PRESSURE DROP FOR A TWO-PHASE FLOW OF STEAM ACROSS VERTICAL TUBE BANKS

by

JANICE HERMAN HEARN

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Approved by:

John E. Kipp

Major Professor

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Special thanks are due to my husband, William, who was and is always there when I need him.

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1. INTRODUCTION

A heat exchanger is a device in which energy is transferred from one fluid to another across a solid surface. One type of heat exchanger widely used is that of the baffled shell-and-tube arrangement. One fluid flows through a series of tube banks while the other fluid is forced through the shell and over the outside of the tubes. The baffles are perpendicular to the tube banks, and they insure that the shell-side fluid will flow more or less normal to the tube banks, thus inducing higher heat transfer. The tubes between each pair of baffles approximate an ideal tube bank. Although various leakage and bypass flow paths and turn-around regions cause significant performance differences between a real heat exchanger and the corresponding ideal tube bank, most exchanger design methods are based on the heat transfer and fluid friction parameters of the ideal tube bank, and the effects of these non-ideal components are considered either correction factors to be applied to the ideal tube bank heat transfer coefficient and pressure drop or as a reduction in the effective flow rate across the tube bank. (1)

In using these design methods the friction factor (f) curve for the ideal tube bank must be known as a function of Reynold's number. This curve is available for most fluids, but there is no corresponding data for two-phase flows, such as wet steam.

A two-phase flow is simply the simultaneous flow of two states of matter, such as liquid-solid, gas-solid, or gas-liquid as is the case of wet steam. Other common examples of two-phase flows are smoke, quicksand, mud slides and coal slurries.

Two-phase flows obey all the basic laws of fluid mechanics, but the equations of state are more complicated or more numerous than those for a one-phase flow. There are many ways to analyse two-phase flows, ranging from simple correlations to complex differential analysis. For simplicity in this study, a homogeneous model was used wherein a detailed description of flow patterns was not necessary. The flow components were treated as a one-phase fluid with properties that were weighted averages of the liquid and gas phases and were not necessarily the same as the properties of either phase.

Because there are no data for two-phase flows across ideal tube banks, design methods employ some estimations to adapt one-phase curves to two-phase, or the two-phase fluid is considered one phase. The purpose of this research was to obtain ideal tube bank data for pressure drop of horizontal two-phase steam flow across vertical tube banks. Test runs for two tube patterns were made in an adiabatic test unit and curves of friction factor as a function of Reynold's number were plotted.

2. LITERATURE REVIEW

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One paper has been published on the pressure drop for two-phase flow of fluids across ideal tube banks. Ishihara, Palen and Taborek (21) reviewed correlations for the prediction of the pressure drop for two-phase flow of fluids across ideal tube banks and compared them to experimental data. However, due to proprietary status of their work, many details were not presented, and their work could not be used in this study.

The remainder of this chapter considers published results in two related areas:

1. The pressure drop of single-phase flows across ideal tube banks.
2. The pressure drop for two-phase flows through tube banks.

2.1 One-Phase Fluid Flow Across Ideal Tube Banks

Chilton and Genereaux (2) were the first to attempt to correlate available data on the pressure drop across tube banks and obtain a representative equation. They recommended separate equations for laminar flow and turbulent flow.

The pressure drop for transverse flow over the Reynold's number range of 2000 to 40,000 was later measured by Pierson. He attempted to determine the effect of varying the spacing of tubes of identical size in a tube bank on the pressure drop across it and found that spacing had a great effect. However, no simple equation was formulated to describe this effect. Pierson's experiment was continued by Huge (4) and included a variation in tube size. The range of tested Reynold's numbers was 2000 to 70,000, and Pierson's results as well as the validity of the application of the similarity principle to flows across tube banks were confirmed. The results of Pierson and Huge were then graphically correlated by Grimison (5) as functions of the Reynold's number and the pitch ratio of the tube bank. Gunter and Shaw (6) demonstrated the use of a friction factor correlation that they proposed for bare tubes as a general method for correlating on a single line, pure crossflow friction over both bare and extended surfaces for a wide range of Reynold's numbers. They used an equivalent volumetric hydraulic diameter in both the Reynold's number and the friction factor, and, to obtain a correlation to present data for staggered and unstaggered tube arrangements on the same curve, the longitudinal pitch was defined as the center-to-center distance from a tube in one row to the nearest tube in the next transverse row.

Gunter and Shaw's equations for pressure drop in cross-flow are:

$$\frac{f}{2} = \frac{144\Delta P g_c D_V}{G^2 L}\left(\frac{\mu}{\mu_\omega}\right)^{0.14}\left(\frac{D_V}{S_T}\right)^{-0.4}\left(\frac{S_L}{S_T}\right)^{-0.6} \qquad (2\text{-}1)$$

and,

$$Re = \frac{D_V G}{\mu} \qquad (2\text{-}2)$$

This correlation was checked with experimental data of tube diameters from 0.02 to 2 inches, Reynold's number from 0.01 to 3×10^5, and transverse and longitudinal pitches from 1.25 to 5 diameters, and the correlation agreed very well.

Viscous flow across three patterns of vertical tube banks, staggered-square, in-line square, and equilateral triangular was studied by Bergelin, Davis, and Hull (7). Other than in arrangement, the banks studied were identical, and a correction for the effect of viscosity gradient on friction during heat transfer was presented. The aforementioned study as expanded by Bergelin, Brown, Hull, and Sullivan (8) with four additional tube banks and additional tube sizes and pitch ratios. Tenative correlations for friction were developed.

The preceding experimental results were extended by Bergelin, Brown, and Doberstein (9) from viscous flow through the transition zone into the turbulent zone ($25<Re<10{,}000$).

Heating, cooling, and isothermal tests were taken and the current standard correlation parameters were presented:

$$Re = \frac{d_t G}{\rho} \qquad (2\text{-}3)$$

$$f = \frac{2\Delta P \rho\, g_c}{4G^2 N} \left(\frac{\mu}{\mu_\omega}\right)^{.14} \qquad (2\text{-}4)$$

Friction design data for flow across circular tube banks with a Reynold's number range 500 to 20,000 (based on tube diameter) was presented by Kays, London, and Lo (10). These data complete the available data as a supplement to the high Reynold's number data of Pierson, Huge and Grimison, and the low Reynold's number data of Bergelin, Brown, et al. Kays et al. employed a transient technique, but several steady state tests were conducted to prove the validity of the transient method. The advantages and disadvantages of the method were reported in detail.

The flow of a non-Newtonian fluid, carboxymethyl-cellulose solution, across three ideal tube banks was studied by Adams and Bell (11). They used several concentrations of a sodium carboxymethycellulose solution, and based on upper laminar and lower transitional flow regimes, suggested that a Reynold's number analogous to the Reed-Metzner Reynold's Number, with the addition of parameters characteristic of tube bank flow, correlates Newtonian and non-Newtonian results for the friction factor.

2.2 Two-Phase Pressure Drop Through Tube Banks

Air-water and pentane vapor pentane liquid horizontal crossflow systems through four tube banks with horizontal tubes were studied by Diehl and Unruh (12). A pressure drop correlation was developed from the study, and this correlation can be used to estimate condensation pressure drop.

Previous studies of two-phase pressure drop in rod bundles were reviewed by Castellana and Bonilla (13). The basic governing equations of two-phase pressure drop to rod bundles were then extended with a single equivalent diameter and the subchannel analysis techniques.

Experimental results of friction factor as a function of Reynold's number for various spacings of several hexagonal tube bank sizes were published by Rehme (14). From the investigation, he concluded that the number of tubes has no effect on the friction factor and that the spacing of the tubes has only a small significance.

Two square tube banks of different sizes were used by Marek, Maubach, and Rehme (15) to compare friction factors with isothermal and nonisothermal flows through the tube banks. No appreciable difference in the friction factors for isothermal and nonisothermal flows was found.

3. EXPERIMENTAL STUDIES

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3.1 Flow Loop

In this study the steam was supplied from a tap off of a main steam line in the Heat Transfer Fluids Laboratory of Seaton Hall, Kansas State University. From a 3/4 inch schedule 40 pipe the steam proceeded through 1 1/4 inch schedule 40 pipe to a 1 foot by 1 foot sheet metal angle with baffles. Then the steam passed through two screens in a 1 foot by 1 foot sheet metal duct. The baffled sweep of ductwork and screens were installed to insure that the flow of steam entering the bank was uniform. From there the steam flowed through 1 1/2 inch schedule 40 pipe, an orifice plate, and down into a metal tank where it condensed in a water supply. When the water in the tank reached a certain level, it drained out into a sink, thus avoiding overflow problems. A 1/2 inch outer diameter copper pipe was installed on the side of the baffled duct which was used for drainage of the ductwork and in quality measurements. All the piping and ductwork up to the orifice plate was insulated with one inch thick fiberglass or one inch thick Dow polystyrene. A flow diagram and drawing of the apparatus are presented in Figure 1 and Figure 2 respectively.

FIGURE 1.

LINE DIAGRAM OF FLOW LOOP

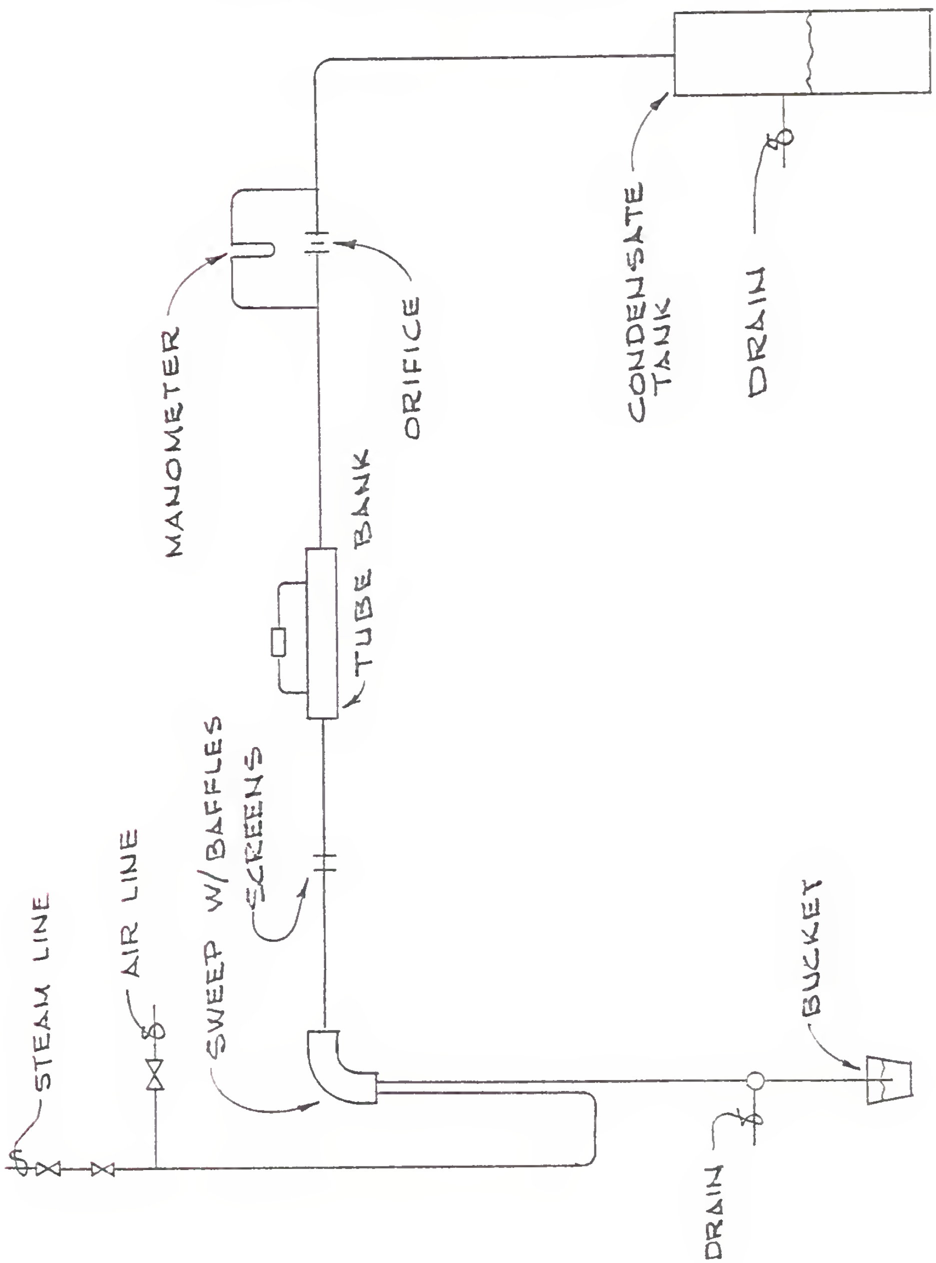
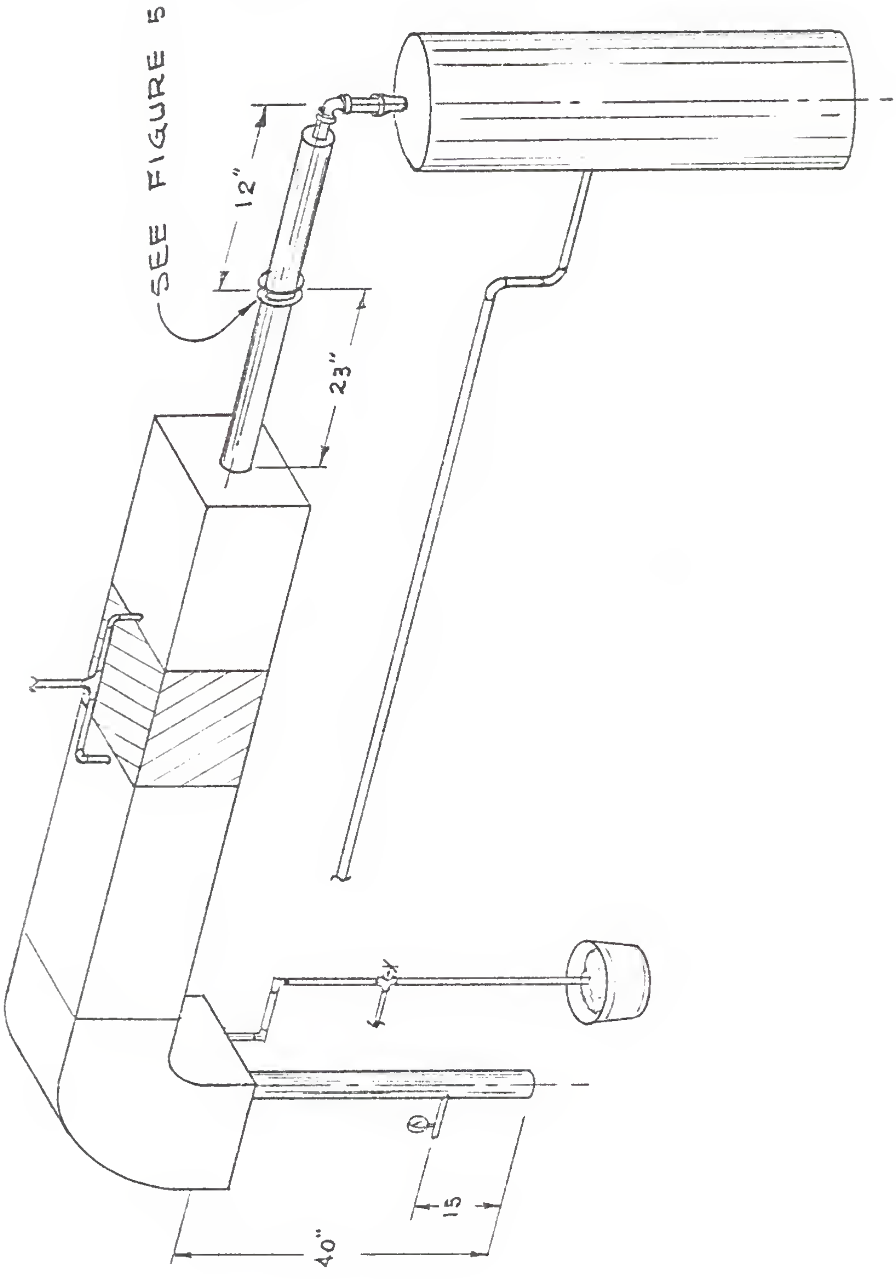


FIGURE 2.

SKETCH OF FLOW LOOP



3.2 Tube Banks

Two tube banks with different layout arrangements were used. The patterns were in-line and staggered square. The dimensions are presented in Table 1, and diagrams of the tube banks are on Figures 3 and 4.

Each tube bank was constructed of two 1/2 inch thick pieces of shellacked particleboard which held 200 3/8 inch outer diameter, 18 BWG copper tubes with six inches of exposed length. The tubes were held in position by two iron rods which passed through a tube at each end of the bank and were clamped tight to the particleboard. Sheet metal flanges were added to the top and bottom particleboard and the sides of the ductwork, insuring that steam would flow only horizontally across the tube bank. Since the pressure differential and not the heat transfer across the tube banks was measured, no tubeside flow was required.

Four pressure taps were located on the top of the ductwork for each tube bank. The first two taps were equidistant from the sides of the duct and 3/4 inch in front of the first tube row, while the second pair were 3/4 inch behind the trailing edge of the last tube row, and also equidistant from the sides of the duct. Copper piping of 5/16 inch outer diameter was screwed to these taps, to which the differential pressure measuring device was attached.

TABLE 1

TUBE BANK DIMENSIONS AND CONSTANTS

	Model Number 1	Model Number 2
Tube Layout	In-Line Square	Staggered Square
Outside Tube Diameter, In.	0.375	0.375
Minimum Tube Clearance, In.	0.375	0.375
Tube Length, In.	6.0	6.0
Exposed Tubes	200	200
Volumetric Hydraulic Diameter, Ft.	0.7674	0.7674
Net Free Volume, $Ft.^3$	0.00157	0.00157
Minimum Cross-sectional Flow Area, $Ft.^2$	0.1406	0.1406
Number of Tube Rows	20	40
Transerve Pitch, Ft.	0.0625	0.0883
Longitudinal Pitch, Ft.	0.0625	0.0442

FIGURE 3.

INLINE TUBE BANK ARRANGEMENT

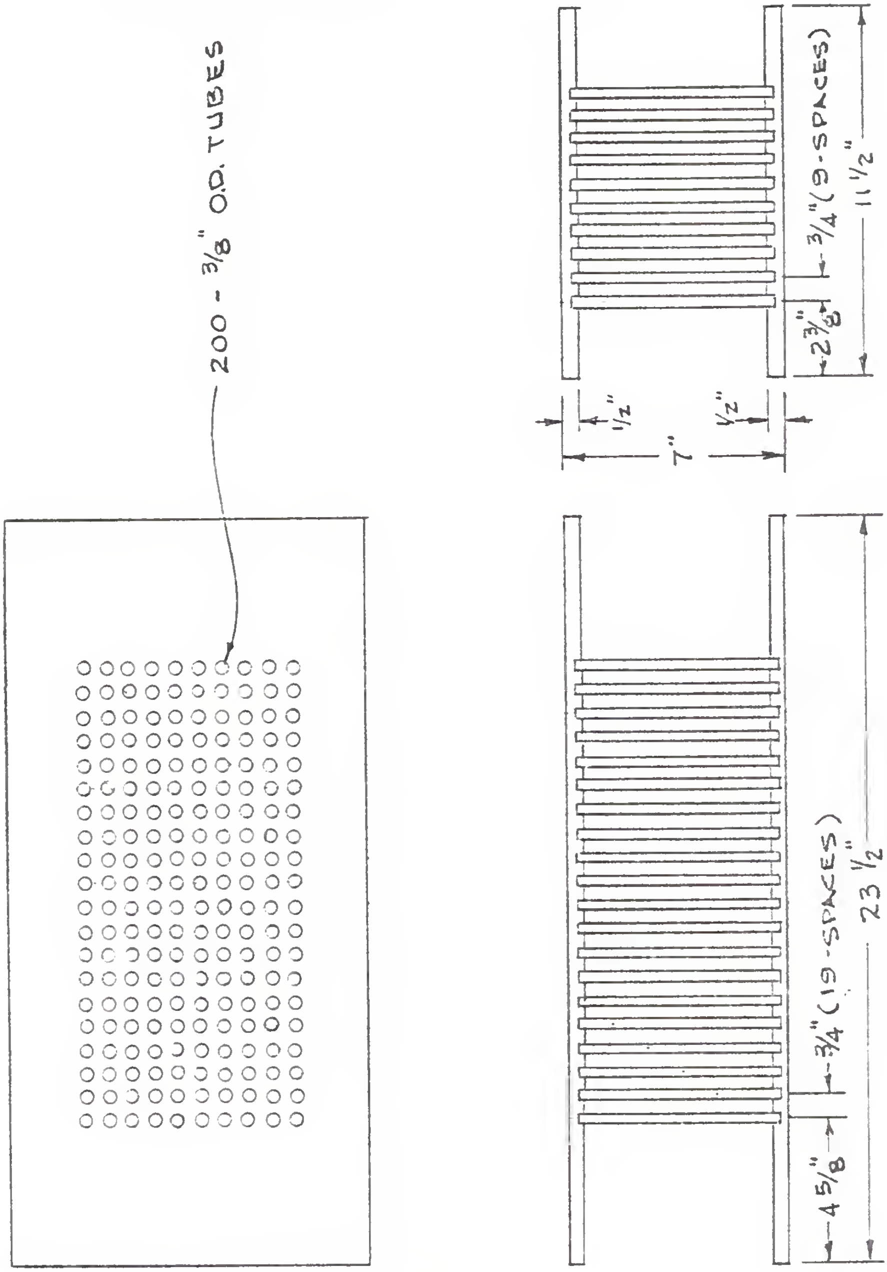
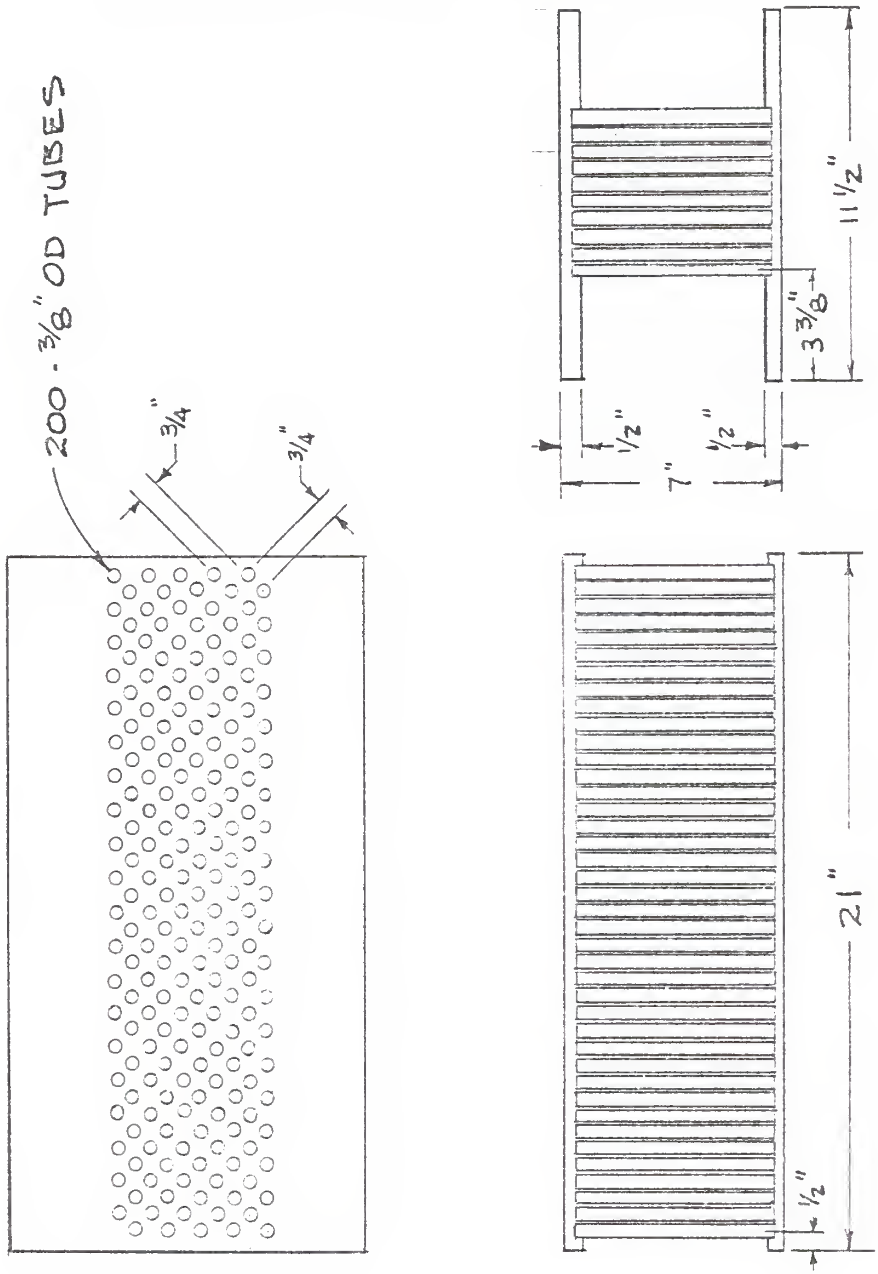


FIGURE 4.

STAGGERED TUBE BANK ARRANGEMENT



3.3 Instrumentation

The steam flow was varied by a throttling valve at the entrance of the piping of the apparatus, while the pressure in the 1 1/4 inch line was measured with a Bourdon pressure gage. The instrumentation involved with the steam quality measurement was a mercury thermometer, Fairbanks Morse & Co. scale with a range of 0 to 120 pounds, and a plastic bucket. The bucket was insulated with 4 inch thick fiberglass building insulation by Johns-Manville Co. and the top covered with Reynolds Co. aluminum foil.

A sharp edge brass orifice with vena contracta pressure tap locations was used to determine the flow rate of the steam. The orifice was designed and installed in the apparatus according to standards specified in References 18 and 19. A diagram and dimensions of the orifice are presented in Figure 5. Depending on the flow, the pressure taps were connected to a mercury manometer, a Meriam Instrument Company water micromanometer or an Ellison Draft Gage Company inclined draft gage filled with petroleum oil of specific gravity 0.834.

The tube bank pressure drop was measured with a Statham 10 volt strain gauge-type pressure transducer. This was connected to a Hewlett Packard Model 8805A Carrier Preamplifier and a Hewlett Packard Model 7702B Recorder. The pressure transducer was calibrated before the test runs using a Meriam Instruments Company water micromanometer.

3.4 Calibration

Before any experimental runs were made, the preamplifier was calibrated with an air supply connected to the high side of the pressure transducer and the low side open to the air. The differential pressure was simultaneously measured on a Meriam Instruments Company water micromanometer for various air flows and a plot of transducer output as a function of measured pressure was drawn. With this plot, the differential pressure recorded on the strip chart was readily converted to a reading in inches of water. The relationship of transducer output and measured pressure is shown in Figure 6.

3.5 Test Procedure

The same test procedure was used for each of the two tube bank patterns. After the tube bank pattern was inserted, the ductwork was soldered and caulked to provide an airtight and watertight seal. Air was supplied to the apparatus with a rubber hose of 1/2 inch inner diameter connected to an air line. The air flow was controlled with a valve in the air line. The pressure differentials across the tube bank and across the orifice were measured and recorded for each flow rate. When the flow rate was changed, a three-minute delay was allowed to insure steady state conditions before measurements were taken. With this data, the flow speed was calculated and used to determine the Reynold's number and friction factor for each flow. Data on the friction factor as a function of the Reynold's number is available in the literature (6). Since the experimental data compared favorably with the published data, tests with steam as the working fluid were undertaken without major alteration to the apparatus.

Before tests were run with steam as the working fluid, the air hose was removed, that connection sealed, and any remaining holes in the ductwork were caulked. Steam flow through the apparatus was regulated using a throttling valve at the inlet pipe. A ten-minute interval was allowed to insure that any condensed steam from the pipes was removed through the drains in the ductwork. These drains were then closed and pressure differentials across the tube bank and orifice were measured as in the air runs.

Steam quality was found by condensing steam into standing water in an insulated bucket. Steam was allowed to flow from the drain in the baffled section of the ductwork into the bucket for fifteen minutes. The temperature and weight of the water in the bucket were recorded before and after the addition of the condensed steam. The quality of the steam which flowed across the tube bank was calculated from the data using the equation derived in Chapter 4.

FIGURE 5.

FLOW ORIFICE

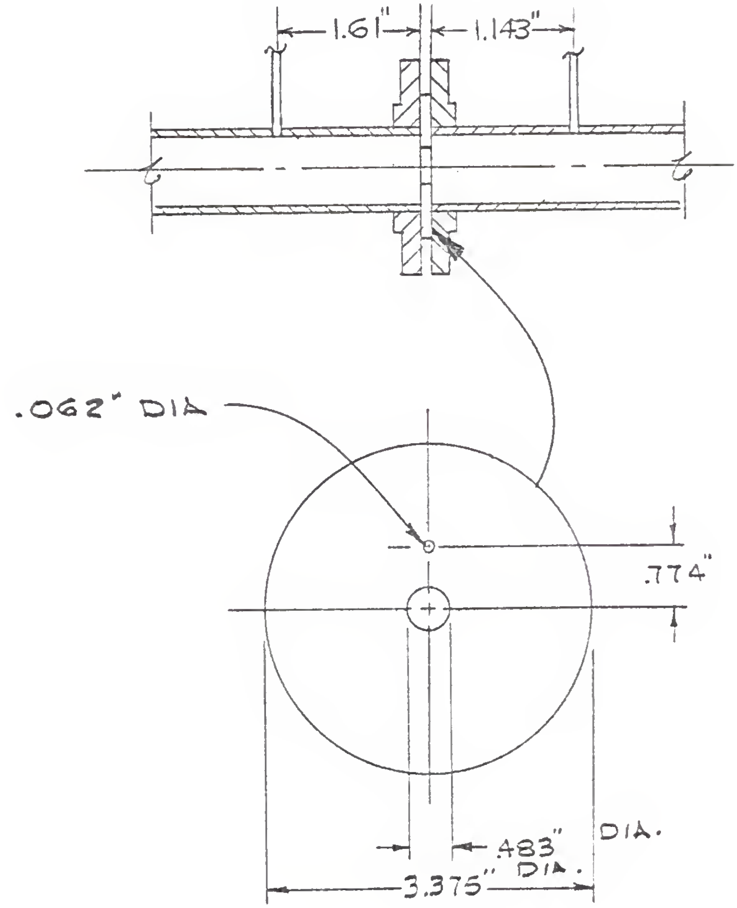
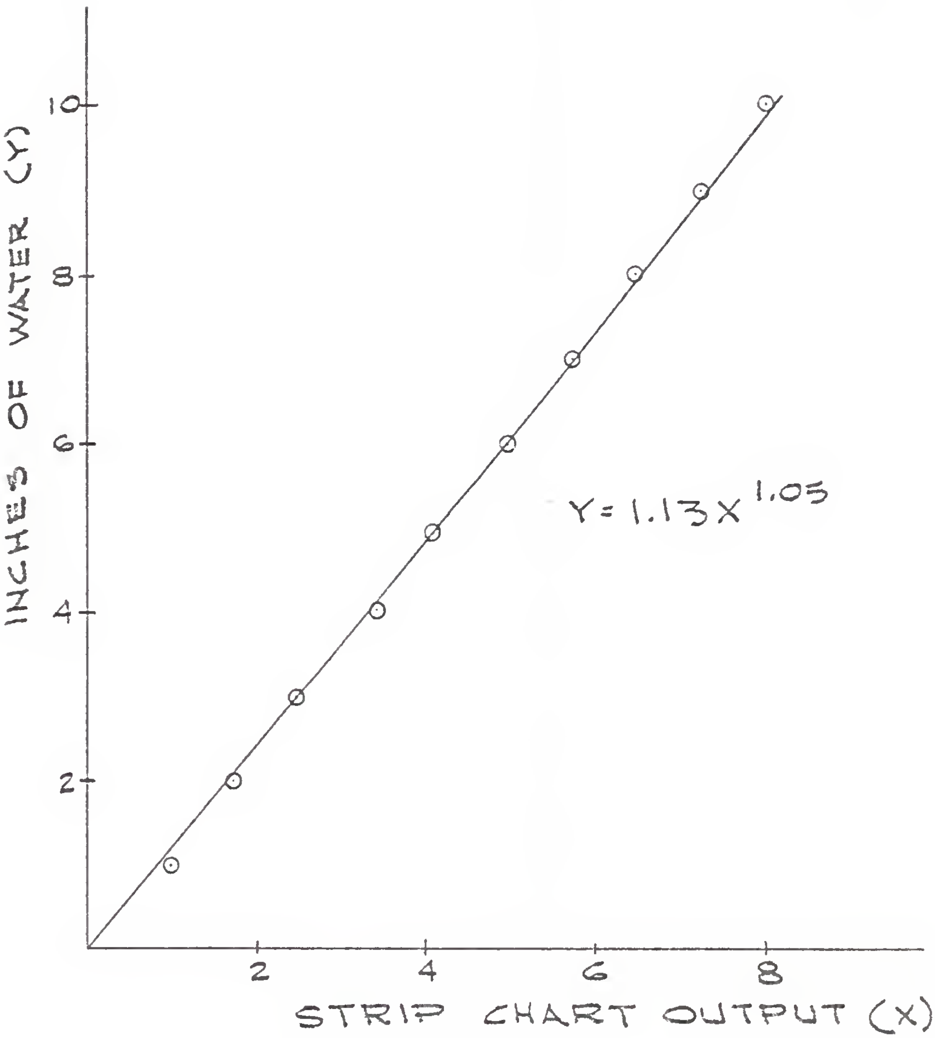


FIGURE 6.

PLOT OF TRANSDUCER OUTPUT VERSUS

MEASURED DIFFERENTIAL PRESSURE



4. DATA REDUCTION

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The pressure drop of a fluid flow in general may be due to four factors: (1) changes in elevation, (2) acceleration of the fluid, (3) friction due to fluid viscosity, and (4) sudden expansions and contractions of the pipe as in inlets and exits. In this study the flow through the tube bank was horizontal so there was no pressure drop due to change in elevation. The differential pressure measured included the drop due to the inlet and exit of the tube bank, friction, and acceleration.

In this study, the steam flow was considered homogeneous with no slip, and the Reynold's number and friction factor were thus weighted functions dependent on the steam quality. Because of this assumption, Gunter and Shaw's equations for the Reynold's number and friction factor for pressure drop in crossflow (equations (2-1) and (2-2)) could be used in analyzing both air and steam flow data.

This study was conducted with no flow through the tube bank tubes. Since the tube temperature was then equal to the flowing fluid temperature, the absolute viscosity at the average main stream temperature (μ) was equal to the absolute viscosity at the tube wall temperature (μ_w); thus $\left(\frac{\mu}{\mu_w}\right)^{.14} = 1$ in equation (2-1).

The maximum fluid mass velocity, G, was based on minimum net free area through the tube bank and the rate of flow through the orifice. The rate of flow of the air or steam through the orifice was calculated from the pressure drop across the orifice with the following equation (21):

$$q_o = YCA\sqrt{2g_c h_L} = YCA\sqrt{\frac{2g_c(144)\,\Delta P}{\rho}} \qquad (4\text{-}1)$$

By use of Bernoulli's Equation, the relationship between the flow through the orifice and flow through the tube bank was found.

$$V_{tb}^2 = V_o^2 + \frac{2g_c(144)}{\rho}(P_o - P_{tb}) + 2g_c(z_o - z_{tb}) + K2g_c\left(\frac{V_o^2}{2g_c}\right) \qquad (4\text{-}2)$$

Then the fluid mass velocity through the tube bank was calculated from

$$G = V_{tb}\,\rho \qquad (4\text{-}3)$$

The air and steam flow data were calculated at atmospheric conditions, but the two-phase aspect of the steam had to be considered in calculating its density and viscosity. The average of the recorded steam qualities was used in determining the steam flow density and viscosity with the following properties of a liquid-vapor mixture:

$$v = v_f + x*(v_g - v_f) = \frac{1}{\rho} \qquad (4\text{-}4)$$

$$\rho = (v_f + x*(v_g - v_f))^{-1} \qquad (4\text{-}5)$$

$$\mu = (1-x)\ \mu_f + x\,\mu_g \qquad (4\text{-}6)$$

Because the bucket used in the steam quality recordings was insulated, heat transfer to the bucket from the steam was negligible. Therefore all the energy of the steam was assumed to be transferred to the standing water until the steam condensed and reached thermal equilibrium with the standing water. The largest change in the temperature of the standing water was from 66°F to 136°F. Since there is only a 0.001 change in the specific heat of water between these temperatures, the specific heat of water was assumed constant at 0.998 BTU $°F^{-1} lbm^{-1}$. Equations to describe the procedure used to calculate steam quality were formulated from the law of conservation of energy.

$$E_{initial}=E_{final}$$

$$m_s*h+m_b*h_1=(m_b+m_s)*h_2$$

$$m_s*(h-h_2)=m_b*(h_2-h_1)$$

$$m_s*(h-h_2)=m_b*c_p*(t_2-t_1)$$

$$h=m_b*c_p*(t_2-t_1)/m_s+h_2 \qquad (4\text{-}7)$$

also, $$h=h_f+x*(h_g-h_f)$$

$$(\text{at } p=14.696 \text{ psia, } t=212°F) \qquad (4\text{-}8)$$

Therefore, from the solution of equations (4-7) and (4-8),

$$x = (m_b*c_p*(t_2-t_1)/m_s+h_2-h_f)/(h_g-h_f) \qquad (4\text{-}9)$$

5. RESULTS

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The experimental data and results of the steam quality tests are shown in Table 2. The average quality of the steam was 51%, and this was used in all calculations. The friction factor versus Reynold's number for air and steam flow for the inline and staggered tube bank arrangements are shown in Figures 7-10.

In the air flow figures, a plot of $\frac{f}{2}$ as a function of Re based on this study's data, and an extrapolation of curves from Gunter and Shaw's study are compared (6). For the same mass flows, the friction factors which were calculated in this study were 25% higher than those extrapolated from Gunter and Shaw's study. This discrepancy was due to leakage in the test apparatus which in turn caused a lower flow through the orifice.

In the steam flow figures, curves A and A^1 are plots of equations which were obtained by regression analysis of the experimental data. Linear, exponential, logarithmic, and power function curve fits were performed on the data, and the best fit for the data of the staggered tube bank arrangement was a power function, with a correlation coefficient of 78%. The inline arrangement data did not correlate well with any simple function. The best fit was an exponential function with a correlation coefficient of 8%.

Curves B and B^1 were generated from Gunter and Shaw's studies. The experimentally determined Reynold's numbers were used with Gunter and Shaw's curves to find their corresponding friction factors. As shown in the figures, the two curves were not similar and hence the homogeneous model does not work in the conditions of this study.

TABLE 2

EXPERIMENTAL DATA AND RESULTS OF STEAM QUALITY TESTS

Test Number	Standing Water Initial (°F)	Temperature Final (°F)	Standing Water Weight (lbf)	Condensed Steam Weight (lbf)	Steam Quality
1.	66	134	15.563	1.875	0.64
2.	68	113	13.938	1.25	0.53
3.	106	172	14.25	2.375	0.54
4.	68	106	17.063	1.375	0.49
5.	98	136	15.875	1.438	0.37

Average Steam Quality = 0.51

FIGURE 7.

FRICTION FACTOR VERSUS REYNOLD'S NUMBER

AIR FLOW AND INLINE ARRANGEMENT

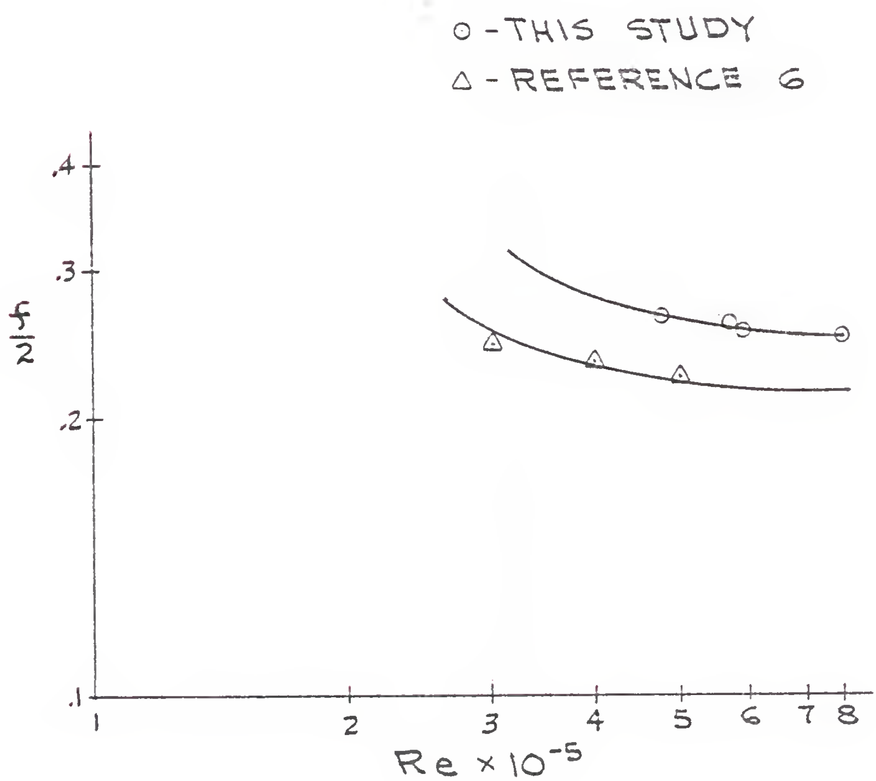


FIGURE 8.

FRICTION FACTOR VERSUS REYNOLD'S NUMBER

AIR FLOW AND STAGGERED ARRANGEMENT

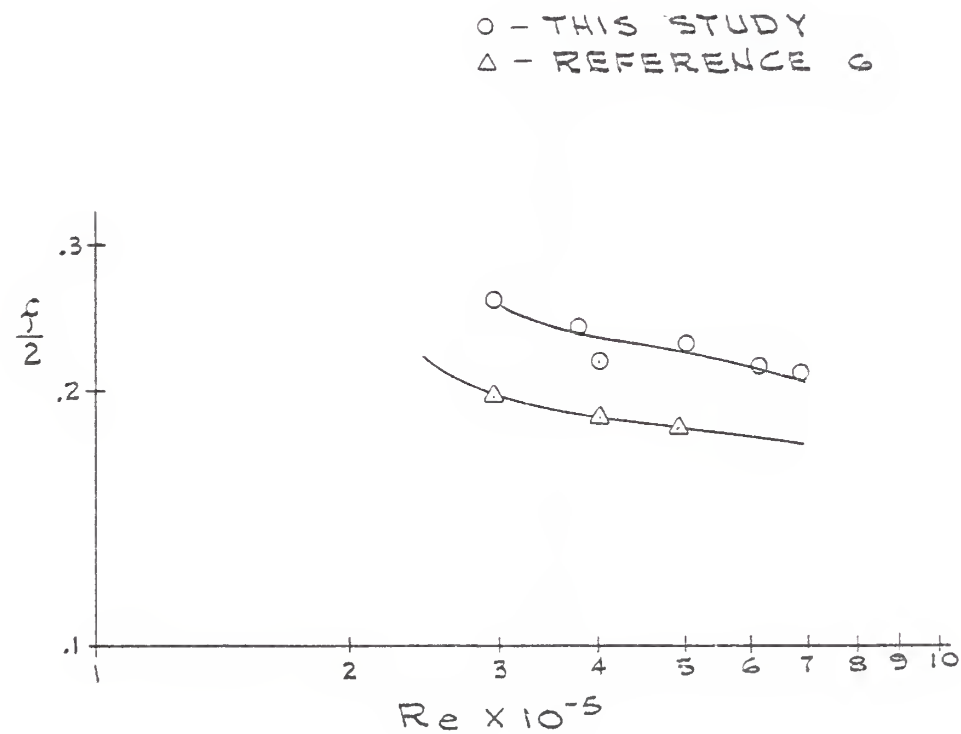


FIGURE 9.

FRICTION FACTOR VERSUS REYNOLD'S NUMBER

STEAM FLOW AND INLINE ARRANGEMENT

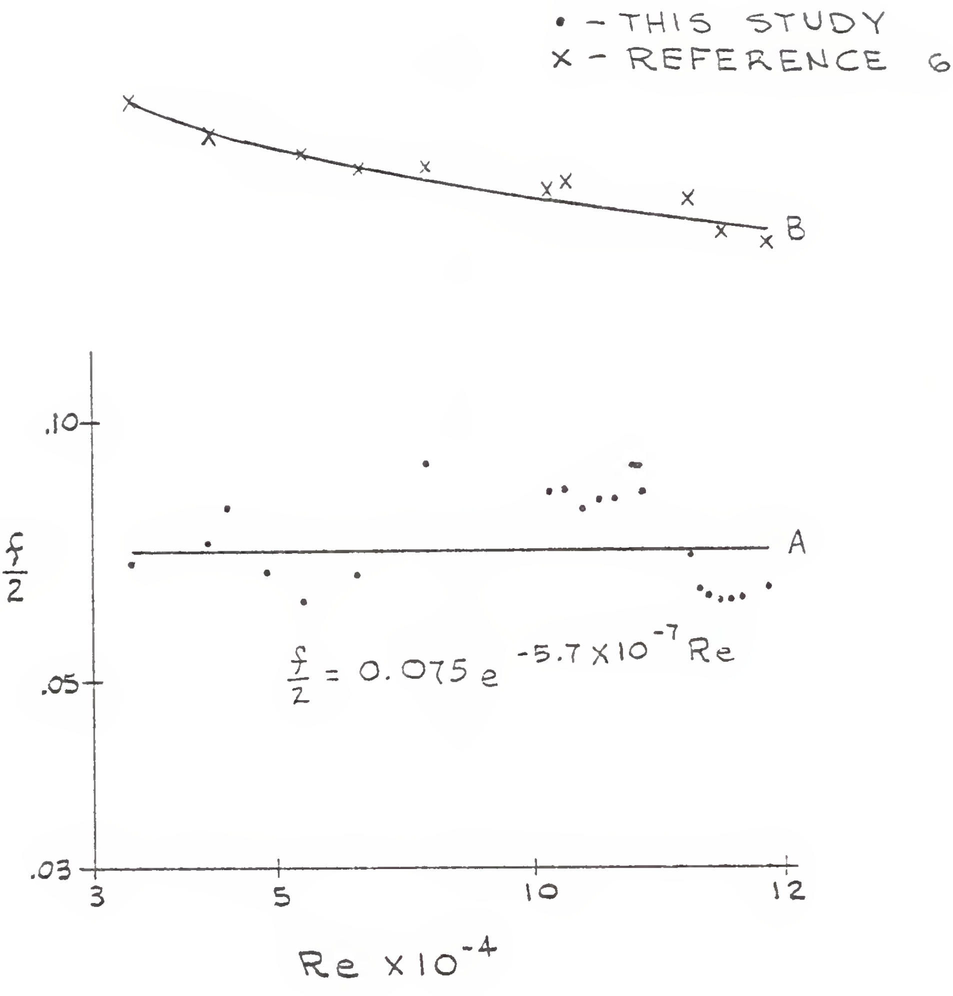
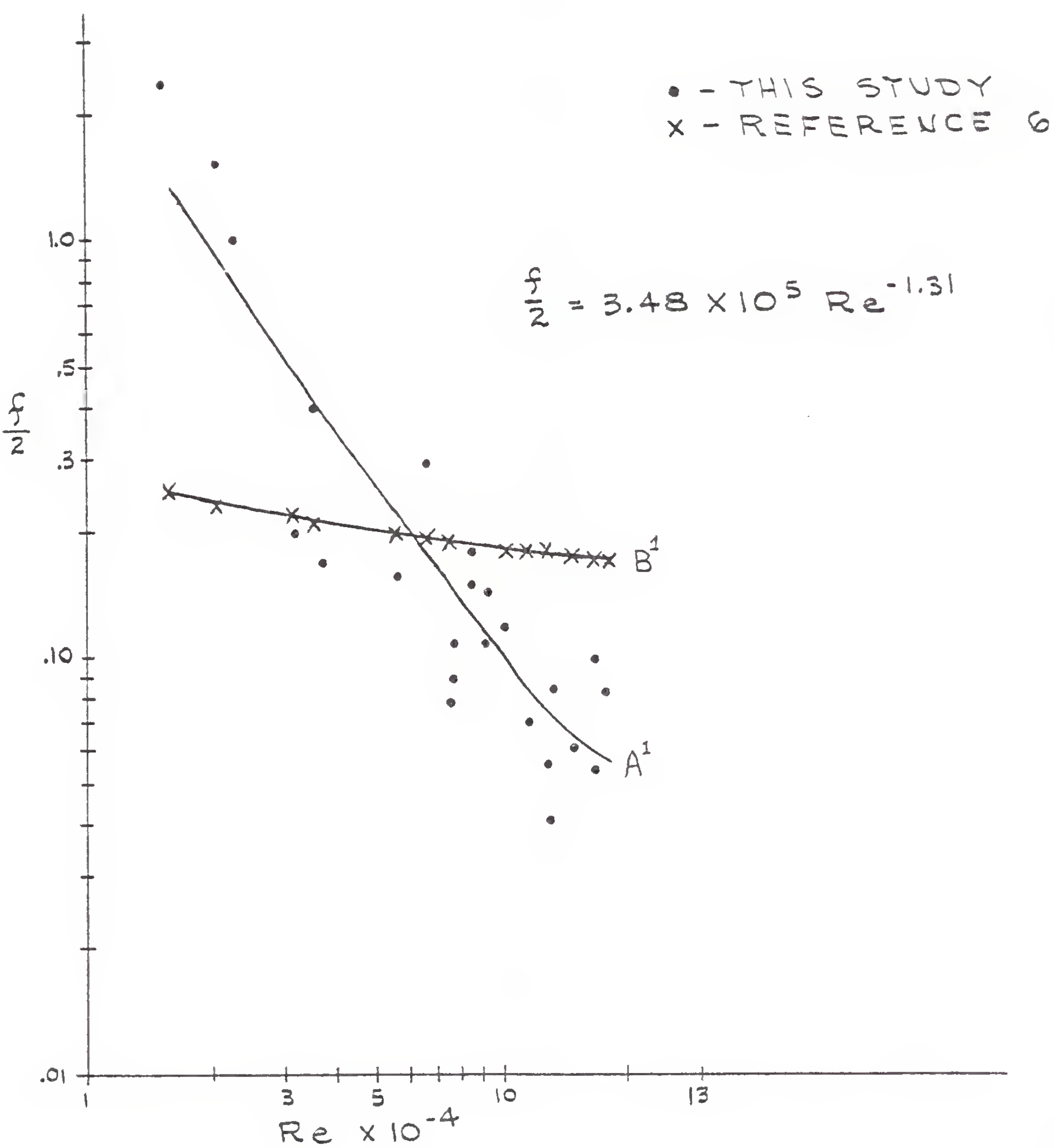


FIGURE 10.

FRICTION FACTOR VERSUS REYNOLD'S NUMBER

STEAM FLOW AND STAGGERED ARRANGEMENT



6.CONCLUSIONS

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The following was shown by examination of the results:

1. The friction factor as a function of Reynold's number for air flow with both inline and staggered patterns of tube banks was 25% higher than extrapolation of the curves in Gunter and Shaw's work (6).
2. In the two-phase flow of steam runs, the relationship between friction factor and Reynold's number was similar to that for air flow. For the same Reynold's numbers, the curves had the same characteristic shapes.
3. The homogeneous model of two phase steam flow presented does not work.

The above ideas are based on a limited amount of data taken exclusively with highly turbulent flow. To better appreciate the results of this study, more data should be taken at lower Reynold's numbers, and with a variety of water-vapor combinations.

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APPENDIX 1

Nomenclature

1. NOMENCLATURE

Symbol	Definition	Units
A	area	ft^2
C	orifice friction factor	dimensionless
C_p	specific heat	$BTU\,°F^{-1} lbm^{-1}$
d_t	outside diameter of tubes	ft
D_v	volumetric hydraulic diameter	ft
E	energy	BTU
$f/2$	half friction factor	dimensionless
g_c	acceleration of gravity	$ft\ sec^{-1}$
G	fluid mass velocity	$lbm\ sec^{-1} ft^{-2}$
h_L	head loss due to friction and contraction in piping	ft
h	enthalpy	$BTU\ lbm^{-1}$
K	resistance of piping	dimensionless
L	fluid flow length	ft
m	mass	lbm
N	number of rows of tubes	dimensionless
ΔP	pressure drop	psi
P	pressure	psi
q	volumetric flow of fluid	$ft^3 sec^{-1}$
Re	Reynold's number	dimensionless
S_T	transverse pitch	ft
S_L	longitudinal pitch	ft
t	temperature	°F
V	velocity of fluid	$ft\ sec^{-1}$
v	specific volume	$ft^3 lbm^{-1}$

1. NOMENCLATURE cont.

Symbol	Definition	Units
x	steam quality	dimensionless
Y	expansion factor	dimensionless
z	elevation	ft
ρ	fluid density	lbm $ft^{-1}sec^{-1}$
μ	absolute viscosity at average main stream temperature	lbm $ft^{-1}sec^{-1}$

Subscripts

o	orifice
tb	tube bank
1	initial point
2	final point
f	liquid
g	gas
w	at surface wall temperature
s	condensed steam
b	standing water in bucket before steam addition

APPENDIX 2

Sample Calculations

2. SAMPLE CALCULATIONS

2.1 Steam Quality

$$x = (m_b*C_p*(t_2-t_1)/m_s+h_2-h_f)/(h_g-h_f)$$

where

m_b = 15.56 lbm

c_p = 0.998 BTU°F^{-1}lbm^{-1}

m_s = 1.88 lbm

t_2 = 134°F

t_1 = 65°F

h_2 = 101.90 BTU lbm^{-1}

h_f = 180.07 BTU lbm^{-1}

h_g = 1150.37 BTU lbm^{-1}

$$x = (15.56*.998*(134-65)/1.88+101.90-180.07)/(1150.37-180.07)$$

$$x = 0.51$$

2.2 Steam Viscosity

$$\mu = (1-x)*\mu_f+x*{}_g$$

where

$x = 0.51$

$\mu_f = 1.931*10^{-4}$lbm ft^{-1}sec^{-1}

$\mu_g = 8.028*10^{-6}$lbm ft^{-1}sec^{-1}

$$\mu = (1.51)*1.931*10^{-4}+.51*8.028*10^{-6}$$

$$\mu = 9.861*10^{-5}\text{lbm ft}^{-1}\text{sec}^{-1}$$

2.3 Steam Density

$\rho = ((1-x)*V_f+xV_g)^{-1}$

where

$x = 0.51$

$v_f = 0.01672\ ft^3 lbm^{-1}$

$v_g = 26.799\ ft^3 lbm^{-1}$

$\rho = ((1-.51)*.01672+.51*26.799)^{-1}$

$\rho = 0.073\ lbm\ ft^{-3}$

2.4 Volumetric Hydraulic Diameter

D_v = 4*net free volume/friction surface

$D_v = 4*(S_T*S_L-(\pi*d_t^2/4))*H/\pi *d_t$

where

$S_T = 0.0883\ ft$

$S_L = 0.0442\ ft$

$d_t = 0.0313\ ft$

$H = 0.50\ ft$

$D_v = 4*(.0883*.0442-(\pi*.0313^2/4))*6/\pi*.0313$

$D_v = 0.7674\ ft$

2.5 Fluid Flow Through the Orifice

$$q_o = Y*C*A_o \sqrt{2*g_c*144*\Delta P_o/\rho}$$

where

$Y = 0.99$ (page A-21, Ref. 20)

$C = 0.6$ (page A-20, Ref. 20)

$A = 0.00127 \text{ ft}^2$

$\rho = 0.073 \text{ lbm ft}^{-3}$

$\Delta P = 0.0372 \text{ psi}$

$g_c = 32.2 \text{ ft sec}^{-2}$

$$q_o = .99*.6*.00127 \sqrt{2*32.2*144*.0372/.073}$$

$$q_o = 0.052 \text{ ft}^3\text{sec}^{-1}$$

2.6 Fluid Flow Through the Tube Bank

$$V_{tb} = \sqrt{V_o^2+2*g_c*144*(P_o-P_{tb})/e+2*g_c*(z_o-z_{tb})+K*V_o^2}$$

where

$V_o = q_o/A_o \quad 40.94 \text{ ft sec}^{-1}$

$g_c = 32.2 \text{ ft sec}^{-2}$

$P_o = P_{tb}$

$\rho = 0.073 \text{ lbm ft}^{-3}$

$z_o = z_{tb}$

$K = 0.788$

$$V_{tb} = \sqrt{40.94^2+.788*40.94^2}$$

$$V_{tb} = 54.74 \text{ ft sec}^{-1}$$

2.7 Fluid Mass Velocity

$G = \rho * V_{tb}$

where

$\rho = 0.073 \text{ lbm ft}^{-3}$

$V_{tb} = 54.74 \text{ ft sec}^{-1}$

$G = 0.073*54.74$

$G = 3.996 \text{ lbm sec}^{-1}\text{ft}^{-2}$

2.8 Reynold's Number

$Re = D_v*G^2/\mu$

where

$D_v = 0.7674 \text{ ft}$

$G = 3.996 \text{ lbm sec}^{-1}\text{ft}^{-2}$

$\mu = 9.861*10^{-5}\text{lbm ft}^{-1}\text{sec}^{-1}$

$Re = 0.7674*3.996/(9.861*10^{-5})$

$Re = 3.110*10^{4}$

2.9 Friction Factor

$$\frac{f}{2} = 144*\Delta P_{tb}*g_c*D_v*\rho*(D_v/S_t)^{-0.4}*(S_L/S_T)^{-0.6}/G^2*L$$

where

ΔP_{tb} =0.0325 psi

g_c = 32.2 ft sec^{-2}

D_v = 0.7674 ft

ρ = 0.073 lbm ft^{-3}

S_t = 0.0883 ft

S_L = 0.0442 ft

G = 3.996 lbm $sec^{-1} ft^{-2}$

L = 1.723 ft

$$\frac{f}{2} = 144*.0325*32.2*.7674*.073*(.7674/.0883)^{-.4}*(.0442/.0883)^{-.6}/3.996^2*1.723$$

$$\frac{f}{2} = 0.1957$$

PRESSURE DROP FOR A TWO-PHASE
FLOW OF STEAM ACROSS VERTICAL
TUBE BANKS

by

JANICE HERMAN HEARN

B.S., Carnegie-Mellon University, 1973

AN ABSTRACT OF A MASTER'S THESIS

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requirements for the degree

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Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

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ABSTRACT

Ideal tube bank data for pressure drop of horizontal two-phase steam flow across vertical tube banks were obtained experimentally for inline and staggered arrangements of tube banks. The apparatus and procedure of the study are discussed, and a complete explanation of the equations and their derivation is provided. Appendices include sample calculations and a list of nomenclature which is used throughout the text.

Results indicate that the two-phase steam flow does not follow a homogeneous model in highly turbulent regions. Areas for further study are discussed.