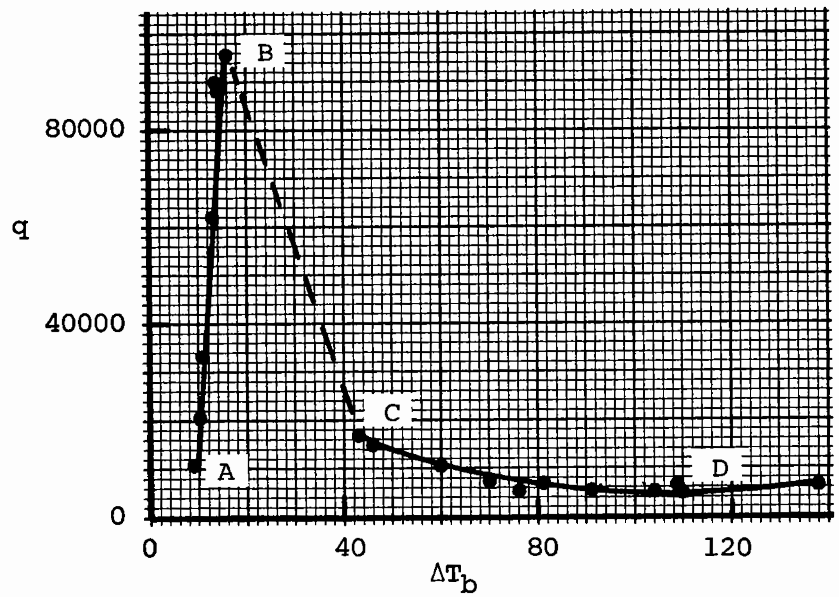

FIGURE 14 Boiling Interface Data by Berenson (1)

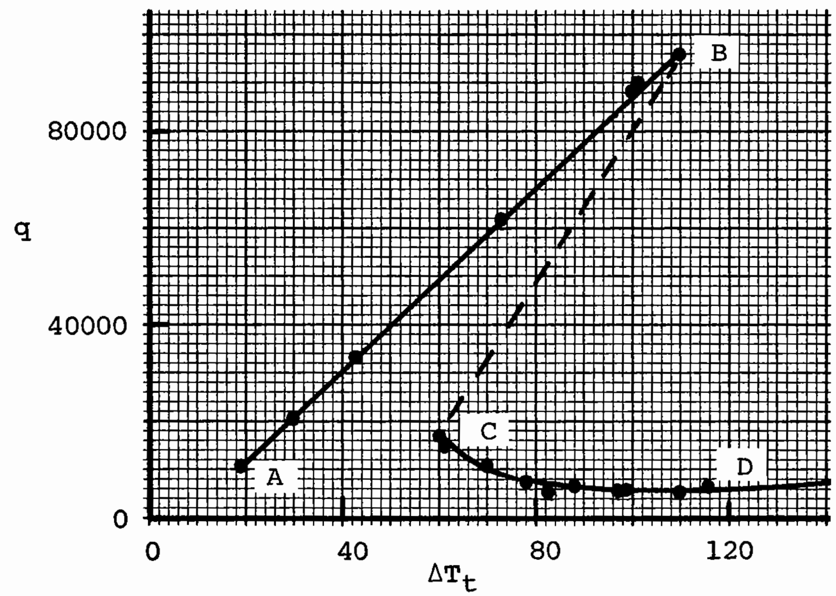

FIGURE 15 Overall Thermal Behavior $q\{\Delta T_t\}$ of the Boiler in Problem 4

TABLE 1

Transforming Berenson's $q\{\Delta T_b\}$ Data (Runs 17/22) to $q\{\Delta T_t\}$ and $Q\{\Delta T_t\}$, Problem 4

q Ref 1	ΔT_b Ref 1	ΔT_w Rel 32	ΔT_c Rel 33	ΔT_t	Q Rel 35
10500	9	9	1	19	230
20600	10	18	2	30	450
33500	11	29	3	43	730
62000	13	54	6	73	1350
88500	14	77	9	100	1930
90000	14	78	9	101	1960
96000	16	84	10	110	2090
16700	43	15	2	60	360
15000	46	13	2	61	330
10500	60	9	1	70	230
7700	70	7	1	78	170
5800	76	5	1	82	130
6400	81	6	1	88	140
5800	91	5	1	97	130
6000	93	5	1	99	130
5500	104	5	1	110	120
6700	109	6	1	116	150
5500	110	5	1	116	120
6450	138	6	1	144	140

Problem 4 cont.

Sol'n
$$Q = q\,A = q\,\frac{\pi(2.0)^2}{4(144)} = .0218\,q \qquad (35)$$

The digital results of this transformation are presented in Table 1, pg 4-26. Figure 16 is presented below.

Ans'r Based on the given boiler design information, the overall thermal behavior of the equipment is described by the curve in Fig 16 below. (Note that the curve in Fig 16 might also be called the "thermal operating characteristic" of the equipment.)

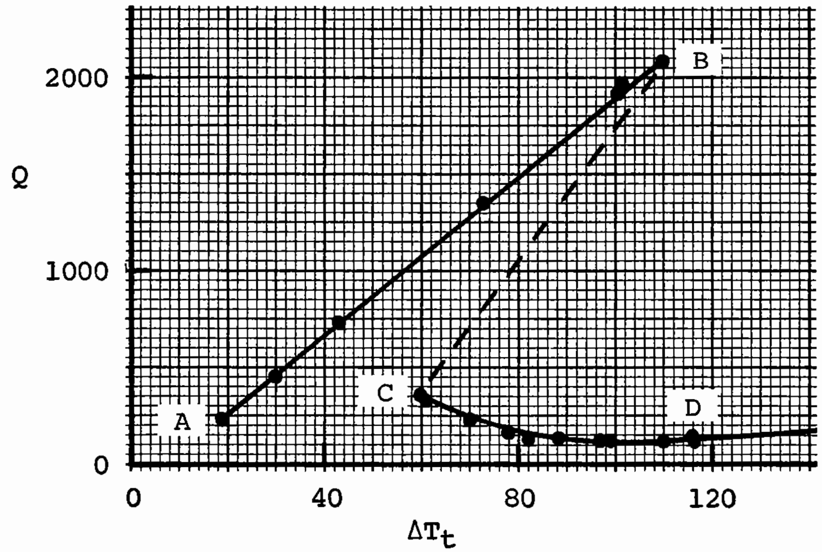

FIGURE 16 Overall Thermal Behavior $Q\{\Delta T_t\}$ of the Boiler in Problem 4

DISCUSSION OF PROBLEM 4

Figure 16 is important because it describes how the equipment behaves in an overall sense, and this is the principal reason for all equipment analysis. Figure 15 is important because it describes local thermal behavior within the boiler, and this is prerequisite to an understanding of overall equipment behavior. Figure 15 tells us that the Problem 4 boiler design is such that MORE THAN ONE VALUE OF HEAT FLUX can result from a particular temperature difference in the range 60F to 110F. Compare this "new" understanding with the understanding expressed by Berenson (1) based on the old heat transfer:

> "The shape of the characteristic boiling curve is such that for a given heat flux three different boiling regimes may be obtained. The transition region is inherently unstable in experiments in which only the heat flux is controlled owing to the negative slope in the transition region. However, an experiment designed to control the temperature difference allows operation within the transition region, as well as the other two regions, SINCE THERE IS ONLY ONE VALUE OF HEAT FLUX ASSOCIATED WITH EACH VALUE OF TEMPERATURE DIFFERENCE."

This old heat transfer view is based on the erroneous supposition that the behavior of a pool boiler can be determined by inspection of a pool boiling curve $q\{\Delta T_b\}$ such as the one in Fig 14. Figure 14 does indeed state that "there is only one value of heat flux associated with each value of temperature difference" ACROSS THE BOILING INTERFACE. But this temperature difference is NOT the one which concerns us! Our proper concern is with the OVERALL temperature difference--our proper concern is with $q\{\Delta T_t\}$ and NOT with $q\{\Delta T_b\}$--our proper concern is with Fig 15 and NOT with Fig 14. And Fig 15 concretely proves that there is MORE THAN ONE VALUE OF HEAT FLUX associated with each value of temperature difference in the boiler.

PROBLEM 5

If the Problem 4 boiler were operated at a total temperature difference of 90F, what local heat flux and total heat flow rate would be expected? (In other words, determine q and Q at $\Delta T_t = 90F$.) Do NOT use Figs 15 and 16 except to check your results.

Sol'n The solution is the same as in Problems 2 and 3--plot $q_{in}\{\Delta T\}$ and $q_{out}\{\Delta T\}$ on the same graph where ΔT refers to the temperature difference across the highly nonlinear interface--in this case, the boiling interface. Thus q_{out} is $q_{out}\{\Delta T_b\}$ and is obtained from Fig 14. $q_{in}\{\Delta T_b\}$ is obtained from rels 32 and 33 by noting that

$$\Delta T_c + \Delta T_w \rightarrow \frac{q_{in}}{10000} + \frac{q_{in}}{1148} \qquad (36)$$

$$\therefore q_{in} \rightarrow 1030(\Delta T_c + \Delta T_w) \qquad (37)$$

$$\Delta T_t = 90 = \Delta T_c + \Delta T_w + \Delta T_b \qquad (38)$$

Combining 37 and 38 gives the desired function

$$q_{in} \rightarrow 1030(90 - \Delta T_b) \qquad (39)$$

Plotting the functions described by Fig 14 and rel 39 on the same graph gives Fig 17, page 4-30. Figure 17 indicates two stable intersections at $q = 78000$ and $q = 6000$ and one unstable intersection at $q = 66000$ Inspection of Fig 17 indicates that the equipment will simply refuse to operate at the unstable intersection--ie even if some transient should somehow bring the system to the unstable intersection, it would simply "drift off" to one of the stable intersections. We

Problem 5 cont.

therefore conclude that the heat flux is either 78000 or 6000 B/hr ft^2 at an overall temperature difference of 90F. The total heat flow rate is

$$Q = q\,A = 78000\,\frac{\pi(2)^2}{4(144)} = 1702 \quad (\text{or } 131) \qquad (40)$$

Ans'r If the Problem 4 boiler were operated at a total temperature difference of 90F, the expected heat flux would be 78000 or 6000 B/hr ft^2, and the expected heat flow rate would be 1702 or 131 B/hr. These same results are obtained by inspection of Figs 15 and 16. The Problem 5 analysis and results are shown in Fig 17 below.

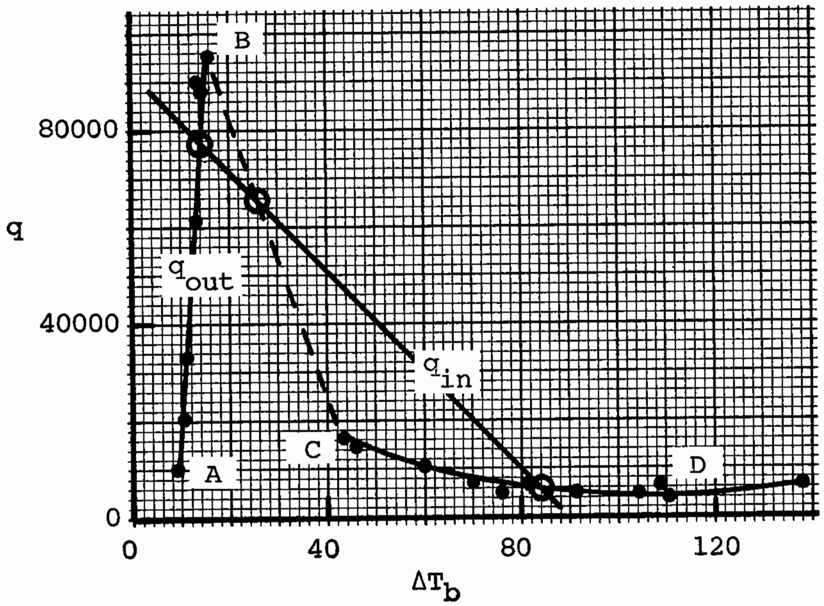

FIGURE 17 Determining the Heat Flux in Problem 5

PROBLEM 6

Based on the old heat transfer, Rohsenow (2) states:

> "With condensing vapor as the heat source on one side of a wall, any point on the entire (pool boiling) curve . . . can be reached under stable conditions."

Based on this old heat transfer view, the Problem 4 boiler should be able to operate stably at any point on the Fig 14 pool boiling curve. Determine whether this old heat transfer view is true or false.

Sol'n & Ans'r From 37 and 38, we may write

$$q_{in} \rightarrow 1030(\Delta T_t - \Delta T_b) \qquad (41)$$

$$\therefore \frac{dq_{in}}{d\Delta T_b} \rightarrow -1030 \qquad (42)$$

The thermal stability criterion requires that

$$\frac{dq_{in}}{d\Delta T_b} < \frac{dq_{out}}{d\Delta T_b} \qquad (43)$$

From 42 and 43, we conclude that the operation is UNstable whenever

$$\frac{dq_{out}}{d\Delta T_b} < -1030 \qquad (44)$$

Inspection of Fig 14 shows that 44 is satisfied throughout the BC region. Thus the Problem 4 boiler is UNstable in the BC region of the Fig 14 PBC and the old heat transfer view is FALSE.

PROBLEM 7

Describe the hysteresis behavior of the Problem 4 boiler.

Sol'n The hysteresis behavior of the Problem 4 boiler is determined by inspection of Fig 16. Figure 16 indicates that a monotonic increase in ΔT_t starting from a value of zero would result in the behavior described in Fig 18, below. A monotonic decrease starting from a value of 140F would result in the behavior described in Fig 19, pg 4-33. Combining Figs 18 and 19 gives Fig 20, pg 4-33, and this Figure is the desired result.

Ans'r The hysteresis behavior of the Problem 4 boiler is described in Fig 20, pg 4-33.

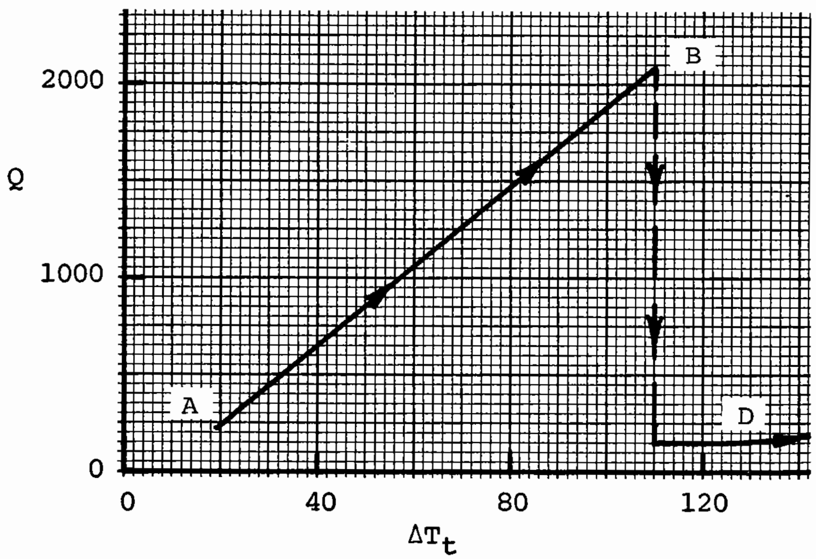

FIGURE 18 The Result of a Monotonic Increase in the Total Temperature Difference

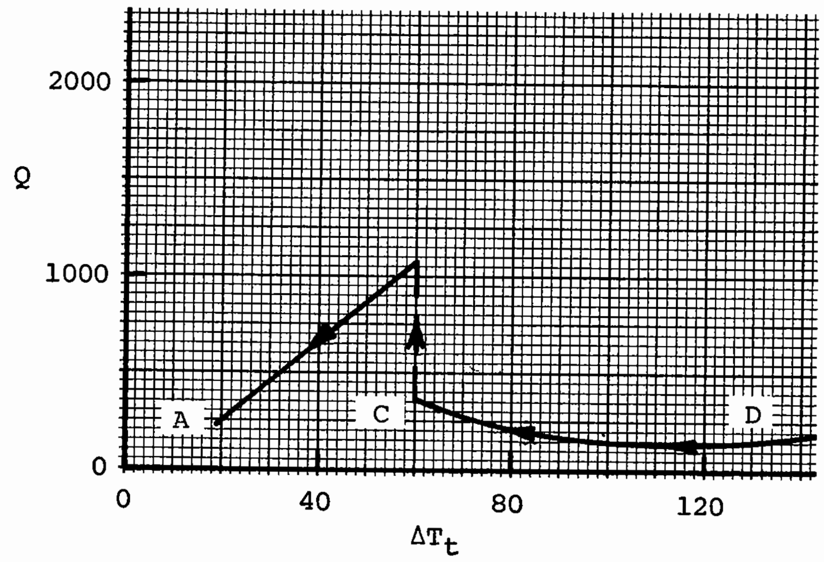

FIGURE 19 The Result of a Monotonic Decrease in the Total Temperature Difference

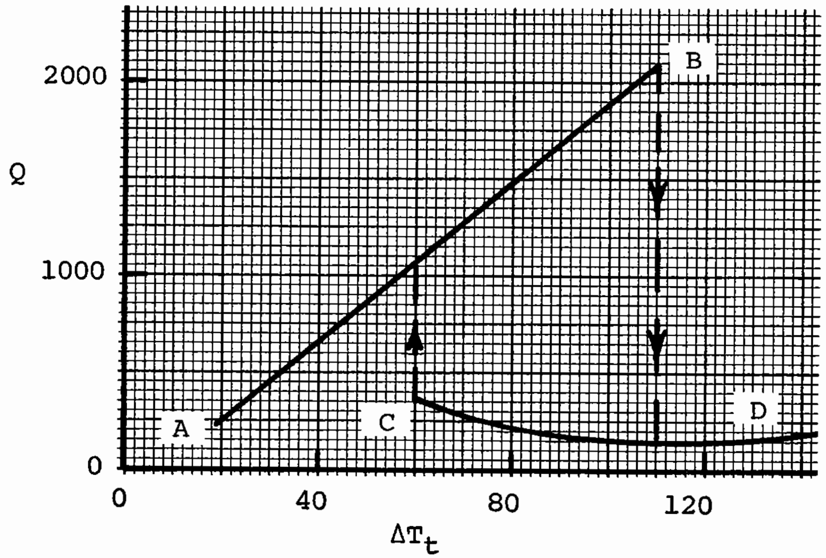

FIGURE 20 The Hysteresis Behavior of the Boiler in Problem 4

PROBLEM 8

It is desired to modify the Problem 4 boiler design in order to eliminate the hysteresis in the operating characteristic, Fig 20. Can this be accomplished by simply changing the thickness of the boiler plate? Can it be accomplished by simply altering the thermal behavior of the condensing interface? Describe the operating characteristic of the boiler after it is modified to eliminate the hysteresis in Fig 20.

Sol'n As we noted in the preceding problems, the hysteresis in Fig 20 is the direct result of thermal instability. Therefore, in order to eliminate the hysteresis, we must eliminate the thermal instability. Since the thermal stability criterion is

$$\frac{dq_{in}}{d\Delta T_b} < \frac{dq_{out}}{d\Delta T_b} \qquad (45)$$

we must somehow modify the equipment design so that $dq_{in}/d\Delta T_b$ will be less than the minimum value of $dq_{out}/d\Delta T_b$. Inspection of Fig 14 indicates that this minimum value is about -3000 B/hr ft^2 F. Thus criterion 45 tells us that the boiler design modification must be such that $dq_{in}/d\Delta T_b$ will be less than -3000 B/hr ft^2 F. Let us set a design target of -5500 B/hr ft^2 F and determine what design modifications will give us this value of $dq_{in}/d\Delta T_b$.

As in the previous analyses, we determine the expression for $q_{in}\{\Delta T_b\}$ by noting that

$$q_w \rightarrow 220 \frac{\Delta T_w}{t} \qquad (46)$$

$$q_c \rightarrow 10000\ \Delta T_c \qquad (47)$$

Problem 8 cont.

Sol'n $\Delta T_t = \Delta T_c + \Delta T_w + \Delta T_b$ (48)

$$q_c = q_w = q_{in} = q \qquad (49)$$

$$\therefore \Delta T_w + \Delta T_c \rightarrow \frac{q\ t}{220} + \frac{q}{10000} \qquad (50)$$

$$\therefore q_{in} \rightarrow \frac{\Delta T_w + \Delta T_c}{t/220 + 1/10000} \qquad (51)$$

$$\therefore q_{in} \rightarrow \frac{\Delta T_t - \Delta T_b}{t/220 + 1/10000} \qquad (52)$$

$$\therefore \frac{dq_{in}}{d\Delta T_b} \rightarrow - \frac{1}{t/220 + 1/10000} \qquad (53)$$

Recalling that the design target for $dq_{in}/d\Delta T_b$ is -5500 B/hr ft^2 F, we write

$$-5500 = - \frac{1}{t/220 + 1/10000} \qquad (54)$$

$$\therefore t = .018 \qquad (55)$$

We therefore conclude that the hysteresis in the operating characteristic of the Problem 4 boiler can be eliminated by simply changing the thickness of the boiler plate. A thickness of .018 ft (.216 in) would eliminate the hysteresis.

(Decreasing the thickness of the boiler plate improves the thermal stability of the boiler by increasing the "thermal coupling" between the heat source and the boiling interface--ie in a

Problem 8 cont.

Sol'n thermal sense, it more closely couples/connects the heat source to the boiling interface.)

In order to improve the boiler thermal stability by modifying the thermal behavior of the condensing interface, we must somehow increase the thermal coupling across the interface. However, there is a physical limitation on the degree to which we can improve this coupling. The physical limit is described by

$$q_c \to \infty \, \Delta T_c \qquad (56)$$

which is another way of saying that there is a negligible temperature difference across the condensing interface. If we were somehow able to accomplish this, but did not alter the thickness of the boiler plate, rel 53 would become

$$\frac{dq_{in}}{d\Delta T_b} \to - \frac{1}{.1917/220 + 1/\infty} = -1148 \qquad (57)$$

Since this is nowhere near the required value of -3000 B/hr ft^2 F, we conclude that the subject hysteresis can NOT be eliminated by modifying the thermal behavior of the condensing interface.

The overall thermal behavior of the Problem 4 boiler with a modified boiler plate thickness of .216 inches is obtained as before, by determining the function $Q\{\Delta T_t\}$ graphically. For a thickness of .216 inches, rel 51 gives

$$q_{in} \to 5500(\Delta T_w + \Delta T_c) \qquad (58)$$

Problem 8 cont.

Sol'n We determine $Q\{\Delta T_t\}$ coordinates as indicated in the following table:

q	ΔT_b from Fig 14	$\Delta T_w + \Delta T_c$ from rel 58	ΔT_t	$Q = qA = .0218\,q$
10000	9	2	11	218
10000	60	2	62	218
20000	10	4	14	436
20000	42	4	46	436

Plotting $Q\{\Delta T_t\}$ coordinates gives Fig 21.

Ans'r The subject hysteresis can be eliminated by changing the boiler plate thickness, but not by altering the thermal behavior of the condensing interface. The operating characteristic of the modified boiler is described in Fig 21, below.

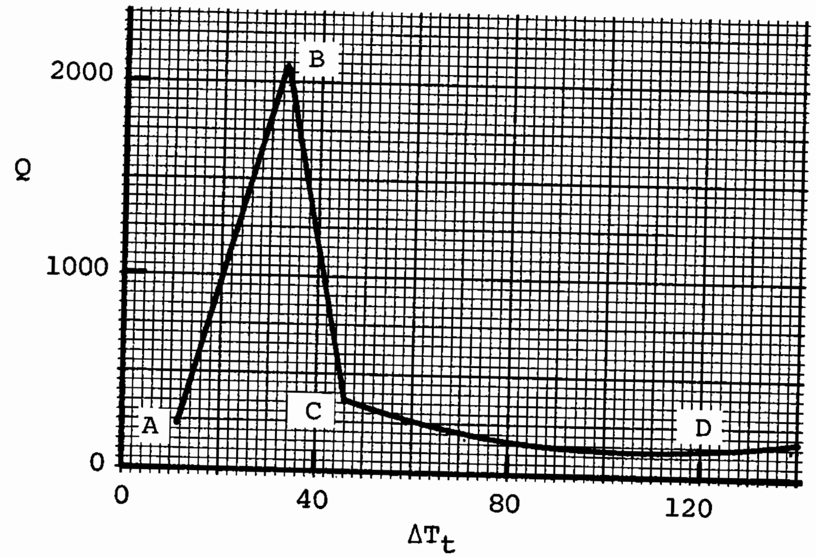

FIGURE 21 The Thermal Operating Characteristic of the Modified Boiler in Problem 8

PROBLEM 9

Describe the thermal behavior of the Problem 4 boiler if the overall temperature difference were 70F and the thermal behavior of the boiling interface were as described in Fig 22, pg 4-39.

Sol'n As in Problem 5, we determine the behavior by plotting $q_{in}\{\Delta T_b\}$ and $q_{out}\{\Delta T_b\}$ on the same graph. The first function is obtained from rel 41 evaluated at $\Delta T_t = 70F$:

$$q_{in} \rightarrow 1030(70 - \Delta T_b) \qquad (59)$$

The second function is obtained from Fig 22. Plotting these functions on the same graph gives Fig 23. Notice that there is only one intersection and that it is thermally UNstable because the slope of the q_{out} curve is more negative than the slope of the q_{in} curve, thus violating the criterion for thermal stability. The fact that the equipment is thermally unstable at the intersection tells us that, should the equipment somehow find itself at this intersection, it will not want to remain there and will actually move AWAY from the intersection. Moreover, since there is only the one intersection, the equipment will not be able to "find" a stable intersection, and so the equipment will go through an unending transient in spite of the fact that the overall temperature difference is maintained at precisely 70F.

If the equipment should somehow arrive at the Fig 23 intersection, it would drift off to either point B or C, and then it would permanently remain on the cyclic loop shown in Fig 24, pg 4-40. The time-dependent behavior of the Fig 24 loop is shown more or less quantitatively in Figs 25 and 26, pgs 4-40 and 4-41.

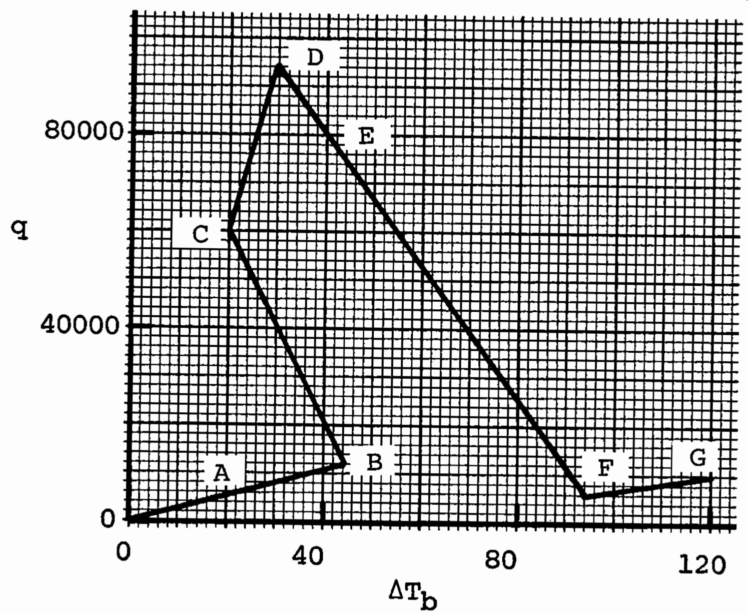

FIGURE 22 The Thermal Behavior of the Boiling Interface in Problem 9

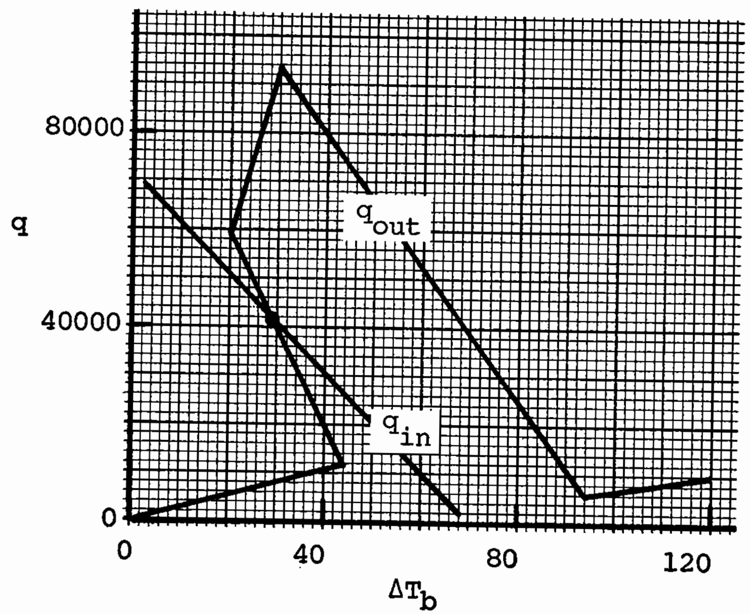

FIGURE 23 Determining the Heat Flux in Problem 9

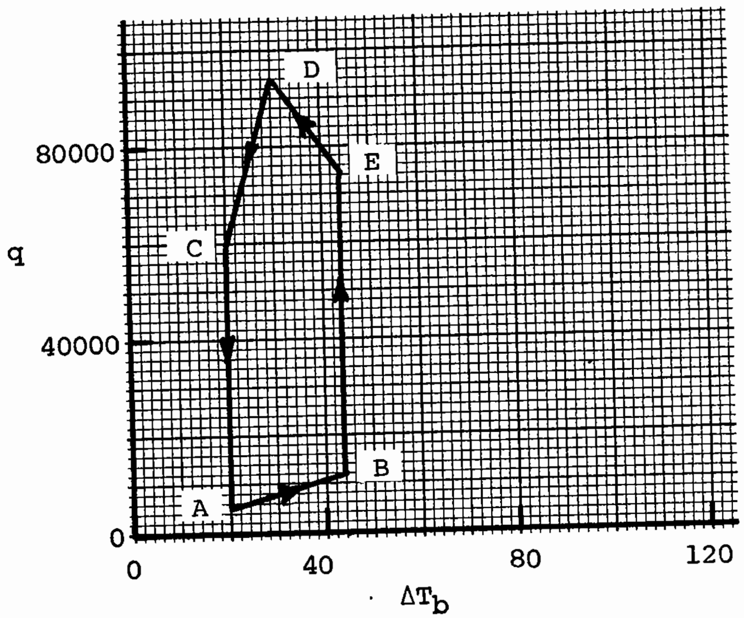

FIGURE 24 Cyclic Thermal Behavior of the Boiler in Problem 9

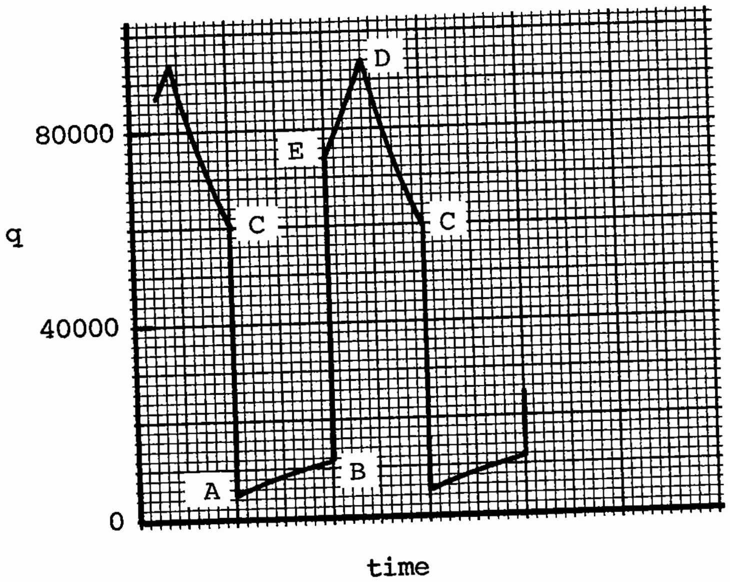

FIGURE 25 Time Dependent Thermal Behavior of the Boiler in Problem 9

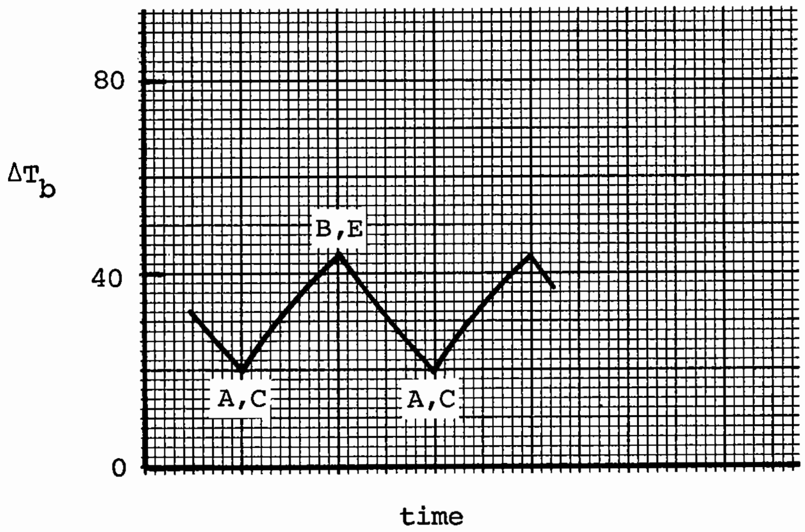

FIGURE 26 Time Dependent Thermal Behavior of the Boiler in Problem 9

The loop in Fig 24 is constructed by inspection of Fig 23. Note that line OAB is a non-boiling region and that q_{in} is greater than q_{out} when the system is at any point on this line. This means that ΔT_b increases whenever the system is on OAB. When ΔT_b increases to the value at point B, boiling begins and the system jumps to point E. Line CDE is a boiling region and, along this line, q_{in} is less than q_{out}. Thus ΔT_b decreases when the system is at any point on this line. When ΔT_b decreases to the value at point C, boiling ceases, the system jumps to point A, and the cycle repeats.

Ans'r The thermal behavior of the subject system is described in Figs 24, 25, and 26.

PROBLEM 10

Describe the overall thermal behavior of the Problem 9 boiler--ie determine $q\{\Delta T_t\}$.

Sol'n As in Problem 4, we determine $q\{\Delta T_t\}$ in the manner suggested in the following table:

q	ΔT_b from Fig 22	$\Delta T_w + \Delta T_c$ from rel 37	ΔT_t
11000	44	11	55
60000	20	58	78
96000	30	93	123
7000	95	7	102
10000	120	10	130

In plotting the $q\{\Delta T_t\}$ coordinates, it must be recalled that intersections on BC of Fig 22 were thermally unstable and resulted in a cyclic heat flux. Since $q\{\Delta T_t\}$ actually refers to steady-state behavior, we have the problem of trying to describe cyclic behavior on a steady-state graph. One way of doing this is to indicate the range of q values which can result from a given value of ΔT_t, and that is the way the BC intersections are described on Fig 27, pg 4-43. Fig 27 is the result of plotting the above $q\{\Delta T_t\}$ coordinates and indicating BC intersections by a range of q values.

Ans'r The overall thermal behavior of the Problem 9 boiler is described in Fig 27 in the form $q\{\Delta T_t\}$.

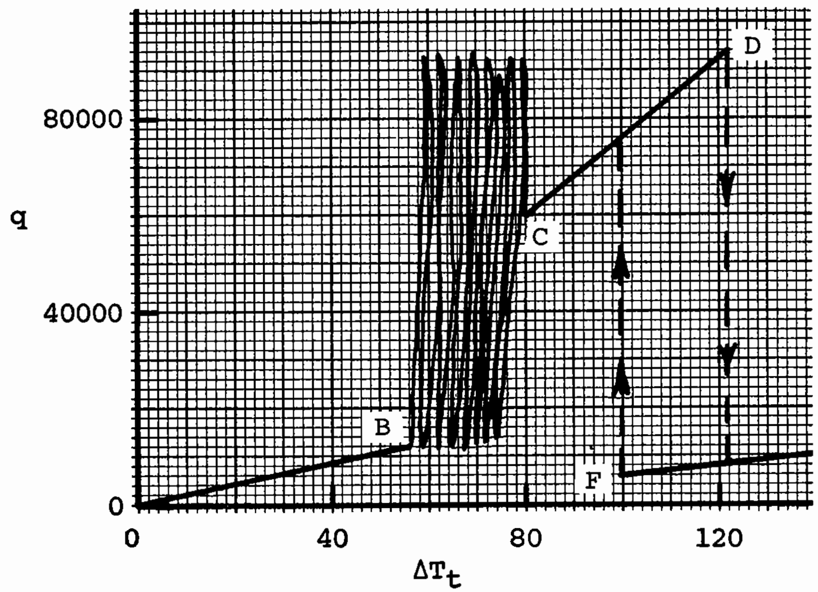

FIGURE 27 Overall Thermal Behavior $q\{\Delta T_t\}$ of the Boiler in Problem 9

DISCUSSION OF PROBLEMS 9 AND 10

Problems 9 and 10 demonstrate that undamped, cyclic operation of a pool boiler strongly suggests that the thermal behavior of the boiling interface resembles Fig 22. Cyclic pool boiler operation has often been observed when boiling liquid metals. However, since the old heat transfer does not effectively deal with highly nonlinear behavior, the researchers who observed this type of boiler operation understandably failed to infer that the "pool boiling curve" contained a negative slope region such as BC in Fig 22. Note from Figs 22 and 27 that oscillatory behavior requires a negative slope region between a maximum and a minimum in $\Delta T_b\{q\}$, whereas hysteresis can result from a negative slope region between a maximum and a minimum in $\Delta T_b\{q\}$ OR $q\{\Delta T_b\}$. Notice also that oscillatory operation and hysteresis are two different EFFECTS which result from the SAME CAUSE--thermal instability.

PROBLEM 11

The oscillatory operation of the Problem 9 boiler (at values of ΔT_t from 55F to 80F) is mechanically and operationally unsatisfactory. The boiler design must be modified so that the boiler will operate without oscillation at all values of ΔT_t. What design modification is necessary to accomplish this? Describe the overall thermal behavior ($q\{\Delta T_t\}$) of a modified boiler.

Sol'n The oscillatory behavior is the result of thermal instability, and this is the result of the fact that the slope of the BC line in Fig 22 is more negative than the slope of the q_{in} line for the Problem 9 boiler design. (See Fig 23.) From Fig 22, the slope of the BC line is -2000 B/hr ft^2 F. Therefore, in order to eliminate the thermal instability and the oscillatory behavior (ie in order to eliminate the cause and its effect), we must modify the boiler design so that $dq_{in}/d\Delta T_b$ is more negative than -2000 B/hr ft^2 F.

The modified boiler design in Problem 8 resulted in a q_{in} function described by

$$q_{in} \rightarrow 5500(\Delta T_c + \Delta T_w) = 5500(\Delta T_t - \Delta T_b) \qquad (60)$$

$$\therefore \frac{dq_{in}}{d\Delta T_b} \rightarrow -5500 \qquad (61)$$

Since -5500 is more negative than -2000, the modified design in Problem 8 should result in operation without oscillation at all values of ΔT_t.

The overall thermal behavior ($q\{\Delta T_t\}$) for the modified boiler design in Problem 8 is obtained

Problem 11 cont.

Sol'n in the manner indicated in the following table:

q	ΔT_b from Fig 22	$\Delta T_c + \Delta T_w$ from rel 60	ΔT_t
11000	44	2	46
60000	20	11	33
94000	30	17	47
6000	95	1	96
10000	120	2	122

The result of plotting the $q\{\Delta T_t\}$ coordinates is shown in Fig 28.

Ans'r The subject oscillatory behavior can be eliminated by modifying the boiler design so that $dq_{in}/d\Delta T_b$ is more negative than -2000 B/hr ft^2 F. Fig 28 describes the behavior of a modified boiler:

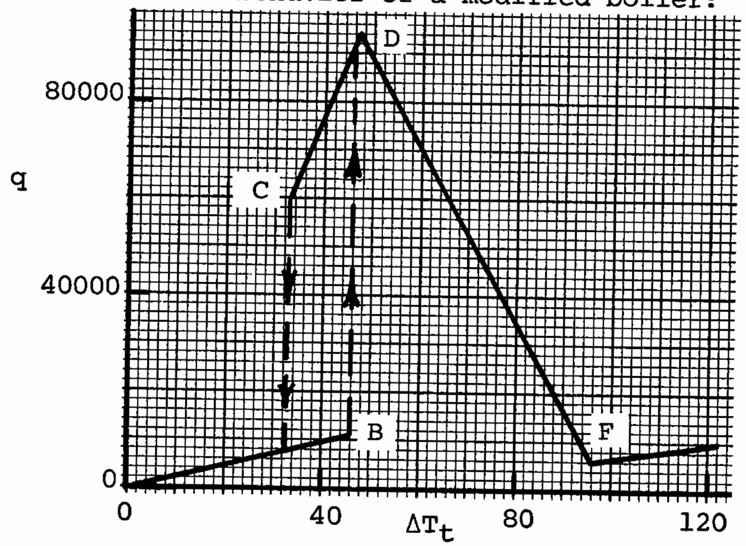

FIGURE 28 The Overall Thermal Behavior of the Modified Boiler in Problem 11

DISCUSSION OF PROBLEM 11

Problem 11 demonstrates that oscillatory operation OR hysteresis WILL RESULT from negative slope regions (such as BC in Fig 22) which lie between a maximum and a minimum in $\Delta T\{q\}$. (Notice that point B in Fig 22 is a maximum and point C is a minimum in $\Delta T\{q\}$.) In other words, we MUST accept either oscillatory operation or hysteresis when we are dealing with an interface whose thermal behavior includes a region in which $dq/d\Delta T$ is less than zero and the region is bounded by a maximum and a minimum in $\Delta T\{q\}$. If the negative slope region and the equipment design are such that

$$\frac{dq_{out}}{d\Delta T} < \frac{dq_{in}}{d\Delta T} \qquad (62)$$

then oscillatory operation WILL result in the negative slope region. However, if the inequality in 62 is NOT satisfied, then hysteresis will result from the negative slope region.

On the other hand, oscillatory operation CAN NOT RESULT from negative slope regions (such as DF in Fig 22) which are bounded by a maximum and a minimum in $q\{\Delta T\}$. In such cases, hysteresis MAY or MAY NOT result. If the inequality in 62 is satisfied, then hysteresis WILL be experienced. If the inequality in 62 is NOT satisfied, then hysteresis will NOT be experienced.

The above results are merely observations and are not intended as substitutes for actual analysis. As shown by Problem 11, it is a simple matter to analyze real systems in order to determine what type of behavior will be experienced in each particular case.

THE THERMAL DESIGN/ANALYSIS OF NON-BOILING KETTLES AND REACTORS

In formulating Problems 4 through 11, the equipment was described as a pool boiler, and Figs 14 and 22 were presented as descriptions of the thermal behavior of the boiling interface. But now let us go back through the problems and formulate them in a somewhat different way. Let us formulate them so that they will in fact deal with the thermal design/analysis of non-boiling kettles and reactors. This is readily accomplished since it requires NO changes in the analyses or the results, and only the following changes in terminology:

1. Substitute the word "kettle" or "reactor" for "pool boiler".

2. Substitute a cooling coil in the kettle liquid for the condenser in the vapor phase.

3. State that a controller in the cooling system maintains the kettle liquid at a constant temperature independent of the heat flow into the liquid. (This is the function of the vent in the pool boiler problems. Thus the controller substitutes for the vent.)

4. Present Figs 14 and 22 as descriptions of the thermal behavior of the kettle wall/liquid interface (or, if the kettle is heated by a submerged heating coil, relate Figs 14 and 22 to the interface between the heating coil and the kettle liquid).

Once we have made these few changes in terminology, it becomes apparent that there is nothing about Problems 4 through 11 which is peculiar to the boiling process, and thus the problems relate to constant temperature kettles in a very general way.

THE THERMAL DESIGN/ANALYSIS OF ONE AND TWO PHASE FORCED CONVECTION HEAT EXCHANGERS AND BOILERS

Flat plate pool boilers are the simplest type of equipment in which highly nonlinear thermal behavior normally occurs. This simplicity is primarily the result of two characteristics normally exhibited by this type of equipment:

1. It is reasonably true that the thermal behavior $q\{\Delta T_t\}$ is independent of location.

2. It is reasonably true that the overall thermal driving force ΔT_t is independent of location.

These two characteristics allow us to write, with reasonable accuracy,

$$Q = \int q\,dA = qA \tag{63}$$

where Q refers to the total heat flow rate in the equipment. Equation 63 is the key to the simplicity of pool boilers. It allows us to integrate the local behavior in the boiler by simply multiplying the local behavior by the heat flow area--ie it allows us to determine the overall equipment behavior by simply multiplying the local behavior by a constant, namely the heat flow area. If it were not for eq 63, the problems would have dealt only with $q\{\Delta T_t\}$ and would have avoided $Q\{\Delta T_t\}$ on the basis that the integration of $q\{\Delta T_t\}$ is a problem which strongly suggests computer solution.

However, the above characteristics also apply to equipment in which the thermal behavior and the heat flux vary with location PROVIDED that we deal with a very small region of such equipment--ie provided we deal with a differential element. Of course, when we do this, we must give up the convenience of eq 63. In its place, we will have to use the more difficult form

$$Q = \int q\,dA = \Sigma\, q_i\, dA_i \tag{64}$$

where q_i is the heat flux in the i element and dA_i is the element heat flow area.

The purpose of the above discussion is to point out that Problems 4 through 11 also apply to one and two phase forced convection heat exchangers and boilers PROVIDED that we go back and make the following changes in terminology:

1. Substitute a differential area of a forced convection heat exchanger for the pool boiler.

2. State that the interfaces which adjoin the heat flow wall can be either one or two phase.

3. Present Figs 14 and 22 as descriptions of interface behavior and leave open the question of whether they resulted from one or two phase flow.

4. Note that, because we are dealing with a differential element, the fluid temperature and flow rate are independent of the heat flow through the wall.

5. Note that Q refers not to the total heat flow in the equipment, but rather to the total heat flow through the element wall--ie Q refers to $q_i\, dA_i$ rather than to qA.

In summary, Problems 4 through 11 apply to forced convection heat exchangers in general, and only the problem of integration/summation is avoided. Since this problem is better handled with the aid of a computer, we will not perform the integration either graphically or analytically. However, we will discuss the computer integration/summation of $\int q\,dA$ in the next chapter.

THE THERMAL DESIGN/ANALYSIS OF EQUIPMENT IN WHICH THE HEAT SOURCE IS NON-THERMAL

Problems 4 through 11 deal with equipment in which the heat source is thermal--ie in which the temperature of the heat source is controlled either manually or automatically. (In the problems, the thermal heat source is the steam and its temperature is controlled manually by an operator/experimenter or automatically by a controller which directly or indirectly senses the temperature of the steam.)

Now let us go back through the problems and substitute a non-thermal heat source for the steam heat source. In this case, the solution of the problems will be affected in that $q_{in}\{\Delta T_b\}$ must be determined in a different and somewhat more difficult way. For non-thermal heat sources, the determination of $q_{in}\{\Delta T_b\}$ involves analysis to determine how q_{in} from the heat source is affected by ΔT_b. In other words, we must determine how the heat generated by the heat source is influenced by the temperature difference across the boiling interface. This obviously requires a considerable understanding of the behavior of the heat source, and explains why the solution of problems like 4 through 11 is somewhat more difficult when the problems involve non-thermal heat sources.

To illustrate, suppose that the boiler plate in the problems is electrically heated and that the heat flux q is controlled by setting the voltage drop across the boiler plate. Further suppose that the electrical "resistivity" of the boiler plate is independent of the boiler plate temperature. In this case, ΔT_b would affect the temperature of the boiler plate, but this would have NO effect on Q or q, and therefore we could write

$$\frac{dq_{in}}{d\Delta T_b} \to 0 \qquad (65)$$

The q_{in} line described by rel 65 would appear as a horizontal line on the $q\{\Delta T_b\}$ graphs, and the value of q would be determined by the value of the controlled voltage drop across the boiler plate. Moreover, rel 65 and the thermal stability criterion tell us that the equipment is thermally unstable if the value of $dq_{out}/d\Delta T_b$ is negative--ie if the thermal behavior of the boiling interface exhibits a negative slope at the operating point of the equipment. As shown in the problems, the result of this thermal instability is oscillatory performance (if the negative slope region is bounded by a maximum and a minimum in $\Delta T_b\{q\}$) or hysteresis (if the negative slope region is bounded by a maximum and a minimum in $q\{\Delta T_b\}$).

Now suppose that the boiler plate material is such that its electrical "resistivity" increases with temperature. In this case, an increase in ΔT_b results in an increase in the boiler plate temperature, thus increasing the "resistance" of the boiler plate and decreasing the electrical current (since the voltage drop across the boiler plate is controlled at a constant value). The decrease in the electrical current results in a decrease in the heat generated in the boiler plate, and this results in a decrease in q and Q. Mathematically, we write

$$Q \to 3.413\ EI = 3.413\ EI_o(1 - \varepsilon(T_{bp} - T_o)) \to q_{in}A \qquad (66)$$

$$\therefore \frac{dq_{in}}{dT_{bp}} \to -3.413\ EI_o\varepsilon/A \qquad (67)$$

where bp refers to the boiler plate and I_o and ε result from linearizing $I\{T\}$. Neglecting the temperature drop through the boiler plate wall, we write

$$T_{bp} = T_{bl} + \Delta T_b \qquad (68)$$

From 67 and 68, we write

$$\frac{dq_{in}}{d\Delta T_b} \rightarrow -3.413\ EI_o\varepsilon/A \qquad (69)$$

The q_{in} line described by rel 69 would have a negative slope on the $q\{\Delta T_b\}$ graphs as indicated by the minus sign, and the value of q would be determined by the value of the controlled voltage across the boiler plate. Relation 69 and the thermal stability criterion indicate that the equipment may or may not be thermally unstable if the value of $dq_{out}/d\Delta T_b$ is negative, and that the stability is decided by the parametric criterion

$$-3.413\ EI_o\varepsilon/A \ < \ \frac{dq_{out}}{d\Delta T_b} \qquad (70)$$

The result of the thermal instability is oscillatory performance or hysteresis as described above.

Notice that if the electrical "resistivity" of the boiler plate material DEcreased with temperature, we would obtain the parametric criterion

$$3.413\ EI_o\varepsilon/A \ < \ \frac{dq_{out}}{d\Delta T_b} \qquad (71)$$

and the equipment could be thermally unstable even if $dq_{out}/d\Delta T_b$ were positive! In this case, the right side of rel 69 would be positive and the q_{in} line on the $q\{\Delta T_b\}$ graphs would show a positive slope.

In summary, if the problems contained a non-thermal heat source, the heat source would have to be analyzed in order to determine $q_{in}\{\Delta T_b\}$ and $dq_{in}/d\Delta T_b$. This requires a good understanding of the nature of the heat source.

UNDERSTANDING AND INTERPRETING POOL BOILER BEHAVIOR

The pool boiling experiment by Berenson was entitled "Transition Boiling Heat Transfer from a Horizontal Surface" (MIT Technical Report No. 17). Yet, as we have noted before, very little data was obtained in the "transition" region (ie the region of negative slope which lies between "nucleate" and "film" boiling) and most of the reported data were actually obtained in the nucleate and film boiling regions. The fact that very little transition boiling data was reported does not in itself prove that the desired data could not have been obtained in the experimental apparatus. (We must allow the possibility that the transition boiling data was simply overlooked.)

However, there is a simple, definitive proof which will tell us whether or not the desired transition boiling data could have been obtained in Berenson's boiler. The proof is based on the fact that q and Q should change in a continuous manner as ΔT_t is increased IF the equipment is such that "there is only one value of heat flux associated with each value of temperature difference" and that "any point on the entire (pool boiling) curve . . . can be reached under stable conditions". (And similarly for a decreasing ΔT_t.) If q and Q change in a continuous manner, then the desired transition boiling data CAN be obtained in the equipment. But if q and Q do not change in a continuous manner--ie if there are step changes in q and Q as ΔT_t is increased or decreased--then the desired transition boiling data CAN NOT be obtained in the equipment because the step changes reflect the fact that the equipment is thermally unstable in part or all of the transition region.

Now look at Figs 18 and 19 which are based on Berenson's reported data, analyzed to determine how Q must have responded to ΔT_t. As ΔT_t is slowly increased from a small value in the nucleate boiling region, q increases in a continuous manner until $q = 96000$ at

$\Delta T_t = 106$. Then, as ΔT_t is increased to 107, q goes through a step decrease from 96000 to 6000. And this step change based on Berenson's reported data is the indisputable proof that the boiler design was such that there was MORE THAN ONE value of heat flux associated with each value of temperature difference and that any point on the entire pool boiling curve COULD NOT be reached under stable conditions. (Note also that there is a step increase in Q as ΔT_t is decreased in Fig 19.)

Notice that the objective in Problem 4 was to determine $Q\{\Delta T_t\}$ and that

$$\Delta T_t = T_{steam} - T_{boiling\ liquid} \qquad (72)$$

Thus, if Berenson had simply reported the steam temperature data, we could have plotted Figs 18 and 19 directly from the data, and we could have settled the question of the thermal stability of the boiler without any analysis at all! The fact that the crucial steam temperature data was not reported is silent testimony to the fact that the old heat transfer simply can not cope with highly nonlinear thermal behavior.

As we noted in Bk 1, Ch 7, a few of Berenson's runs were unusual in that the data did NOT "lie along a straight line connecting the burnout point and the film-boiling minimum point on log-log graph paper". The reason for this unusual behavior was that these runs contained data in the transition region whereas most of the runs contained little or no transition region data. Now let us determine $Q\{\Delta T_t\}$ for one of the "unusual" runs in order to determine whether or not step changes in Q are indicated. Analyzing Berenson's "Run 9" data as in Problem 4 gives the $Q\{\Delta T_t\}$ function described in Fig 29, pg 4-55. Notice that Fig 29 does not indicate that increases/decreases in ΔT_t would result in step changes in Q. However, the fact that the curve becomes almost vertical indicates that

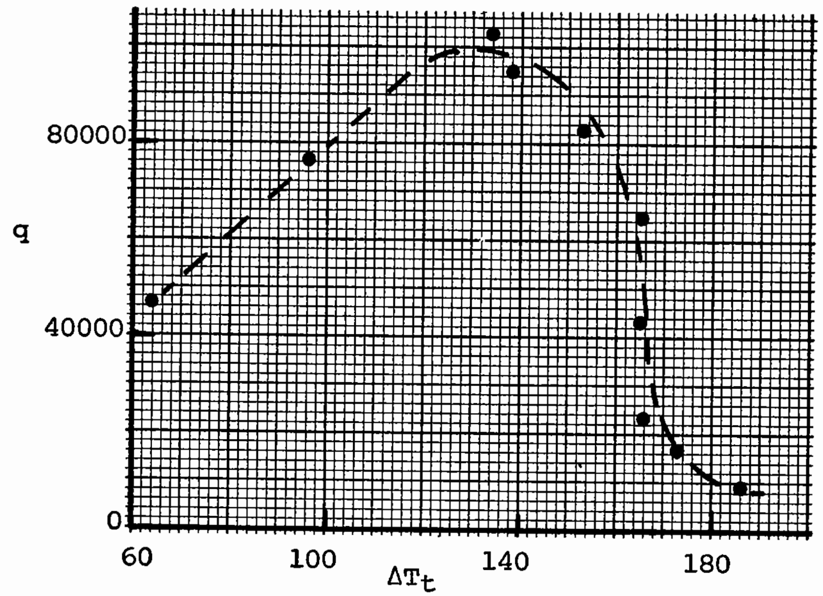

FIGURE 29 Overall Thermal Behavior of Boiler in Ref 1, Run 9 (estimated)

Q was on the brink of a step change--ie a step change would have resulted had $dq_{out}/d\Delta T_b$ in Run 9 been only slightly more negative.

The conclusion to be reached from the above is that Berenson's unreported steam temperature data would indicate quite simply whether or not the desired data could have been obtained with the boiler used in the experiment. Our analysis of the reported boiling interface data strongly suggests that the steam temperature data indicates step changes in Q and thus that the desired data could not have been obtained with the subject boiler. A less obvious conclusion is that the overall thermal behavior of heat flow equipment reveals a great deal about the phenomena occurring within the equipment. Therefore, when performing heat flow experiments, the overall thermal behavior of the equipment should be monitored and the related data should be reported.

IMPROVING POOL BOILER PERFORMANCE BY INFLUENCING THE THERMAL BEHAVIOR OF POOL BOILING INTERFACES

Several of the problems deal with equipment design changes which will improve the pool boiler performance by affecting $q_{in}\{\Delta T_b\}$. As shown in the problems, this requires design changes UPSTREAM (thermally) of the boiling interface. For example, if we were given the boiler in Problem 4, we could make $dq_{in}/d\Delta T_b$ more negative (and thus improve the thermal stability of the boiler) by decreasing the thickness of the boiler plate or by improving the thermal coupling across the condensing interface.

However, it is important to realize that designers can improve pool boiler performance by affecting $q_{out}\{\Delta T_b\}$. In other words, the thermal behavior of a boiling interface is NOT uniquely determined by the boiling liquid--it is determined by the boiling liquid AND several designer controlled parameters. For example, Berenson reports that "The nucleate-boiling heat-transfer coefficient varied by 600 per cent owing to variations in surface finish." and that the "heat-transfer coefficient" increases with increasing surface roughness. His experiments also indicated that $dq_{out}/d\Delta T_b$ in the transition region was strongly dependent on the surface cleanliness, and that $dq_{out}/d\Delta T_b$ was more negative for a clean surface than for an uncleaned surface, the difference being as large as a factor of five or six. Thus, since the Problem 4 conditions were actually based on a clean heat flow surface, we could have markedly improved the thermal stability of the boiler by simply allowing the heat flow surface to become "unclean"!

The important point is that equipment designers CAN CONTROL the thermal behavior of boiling interfaces as well as the thermal behavior of the hardware upstream of the boiling interface. The ability to control the thermal behavior of boiling interfaces adds a whole new dimension to the design of boiling equipment--and markedly improves the designer's ability to cope with

boiling phenomena. To cite only one example, the design/analysis of Pressurized Water Reactors (PWR's) must reflect the fact that the coolant pumps may stop and that the coolant flow rate will decrease considerably before the control rods insert, causing the power to decrease to a small value. During the flow coastdown, some of the fuel elements may enter the transition region and the heat generated in the fuel may exceed the heat flow rate through the boiling interface, resulting in a rapidly increasing fuel temperature. Berenson's results indicate that, at a given ΔT_b in the transition region, the heat flow rate through the boiling interface strongly depends on the "cleanliness" of the interface wall. Thus the "cleanliness" of the fuel element surface strongly influences the rate of fuel temperature rise following a "loss of flow accident" and a "clean" fuel element could experience a rate of temperature rise several times greater than that of an "unclean" fuel element! Thus Berenson's results raise the possibility that the reactor designer could markedly improve the fuel element response to a loss of flow accident by somehow ensuring that the fuel element surfaces are "unclean".

CONCLUDING REMARKS

It is important to recognize that the real subject of this chapter is NOT the thermal design/analysis of pool boilers. The real subject of this chapter is the thermal design/analysis of heat flow equipment IN GENERAL including the design/analysis of pool boilers, one and two phase forced convection heat exchangers, boilers, evaporators, condensers, etc etc. The problems deal with highly nonlinear thermal behavior, but it is obvious that these same methods which effectively deal with highly nonlinear behavior can also deal effectively with linear behavior and also with proportional behavior. The point is that, although the terminology in the problems suggests that the problems deal with pool boilers, they actually deal with heat flow equipment in a very general way. As we noted before, the problems deal with local behavior and avoid the integration/summation which is normally required in order to determine the overall behavior of equipment. This integration/summation is discussed in the next two chapters.

The fact that the analysis in this chapter applies equally to pool boilers and forced convection equipment is evidenced by the fact that this type of analysis was first published in reference 3 where it was applied to the thermal design/analysis of a forced convection, liquid metal boiler.

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CHAPTER 5 THE THERMAL DESIGN/ANALYSIS OF "NONSIMPLE", ONE PHASE, FORCED CONVECTION HEAT FLOW EQUIPMENT

INTRODUCTION

In Chapter 3, we used the new form $q\{\Delta T\}$ to design and analyze forced convection equipment in which the thermal behavior of the heat flow phenomena was strictly proportional. This simple behavior made it possible to integrate the expression for local behavior and thus to obtain an analytical expression for the function $Q\{\Delta T_{OA}\}$ (which we have called the "equipment overall thermal behavior" and also the "equipment thermal operating characteristic"). In Chapter 4, we progressed to the consideration of nonlinear thermal behavior in very simple equipment--ie equipment in which the thermal behavior and the thermal driving force do not vary within the equipment. In this case, the simplicity of the equipment made it possible to integrate the local behavior quite simply and thus to obtain an expression for $Q\{\Delta T_{OA}\}$. In other words, integration was a simple matter in Ch 3 because we dealt with simple phenomena; integration was a simple matter in Ch 4 because we dealt with simple equipment.

In this chapter, the problems involve neither simple phenomena nor simple equipment, and thus integration is not a truly simple matter. Instead of integrating analytically as in Chapters 3 and 4, we will have to integrate by summing up the behavior of differential elements of the equipment. And each summation result will yield the coordinates of only a single operating point on the $Q\{\Delta T_{OA}\}$ function. Thus, if we were to break down a particular exchanger into 100 differential elements, we would have to perform 100 element analyses and to determine 100 sets of boundary conditions in order to determine a single pair of $Q,\Delta T_{oa}$ coordinates. Moreover, if we wished to define the shape of the $Q\{\Delta T_{OA}\}$ function by determining the coordinates of 20 or 30 points on the function, we would have to work with 2000 or 3000 elements and an equal number of boundary conditions.

Obviously, we are not going to analyze thousands of differential elements in this book. Obviously, we are discussing problems which can conveniently be solved only with the aid of a computer. This brings up the question of how to best describe the computer solution of thermal design/analysis problems. One way would be to present the logic and the listing for a program which could solve the problems of interest. A more revealing way would be to present the engineering analysis involved in the computer solution of specific problems of interest, and this is the method used in this chapter and in the remainder of the book. In the problems, the equipment is broken up into differential elements, the elements are analyzed one at a time, and the results obtained in one element establish the boundary conditions for the next element. Of course the problems do not present the analysis of every element in the equipment, but enough elements are analyzed in each problem to completely define the analysis on which the computer program would be based. The particular advantage of this method is that it completely defines the computer analysis and yet it is readily intelligible even to those who are unfamiliar with computer programming methods/conventions.

Just as in Chapter 3, the problems in this chapter deal primarily with double pipe heat exchangers because their simple geometry lends itself to analysis. However, the problems also illustrate the design/analysis of shell and tube heat exchangers, the only difference being that the summation illustrated in the problems is carried out over a number of tubes and then summed. It should be noted that the q_{avg}/q_{LM} and Q/Q_{max} graphs we used to avoid integration/summation when dealing with shell and tube heat exchangers in Ch 3 are of NO USE in this chapter. Here we are dealing with nonsimple thermal behavior, and there is no shortcut alternative to integration/summation.

PROBLEM 1

A double pipe heat exchanger is to be designed to heat Fluid B from an initial temperature of 160F to a final temperature of 300F. Based on the equipment specifications given below, describe a differential element solution which can be used to determine the heat exchanger length.

PROBLEM 1 GIVEN

The heat exchanger is to be a double pipe heat exchanger with Fluid A on the tube side and Fluid B on the shell side. The fluids flow in the same direction--ie the heat exchanger fluids are cocurrent.

The pipe diameters are:

inner pipe	$D_i = .06$	$D_o = .07$
outer pipe	$D_i = .09$	$D_o = .10$

The thermal behavior of the inner pipe wall material is described by

$$q_w \rightarrow 13 \frac{dT_w}{dx} \qquad (1)$$

$$\therefore\ q_{w,i} \rightarrow 2811\ \Delta T_w \qquad \text{(see pg 3-6)} \qquad (2)$$

The thermal behavior of the wall/Fluid A interface is described by

$$q_A \rightarrow \frac{G^{.8}}{D^{.2}} (a + bT + cT^2)\ \Delta T_A \qquad (3)$$

where $a = 4.76 \times 10^{-3}$, $b = 6.05 \times 10^{-5}$, $c = -5.10 \times 10^{-8}$ and

T refers to the bulk temperature of the fluid. The Fluid A flow rate is 1200 lbs/hr, the inlet temperature is 480F, and the heat capacity is as described in Fig 1, below.

The thermal behavior of the wall/Fluid B interface is described by

$$q_B \rightarrow \frac{G^{.8}}{D^{.2}} (a + bT)\ \Delta T_B^{1.25} \quad (4)$$

where $a = 2.13 \times 10^{-3}$ and $b = 3.79 \times 10^{-5}$. The Fluid B flow rate is 1000 lbs/hr, the inlet temperature is 160F, and the heat capacity is as described in Fig 1.

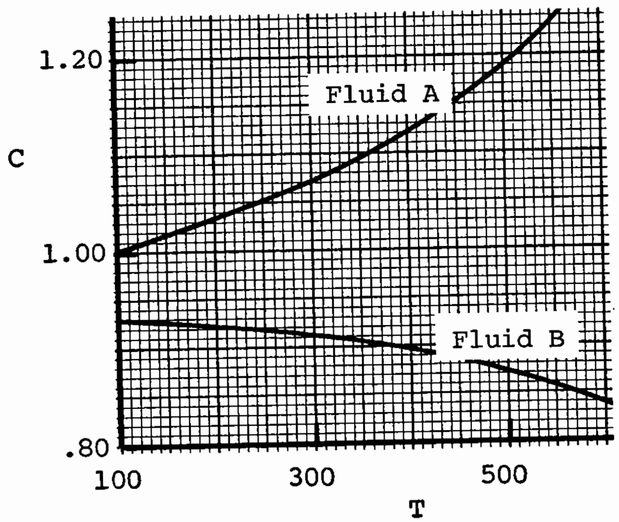

FIGURE 1 Heat Capacity of Fluids A and B

PROBLEM 1 DIFFERENTIAL ELEMENT SOLUTION

We must break down the heat exchanger into differential elements (nodes) small enough that the heat flux will not vary appreciably within the node--let us arbitrarily

set a maximum allowable variation of 5% in heat flux. In other words, the nodes must be small enough that the criterion

$$\frac{q_{max}}{q_{min}} < 1.05 \quad (5)$$

is satisfied within every node. Let us arbitrarily select a node length of 0.1 ft and, in the analysis, verify that criterion 5 is satisfied in each node. Figure 2 shows the nodal arrangement at the inlet end of the exchanger and describes the terminology we will use to identify location in the heat exchanger.

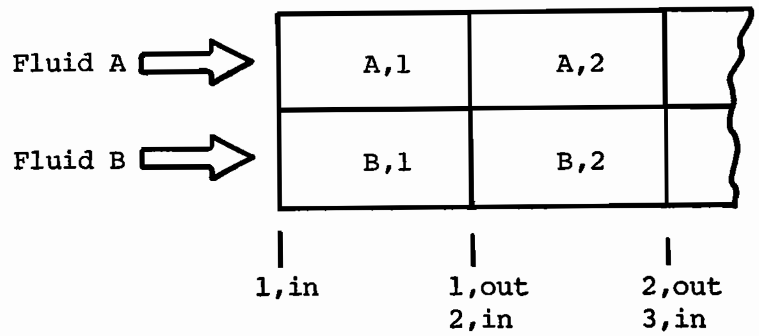

Figure 2 Nodal Nomenclature, Problem 1

What we wish to do is to determine the heat flux and the total heat flow rate in node 1. Since we are given the fluid temperatures at the inlet end of node 1, we can immediately calculate the heat flux at that end. We could then ignore the variation in q within node 1 and obtain the total heat flow rate in node 1 from

$$Q_1 \approx q_{1,in,i} A_{1,i} \quad (6)$$

where q with the subscript 1,in,i refers to the heat flux at the inlet end of node 1 evaluated at the inner diameter of the heat flow wall and A with the subscript 1,i refers to the heat flow area in node 1 evaluated at the inner diameter of the heat flow wall.

However, we could improve the accuracy considerably by using $q_{1,in,i}$ to determine the axial temperature gradient in Fluids A and B and then using this gradient to estimate T_A and T_B in the middle of node 1 from the expressions

$$T_{A,1,mid} \simeq T_{A,1,in} + (dT_A/dx)_{1,in}(\Delta x_1/2) \quad (7)$$

$$T_{B,1,mid} \simeq T_{B,1,in} + (dT_B/dx)_{1,in}(\Delta x_1/2) \quad (8)$$

From the fluid temperatures at the middle of node 1, we can quite accurately estimate the average heat flux in node 1--ie the heat flux based on the temperature estimates in 7 and 8 will closely agree with the true heat flux because of criterion 5 and because the heat capacities of the fluids are well behaved as shown in Fig 2. Using the heat flux at the middle of node 1, we can quite accurately estimate the total heat flow in node 1 from

$$Q_1 \simeq q_{1,mid,i} A_{1,i} \quad (9)$$

The method of eqs 7, 8, and 9 is the method used in the analysis below.

In order to evaluate the heat flux at the inlet end of node 1, we write

$$\Delta T_{t,1,in} = \Delta T_{A,1,in} + \Delta T_{w,1,in} + \Delta T_{B,1,in} \quad (10)$$

Next we determine the relationship between q and each of the ΔT's on the right side of eq 10:

$$G_A = \frac{1200}{(\pi/4)(.06)^2} = 424000 \quad (11)$$

Solving rel 3 for $G_A = 424000$, $D = .06$, $T = 480$ gives

$$q_A \rightarrow 1232\ \Delta T_A \quad (12)$$

$$G_B = \frac{1000}{(\pi/4)(.09^2 - .07^2)} = 398000 \quad (13)$$

Solving rel 4 for $G_B = 398000$, $D = .02$, $T = 160$ gives

$$q_B \rightarrow 541\ \Delta T_B^{1.25} \quad (14)$$

Substituting 2, 12, and 14 into eq 10 gives

$$\Delta T_{t,1,in} \rightarrow \frac{q_{A,1,in}}{1232} + \frac{q_{w,i,1,in}}{2811} + \frac{q_{B,1,in}^{.8}}{153.7} \quad (15)$$

From the pipe diameters and Fig 2, we can write

$$q_{A,1,in} = q_{w,i,1,in} = \frac{.07}{.06} q_{B,1,in} = q_{1,in,i} \quad (16)$$

Combining 15 and 16 gives

$$\Delta T_{t,1,in} \rightarrow \frac{q_{1,in,i}}{1232} + \frac{q_{1,in,i}}{2811} + \frac{q_{1,in,i}^{.8}}{135.9} \quad (17)$$

From the given information and Fig 2,

$$\Delta T_{t,1,in} = 480 - 160 = 320 \qquad (18)$$

Substituting 18 into 17 and solving gives

$$q_{1,in,i} = 175300 \qquad (19)$$

Now that we have determined the heat flux at the inlet end of node 1, we can determine the Fluid A temperature gradient at this same location from

$$\frac{dT_{A,1,in}}{dx} = -\frac{q_{1,in,i}\,\pi D_{1,i}}{W_A C_A} \qquad (20)$$

$$'' \quad = -\frac{175300(.06)}{1200(1.17)} = -23.5 \qquad (21)$$

The temperature of Fluid A at the mid point of node 1 is obtained from this temperature gradient and eq 7:

$$T_{A,1,mid} = 480 - (23.5)(.10/2) = 478.8 \qquad (22)$$

The temperature of Fluid B at the mid point of node 1 is obtained in a similar manner from eq 8:

$$(dT_B/dx)_{1,in} = \frac{175300(.06)}{1000(.926)} = 35.7 \qquad (23)$$

$$T_{B,1,mid} = 160 + 35.7(.10/2) = 161.8 \qquad (24)$$

Now that we have estimated the fluid temperatures at the mid point of node 1, we can determine the heat flux at this location from a relation similar to rel 17 except that it is written for the mid point:

$$\Delta T_{t,1,mid} \rightarrow \frac{q_{1,mid,i}}{1232} + \frac{q_{1,mid,i}}{2811} + \frac{q_{1,mid,i}^{.8}}{135.9} \qquad (25)$$

(Note that the fluid temperatures at the mid point are different than those at the inlet, and this difference would affect the constants 1232 and 135.9 in rel 17. In writing rel 25, we are neglecting this effect because it is obviously small.) From 22 and 24,

$$\Delta T_{t,1,mid} = 478.8 - 161.8 = 317.0 \qquad (26)$$

Combining 25 and 26 and solving gives

$$q_{1,mid,i} = 173500 \qquad (27)$$

$$Q_1 \simeq q_{1,mid,i}\,\pi D_{1,i}\,\Delta x_1 \qquad (28)$$

$$\therefore Q_1 \simeq 173500\pi(.06)(.10) = 3270 \qquad (29)$$

$$\therefore T_{A,1,out} = 480 - \frac{3270}{1200(1.17)} = 477.67 \qquad (30)$$

$$T_{B,1,out} = 160 + \frac{3270}{1000(.926)} = 163.53 \qquad (31)$$

Writing a relation similar to rel 17 except that it is written at the outlet of node 1, and using the fluid temperatures in 30 and 31 to establish $\Delta T_{t,1,out}$, we obtain

$$q_{1,out,i} = 171900 \quad (32)$$

Using the results in 19 and 32 to verify that crit 5 is satisfied, we obtain

$$\frac{q_{max,1}}{q_{min,1}} = \frac{175300}{171900} = 1.020 < 1.05 \quad (33)$$

from which we conclude that the length of node 1 is short enough to yield an accurate result from our approximate methods. Equation 33 completes our analysis of node 1.

Going on to node 2, we note that the fluid temperatures out of node 1 are also the fluid temperatures in to node 2, and this establishes the boundary conditions for node 2. Repeating the above analysis for node 2, we obtain the following results:

$$q_{A,2,in} \rightarrow 1228\ \Delta T_{A,2,in} \quad (34)$$

$$q_{B,2,in} \rightarrow 550\ \Delta T_{B,2,in} \quad (35)$$

$$q_{2,in,i} = 172400 \quad (36)$$

$$\frac{dT_{A,2,in}}{dx} = -23.15 \quad (37)$$

$$\frac{dT_{B,2,in}}{dx} = 35.09 \quad (38)$$

$$T_{A,2,mid} = 476.51 \quad (39)$$

$$T_{B,2,mid} = 165.28 \quad (40)$$

$$q_{2,mid,i} = 170600 \quad (41)$$

$$Q_2 = 3216 \quad (42)$$

$$T_{A,2,out} = 475.38 \quad (43)$$

$$T_{B,2,out} = 167.00 \quad (44)$$

$$q_{2,out,i} = 169000 \quad (45)$$

$$\frac{q_{max,2}}{q_{min,2}} = 1.020 \quad (46)$$

The above analysis is repeated for node 3 and so on until finally $T_{B,n,out}$ reaches the desired temperature, 300F. At that point, the nodal analysis is complete and the required heat exchanger length is obtained from

$$\text{Heat exchanger length} = n\ \Delta x = n(.10) \quad (47)$$

where n is the number of nodes required for $T_{B,out}$ to reach 300F.

PROBLEM 1 ANSWER

The above analysis describes a differential element solution of a design problem dealing with nonsimple, one phase, cocurrent, forced convection heat exchangers.

DISCUSSION OF PROBLEM 1

Problem 1 deals with a specific design problem, but its main purpose is to illustrate the design/analysis of nonsimple, forced convection, one phase heat flow equipment. Although the problem deals with nonlinear thermal behavior and involves equipment in which both the thermal behavior and the overall thermal driving force vary with location, the analysis of the problem is quite simple. This simplicity is the result of basing the analysis on nodes which are sufficiently small to permit certain simplifying assumptions/ approximations regarding thermal behavior and overall thermal driving force. For example, the Problem 1 analysis is based on the simplifying assumptions that

1. The thermal behavior of the interfaces does not vary within each node.

2. The overall thermal driving force varies linearly within each node.

The inaccuracy introduced by assumptions such as these can be controlled in a variety of ways. In Problem 1, this inaccuracy was controlled by limiting the allowable variation in heat flux within any node to 5%. It might also have been controlled by limiting the fluid temperature change in any node, or by limiting the variation in overall thermal driving force, or by some combination of limits.

The magnitude of the error introduced into Problem 1 by the first simplifying assumption is indicated by the difference between the heat flux calculated at the exit end of node 1 and at the inlet end of node 2. Obviously the heat flux is continuous at the boundary between nodes 1 and 2 and therefore there can be no real discontinuity in heat flux at this boundary. The calculated discontinuity reflects the effect of the variation in thermal behavior across the entire node. Since the discontinuity in heat flux at the node 1/ node 2 boundary is only 0.3% ($q_{1,out,i} = 172400$ and

$q_{2,in,i} = 171900$), we may conclude that assumption 1 causes the average heat flux in node 1 to be in error by about 0.3%/2 = 0.15%.

The magnitude of the error introduced into Problem 1 by the second simplifying assumption is indicated by the fact that the heat flux changes only 2% across node 1. Since we are dealing with well-behaved functions, it seems likely that the error introduced by assumption 2 is something less that 2%/10 = 0.2%. It therefore seems reasonable to conclude that the Problem 1 solution is accurate to within about 0.5%.

Notice that Problem 1 deals with difficult equipment (in that the thermal behavior and thermal driving force vary within the equipment) and yet the solution of the problem is no more difficult than the solution of the problems in Ch 4 which dealt only with simple equipment (in that the thermal behavior and thermal driving force did NOT vary within the equipment). All the problems in this chapter and in Ch 4 are simple to solve. The solutions require little more than an understanding of the two "workhorses" of the new heat flow--namely

1. The heat flow is continuous--ie "heat" does not disappear.

2. The total temperature difference is the sum of the individual temperature differences.

Virtually every equipment design/analysis problem involving heat flow by convection and transmittance is solved by applying these two "obvious" requirements. In Problem 1, they are described analytically in eqs 10 and 16, and these two equations are the crux of the Problem 1 solution.

PROBLEM 2

Repeat Problem 1 for the case where the equipment specifications call for a countercurrent heat exchanger and a Fluid A inlet temperature of 400F.

PROBLEM 2 GIVEN

The equipment specifications for this problem are the same as those in Problem 1 with the single exception noted above.

Because of the countercurrent flow of the fluid streams, the in/out subscripts used in Problem 1 would be rather confusing, and so we will use Left/Right subscripts and the terminology indicated in Figure 3:

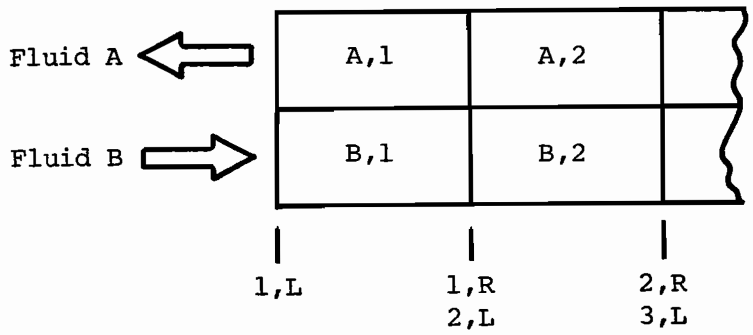

FIGURE 3 Nodal Nomenclature, Problem 2

PROBLEM 2 SOLUTION

As in Problem 1, we wish to solve for the heat flux and the heat flow rate in node 1. As shown in Fig 3, the Fluid B temperature at the left side of node 1 is the same as the Fluid B temperature into the exchanger, 160F. However, the Fluid A temperature at the left

side of node 1 is not part of the given information. Inspection of Fig 3 shows that

$$T_{A,1,L} = T_{A,\text{exchanger outlet}} \qquad (48)$$

The right side of eq 48 can be determined from a heat balance on Fluid B, thus establishing $T_{A,1,L}$. This gives us both fluid temperatures at the left side of node 1 and allows us to determine the heat flux at this location. As in Problem 1, we must also select the simplifying assumptions on which the nodal analysis is to be based, the node size, and the manner in which the accuracy of the result is to be controlled. We could of course make the same two assumptions we made in Problem 1, but for the sake of variety and brevity, let us instead make the following assumptions:

1. Within each node, the thermal behavior does does not vary.

2. Within each node, the overall thermal driving force does not vary.

Note that these assumptions are the equivalent of the single assumption that, within each node, the heat flux does not vary. Let us again use a node length of 0.1 ft, and this time let us limit the inaccuracy in the result by requiring that the difference in heat flux between adjacent nodes not exceed 2%. Should this difference be exceeded, we would reduce the node length and repeat the analysis.

In order to make a heat balance on Fluid B, we must graphically integrate the Fluid B heat capacity in Fig 1 between the limits of 160F and 300F:

$$\int_{160}^{300} C_B \, dT_B = 128.7 \quad \text{(from Fig 1)} \qquad (49)$$

$$\therefore\ Q = 128.7\ W_B = 128.7(1000) = 128700 \qquad (50)$$

$$\therefore\ C_A\ \Delta T_A = \frac{Q}{W_A} = \frac{128700}{1200} = 107.3 \qquad (51)$$

$$\therefore\ \int_{400}^{T_{A,exch\ out}} C_A\, dT_A = 107.3 \qquad (52)$$

From inspection of Fig 1, eq 52 is satisfied if $T_{A,exch\ out} = 302F$.

Now that we have T_A and T_B at the left side of node 1, we are ready to calculate the node 1 heat flux. As in Problem 1, we write

$$\Delta T_{t,1,L} = \Delta T_{A,1,L} + \Delta T_{w,1,L} + \Delta T_{B,1,L} \qquad (53)$$

Next, we determine the relationship between q and each of the ΔT's on the right side of eq 53. Solving rel 3 for $G_A = 424000$, $D = .06$, $T = 302$ gives

$$q_A \rightarrow 1025\ \Delta T_A \qquad (54)$$

Solving rel 4 for $G_B = 398000$, $D = .02$, $T = 160$ gives

$$q_B \rightarrow 541\ \Delta T_B^{1.25} \qquad (55)$$

Substituting rels 2, 54, and 55 into eq 53 gives

$$\Delta T_{t,1,L} \rightarrow \frac{q_{A,1,L}}{1025} + \frac{q_{w,i,1,L}}{2811} + \frac{q_{B,1,L}^{.8}}{153.7} \qquad (56)$$

From the pipe diameters and Fig 3, we can write

$$q_{A,1,L} = q_{w,i,1,L} = \frac{.07}{.06}\, q_{B,1,L} = q_{1,L,i} \qquad (57)$$

Combining 56 and 57 gives

$$\Delta T_{t,1,L} \rightarrow \frac{q_{1,L,i}}{1025} + \frac{q_{1,L,i}}{2811} + \frac{q_{1,L,i}^{.8}}{135.9} \qquad (58)$$

$$\Delta T_{t,1,L} = 302 - 160 = 142 \qquad (59)$$

The solution of 58 and 59 gives

$$q_{1,L,i} = 66800 \qquad (60)$$

The total heat flow in node 1 and the fluid temperatures at the right side of node 1 are obtained by noting that

$$Q_1 = q_{1,L,i}\,\pi D_{1,i}\,\Delta x = 66800\pi(.06)(.10) \qquad (61)$$

$$Q_1 = 1259 \qquad (62)$$

$$T_{A,1,R} = T_{A,1,L} + \frac{Q_1}{W_A C_A} \qquad (63)$$

$$T_{A,1,R} = 302 + \frac{1259}{1200(1.072)} = 302.98 \qquad (64)$$

$$T_{B,1,R} = 160 + \frac{1259}{1000(.926)} = 161.36 \qquad (65)$$

Noting that eqs 64 and 65 establish the fluid temperatures at the left end of node 2, we repeat the above analysis and obtain the following results:

$$q_A \rightarrow 1026\ \Delta T_A \quad (66)$$

$$q_B \rightarrow 544\ \Delta T_B \quad (67)$$

$$\Delta T_{t,2,L} = 302.98 - 161.36 = 141.62 \quad (68)$$

$$q_{2,L,i} = 66600 \quad (69)$$

$$Q_2 = 1255 \quad (70)$$

$$T_{A,2,R} = 303.96 \quad (71)$$

$$T_{B,2,R} = 162.72 \quad (72)$$

Notice that the heat flux at the left side of node 1 is 66800 B/hr ft^2 and at the left side of node 2 is 66600, a difference of only 0.3%. This small difference indicates that the assumption that the heat flux does not vary within the node is quite accurate. It also satisfies the criterion that, between adjacent nodes, the heat flux must not differ by more than 2%.

The analysis is repeated for node 3 and so on until finally $T_{B,n,R}$ reaches 300F, signaling the end of the nodal analysis. (When $T_{B,n,R}$ reaches 300F, $T_{A,n,R}$ should reach 400F if the integrations in eqs 49 and 52 were correct.) The required length of the countercurrent heat exchanger is obtained from eq 47.

PROBLEM 3

Suppose that a 15 foot long, double pipe heat exchanger like the one in Problem 1 is installed and operating with the fluid streams countercurrent. If the incoming fluid streams are as described in Problem 1, what are the fluid outlet temperatures?

PROBLEM 3 SOLUTION

This problem is more difficult than either Problem 1 or Problem 2 because it can not be solved in a direct manner. The difficulty arises because neither fluid outlet temperature is given in the problem statement and the streams are countercurrent. (If the streams were cocurrent, this problem could be solved in a direct manner.) Therefore there is no location in the exchanger at which we can determine the temperature of both fluids in order to establish the overall thermal driving force and thus the heat flux. In other words, we need the thermal driving force to calculate the heat flux, and we need the heat flux to calculate the thermal driving force. That is why we must solve this problem in an indirect manner, for example as described in the following:

1. Arbitrarily pick a value of the outlet temperature of Fluid B. For example, the Fluid B inlet temperature is 160F, and so we might pick a Fluid B outlet temperature of 250F.

2. Determine the corresponding outlet temperature of Fluid A by making a heat balance on Fluid B using Fig 1.

3. Perform the nodal analysis described in Problem 2 until either $T_{B,n,R}$ exceeds 250F or until the end of the heat exchanger is reached at node 150.

4. If T_B reaches 250F before node 150, this indicates that the true outlet temperature of Fluid B would

exceed 250F. Therefore we must pick a value greater than 250F for the outlet temperature of Fluid B and return to step 2.

5. If the analysis reaches node 150 before T_B reaches 250F, this indicates that the true outlet temperature of Fluid B would be less than 250F. Therefore we must pick a value lower than 250F for the outlet temperature of Fluid B and return to step 2.

6. If the analysis reaches node 150 and if the value of $T_{B,150,R}$ is within 1 or 2 degrees of the last value picked for the outlet temperature of Fluid B, the last value picked is the desired result and the nodal analysis is complete.

7. Determine the outlet temperature of Fluid A as in step 2.

As noted above, the nodal analysis in this problem is exactly like that in Problem 2 and we will not bother to repeat it here.

DISCUSSION OF PROBLEMS 2 AND 3

Problem 1 dealt with cocurrent flow of the fluid streams because this arrangement is somewhat easier to deal with. The simplicity results because, with cocurrent flow, we are given the inlet fluid temperatures at the same end of the exchanger. This establishes the overall thermal driving force at a particular location, thus making it possible to determine the heat flux there and elsewhere in a direct manner. With countercurrent flow, we are given the inlet fluid temperatures at opposite ends of the exchanger, and this does not establish the overall thermal driving force at any particular location. Problems 2 and 3 demonstrate how to handle the two types of problems which arise with countercurrent flow.

PROBLEM 4

Repeat Problem 2 with the single exception that the thermal behavior of the wall/Fluid B interface is described by

$$q_B \rightarrow \frac{G^{.85}}{D^{.2}} F\{T,\Delta T\} \qquad (73)$$

where the function $F\{T,\Delta T\}$ is as described in Fig 4.

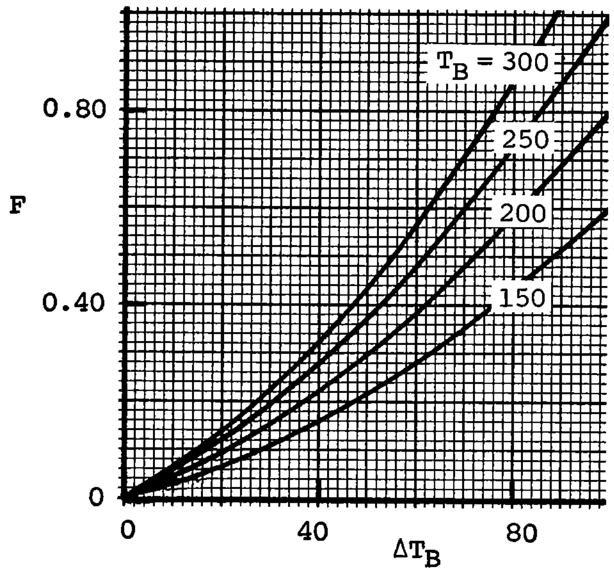

FIGURE 4 The F Function in Relation 73

PROBLEM 4 SOLUTION

Recalling from Problems 1 and 2 that $G_B = 398000$ and that $D = .02$, rel 73 becomes

$$q_B \rightarrow 125800\ F\{T,\Delta T\} \qquad (74)$$

where F{T,ΔT} is described in Fig 4. Relation 74 is shown graphically in Fig 5, and this figure is in the form we require for analysis.

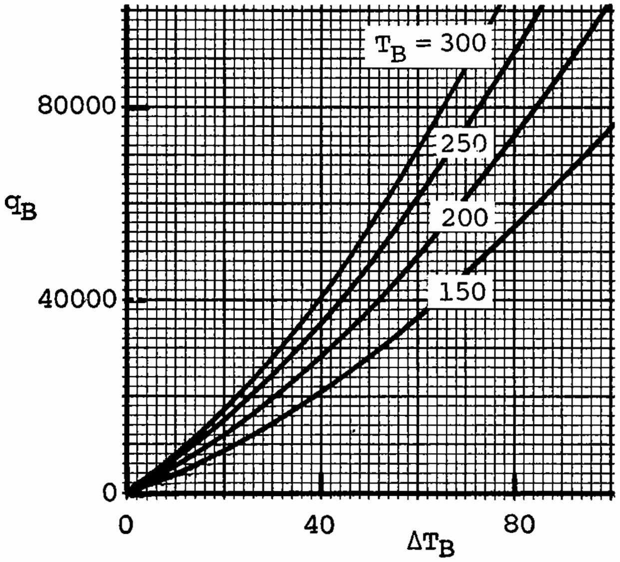

FIGURE 5 Relation 74 in Graphical Form

The solution of this problem is identical to the Problem 2 solution through eq 52 and the determination that $T_{A,exch\ out} = 302F$. The heat flux in node 1 is then determined by noting that Fig 5 describes the heat flux out of the wall/Fluid B interface, and basing the solution on the requirement that the heat flux into this interface must equal the heat flux out of it. In other words, we are going to solve the problem by adding another curve to Fig 5, and this other curve will describe how q into the interface is affected by the temperature difference across the interface. The intersection of these two curves, $q_{in}\{\Delta T_B\}$ and $q_{out}\{\Delta T_B\}$, defines the heat flux.

The $q_{in}\{\Delta T_B\}$ expression for node 1 is obtained by noting that

$$\Delta T_{A,1,L} + \Delta T_{w,1,L} = \Delta T_{t,1,L} - \Delta T_{B,1,L} \quad (75)$$

$$\Delta T_{A,1,L} + \Delta T_{w,1,L} = (302 - 160) - \Delta T_{B,1,L} \quad (76)$$

Combining rels 2 and 54 with eq 76 gives

$$\frac{q_{A,1,L}}{1025} + \frac{q_{w,1,L}}{2811} \rightarrow 142 - \Delta T_B \quad (77)$$

Noting that

$$q_{A,1,L} = q_{w,i,1,L} = \frac{.07}{.06} q_{B,1,L} \quad (78)$$

and combining 77 and 78 gives

$$q_{B,1,L} \rightarrow 91400 - 644\ \Delta T_B \quad (79)$$

Relation 79 is the expression we have been striving for--it describes how the heat flow into the wall/Fluid B interface responds to the thermal driving force across the interface. Plotting rel 79 on a graph which already contains the curves in Fig 5, we obtain Figure 6, pg 5-24. Interpolating linearly between the intersections at T = 150 and T = 200 gives

$$q_{1,L,o} = 47800 \quad (80)$$

where the subscript 1,L,o indicates node 1, Left side, based on outer diameter.

The total heat flow in node 1 and the fluid temperatures at the right side of node 1 are obtained by noting that

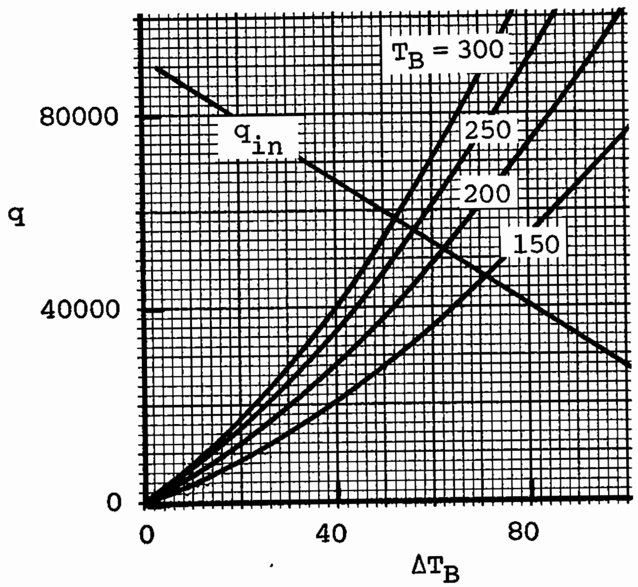

FIGURE 6 Determining the Heat Flux in Problem 4

$$Q_1 = q_{1,L,o} \pi D_{1,o} \Delta x = 47800 \pi (.07)(.10) \quad (81)$$

$$Q_1 = 1051 \quad (82)$$

$$T_{A,1,R} = 302 + \frac{1051}{1200(1.072)} = 302.82 \quad (83)$$

$$T_{B,1,R} = 160 + \frac{1051}{1000(.926)} = 161.13 \quad (84)$$

Repeating the analysis for node 2, we obtain

$$Q_{B,2,L} \rightarrow 91200 - 644\ \Delta T_B \quad (85)$$

$$q_{2,L,o} = 47700 \quad (86)$$

$$Q_2 = 1049 \quad (87)$$

$$T_{A,2,R} = 303.64 \quad (88)$$

$$T_{B,2,R} = 162.26 \quad (89)$$

(Note the very small difference between $q_{1,L,o}$ and $q_{2,L,o}$, demonstrating the accuracy of the assumption that the heat flux does not vary within the node and satisfying the heat flux criterion.) As in Problem 2, the nodal analysis is repeated until $T_{B,n,R}$ reaches 300F, at which point the required length of the heat exchanger is obtained from eq 47.

DISCUSSION OF PROBLEM 4

The main purpose of Problem 4 is to illustrate the nodal solution of one phase, forced convection heat flow problems when the thermal behavior of an interface is described graphically. The nodal/graphical analysis described in Problem 4 can be readily programmed, and would require that the program input include a number of Fig 4 coordinates and that the program contain an interpolation scheme capable of accurately determining $q\{T,\Delta T\}$ coordinates from those obtained by analysis of the input data.

SHELL AND TUBE HEAT EXCHANGERS

The problems in this chapter deal specifically with double pipe heat exchangers, but it should be apparent that they illustrate the thermal design/analysis of nonsimple, one phase, forced convection, shell and tube heat exchangers in a general way. Note that the double pipe heat exchangers in the problems are also modules of a shell and tube heat exchanger, and that a shell and tube heat exchanger is logically designed/analyzed by summing up the performance of the exchanger modules--ie the individual tubes in the exchanger.

For example, suppose we were dealing with the following shell and tube heat exchanger:

1. Number of tubes: 10
2. Number of tube passes: 1
3. Number of shell passes: 1
4. Tube side flow distribution: uniform
5. Shell side thermal behavior varies axially but not radially--ie at a given axial location in the exchanger, $q\{\Delta T_B\}$ does not vary from tube to tube.
6. There is infinite radial mixing in the shell side fluid and therefore there are no temperature gradients in the shell fluid in the radial direction.

Because of items 4 to 6, the thermal performance of each of the ten tubes would be identical. Consequently we would analyze a nonsimple heat exchanger of the above type by performing a nodal analysis of one tube and then multiplying the result by ten. And of course the point is that the analysis is essentially a one tube analysis exactly like the analyses in Problems 1 to 4.

Or suppose that we were dealing with the above exchanger except that item 4 indicated that the tube side flow distribution was nonuniform and that each tube had a different flow rate. In this case, we would analyze the ten tubes separately, one at a time as in the sample problems, and then we would add the results in order to determine the overall performance of the heat exchanger. It should also be noted that, in this case, the axial

temperature gradient in the shell side fluid can be determined only by analyzing ALL the tubes. Note that in the above example, the tube side flow distribution was uniform and thus items 5 and 6 allowed us to write

$$\frac{dT_{shell,n}}{dx} = \frac{10\, q_n\, \pi D_n}{(WC)_{shell}} \qquad (90)$$

where q_n is the heat flux in the nth node of any of the ten tubes. On the other hand, since the flow distribution is NOT uniform in the present case, we must write

$$\frac{dT_{shell,n}}{dx} = \frac{\pi D_n\, \Sigma q_{i,n}}{(WC)_{shell}} \qquad (91)$$

where $\Sigma q_{i,n}$ indicates the summation of $q_{i,n}$ from $i = 1$ to $i = 10$. Equation 91 states that we must analyze the first node of all ten tubes in order to determine the shell side fluid temperature gradient and thereby the shell side fluid temperature in node 2 which we require in order to determine the heat flux in node 2. In other words, we must analyze the first node of all the tubes before we can analyze the second node of any tube.

Or suppose that we were dealing with the exchanger described on pg 5-26 except that item 3 indicated that there were two shell passes and that the first shell pass was cocurrent with the tube pass. In this case, we would analyze two tubes--two "double pipe heat exchangers". One tube would be analyzed as a double pipe, cocurrent heat exchanger with fluid inlet temperatures the same as those specified for the shell and tube exchanger. The second tube would be analyzed as a double pipe, countercurrent heat exchanger and the shell fluid inlet temperature would be the shell fluid outlet temperature from the first tube. The total heat flow in the shell and tube heat exchanger would then be obtained by multiplying the total heat flow in the two analyzed tubes by a factor of ten.

Or suppose that we were dealing with the exchanger described on pg 5-26 except that item 2 indicated there were two tube passes and that the first tube pass was cocurrent with the shell pass. In this case, the analysis would be similar to the above analysis for one tube pass, two shell passes. However, we would have the added complication that the temperature profile of the shell side fluid is determined by the heat flow in both tube passes--both analyzed tubes. Therefore we can not complete the analysis of the first tube until we have completed the analysis of the second tube, and the end result of this round robin is that we must solve the problem in an indirect manner.

This discussion is intended to illustrate that double pipe heat exchangers are in fact modular units of shell and tube heat exchangers, and that the analysis in the double pipe example problems is readily extended to the design/analysis of shell and tube heat exchangers.

THE THERMAL DESIGN/ANALYSIS OF ONE PHASE, FORCED CONVECTION HEAT FLOW EQUIPMENT CONTAINING NONTHERMAL HEAT SOURCES

So far in this chapter, we have considered only heat flow equipment which contains thermal heat sources such as hot fluids. However, nonthermal heat sources such as nuclear or electrical are of great practical importance, and the design/analysis of equipment containing such heat sources is somewhat different than what we have so far discussed.

Nonthermal heat sources may be divided into two types:

1. Those which behave in such a way that the heat flow from the heat source is independent of the temperature of the heat source.

2. Those which behave in such a way that the heat flow from the heat source is weakly or strongly dependent on the temperature of the heat source.

(As we discussed in Book 1, the old heat transfer does not recognize the second type, as evidenced by the old view that the phenomenon called "burnout" can not be avoided in boiling equipment which contains a nonthermal heat source.)

The first type results in a very simple thermal analysis because the heat flux distribution in the equipment is unaffected by thermal considerations and thus the heat flux distribution is one of the boundary conditions for the thermal analysis. For example, suppose that the heat exchanger tube in Problem 1 were an electrically heated rod and that the "resistivity" (using the old electrical engineering) of the rod material were independent of temperature. In this case, the heat flux along the length of the rod would be uniform, and the value of the heat flux would in no way depend on any thermal considerations. The heat flux would be determined by the nonthermal characteristics of the heat source, and thus the heat flux would be specified as a boundary condition for the thermal analysis. With the heat flux specified, the thermal analysis would of course be quite trivial.

But now suppose that the "resistivity" of the rod material is strongly dependent on its temperature. In this case, the heat flux profile is NOT one of the boundary conditions for the thermal analysis, and the thermal analysis must be based on the following input:

1. A description of the functionality between the "resistivity" and the temperature of the rod material.

2. A description of the functionality between power supply voltage and power supply current.

3. A description of the "setting" on the power supply.

For example, suppose that the functionality between the resistivity and the temperature of the rod material is described by

$$\rho = \rho_o(1 + \alpha(T - T_o)) \tag{92}$$

Also suppose that the power supply voltage is independent of the power supply current, and that the power supply "setting" is 57 volts. In order to simplify the analysis, let us assume that the radial temperature gradient in the rod is small and can be neglected. Now suppose that we wish to determine the axial temperature profile of the tube/rod in Problem 1. We again begin at node 1, but this time we must determine the rod temperature from:

1. The electrical current through the rod.

2. The resistivity of the rod material in node 1.

3. The thermal behavior of the rod/fluid interface in node 1.

4. The temperature of the fluid in node 1.

The electrical current is determined by dividing the power supply voltage, 57 volts, by the overall electrical resistance of the rod. However, the overall resistance of the rod is determined by the temperature profile in the rod, and that is the thing we are trying to find. Thus we would have to solve the problem in the following indirect way:

1. Assume a value for the electrical current.

2. Solve for the rod temperature profile based on the assumed electrical current. This involves performing a nodal analysis similar to that in Problem 1 except that q_n, Q_n, and $T_{w,n}$ are obtained by the solution of the following relations and also eq 92:

$$R_n = \frac{\rho_n\{T_{w,n}\}\Delta x_n}{A_{xs}} \tag{93}$$

$$Q_n = I^2R_n \tag{94}$$

$$q_n = \frac{Q_n}{\pi D_n \Delta x_n} \tag{95}$$

$$T_{w,n} = T_{B,n} + \Delta T_{B,n} = T_{B,n} + f\{q_n\} \tag{96}$$

where A_{xs} refers to the cross-sectional area of the rod, R_n refers to the electrical resistance of the nth node, and $f\{q_n\}$ is some unspecified function of q_n.

3. Solve for the voltage drop ΔV in each node and for the voltage drop V across the entire length of the rod from

$$\Delta V_n \rightarrow IR_n \tag{97}$$

$$V = \Sigma\ \Delta V_n \tag{98}$$

4. Compare V with the stated value, 57 volts. If $V > 57$, assume a lower value for the electrical

current and return to step 2. If the converse is true, assume a higher value etc.

5. If V is about 57 volts, the last calculated temperature profile is correct.

The two problems discussed above illustrate that problems involving nonthermal heat sources are sometimes simpler and sometimes more difficult to solve than problems involving thermal heat sources. However, it should be noted that whether the heat source is thermal or nonthermal, the problems are solved in basically the same way--by noting that "heat" does not disappear and that the total temperature difference is the sum of the individual temperature differences.

CONCLUDING REMARKS

The problems in this chapter deal with the practical solution of nonsimple problems involving one phase, forced convection heat flow equipment. The problems are "solved" on the basis of nodal solutions which closely resemble the manner in which the problems would be solved by a computer. The intent is that the problems illustrate the engineering analysis involved in the solution of the problems, and at the same time describe the program logic which would serve as the basis for a computer program. In this way, the examples should be useful for those who are and those who are not familiar with logic diagrams. Also, since numerical results are obtained for one or two nodes in each problem, the example problems should be useful for program debugging.

We have avoided discussing thermal stability in this chapter because our concern is with one phase heat flow and, as we know from the old heat transfer, this form of heat flow is well-behaved. In the next chapter, we take up the subject of thermal stability in forced convection heat flow equipment.

CHAPTER 6 THE THERMAL DESIGN/ANALYSIS OF TWO PHASE, FORCED CONVECTION HEAT FLOW EQUIPMENT

INTRODUCTION

As we know from the old heat transfer, two phase, forced convection heat flow is highly nonlinear. Yet in the old heat transfer, this type of heat flow is usually described in terms of the proportional concept of the "heat transfer coefficient". For example, on page 13-40 of reference 1, Rohsenow discusses forced convection, boiling heat transfer correlations and states:

> Various investigators have suggested a variety of correlations which are gathered together in Table 4. Some equations refer the actual h in the two-phase flow region to . . . The Chen equation is recommended for the lower quality region and Dengler and Addoms for the higher quality region. (Both the Chen equation and the Dengler/Addoms equation are "h" correlations. In fact, ALL the correlations in Rohsenow's Table 4 are "h" correlations. Note also that reference 1 is dated 1973.)

If it seems surprising that the old heat transfer should describe forced convection, boiling heat flow in terms of h{ΔT} and pool boiling heat flow in terms of q{ΔT}, recall that the q{ΔT} pool boiling curves of the old heat transfer are NOT intended to be used for equipment design/analysis. As we discussed in Bk 1, Ch 7, the pool boiling curves of the old heat transfer were presented in the form q{ΔT} because this form makes it possible "to isolate any possible error in the abscissa" as noted by McAdams.

Although pool boiling data are often presented and discussed in the old heat transfer in the form q{ΔT}, the intent is that pool boiling heat flow equipment is to be designed on the basis of heat transfer coefficients. For example, Berenson (2) presents his pool boiling data in the form $q\{\Delta T\}$, then summarizes the data in the form useful for design/analysis in the old heat transfer:

> The nucleate-boiling heat transfer coefficient varied by 600 per cent owing to . . .

Condensing heat flow is another type of two phase heat flow. In the old heat transfer, it is usually dealt with in terms of $h\{q\}$ and $h\{\Delta T\}$, although condensing heat flow data are sometimes presented in the form $q\{\Delta T\}$. Condensing heat flow is much simpler than boiling heat flow because it is only slightly nonlinear. In fact, the condensing heat flow data reported by LeFevre and Rose (3) indicate a highly linear relationship between q and ΔT. In discussing their data, the authors state:

> Graphs showing the steam-to-surface temperature difference as a function of the heat flux have been chosen for presenting the results. THE TRADITIONAL PLOT OF HEAT-TRANSFER COEFFICIENT AGAINST HEAT FLUX HAS BEEN DISCARDED.

In defense of this break with "tradition", the authors state:

> In any case, the steam-to-surface temperature difference and the heat flux are the measured quantities.

To explain why they present and discuss their data in the unusual form $\Delta T\{q\}$ rather than the traditional form $h\{q\}$, the authors state:

> Furthermore, a heat-transfer coefficient vs heat flux plot can be misleading when the error in the steam-to-surface temperature difference is appreciable, since the heat-transfer coefficient involves the reciprocal of this quantity. When the range of error in the steam-to-surface temperature difference is constant over the heat-flux range, the scatter of the heat-transfer coefficients increases as the heat flux falls. Moreover, a symmetric distribution of temperature-difference errors leads to . . .

Note that this argument by LeFevre and Rose is essentially the same argument used by McAdams almost twenty years before to demonstrate that pool boiling data should be presented in the form $q\{\Delta T\}$ and NOT in the form $h\{\Delta T\}$!!! Both arguments are based on the fact that errors in temperature measurement tend to distort the experimental results if the data analysis is based on h--ie if h is one of the coordinates--ie if $q/\Delta T$ is one of the coordinates. Note also that neither argument has anything to do with equipment design/analysis--the sole concern of both arguments is the prevention of distortion in the experimental results. And when the undistorted experimental results have been obtained, the intent is to transform these results to the form $h\{\Delta T\}$ or $h\{q\}$, since these are the forms required for equipment design/analysis in the old heat transfer.

LeFevre and Rose present and correlate their data in the form $\Delta T\{q\}$, and they find that their data are highly correlated by linear, dimensional equations of the form

$$\Delta T = mq + c \tag{1}$$

where m and c are dimensional constants. With regard to these linear, dimensional $\Delta T\{q\}$ equations, the authors very carefully state:

> It is stressed that these linear equations do no more than *represent* the results . . . .

In other words, the authors stress that these equations have NO USE other than to facilitate the discussion of the data--no use for design/analysis. In the abstract of their report, LeFevre and Rose summarize their results in the form useful for design/analysis in the old heat transfer:

> The steam-side heat-transfer coefficient was found to increase with heat flux over the (range 100000 to 570000 B/hr ft^2), the maximum coefficient being about (53000 B/hr ft^2 F).

In this chapter, we deal with forced convection, two phase heat flow using the new heat flow, and this of course means that we will have no use for h{q} or for h{ΔT}. We will use only the form q{ΔT}--the same form we use to present and correlate ALL types of convective heat flow data--the same form we use to design/analyze ALL types of convective heat flow equipment. Although two phase heat flow includes both boiling and condensing, the emphasis in this chapter is on boiling because it is so highly nonlinear and thus presents a more difficult problem to the equipment designer/analyst.

In order to illustrate the design/analysis of boiling equipment, it is of course necessary to have some understanding of the thermal behavior of boiling interfaces. Unfortunately, we cannot obtain this understanding by translating the correlations of the old heat transfer because the old heat transfer simply can not cope with boiling heat flow. Instead, we will qualitatively determine the thermal behavior q{ΔT} of forced convection, boiling interfaces by interpreting certain qualitative observations which are widely accepted in the old heat transfer. We will then use this qualitative thermal behavior as the basis for the equipment design/analysis problems in this chapter.

As in Chapter 5, we are again dealing with complex thermal behavior in complex equipment and therefore there is no shortcut alternative to integration/

summation. For that reason, the problems in this chapter are "solved" in more or less the manner of Ch 5, and again the intent is to present solutions which are useful for those who are and those who are not familiar with computer programming.

THE OLD FORCED CONVECTION, BOILING HEAT TRANSFER CORRELATIONS

Forced convection, boiling heat flow has had great practical importance for approximately 100 years. Yet in all that time, the old heat transfer has not led to even a qualitative description of the functional relationship between q and ΔT during forced convection boiling--not even for water! Nor is it possible to determine this functionality by translating the h correlations of the old heat transfer! And if this seems incredible, the reader should verify it by attempting to determine this functionality by translating the h correlations of the old heat transfer. Those readers who make the attempt will find it to be an exercise in futility.

The truth of the matter is that the old forced convection, boiling heat transfer "h" correlations simply AVOID the nonlinear character of boiling heat flow, and that is why they are of NO USE in attempting to describe the real world behavior of forced convection, boiling heat flow. And if this seems incredible, recall from Bk 1, Ch 6 and Bk 2, Ch 10 that the x/Ms correlations of the old heat transfer simply avoid the nonlinear character of film cooling--and that is why they are of NO USE in dealing with the real world behavior of film cooling.

These unsatisfactory correlations of the old heat transfer have resulted NOT because the methods of the old heat transfer have been improperly applied, but because the old methods themselves are not useful.

Using these old methods, it is readily possible for researchers to "describe" nonlinear heat flow phenomena without describing their nonlinear character. How?

By inventing proportional "parameters" like the "heat transfer coefficient h" and the "film cooling effectiveness η". Parameters such as these make it seem rational to perform experiments over a narrow range and then to apply the results over a wide range. For example, the film cooling experiment we discussed in Bk 1, Ch 6 and Bk 2, Ch 10 resulted in η correlations which related η to the "parameter" x/Ms. The experiment was based on temperature differences of the order of 100F, yet the authors recommend their correlation for "practical applications" in which the temperature differences are often thousands of degrees! By restricting the experiment to small values of temperature difference, the researchers altogether avoided the nonlinear behavior experienced at large temperature differences--and guaranteed that their results would have little usefulness when dealing with practical applications. (Recall that this is essentially the observation made by Papell with regard to the results of the subject film cooling experiment.)

By inventing dimensionless "parameters" like x/Ms and the "Nusselt number" which make it possible to determine the effect of a parameter without varying it. For example, in the film cooling experiment we discussed in Bk 1, Ch 6 and Bk 2, Ch 10, the authors "measured" the effect of M without varying M!!! They varied only the parameter x!!! And this substitution was possible because the "parameter" x/Ms tells us that the effect of M is inversely proportional to the effect of x--and therefore we can measure the effect of M by measuring the effect of x and then inverting it! In this way, the authors altogether avoided the nonlinear character of the relationship between η and M--and ensured that their results would have little usefulness in real world applications!

By casually inventing "regimes" like the "nucleate boiling regime" and the "away from slot regime". Regimes divide nonlinear phenomena into more or less linear fragments and thus make it possible for the researcher to concentrate his attention on the regime of his choice--ie he can concentrate on the trunk or the tail or any small part in between. Unfortunately, this free choice is not available to designers/analysts/blind men who must deal with whole elephants, and regimes are nothing more than an unnecessary, undesirable complication for designers/analysts. (As we saw in Bk 1, Ch 6, regimes sometimes serve to disguise the fact that there is little agreement between the data and the correlation which supposedly describes it.)

In the new heat flow, we have no use for these clever inventions or others like them. For that reason, there is only one way to describe nonlinear phenomena in the new heat flow. And that one way is to actually describe the nonlinear character of nonlinear phenomena.

A QUALITATIVE DESCRIPTION OF THE HIGHLY NONLINEAR CHARACTER OF FORCED CONVECTION BOILING HEAT FLOW

In order to actually design/analyze forced convection boiling equipment, we must first have a quantitative description of the functional relationships among the parameters which define the behavior of forced convection boiling heat flow These parameters would include: pressure, P; enthalpy, H; mass flow rate, G; tube diameter, D; distance from tube inlet, L; heat flux, q; thermal driving force, ΔT. In other words, the design/analysis of forced convection boiling equipment requires a knowledge of the function $q\{P, H, G, D, L, \Delta T\}$ for the fluid of interest.

In this chapter, we are going to set up and "solve" design/analysis problems involving forced convection boiling heat flow equipment, and it would be highly desirable to base the problems on a quantitative description of the thermal behavior of the boiling interfaces in terms of the function q{P, H, G, D, L, ΔT}. At first glance, one might suppose that this function could be obtained by translating the correlations of the old heat transfer. Unfortunately, this is not the case. The correlations of the old heat transfer are the result of avoiding/overlooking/ignoring the nonlinear character of boiling heat flow and therefore this function can NOT be determined quantitatively or even qualitatively by translating the correlations of the old heat transfer.

However, we can qualitatively determine the function q{P, H, G, D, L, ΔT}, at least in part, by interpreting certain observations which have been widely reported and accepted in the old heat transfer. (Recall that this is essentially the method we used in Bk 1, Ch 7 to qualitatively determine the shape of the pool boiling curve for liquid metals.) In this way, we can obtain a partial, qualitative description of q{P, H, G, D, L, ΔT} which includes a description of the nonlinear relationship between q and ΔT during forced convection boiling heat flow. This qualitative description then serves as the basis for the problems in this chapter.

We will not attempt to determine the complete function q{P, H, G, D, L, ΔT} because the complete function is not necessary in order that the problems accomplish their purpose of illustrating the thermal design/analysis of forced convection equipment in which highly nonlinear heat flow phenomena take place. The highly nonlinear character of boiling heat flow is inherent in the partial function q{H,ΔT} and therefore we need not concern ourselves here with the manner in which P, G, D, and L affect the thermal behavior of boiling interfaces. In this chapter, we avoid dealing with the effects of P, G, D, and L by formulating the problems in such a way that P, G, and D are given in the equipment specifications and by making the

simplifying assumption that entrance effects (L effects) are negligible. In this way, the design/analysis problem is simplified considerably, yet we retain the highly nonlinear aspect of the problems since we concern ourselves with the highly nonlinear function, q{H,ΔT}.

We can qualitatively describe the function q{H,ΔT} by interpreting the following widely accepted, empirical observations from the old heat transfer:

1. If the wall temperature is less than the saturation temperature of the fluid, the heat flow is one phase.

2. Boiling heat flow requires that the wall temperature exceed the saturation temperature by a finite amount. In other words, boiling requires that

$$T_{wall} > (T_{sat} + \Delta T_{sh}) \qquad (1)$$

   where ΔT_{sh} is finite and its value is determined by the fluid and the system parameters such as P, G, q, etc.

3. Once boiling begins, small increases in wall temperature result in large increases in heat flux--ie the slope of the q{ΔT} curve increases markedly at the point where boiling begins.

4. The phenomenon of "burnout" is experienced in forced convection boiling as well as in pool boiling. In other words, as the heat flux is monotonically increased beyond the onset of boiling, a point will oftentimes be reached at which a differential increase in heat flux will result in a large increase in wall temperature, indicating a discontinuity in the function ΔT{q}. This discontinuity tells us that there is a maximum in the function q{ΔT}. We will refer to the heat flux at this maximum point with the expression q_{max}.

5. The temperature discontinuity described in 4 is finite and the wall will remain in tact if it has a sufficiently high melting point. The fact that the temperature discontinuity is finite tells us that there must be a minimum in the function $q\{\Delta T\}$ in the region beyond ΔT_{max}. (ΔT_{max} is the ΔT at q_{max}.)

6. As the vapor quality of the fluid approaches 100%, there ceases to be a maximum in $q\{\Delta T\}$.

7. If the vapor quality of the fluid is 100% or if the fluid is superheated, the heat flow is one phase.

8. One phase, forced convection heat flow is such that q is essentially proportional to ΔT.

Now let us use these observations to qualitatively describe the thermal behavior of a boiling interface of a hypothetical fluid which we will call "botherm". Botherm is such that, at a pressure of 400 psia,

$$H_{L,sat} = 500 \qquad T_{sat} = 600$$

$$H_{V,sat} = 1000$$

Since T_{sat} is 600F, observation 1 tells us that there will be one phase, liquid heat flow at 400 psia if the fluid subcooling and the heat flux are such that the wall temperature is less than 600F. Observation 8 tells us that, at these conditions, q is proportional to ΔT and therefore we may conclude that $q\{H,\Delta T\}$ for botherm at 400 psia is graphically described in part by Fig 1, page 6-11.

As q and/or H are increased, boiling begins when

$$T_{wall} = T_{sat} + \Delta T_{sh} \tag{2}$$

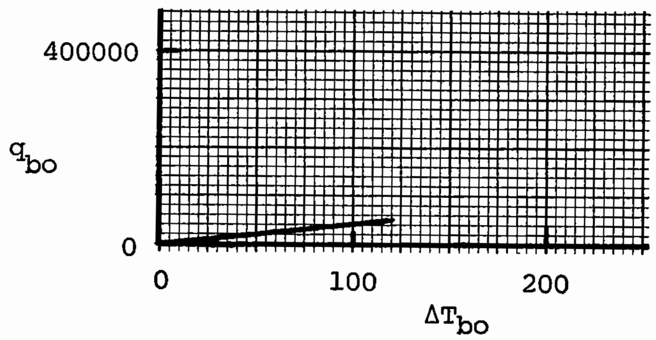

FIGURE 1 Thermal Behavior of Botherm Interface at $P = 400$, $T_w < 600$, Fixed Values of D and G

is first satisfied. When boiling begins, observation 3 tells us that $q\{\Delta T\}$ breaks away from the line in Fig 1, and the breakaway point depends on the temperature of the botherm as shown in Fig 2:

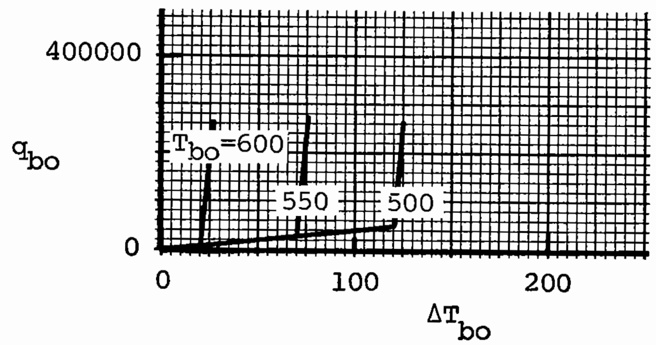

FIGURE 2 Thermal Behavior of Botherm Interface at $P = 400$, Fixed Values of D and G

Observation 4 tells us that each of the boiling curves in Fig 2 passes through a maximum. We do not know the shape of the curves in the region between the onset of boiling and the maxima, nor do we know how the temperature of the botherm affects the value of q_{max}. However, let us arbitrarily extend Fig 2 as shown in Fig 3:

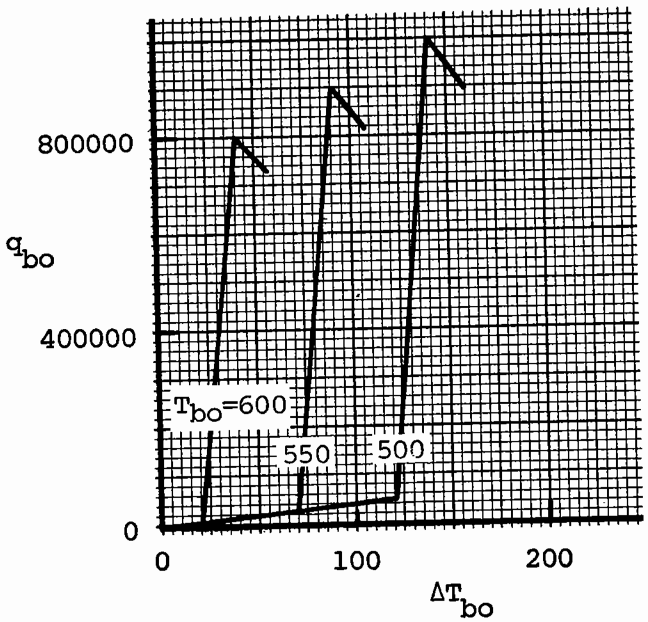

FIGURE 3 Thermal Behavior of Botherm Interface at P = 400, Fixed Values of D and G

In Fig 3, we are SUPPOSING that q{ΔT} is linear in the region between the onset of boiling and q_{max}. (The fact that the pool boiling curve is generally linear in the region between the onset of boiling and the maximum in q suggests that this supposition may in fact be true.) In Fig 3, we are also supposing that q_{max} and T_{bo} are negatively correlated--ie q_{max} INcreases as T_{bo} DEcreases. (This type of behavior has been observed and reported by many experimenters who used the old heat transfer.)

Observation 5 tells us that each of the boiling curves in Fig 3 passes through a minimum in the region beyond ΔT_{max}. We do not know the shape of the curves in the region between the maximum and the minimum, nor do we know how q_{min} is affected by T_{bo}. However, let us arbitrarily extend Fig 3 as shown in Fig 4:

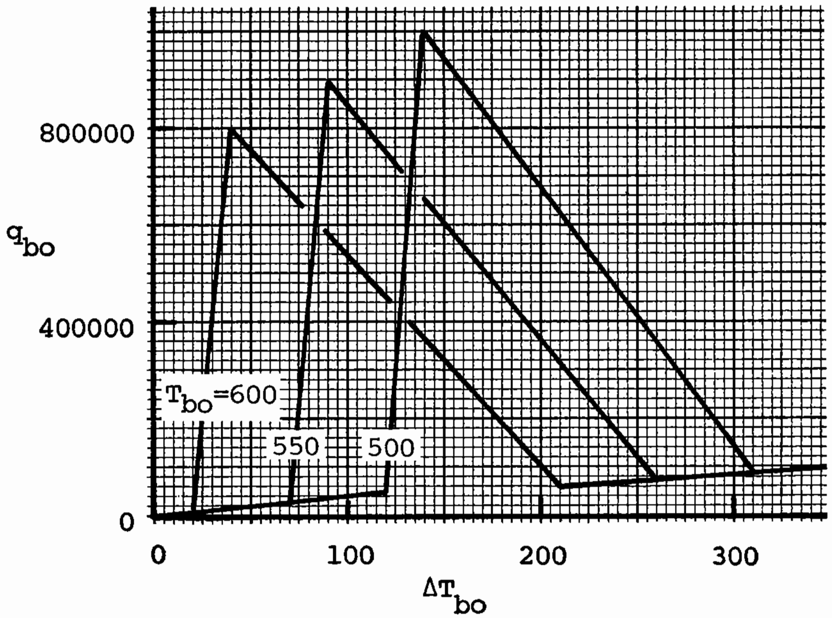

FIGURE 4 Thermal Behavior of Botherm Interface at P = 400, Fixed Values of D and G

Note in Fig 4 that we are SUPPOSING that q{ΔT} is linear in the region between the maxima and the minima (and recall from Bk 1, Ch 7 that q{ΔT} is quite linear in the region between the maximum and the minimum of the pool boiling curve). Note also in Fig 4 that, in the region beyond the minima, we are SUPPOSING that q{ΔT} is linear and is unaffected by T_{bo}.

Figure 4 is a qualitative description of the highly nonlinear thermal behavior of boiling interfaces when the fluid is a subcooled or saturated liquid. Now let us turn our attention to the case where the enthalpy of the fluid equals or exceeds that of saturated liquid--ie where the vapor quality is equal to or greater than zero.

If the vapor quality is zero--ie if the enthalpy of the botherm is 500 B/#--the thermal behavior of the interface is as described by the 600F curve in Fig 4. If the vapor quality is 100% or if the vapor is superheated, observations 7 and 8 tell us that the heat flow is one phase and that the thermal behavior of the interface is such that q is proportional to ΔT. The graphical interpretation of this information is presented in Fig 5:

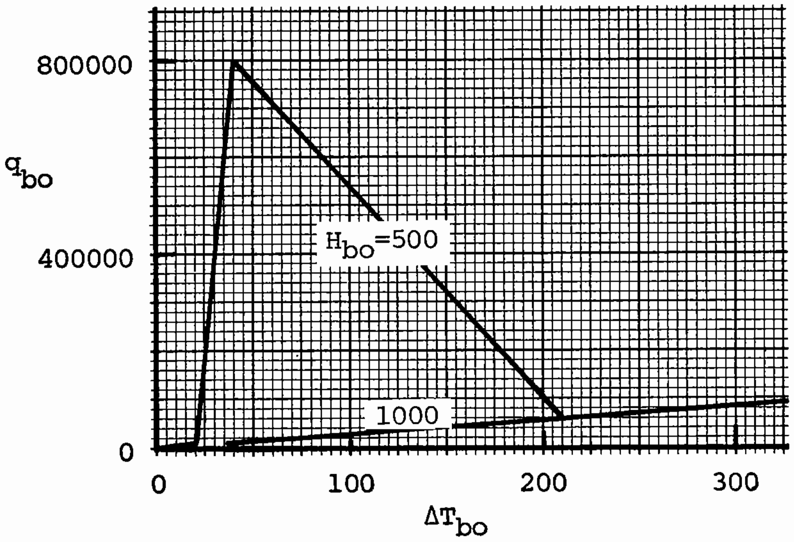

FIGURE 5 Thermal Behavior of Botherm Interface at P =400, Fixed Values of D and G

We do not know how vapor quality (ie how enthalpy in the range 500 to 1000 B/#) affects q{ΔT} for botherm. However, let us arbitrarily extend Fig 5 as shown in Fig 6:

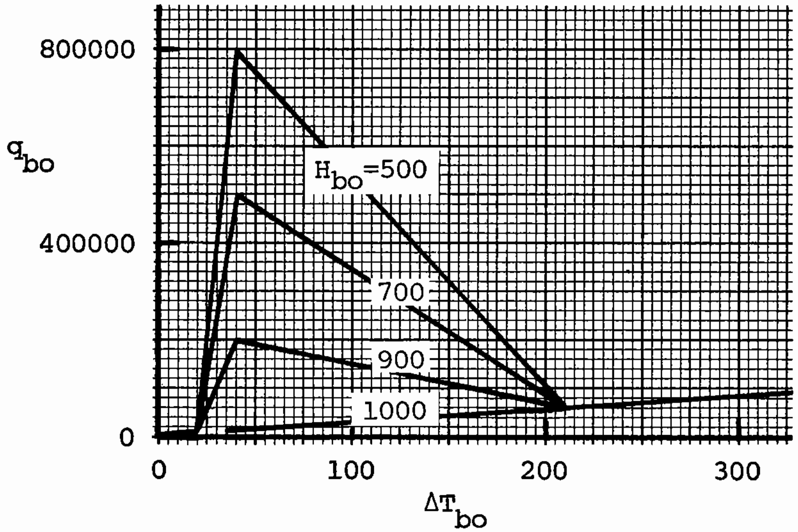

FIGURE 6 Thermal Behavior of Botherm Interface at P = 400, Fixed Values of D and G

Note in Fig 6 that we are SUPPOSING that q_{max} decreases monotonically with H and that ΔT_{max} is unaffected by H in the range between saturated liquid and saturated vapor.

Figures 4 and 6 are what we have been striving for--a qualitative description of the highly nonlinear behavior characteristic of forced convection boiling heat flow. The nonlinear aspect is inherent in q{ΔT} and, as indicated in Figs 4 and 6, fluid temperature/enthalpy has a first order effect on q{ΔT}. The q{ΔT} maxima and minima in Figs 4 and 6 are NOT the result of conjecture/supposition--their existence is based on repeated observations widely reported in the literature of the old heat transfer. Aside from the existence of the maxima and minima, much of Figs 4 and 6 are the result of supposition and analogy to pool boiling behavior. It therefore is quite likely that some of the details in Figs 4 and 6 differ considerably from the real world behavior of forced convection boiling heat flow. However, it should be noted that the primary purpose of Figs 4 and 6 is to provide a description of boiling heat flow behavior which can be used to illustrate the thermal design/analysis of equipment involving highly nonlinear heat flow. Figures 4 and 6 fulfill this purpose quite nicely, and the fact that they differ from real world behavior in certain fine details does not at all compromise their usefulness.

PROBLEM 1

A double pipe heat exchanger is to be used to evaporate botherm. Given the equipment specifications described below, what heat exchanger length would be required in order to produce an exit vapor quality of 80%?

PROBLEM 1 GIVEN

1. The pipe diameters are:

   inner pipe $D_i = .06$ $D_o = .07$

   outer pipe $D_i = .12$ $D_o = .14$

2. The botherm flows through the inner pipe at the following conditions:

   Inlet pressure = 400 Psia

   Inlet temperature = 500F

   Flow rate = 1000 #/hr

   At these conditions, the pressure drop is negligible and so the botherm outlet pressure is also 400 psia.

3. The heat capacity of liquid botherm is 0.7 B/# F.

4. The heat source is a saturated vapor which is admitted to the annulus between the two pipes and condenses on the surface of the inner pipe. The temperature of the saturated vapor is 900F. The temperature of the condensate drained from the annulus is 900F.

5. The thermal behavior of the botherm interface is described by Figs 4 and 6.

6. The thermal behavior of the inner pipe wall is described by

$$q_w \rightarrow 13 \frac{dT_w}{dx} \tag{3}$$

$$\therefore q_{w,i} \rightarrow 2811 \, \Delta T_w \text{ (see page 3-6)} \tag{4}$$

7. The thermal behavior of the condensing interface is described by

$$q_c \rightarrow 15000 \, \Delta T_c \tag{5}$$

PROBLEM 1 SOLUTION

As in the Ch 5 problems, we will analyze small nodes of the heat exchanger. We will not analyze every node, but merely enough to completely describe the solution of the problem. Let us arbitrarily set the node length at 0.1 ft.

Node 1 is at the botherm entrance and therefore T_{bo} is 500F and T_c is 900F. Therefore we may write

$$\Delta T_t = 900 - 500 = \Delta T_c + \Delta T_w + \Delta T_{bo} \tag{6}$$

Combining 4, 5, and 6, we obtain

$$\frac{q_c}{15000} + \frac{q_{w,i}}{2811} + \Delta T_{bo} \rightarrow 400 \tag{7}$$

Since the heat flux through the pipe wall must be continuous, the pipe diameters tell us that

$$\frac{.07}{.06} q_c = q_{w,i} = q_{bo} = q_i \tag{8}$$

Combining 7 and 8 and simplifying, we obtain

$$q_i \rightarrow 2421\ (400 - \Delta T_{bo}) \tag{9}$$

Relation 9 is actually the expression for $q_{in}\{\Delta T_{bo}\}$ in node 1, and the T = 500 curve in Fig 4 is the expression for $q_{out}\{\Delta T_{bo}\}$ in node 1. Since q_{in} must equal q_{out} in the steady-state, we can determine q in node 1 by plotting relation 9 on top of the T = 500 curve from Fig 4 and determining the value(s) of q at the intersection(s). This graphical determination of q in node 1 is shown in Fig 7, page 6-19.

Figure 7 indicates that the steady-state heat flux in node 1 is either 640,000 or 100,000 B/hr ft^2. (Note that we ignore the intermediate value because it is obviously unstable--ie because inspection of the intermediate intersection shows that

$$\frac{dq_{out}}{dT} < \frac{dq_{in}}{dT} \tag{10}$$

and thus the criterion for thermal stability is not

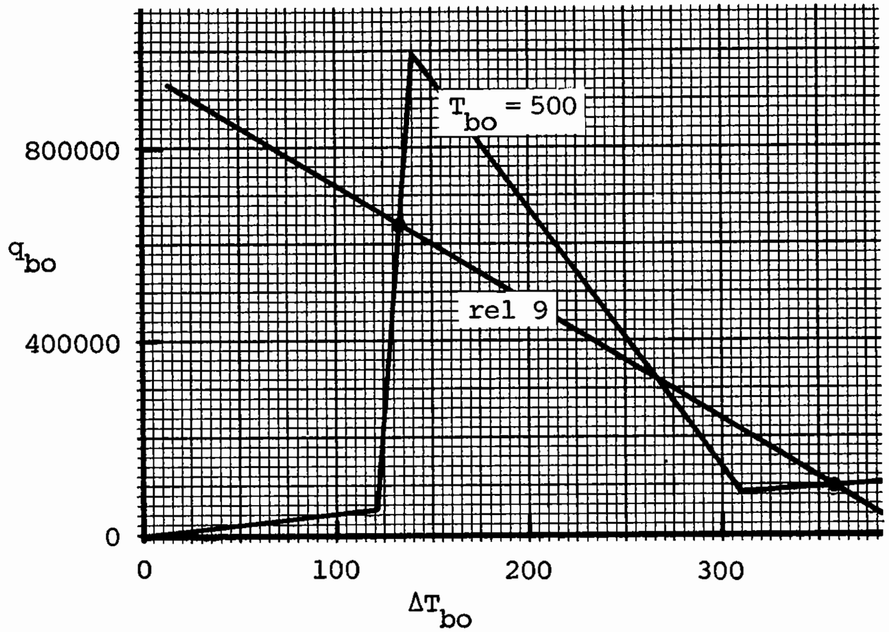

FIGURE 7 Determining the Heat Flux at the Node Where T_{bo} = 500, Problem 1

satisfied.) Both heat flux values are correct in that each one represents a possible operating condition of the equipment. Which operating condition will prevail at any given time will depend on the prior operating history of the equipment. (Recall from Ch 4 that the heat flux in a pool boiler sometimes also depends on the prior operating history of the boiler.) In order to distinguish between conditions resulting from the large and small values of heat flux, let us use the subscripts "hi" and "lo". The total heat flow rate, temperature rise, and botherm outlet temperature for node 1 are obtained as follows:

$$Q = q\ A = q(\pi)(.06)(.1) = .01885\ q \tag{11}$$

$$\therefore Q_{hi} = .01885(640000) = 12060 \tag{12}$$

$$Q_{lo} = .01885(100000) = 1885 \quad (13)$$

$$\Delta T_{bo,hi} = \frac{Q}{WC} = \frac{12060}{1000(.70)} = 17.2 \quad (14)$$

$$\Delta T_{bo,lo} = \frac{1885}{1000(.70)} = 2.69 \quad (15)$$

$$T_{bo,out,hi} = 500 + 17.2 = 517.2 \quad (16)$$

$$T_{bo,out,lo} = 500 + 2.69 = 502.69 \quad (17)$$

Now let us skip a few nodes and analyze the node whose inlet temperature is 550F. For this node, the total temperature difference is (900 - 550) and so rel 9 becomes

$$q_i \rightarrow 2421\ (350 - \Delta T_{bo}) \quad (18)$$

Since the inlet botherm temperature is 550F, we plot rel 18 on top of the T = 550 curve from Fig 4 in order to determine the values of q at the intersections. The resultant plot is shown in Fig 8, page 6-21.

Figure 8 indicates that the heat flux is either 640,000 or 90,000 for the node whose inlet botherm temperature is 550F. By analysis similar to that in rels 11 to 17, we obtain the following results:

$Q_{hi} = 12060$ $\quad \Delta T_{hi} = 17.2$

$Q_{lo} = 1697$ $\quad \Delta T_{lo} = 2.42$

$T_{bo,out,hi} = 567.2$

$T_{bo,out,lo} = 552.42$

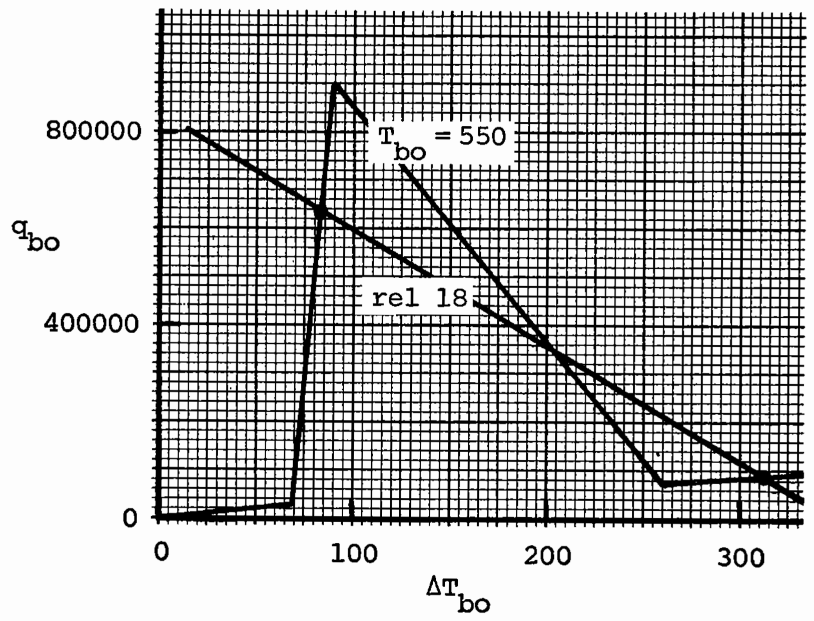

FIGURE 8 Determining the Heat Flux at the Node Where $T_{bo} = 550$, Problem 1

Now let us analyze the node in which the incoming botherm is saturated liquid--ie the botherm temperature is 600F and its enthalpy is 500 B/#. For this node, q_{in} is given by

$$q_i \rightarrow 2421\ (300 - \Delta T_{bo}) \quad (19)$$

and q_{out} is given by the H = 500 curve of Fig 6. The intersections of rel 19 and the H = 500 curve of Fig 6 indicate that the heat flux is either 640,000 or 80,000 B/hr ft^2. By analysis similar to that in rels 11 to 17, we obtain the following results:

$Q_{hi} = 12060$ $\quad \Delta H_{hi} = 12.06$

$Q_{lo} = 1508$ $\quad \Delta H_{lo} = 1.508$

$H_{out,hi}$ = 512.06

$H_{out,lo}$ = 501.51

Turning now to the node in which the enthalpy of the incoming botherm is 600 B/#, q_{in} is again given by rel 19 (since the temperature of the botherm is again 600F). q_{out} is obtained by interpolating between the H = 500 and the H = 700 curves of Fig 6. Plotting q_{in} and q_{out} on the same graph gives Fig 9:

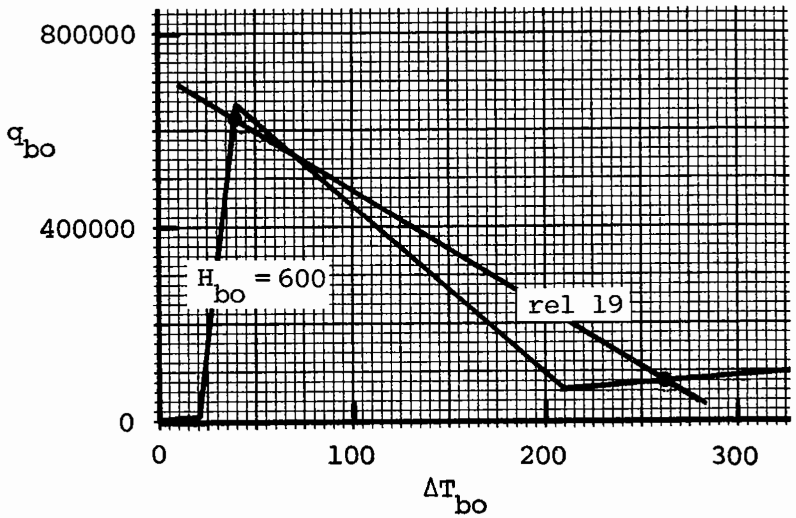

FIGURE 9 Determining the Heat Flux at the Node Where H_{bo} = 600, Problem 1

Figure 9 indicates that the heat flux is either 630,000 or 80,000 when the botherm enthalpy is 600 B/#. By analysis similar to that in rels 11 to 17, we obtain:

Q_{hi} = 11880 ΔH_{hi} = 11.88

Q_{lo} = 1508 ΔH_{lo} = 1.508

$H_{out,hi}$ = 611.88

$H_{out,lo}$ = 601.51

Figure 9 indicates that, for H values larger than about 600 B/#, only the low heat flux solution will be obtained (since the intersection in Fig 9 is near q_{max} and since Fig 6 indicates that q_{max} decreases with increasing H). For example, at H = 700, the graphical solution for q gives Fig 10:

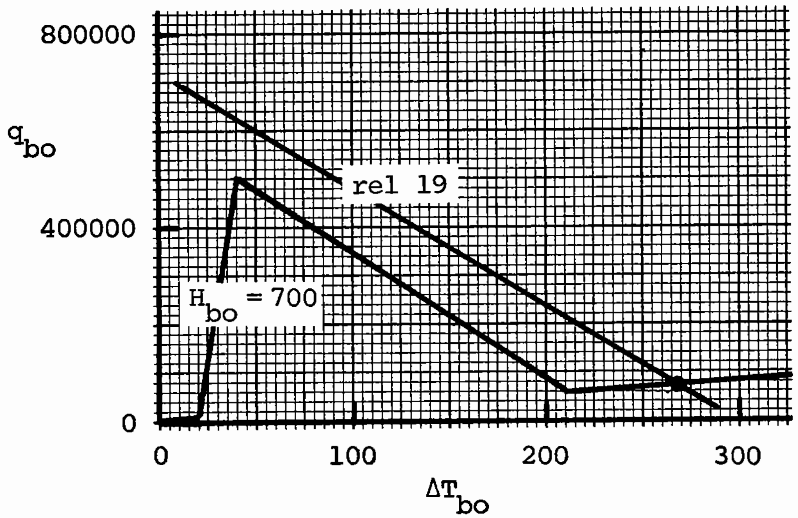

FIGURE 10 Determining the Heat Flux at the Node Where H_{bo} = 700, Problem 1

We therefore conclude that, for H values between about 610 and 900 B/#, the heat flux is single valued and equal to 80,000 B/hr ft^2.

Note that in the above analyses, we have been determining q{T_{bo}} and q{H_{bo}}. These results are summarized in Fig 11, page 6-24. (Note in Fig 11 that the subcooled q{T_{bo}} results have been converted to q{H_{bo}} in order to put all the results on the same basis.) The vertical lines in Fig 11 are intended to indicate that either the hi or the lo heat flux may result at a given H,

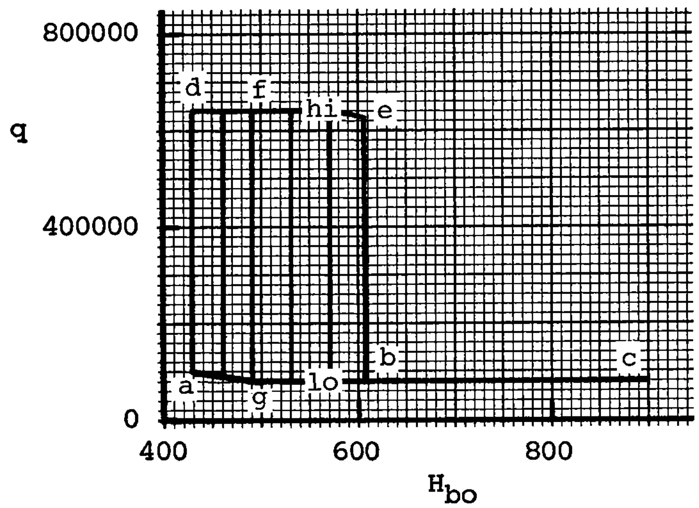

FIGURE 11 Heat Flux Profiles in the Problem 1 Boiler

depending on the prior operating history of the equipment. The vertical lines indicate that there are an infinite number of heat flux profiles and operating modes for the equipment. The extreme operating modes are described by debc and abc in Fig 11.

Figure 11 is particularly useful because it allows us to determine the required length of the heat exchanger. The shortest required length will result if the heat exchanger is operated in such a way that the heat flux profile will be as described by debc in Fig 11. The longest required length will result if the exchanger is operated on line abc. Let us bracket the required length (for an exit vapor quality of 80%) by determining the heat exchanger lengths which correspond to debc and to abc in Fig 11. This determination is based on noting that

$$\frac{dH}{dx} = \frac{q \pi D}{W} = \frac{q(\pi)(.06)}{1000} = .0001885\, q \qquad (20)$$

In the region $430 < H < 610$, Fig 11 indicates that q_{hi} is about 640,000 B/hr ft^2 and therefore

$$\frac{dH}{dx} = .0001885(640000) = 120.6 \qquad (21)$$

$$\Delta x = \frac{\Delta H}{dH/dx} \qquad (22)$$

$$\Delta H = 610 - 430 = 180 \qquad (23)$$

$$\therefore \Delta x = 180/120.6 = 1.5 \qquad (24)$$

In the region $610 < H < 900$, Fig 11 indicates that q is single-valued and equal to 80,000 B/hr ft^2. Therefore

$$\frac{dH}{dx} = .0001885(80000) = 15.08 \qquad (25)$$

$$\Delta H = 900 - 610 = 290 \qquad (26)$$

$$\Delta x = 290/15.08 = 19.2 \qquad (27)$$

We therefore conclude that, with the heat exchanger operating along debc of Fig 11, the length required to produce 80% vapor is (1.5 + 19.2) = 20.7 ft.

Repeating the above analysis along line abc of Fig 11, we obtain the result that the required heat exchanger length is 31.2 ft. We therefore conclude that all the heat flux profiles in Fig 11 are bracketed by heat exchanger lengths of 20.7 and 31.2 ft.

PROBLEM 1 ANSWER

The system parameters described in the equipment specifications do NOT uniquely define the required heat exchanger length because the performance of the heat exchanger is in part determined by the manner in which the equipment is operated. If it is certain that the equipment will always be operated IN THE OPTIMUM MANNER at the conditions listed in the specifications, then a heat exchanger length of 20.7 ft is adequate to ensure that the desired exit vapor quality of 80% will be met. On the other hand, if the possibility exists that the equipment may sometimes be operated in a manner which is LESS THAN OPTIMUM, then a heat exchanger length of 31.2 ft is required to ensure that the exit vapor quality will be at least 80% independent of the manner in which the equipment is operated. (The operating procedure which results in optimum performance is described below.)

DISCUSSION OF PROBLEM 1

Problem 1 demonstrates the new way solution of equipment design/analysis problems involving forced convection heat flow which is highly nonlinear. In spite of the pronounced nonlinearity in Problem 1 (as evidenced by the maxima and minima in Figs 4 and 6), the solution of Problem 1 is extremely simple. In fact, the solution of Problem 1 requires little more than an understanding of three simple ideas:

1. "Heat" does not disappear.

2. The total temperature difference is the sum of the individual temperature differences.

3. Thermal stability requires that dq_{in}/dT be less than dq_{out}/dT.

Problem 1 makes the important point that, when dealing with highly nonlinear thermal behavior, a particular set of boundary conditions may have a number of valid solutions. Note in Fig 11 that there are an INFINITE number of heat flux profiles--ie an infinite number of total heat flow rates--all resulting from the SAME set of boundary conditions! And note that the heat transfer coefficients of the old heat transfer make it virtually impossible to obtain or to understand such multiple solutions.

The solution of Problem 1 is essentially the same as the solution of the pool boiler problems in Ch 4. Each node in Problem 1 is treated like a separate pool boiler with the single exception that the heat flow into the node raises the fluid enthalpy. (When dealing with pool boilers, the heat flow determines the vaporization rate and we are not concerned with the idea of fluid enthalpy.)

It is certainly true that the solution of Problem 1 is quite simple. But it is equally true that this simplicity results because we have dealt with the problem within the framework of the new heat flow--and that the correct solution of the problem would be virtually impossible to obtain within the framework of the old heat transfer.

THE SIGNIFICANCE OF FIGURES 4 AND 6

Figures 4 and 6 present a hypothetical and somewhat simplified description of the thermal behavior of a forced convection boiling interface. We invented this hypothetical behavior NOT because we wanted to, but because we had to in order to provide the proper input for Problem 1. And the reason it was necessary to invent this hypothetical behavior is because the old heat transfer has NOT YET led to even a qualitative understanding of the thermal behavior of forced convection boiling heat flow.

The old heat transfer is based on concepts/methods such as heat transfer coefficients, dimensionless parameters, regimes--all of which make it virtually impossible to describe or to understand or to deal effectively with highly nonlinear forms of heat flow such as boiling. And by simply abandoning these old methods--by simply forgetting about heat transfer coefficients and dimensionless parameters and regimes--it becomes quite possible to describe highly nonlinear heat flow (as demonstrated in Figs 4 and 6) and to understand and deal effectively with it as shown by Problem 1.

Figures 4 and 6 are important in themselves because, although they are hypothetical, they represent a qualitative description/understanding of the true character of forced convection boiling and this has never resulted from the old heat transfer. Figures 4 and 6 are also important because they indicate the direction boiling research should take in order to provide correlations which would be useful for equipment design/analysis. They indicate that boiling research equipment should be designed so that $q\{H,\Delta T\}$ could be measured at various values of P, G, D, L, and these measurements should be the basis for graphical and/or analytical correlations which describe $q\{H,\Delta T\}$ and how this function is affected by P, G, D, and L.

APPRAISING THE EFFECT OF OPERATING HISTORY ON EQUIPMENT PERFORMANCE

In Problem 1, it was mentioned repeatedly that the heat flux profile in the exchanger and the total heat flow rate are in part determined by the prior operating history of the equipment. The determination of the actual heat flux profile and heat flow rate which would result from a given operating history require that we analyze differential elements of the exchanger in much the same manner we analyzed pool boiler behavior in Ch 4.

For illustration, let us analyze the differential element where $H_{bo} = 500$ in order to determine the type of operating history which would result in the hi heat flux at that location. To simplify the problem somewhat, let us suppose that when the system was started up, the botherm flow rate, pressure, and inlet temperature were set at their specification values and that they have been maintained at those values ever since. AFTER the botherm system was on line, the condensing heat source system was started up. For this case, the question of hi or lo heat flux is decided by:

1. The temperature history of the heat source system, $T_c\{time\}$.

2. The thermal behavior at the location of interest, $q\{H=500, \Delta T_t = T_c - 600\}$.

We can determine the function in 2. from the intersections of q_{in} and q_{out} where q_{in} is given by

$$q_{in} \rightarrow 2421(\Delta T_t - \Delta T_{bo}) \quad (28)$$

$$\therefore\ q_{in} \rightarrow 2421(T_c - 600 - \Delta T_{bo}) \quad (29)$$

and q_{out} is given by the $H = 500$ curve in Fig 6. The results of this analysis are:

T_c	q_{hi}	q_{lo}
700	180000	NA
800	410000	NA
835	490000	60000
900	640000	80000
965	800000	90000
1000	NA	100000

Plotting these results gives Fig 12:

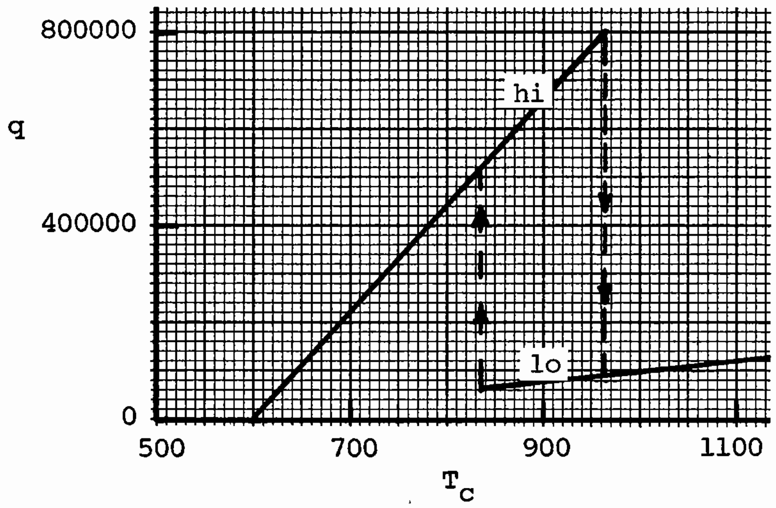

FIGURE 12 Thermal Behavior $q\{T_c\}$ at the Location Where $H_{bo} = 500$

Figure 12 indicates that there is pronounced hysteresis in $q\{T_c\}$ at the location of interest. From Fig 12, we can readily determine whether a given operating history $T_c\{time\}$ would result in the hi or the lo heat flux at the location where $H_{bo} = 500$. For example, if T_c has not exceeded 965F since the last startup, Fig 12 indicates that the hi heat flux would prevail at the location where $H_{bo} = 500$. On the other hand, if T_c has exceeded 965F at any time since the last startup, and if since that time T_c has never been less than 835F, then Fig 12 indicates that the lo heat flux would prevail at the location where $H_{bo} = 500$.

Figure 12 describes the hysteresis behavior at only one location in the heat exchanger. In order to determine the heat flux profile over the entire length of the heat exchanger, the above type of analysis would have to be repeated throughout the double-valued region of the heat exchanger. We are not going to perform

these repeat analyses here. Instead, let us discuss the heat flux profiles which would result from three specific $T_c\{time\}$ operating histories:

1. At startup, T_c was monotonically increased to 900F and has been maintained at that value ever since the startup. This operating history leads to the heat flux profile debc in Fig 11. (Notice that this is also the optimum heat flux profile in that it results in the greatest total heat flow rate.)

2. At startup, T_c was monotonically increased to 900F and maintained there for some time. Twelve hours ago, T_c increased to 965F where it remained for one hour before returning to 900F. Since that temperature excursion, T_c has remained at 900F. This operating history leads to heat flux profile dfgbc in Fig 11. (This result is obtained by inspection of Fig 12 which indicates that a heat source temperature of 965F will result in the lo heat flux for all locations where $H > 500$ and will permit the hi heat flux at all locations where $H < 500$.)

3. At startup, T_c was monotonically increased to 900F and maintained there for some time. Five hours ago, T_c increased to 1100F where it remained for fifteen minutes before returning to 900F. Since that temperature excursion, T_c has remained at 900F. This operating history leads to heat flux profile abc in Fig 11. (Notice that this is the worst possible heat flux profile in that it results in the smallest total heat flow rate.)

By an analogous line of reasoning, the heat flux profile abc in Fig 11 would also result if the heat source system were brought on line first and the botherm flow rate was then monotonically increased from 0 to 1000 lbs/hr. In other words, the order in which the systems are brought on line can have an important effect on the performance of the heat flow equipment when one is dealing with highly nonlinear thermal behavior!

PROBLEM 2

Referring to Problem 1, it was decided that the system would be operated in the optimum manner and at the conditions described in the specifications. Therefore a heat exchanger length of 20.7 ft was ordered and installed. After installation, it was discovered that the desired exit vapor quality was 85% rather than the 80% which was the basis for the heat exchanger design. In order to obtain the higher vapor quality without decreasing the botherm flow rate, it was of course necessary to modify the system so as to increase the total heat flow rate in the exchanger. Toward this end, one of the project engineers recommended that the temperature of the condensing heat source system be increased from the specification value (900F) to the maximum allowable working temperature (1000F). The intent of this recommendation was to obtain the maximum possible heat flow rate from the installed hardware.

Comment on this recommendation in view of the fact that the subject system involves highly nonlinear thermal behavior. Would it be possible to make a better recommendation?

PROBLEM 2 ANSWER

The recommendation to increase the temperature of the heat source system would be just fine IF we were dealing with proportional or with linear thermal behavior. But since we are dealing with highly nonlinear thermal behavior, there is a strong possibility that an INcrease in the temperature of the heat source will result in a DEcrease in the heat flow rate! Thus the suggestion to increase the heat source temperature to 1000F may result in a decrease in the heat flow rate and a DEcrease in the exit vapor quality rather than the intended increase!

When dealing with highly nonlinear thermal behavior,

conclusions which seem "obvious" are oftentimes wrong! And the reason these wrong conclusions seem "obvious" is because we approach them from the standpoint of the old heat transfer and its proportional concept of the heat transfer coefficient. The old heat transfer would have us think in terms of

$$q = h\ \Delta T \qquad (30)$$

$$q = U\ \Delta T \qquad (31)$$

And certainly these equations make it seem "obvious" that the heat flow rate will increase if we increase ΔT by increasing the temperature of the heat source or decreasing the temperature of the heat sink. And these equations make it seem equally "obvious" that the heat flow rate is a single-valued function of ΔT and thus that the specification of the boundary conditions should uniquely define the heat flow rate in a given heat exchanger. These conclusions are "obvious" in the old heat transfer, but they are also wrong. These "obvious" conclusions bear no resemblance to real world behavior when we are dealing with highly nonlinear thermal behavior such as forced convection boiling or pool boiling.

In the new heat flow, we have NO USE for eqs 30 and 31. In the new heat flow, we think in terms of

$$q \rightarrow f\{\Delta T\} \qquad (32)$$

Relation 32 tells us that the heat flow rate is some unspecified function of ΔT. And since the function is unspecified, rel 32 leads to no "obvious" and wrong conclusions about the functionality between heat flow rate and ΔT. Relation 32 says that q may increase or decrease as the result of an increase in ΔT. It says that q may be a single-valued or a multi-valued function of ΔT and thus that the specification of the boundary conditions may or may not uniquely define the heat flow rate in a given heat exchanger. In short, rel 32 says simply that convective heat flow is related to temperature difference, and one advantage of this new way expression is that it does not promote "obvious" conclusions which are wrong.

From the standpoint of the new heat flow and rel 32, it is apparent that increasing the heat source temperature may not increase the heat flow rate in the heat exchanger and therefore we reject the recommendation to increase T_c. A better recommendation would be to perform additional analysis to determine whether some value of T_c other than 900F would result in a higher heat flow rate. However, since the equipment has already been installed, an even better recommendation would be to use the system to experimentally determine how the total heat flow rate is related to the heat source temperature--ie to experimentally determine $Q\{T_c\}$. This could be accomplished quite simply by bringing the botherm system on the line and then monotonically increasing T_c in steps while monitoring the total heat flow rate. The experimental results would then establish beyond any doubt the optimum value of T_c and the maximum values of total heat flow rate and exit vapor quality which can be obtained with the installed equipment.

PROBLEM 3

Suppose that the specifications in Problem 1 had stated that the temperature of the heat source system could be set at any value recommended by the heat exchanger designer in the allowable range (less than 1000F). What heat source temperature would permit the shortest possible heat exchanger? What would be the length of this shortest possible heat exchanger?

PROBLEM 3 SOLUTION AND ANSWER

The heat exchanger length L is in part determined by T_c, and the intent in this problem is to determine the value of T_c which results in the minimum L. We can determine this by evaluating the function $L\{T_c\}$ and then examining the results to determine the minimum L in the allowable range, $T_c < 1000$. Recall that in Problem 1, we determined $L\{T_c = 900\}$. The solution of the present problem requires that we repeat the Problem 1 analysis for several values of T_c in order to obtain a more or less continuous description of the function $L\{T_c\}$ in the allowable range. The results obtained by repeating the Problem 1 analysis are

T_c	L
600	∞
650	36
700	14
750	9
800	7.5
850	20
900	21
1000	23
1100	20
∞	0

These results indicate that, in the allowable range, a heat source temperature of about 800F would permit the shortest possible heat exchanger in Problem 1. For this heat source temperature, the required heat exchanger length is 7.5 ft.

DISCUSSION OF PROBLEM 3

In Problem 1, we determined that a heat source temperature of 900F required a heat exchanger length of 20.7 ft in order to obtain an exit vapor quality of 80%. In Problem 3, we determined that a heat source temperature of 800F required a heat exchanger length of only 7.5 ft to obtain the same exit vapor quality. Thus by simply LOWERING the temperature of the heat source from 900F to 800F, we DEcreased the required heat exchanger length by almost a factor of three!

The tabular results in Problem 3 also tell us that, had the heat source temperature been increased to 1000F as recommended in Problem 2, the result would have been a slight DEcrease in the total heat flow rate and in the exit vapor quality (since the tabular results indicate that a heat exchanger length of 23 ft would be required to attain an exit vapor quality of 80%). On the other hand, the tabular results also indicate that the exit vapor quality would have increased as desired if the heat source temperature had been decreased to 800F--and in fact the exit vapor quality would then have exceeded the desired 85% by a comfortable margin!

PROBLEM 4

Suppose that it were somehow possible to vary the temperature of the heat source along the axis of the heat exchanger. (For example, suppose that the heat exchanger annulus were partitioned into many compartments and that the heat source temperature in each compartment could be varied independently.) What heat source temperature profile $T_c\{H_{bo}\}$ would permit the shortest possible heat exchanger length? What heat flux profile $q\{H_{bo}\}$ would result from this heat source temperature profile? What would be the length of this shortest possible heat exchanger? Note: Assume that the maximum allowable working temperature of the heat sourc system is 1100F.

PROBLEM 4 SOLUTION

The solution of Problem 4 requires that, at each location in the heat exchanger, we determine the heat source temperature which would result in the maximum local heat flux. This in turn requires that we analyze $q_{in}\{T_{bo},T_c,\Delta T_{bo}\}$ and $q_{out}\{H_{bo},\Delta T_{bo}\}$ to determine what value of T_c will result in the largest value of q. The q_{in} function is obtained in the same manner we obtained rel 9 and is described by

$$q_{in} \to 2421(T_c - T_{bo} - \Delta T_{bo}) \qquad (33)$$

The q_{out} functions are described by the curves in Figs 4 and 6.

Now that we have defined the problem and the required relations, let us analyze the location at the heat exchanger entrance--ie at the location where $T_{bo} = 500$. At this location, rel 33 indicates that q_{in} is described by

$$q_{in} \to 2421(T_c - 500 - \Delta T_{bo}) \qquad (34)$$

The q_{out} function at this location is described by the $T_{bo} = 500$ curve in Fig 4. What we must now determine is what value of T_c will cause the q_{in} function to pass through the maximum in the q_{out} function. This value of T_c is readily determined by noting that the coordinates of the maximum in the q_{out} function are

$$q_{out} = 1.0 \times 10^6 \text{ at } \Delta T_{bo} = 140$$

Noting that $q_{in} = q_{out}$ and combining these coordinates with rel 34, we obtain

$$T_c = \frac{1.0 \times 10^6}{2421} + 500 + 140 = 1053 \qquad (35)$$

Repeating this analysis at other locations in the heat exchanger, we obtain the following results:

T_{bo}	H_{bo}	q_{max}	ΔT_{max}	$T_c\{q_{max}\}$
500	430	1000000	140	1053
550	465	900000	90	1012
600	500	800000	40	970
"	600	650000	"	908
"	700	500000	"	847
"	800	350000	"	785
"	900	200000	"	723

Plotting T_c vs H_{bo} and q_{max} vs H_{bo} gives the desired profiles shown in Figs 13 and 14:

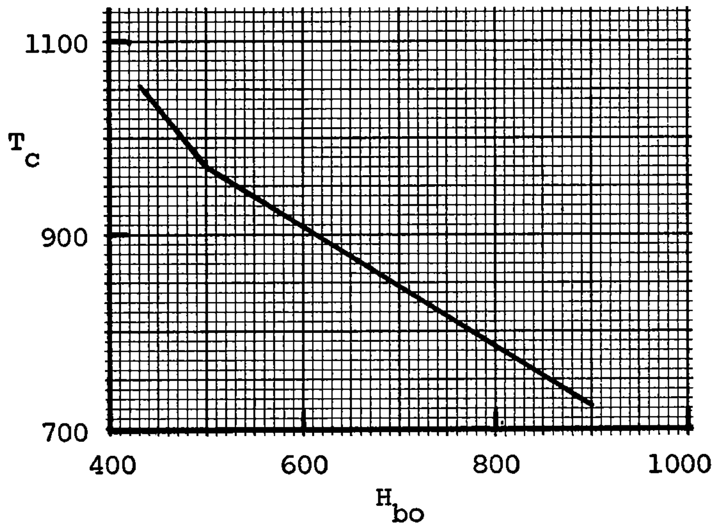

FIGURE 13 The Heat Source Temperature Profile Which Gives the Shortest Possible Heat Exchanger, Problem 4

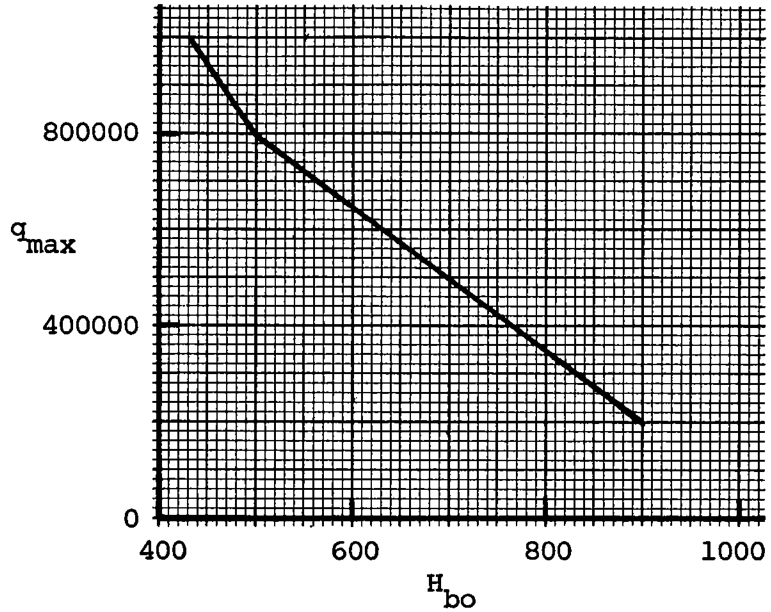

FIGURE 14 The Heat Flux Profile Which Gives the Shortest Possible Heat Exchanger, Problem 4

The heat exchanger length corresponding to the optimum profiles in Figs 13 and 14 is obtained by recalling from page 6-24 that

$$\frac{dH}{dx} \rightarrow .0001885q \qquad (36)$$

and noting from Fig 14 that $q_{max}\{H_{bo}\}$ is described by

$$q_{max} \rightarrow 2.23 \times 10^6 - 2857H_{bo} \qquad 430 < H_{bo} < 500 \qquad (37)$$

$$q_{max} \rightarrow 1.55 \times 10^6 - 1500H_{bo} \qquad 500 < H_{bo} < 900 \qquad (38)$$

Substituting 37 and 38 into 36 and integrating between the limits $H_{bo} = 430$ and $H_{bo} = 900$ results in a heat exchanger length of 5.3 ft.

PROBLEM 4 ANSWER

The heat source temperature profile in Fig 13 is optimum in that it results in the maximum possible heat flux at each location in the Problem 1 heat exchanger and therefore results in the shortest possible length, 5.3 ft. Given the thermal behavior of the botherm interface described in Figs 4 and 6, there is no way a shorter heat exchanger can be made to satisfy the specifications and to give the required performance.

The optimum heat source temperature profile in Fig 13 results in the optimum heat flux profile in Fig 14. Figure 14 also describes how q_{max} from Figs 4 and 6 is related to H_{bo}.

PROBLEM 5

Referring to Problem 1, suppose that the equipment specifications state that the heat source temperature MUST be 1100F and that the design intent is to make the heat exchanger as short as possible. Recalling from Problem 3 that a heat source temperature of 1100F required a heat exchanger length of 20 ft, would it be possible to design the heat exchanger in such a way that a length of 10 ft would give the desired exit vapor quality in spite of the 1100F heat source? How?

PROBLEM 5 SOLUTION

In designing heat exchangers such as the one in Problem 1, the thermal designer often has design influence on:

1. Heat source temperature
2. Thermal behavior of heat source interface
3. Thermal transmittance behavior of heat flow wall material
4. Thickness of heat flow wall

If one is dealing with proportional or linear heat flow, and if the design intent is to maximize the heat flux, this design intent will be satisfied by the wall material which results in the smallest ΔT across the wall. However, if one is dealing with highly nonlinear heat flow, it is not at all certain that minimizing the ΔT across the wall is consistent with maximizing the heat flux!

The present problem is an example of a case where the minimum ΔT across the heat flow wall does NOT result in the maximum heat flux. In the present problem, the wall material and geometry are defined by specification and the difficulty is that the specified wall results in a ΔT_w which is LESS THAN the ΔT_w required by the maximum heat flux! And this means that we must somehow INcrease the temperature drop through the wall in order to INcrease the heat flux!!! In other words, we must add INSULATION to the pipe wall in order to INcrease the heat flux!!!

The reason for this seeming paradox is that, without insulation, the heat source temperature of 1100F results in values of ΔT_{bo} in excess of the values corresponding to the maxima in Figs 4 and 6. By adding insulation to the pipe wall, we could reduce the values of ΔT_{bo} and thereby increase the heat flux profile. Moreover, if we were permitted to vary the thickness of the insulation along the length of the heat exchanger, it would be possible to obtain q_{max} along the entire length of the exchanger--ie it would be possible to obtain the heat flux profile shown in Fig 14. Thus, by everywhere adding the optimum amount of insulation to the pipe wall, it would be possible to design the heat exchanger so that a length of only 5.3 ft (the length corresponding to the heat flux profile in Fig 14 as shown in Problem 4) would give the required performance in spite of the fact that the heat source temperature is 1100F.

For illustration, let us determine the optimum amount of insulation at the location where $H_{bo} = 500$. Because of the temperature drop through the insulation, rel 33 becomes

$$q_{in} \rightarrow 2421(T_c - T_{bo} - \Delta T_{ins} - \Delta T_{bo}) \qquad (39)$$

Since $T_{bo} = 600$ at $H_{bo} = 500$, and since we are given that $T_c = 1100$, rel 39 becomes

$$q_{in} \rightarrow 2421(500 - \Delta T_{ins} - \Delta T_{bo}) \qquad (40)$$

The coordinates of the maximum in the $H_{bo} = 500$ curve of Fig 6 are

$$q_{out} = 800000 \quad \text{at} \quad \Delta T_{bo} = 40$$

Noting that $q_{in} = q_{out}$ and substituting these coordinates into rel 40, we obtain the result

$$\Delta T_{ins} = 460 - \frac{800000}{2421} = 130 \qquad (41)$$

In other words, the optimum amount of insulation at the location where $H_{bo} = 500$ is whatever thickness would

result in a temperature drop of 130F across the insulation. For example, if the thermal behavior of the insulating material were described by

$$q \rightarrow 13 \frac{dT}{dx} \qquad (42)$$

the required insulation thickness would be given by

$$x_{ins} \rightarrow \frac{13(130)}{800000}(.0725/.060) = .0025 \qquad (43)$$

(The factor .0725/.060 is approximate and is required because q is based on the inner pipe diameter (.06) rather than the mean diameter of the insulation.)

Repeating the above analysis at other locations in the exchanger, we obtain the following insulation thickness profile:

T_{bo}	H_{bo}	q_{max}	ΔT_{ins}	x_{ins} (approx)
500	430	1000000	147	.0023
550	465	900000	138	.0024
600	500	800000	130	.0025
"	600	650000	192	.0048
"	700	500000	253	.0086
"	800	350000	315	.0169
"	900	200000	377	.0413

Notice that rel 42 is identical to the relation which describes the thermal behavior of the pipe wall, rel 3. Thus by adding the pipe wall thickness (.005) to the values of x_{ins} in the above table, we obtain the pipe wall thickness profile which would result in a heat exchanger of the shortest possible length.

PROBLEM 5 ANSWER

It would be readily possible to design the Problem 1 heat exchanger in such a way that a length of 10 ft would give the desired exit vapor quality with a heat source temperature of 1100F. This could be accomplished by insulating/contouring/stepping the pipe wall as indicated in the table on page 6-43.

PROBLEM 6

Referring to Problem 1, suppose that the heat source is specified to be a one phase, forced convection fluid and that its inlet and outlet temperatures are to be 950F and 750F. If the design intent is to make the shortest possible heat exchanger without insulating/contouring/stepping the heat flow pipe wall, should the exchanger be designed as a cocurrent or as a countercurrent heat exchanger? Why?

PROBLEM 6 SOLUTION AND ANSWER

The flow direction of the heat source fluid is important because it has a strong influence on the heat source temperature profile which in turn has a strong influence on the heat flux profile which in turn has a strong influence on the required length of the heat exchanger. As we saw in Problem 4, the optimum heat source temperature is described by Fig 13. Notice that the profile in Fig 13 resembles the temperature profile which would result from cocurrent flow and that it is just the opposite of what would result from countercurrent flow. We therefore conclude that a cocurrent heat exchanger would give the required performance with a much shorter heat exchanger length than a countercurrent exchanger. (In fact, the cocurrent heat exchanger would be several times shorter than the countercurrent exchanger.)

CONCLUSIONS

The problems in this chapter are based on Figs 4 and 6 which describe the thermal behavior of a hypothetical, forced convection boiling interface. It should be noted that this new heat flow description of a forced convection boiling interface bears NO RESEMBLANCE to the old heat transfer description. This lack of resemblance between new and old results because the old heat transfer does not and indeed can not come to grips with the real world behavior of highly nonlinear phenomena such as boiling. And the principal reason the old heat transfer can not deal effectively with highly nonlinear phenomena is because the old heat transfer is based on the proportional concept of the heat transfer coefficient. Proportional concepts lead to proportional thinking, proportional correlating, proportional analyzing--and make it impossible to deal effectively with nonlinear phenomena like boiling.

On the other hand, the new heat flow can come to grips with nonlinear phenomena like boiling because the new heat flow is based on the free form, nonlinear concept of thermal behavior. This new concept neither dictates nor suggests a particular mode of thinking, correlating, analyzing. On the contrary, this new concept is like putty in that it adapts to whatever form is suggested by the real world behavior of the phenomenon of interest.

The problem solutions in this chapter bear NO RESEMBLANCE to the old way solution of thermal design/analysis problems involving forced convection boiling equipment. The problems in this chapter are solved using the new heat flow, and this new type of solution is the reason the solution of these problems is so simple. However, it should be noted that these same simple problems are VIRTUALLY IMPOSSIBLE to solve using the methods of the old heat transfer. (This can be readily verified by attempting to describe forced convection boiling behavior using the correlations of the old heat transfer, and then trying to solve problems like those in this chapter using heat transfer coefficients, regimes, etc.)

The problems in this chapter do not cover the complete spectrum of highly nonlinear thermal design/analysis problems. However, it is my feeling that these simple problems describe and illustrate the essential features involved in solving highly nonlinear problems using the new heat flow. I also feel that any reader who has a real understanding of the solution of the problems in this chapter will experience little difficulty in solving other thermal design/analysis problems involving proportional/linear/nonlinear/highly nonlinear phenomena.

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