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HEAT TRANSFER AND PRESSURE DROP IN AMMONIA EVAPORATORS

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INTRODUCTION

Considering the wide use of ammonia in the past and at present, it is remarkable how little information pertaining to the design of ammonia evaporators is available. For heat transfer and pressure drop calculations, experimental data or correlations with at least mass flow rate, vapor quality, and temperature as variables are needed, even if possible parameters like flow patterns are ignored.

In this article, data from author's earlier experimental work (1) on an industrial sized ammonia evaporator are analyzed. Analysis of single phase data provided some surprising results. A new correlation, which is likely to have general applicability, is proposed for two-phase heat transfer. Methods for calculating two-phase pressure drop are also discussed. A theory is put forward suggesting that the phenomena due to interaction of phases are independent of single phase effects.

EXPERIMENTAL SETUP

The setup is shown schematically in Fig. 1. The evaporator was made from commercial grade steel pipe 26.2mm ID and 33.0mm OD. The evaporator was hung horizontally and was divided into 12 sections, the first being 10.6m long while the rest were each 11.7m long. Short sight glasses were inserted between sections for visual observations. Pressures and temperatures were measured at beginning and end of each section, thus giving 13 measuring stations. At each station, a pressure tap, a thermocouple inserted into the pipe, and 7 thermocouples soldered to half the circumference of the pipe were provided.

The pressure taps were connected to a set of U-tube manometers filled with mercury and paraffin oil. Pressure drop in each section was read from the appropriate manometer depending on range of pressure drop. The pressures at beginning, middle and end of coil were also measured directly with a manometer open to atmosphere. Thus, the absolute pressure, temperature of ammonia and wall temperature at each station were known and local heat transfer and pressure drop rates could be estimated.

Heating was done by 3 resistance cables at 120 deg to one another, running along the length of the evaporator. Direct current was supplied to these cables with a transformer-rectifier set. 70mm-thick insulation was given on the evaporator and it was covered with an impermeable plastic sheet to prevent moisture from air being trapped into the insulation.

The refrigeration system consisted of 3 compressors, one of them with a variable speed DC motor. The compressors could be run separately or in parallel. Standard oil separators were provided in discharge lines. Mobil Oil corporation's Arctic 300 oil was used in the compressors. The condenser was a water-cooled shell and tube type with sufficient instrumentation for taking heat balance. Ammonia was fed to an accumulator through a level controlling valve. Two pumps, one small and one large, were used to pump liquid ammonia into the evaporator. The pump discharge rate was measured with standard orifice plates connected to mercury and glycerin manometers.

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Experimental Range

Mass flow rate	60 to 3000 kg/hr
Heat flux	500 to 2000 kcal/m ² -hr
Ammonia temperature	0 to -40 C
Vapor quality	up to 100%

Preliminary Data Reduction

Absolute pressures at each station were calculated by successively adding the pressure drop in each length to the pressure measured at entrance to coil. Local pressure drop rate at each station was estimated using a second order correct finite difference analog and the results matched closely with graphical estimations. No curve smoothing was done and no data were eliminated. It is to be noted that pressure often fluctuated rapidly and in such cases, a true pressure or pressure drop does not exist. Mean values were taken according to observer's judgment alone.

The wall temperature was assumed to be the arithmetic mean of the 7 thermocouple readings. In the non-boiling region, these temperatures generally plotted as a smooth line; but in the boiling region, it was rarely the case. Fig. 2 shows the results of a typical run. It is seen that in the boiling region, no rational basis for drawing the mean wall temperature curve is available. The curve was drawn by personal judgment and the calculated heat transfer coefficients in the boiling region have to be regarded as merely indicative of the range.

For calculating two-phase heat transfer coefficients, saturation temperatures were used. Maximum difference between temperatures measured by thermocouples and saturation temperature was 0.5 deg C. The inside pipe area was used for calculating heat transfer coefficients.

Property data were taken from VDI Kaltmaschinen Regeln. These agree closely with other sources with one notable exception. The values for liquid viscosity given in 1972 ASHRAE Handbook of Fundamentals are about 20% lower than values used in these calculations.

SINGLE PHASE PRESSURE DROP

The well known Darcy equation

$$\frac{\Delta P}{\Delta L} = f \cdot \frac{\rho V^2}{2gD} \quad (1)$$

is widely used for calculating single phase pressure drop. The friction factor 'f' is obtained from the equally well known Moody Chart.

Calculation of pressure drop in this manner gave values differing widely from measurements, generally much higher.

Friction factors were calculated with Eq 1 using measured pressure drop rates. In Fig. 3, data at ammonia temperatures between 0 and -15 deg C are shown. For comparison, the values of 'f' from Moody Chart for a commercial grade steel pipe of this size are also shown. It is seen that the calculated friction factors are invariably lower than standard values and a mean line can be drawn at $f = .018$

This drag reduction phenomena was totally unexpected and no satisfactory explanation could be found. The effect seems similar to drag reduction caused by addition of polymers. The explanation generally given there is that the polymers cause the liquid to slip past the laminar sublayer. For want of any other theory, we may tentatively assume that oil causes the same effect in ammonia evaporators.

In Fig. 4, friction factors for ammonia temperatures between -22 and -37 deg C are shown. Here the data are widely scattered and friction factors as high as twice Moody values occur. This is believed to be due to formation of highly viscous oil films whenever the temperatures are sufficiently low and sufficient amount of oil is present. (The viscosity of oil varies from 1200 centistokes at 0 deg to 100,000 at -30 deg C.) This belief was confirmed by visual observations. Thick, apparently solid films were seen in sight glasses

and simultaneous measurements showed excessive pressure drops and exceptionally low heat transfer coefficients.

It will be noticed that in the lower limit, friction factors in Fig. 4 approach those in Fig. 3. It is concluded that though the friction factors at these lower temperatures cannot be predicted with certainty, under normal conditions of oil content and flow, they will also approximate to about .018.

Single Phase Heat Transfer

It is generally accepted that for small wall superheats, single phase heat transfer coefficients during turbulent flow through heated pipes are well predicted by the Dittus-Boelter equation

$$Nu = .023 Re^{0.8} Pr^{0.4} \quad (2)$$

The measured heat transfer coefficients as well as those calculated by this equation are shown in Fig. 5. It is noticed that measured values are considerably lower than calculated values. The mean line through the data can be represented by the equation

$$Nu = .1825 Re^{.509} Pr^{0.4} \quad (3)$$

A great majority of data are within 30% of this line. All the data points below the -30% line are at -30 deg C or lower, and in each case insulating oil films were visually observed. However, the reason for the low heat transfer coefficients at other points cannot be given definitely as, in many cases, oil films were not observed. Fig. 6 shows the thickness of oil film required to account for the discrepancy between Eqs 2 and 3. (Thermal conductivity of oil used in these calculations was .113 kcal/m hr deg C). Indeed, the thicknesses are so small that the almost colorless oil may well have escaped observation.

TWO-PHASE PRESSURE DROP

(a) Theoretical Development

In the general case, total two-phase pressure drop during flow through pipes may be written as

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_{TPF} + \left(\frac{\Delta P}{\Delta L}\right)_A + \left(\frac{\Delta P}{\Delta L}\right)_S \quad (4)$$

where the 3 terms on the right hand side are the frictional, acceleration and static pressure drop respectively.

Confining attention to frictional pressure drop, we define a variable ϕ_1 such that

$$\left(\frac{\Delta P}{\Delta L}\right)_{TPF} = \phi_1^2 \left(\frac{\Delta P}{\Delta L}\right)_1 \quad (5)$$

where $\left(\frac{\Delta P}{\Delta L}\right)_1$ is the frictional pressure drop for the liquid phase flowing alone.

We further define a parameter X such that

$$X^2 = \left(\frac{\Delta P}{\Delta L}\right)_1 / \left(\frac{\Delta P}{\Delta L}\right)_g \quad (6)$$

where the denominator on the right side is the pressure drop for gas phase flowing alone.

It is intuitively evident that X is in some way connected with the two-phase pressure drop. It is now asserted that ϕ_1 is a function of X. It implies that if the relation between ϕ_1 and X is experimentally determined for any particular system, it will also apply to other similar systems.

It will be noticed that the relation derived here is the well known correlation of Martinelli and coworkers (2, 3). However, the interpretation given here has important consequences and its utility will soon be evident.

Lockhart and Martinelli (2) established the relation between X and ϕ_1 mostly on the basis of experiments on smooth pipes. Hence, if their data curve is to be used, X as given by Eq 6 should be evaluated with smooth pipe friction factors. Following their definitions of various dissipative flow patterns, Eq 6 may be rewritten for the turbulent-turbulent regime as

$$X_{tt}^2 = \frac{\rho_g}{\rho_l} \left(\frac{\mu_l}{\mu_g} \right)^{0.2} \left(\frac{1}{x} - 1 \right)^{1.8} \quad (7)$$

For the viscous-turbulent regime, we may write

$$X_{vt}^2 = \frac{\rho_g}{\rho_l} \left(\frac{\mu_l}{\mu_g} \right) \left(\frac{1}{x} - 1 \right) \left(\frac{348}{Re_g^{0.8}} \right) \quad (8)$$

Hence, knowing the vapor quality and fluid properties, X can be calculated and the multiplier ϕ_1 can be estimated from Table 4 of Ref 2. It remains to evaluate $(\Delta P / \Delta L)_1$ in Eq 5. This is to be evaluated under actual flow conditions. If the flow is in a rough pipe, rough pipe friction factors have to be used or, for special circumstances, drag reducing additives, internally finned tubes etc., $(\Delta P / \Delta L)_1$ has to be experimentally determined. In essence, an important principle is being asserted which is:

The relation between X and ϕ_1 is independent of variations in single phase pressure drop.

For determining the acceleration pressure drop, Martinelli (3) has presented two methods. If slip is assumed between the phases, calculations require the knowledge of void fraction. If, however, it is assumed that the two phases have the same velocity, calculations are much simplified and the following equation may be written;

$$\left(\frac{\Delta P}{\Delta L} \right)_A = \frac{16W^2}{\pi^2 D^4 \cdot \Delta L} \left[\frac{(1-x_e)}{\rho_{1e}} - \frac{(1-x_i)}{\rho_{1i}} + \frac{x_e}{\rho_{ge}} - \frac{x_i}{\rho_{gi}} \right] \quad (9)$$

As the acceleration pressure drop in the present case was only a small fraction of total pressure drop and void fraction data for ammonia evaporators were unavailable, all calculations were done with Eq 9, i.e., no slip between the phases was assumed.

The static pressure drop may be estimated as

$$\left(\frac{\Delta P}{\Delta L} \right)_S = R_1 \cdot \rho_1 \left(\frac{\Delta S}{\Delta L} \right) \quad (10)$$

As the experimental evaporator was horizontal, this contribution was zero.

(b) Experimental Results

Local two phase pressure drops were first calculated using Eqs 1 and 4 to 9. Friction factors were obtained from Moody Chart. The results as given in Fig. 7 show the calculated pressure drop to be far greater than measured values.

The calculations were repeated in the same manner but using a friction factor of .018. The results for the turbulent-turbulent region, in which most of the data are, are shown in Fig. 8. It is evident that, though some points show greater scatter, the great majority of data are predicted within $\pm 30\%$. With the wide fluctuations that occurred during flow, absence of any scatter would indeed be surprising. It will be remembered that pressure drop data have been used raw, without any smoothing.

In many cases, the designer is concerned with the total pressure drop only. By numerically integrating the local pressure drops over the length of the evaporator, total pressure drops were calculated. It is seen in Fig. 9 that use of standard friction factors gives errors up to 200%. It is seen in Fig. 10 that with $f = .018$, total calculated pressure drop never has error greater than 40%.

Figs. 7 to 10 have two important consequences:

1. The assumption of constant friction factor of .018 down to a Reynolds number of 1,000 is justified. As seen in Fig. 3, our measurements extended no lower than $Re = 30,000$.
2. The theory that relation between X and ϕ_1 does not depend on single phase pressure drop is upheld.

In the viscous-turbulent region, neither the standard friction factors nor $f = .018$ gave satisfactory correlation, though the results were better with standard factors. It is interesting to note for qualities between about 85 to 100%, reasonable estimate of two-phase pressure drop may be made by simply calculating the single phase drop for vapor flowing alone.

Two Phase Heat Transfer

No generally accepted heat transfer correlation is available. One is derived here based on arguments similar to those made in interpreting the Martinelli correlation.

A variable ψ is defined such that

$$h_{TP} = \psi \cdot h_1 \quad (11)$$

Further, a parameter Y is defined such that

$$Y = h_1/h_g \quad (12)$$

It is now postulated that ψ is a function of Y . The principle that phase interaction phenomena are independent of single phase phenomena is again asserted. It is implied that if the relation between Y and ψ is established for a reference system (round smooth pipe for convenience), it will also apply to cases where single phase heat transfer is affected by circumstances like scale, oil films, turbulators, etc.

Using Eq 2, we may write for each phase flowing alone

$$h_1 = 0.023 Re_1^{0.8} Pr_1^{0.4} \frac{k_1}{D} \quad (13)$$

$$h_g = 0.023 Re_g^{0.8} Pr_g^{0.4} \frac{k_g}{D} \quad (14)$$

$$\text{Hence, } Y = \left(\frac{Re_1}{Re_g} \right)^{0.8} \left(\frac{Pr_1}{Pr_g} \right)^{0.4} \frac{k_1}{k_g} \quad (15)$$

This may be expanded to yield

$$Y = \left(\frac{1}{x} - 1 \right)^{0.8} \left(\frac{\mu_g}{\mu_1} \right)^{0.4} \left(\frac{c_{p1}}{c_{pg}} \right)^{0.4} \left(\frac{k_1}{k_g} \right)^{0.6} \quad (16)$$

It will be noted that starting with Dengler and Addoms (9), several somewhat similar correlations using X_{tt} as the correlating parameter and using a correction factor for effects of nucleation have been proposed. It is pointed out that such correlations ignore the influence of Prandtl number whose significance in heat transport phenomena could hardly be denied. It is further suggested that the influence of nucleation will be small, if any, in the turbulent-turbulent region and would be accounted for while experimentally establishing the relation between Y and ψ .

With Eq 15 to calculate Y , Eq 3 for h_1 , ψ is calculated with Eq 11 using measured values of h_{TP} . The results are plotted in Fig. 11. Ignoring the data in the viscous-turbulent region, the dotted line would appear to be the mean through the data. However, due to the difficulties in estimating wall temperatures pointed out earlier, it would at best be a mere indication of the true curve. Hence some independent method of establishing the Y - ψ relation is needed.

The same was attempted by analyzing available data for other refrigerants. We note that if no subcooled flow or liquid deficiency occurs in the evaporator, conditions are sufficiently removed from the peak of heat transfer curve, and variations of quality with length are linear, Eqs 12, 13 and 15 can also be applied to mean heat transfer data. On this basis data on R-11 (4), R-12 (5), R-22 (6, 7), and R-113 (8) were analyzed. For the last mentioned, local data were read from small graphs and the property values are uncertain. All property data were taken from 1972 ASHRAE Handbook of Fundamentals. The results are shown in Fig. 12. Data for Y less than one have been ignored. The relation in this region has to be established with local data as peak heat transfer and liquid deficiency are likely to occur here, involving sharp changes in slope of curve, and hence making average methods too erroneous.

It is seen from Fig. 12 that for $Y = 1$ to 10, the $Y-\psi$ curve can be established with much confidence, most of the points being well within 15%. Hence the same line has been superimposed on Fig. 11 and extrapolated to higher values of Y . It is seen that this line lies near the upper limit of author's data. Now, according to the arguments used in establishing our correlation, there is no reason for ammonia to have a curve different from other refrigerants. Hence the solid line in Fig. 11 is recommended for design. It may be mentioned that if, in calculating the heat transfer coefficients for ammonia, the bulk temperatures had been used instead of saturation temperatures, the two lines would have been much closer.

Confirmation From Grasso Data

The single phase heat transfer and pressure drop results presented here are rather surprising and it is desirable to check them against independently collected data. Unfortunately, no such data are available in literature. However, at Grasso Company in Holland, Van Maale and Cosijn (10) measured total pressure drop and mean heat transfer coefficients on an ammonia evaporator. As the two phase calculations also involve the single phase relations developed here, analysis of their data will give some indication of the validity of our conclusions.

Their evaporator consisted of 5 serpentine coils, made from 15mm ID, 18mm OD, horizontal, commercial grade steel pipe, connected in parallel. Ammonia was circulated with a pump and heating was provided by R-11 condensing on the outside surface of coil. Inside heat transfer coefficients were estimated by deducting the calculated resistance of condensing R-11 from total measured resistance. Total pressure drop was measured across the inlet and outlet from evaporator.

Total height of evaporator was estimated to be 568mm. For each bend, an additional equivalent pipe length of 1m was assumed. Using Eqs 5, 7, 9, 10 and assuming a constant friction factor of .018, total two-phase pressure drops were calculated. Fig. 13 shows the correlation to be within +20 and -10%. If standard friction factors had been used, calculated pressure drops would have been much higher in most cases. Average heat transfer coefficients were next estimated using Eqs 3, 11, and 16. The results as shown in Fig. 14 are quite satisfactory. Use of Eq 13 for estimating h_1 would have resulted in a gross over-estimation of two-phase heat transfer coefficients.

Design Recommendations

Based on the data and analysis presented in the foregoing, some recommendations for the design of ammonia evaporators containing oil are made. Evaporators with other refrigerants are also mentioned in passing. While no data are available for ammonia evaporators without oil, it is suggested that these can be designed in the same way as for other refrigerants.

1. Single Phase Pressure Drop

A friction factor of .018 may be used as long as oil is likely to remain fluid. A safe maximum limit would seem to be 40,000 centistokes for the oil viscosity. For higher oil viscosities, standard friction factors may be used, noting that higher or lower values may actually occur.

2. Single Phase Heat Transfer

Eq 3 may be used down to a Reynolds Number of about 3,000. Alternatively, heat transfer coefficients may be calculated by Eq 2 and the resistance of oil film may be estimated from thickness of oil film given by Fig. 6. For conditions where oil is near its set-point, even lower values may prevail.

3. Two-Phase Pressure Drops

In the turbulent-turbulent region, $f = .018$ is to be used throughout. For viscous-turbulent regime, no recommendations can be made. However, for vapor qualities greater than about 85%, the pressure drop of vapor phase alone was found to approximate to two-phase pressure drop.

4. Two-Phase Heat Transfer

h_1 is estimated as in Par. 2 above. Y is estimated with Eq 15 or 16. Obtain ψ from Fig. 11. Then use Eq 11 to calculate h_{TP} . Do not extrapolate Fig. 11 for Y less than 1.

5. Other Refrigerants and Oil-Free Ammonia

The same procedure is followed but for single phase pressure drop, standard friction factors are used and Eq 13 is used for estimating h_1 .

6. Complete Design of Evaporator

Complete design of evaporator involves an iterative procedure employing the methods outlined in the foregoing. Knowing the coil exit temperature and outside heat transfer coefficient or heat flux, a certain exit quality is assumed. The mass flow rate is now fixed by consideration of total heat load to be carried. Heat transfer and pressure drop calculations are now made on a short length of coil, thus determining the change in quality and pressure. Calculations are repeated on similar elements, working backwards until the required heat load is met. Optimization is achieved by assuming different exit qualities until the best balance between coil length and pressure drop is obtained.

Approximate estimates of heat transfer and pressure drop may be quickly made working with mean quality. In many cases, the results will be sufficiently accurate for practical purposes.

Concluding Remarks

1. Though much more data analysis is needed, it appears that a generally applicable method for calculating two-phase heat transfer coefficients has been found. Local data on many systems, including two-phase two component systems, should be analyzed to establish the relation between Y and ψ . Specially needing attention is the region for Y around and less than 1. A further refinement would be to plot the curves for each flow pattern separately. Conditions near critical point may also pose problems.

2. The results for single phase flow of ammonia presented here are of much practical significance. Though these are to some extent confirmed by successful correlation of two-phase data from Grasso evaporator, direct experimental verification is highly desirable.

3. The theory advanced here that the enhancement of single phase heat transfer and pressure drop due to simultaneous presence of two phases is not affected by variations in single phase heat transfer and pressure drop, finds confirmation through the successful correlation of author's own and Grasso data. Indeed, its validity is evident in being able to use the correlation of Lockhart and Martinelli, based on smooth pipe data, for calculating pressure drop in rough pipes. Anticipated further applications of this principle are non-circular ducts, internally finned pipes, those with internal turbulators, pipes with scale, addition of drag reducers, etc. Only through experiments can its general validity be ascertained.

ACKNOWLEDGMENT

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Thanks are also due to Mr. E. A. Cosijn of Grasso Koninklijke Machinenefabriek who made their experimental data on ammonia evaporators available from their original test records.

NOTATION (Any consistent system of units may be employed.)

- C_p = Specific heat at constant pressure.
 D = Internal diameter of pipe.
 f = Friction factor in Darcy equation.
 g = Acceleration due to gravity.
 h = Heat transfer coefficient.
 k = Thermal conductivity.
 R_1 = Area of pipe section occupied by liquid phase/Area of section.
 W = Total mass flow rate.
 X = Martinelli parameter for pressure drop.
 x = Thermodynamic vapor quality.
 Y = Parameter for heat transfer correlation = h_1/h_g
 Re = Reynolds Number.
 Pr = Prandtl Number.
 Nu = Nusselt Number.
 V = Velocity through the pipe.
 μ = Dynamic viscosity.
 ρ = Density.
 ϕ_1 = Multiplier for two phase pressure drop.
 ψ = Multiplier for two phase heat transfer.
 $(\Delta P/\Delta L)$ = Pressure drop per unit length.
 $(\Delta S/\Delta L)$ = Change in elevation per unit length.

Subscripts

- e = At exit from element of pipe.
 i = At inlet to element of pipe.
 l = For liquid phase flowing alone.
 g = For gas phase flowing alone.
 A = Acceleration component.
 S = Static component due to change in elevation.
 TPF = Two phase frictional component.
 TP = Total two phase.
 tt = Turbulent- turbulent regime.
 vt = Viscous- turbulent regime.

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APPENDIX

Since this paper was written in late 1972, some additional data analysis has been done. This appendix summarizes the pertinent results.

- (a) The two-phase heat transfer correlation presented here was tested against more extensive data on boiling refrigerants and one set of data on condensing R-11.

In the turbulent-turbulent region with exit vapor quality less than 85% (for evaporators), the following equations correlate the vast majority of data to within +30%.

$$\psi = 16/Y^{0.8}, Y < 20 \quad (17)$$

$$\psi = 1.3, Y > 20 \quad (18)$$

Eq 17 differs slightly from the correlating curve of Fig. 12.

- (b) The author's colleague, Mr. Rolf Grønnerud, had earlier carried out extensive tests with R-12 on the same evaporator used later by this author for the tests on ammonia. His data for all liquid flow, covering a range of +5 to -30C, were analyzed. Heat transfer coefficients were found to be in good agreement with predictions of the Dittus-Boelter equation. As R-12 is miscible with oil, no insulating oil films were expected and this agreement dispels to a great extent possible doubts about accuracy of instrumentation and measuring techniques.

The results of friction factor calculations were almost exactly the same as shown in Fig. 3 throughout the temperature range of +5 to -30C. It is hence concluded that the low friction factors are not peculiar to the oil-ammonia mixtures as is tentatively suggested in the paper.

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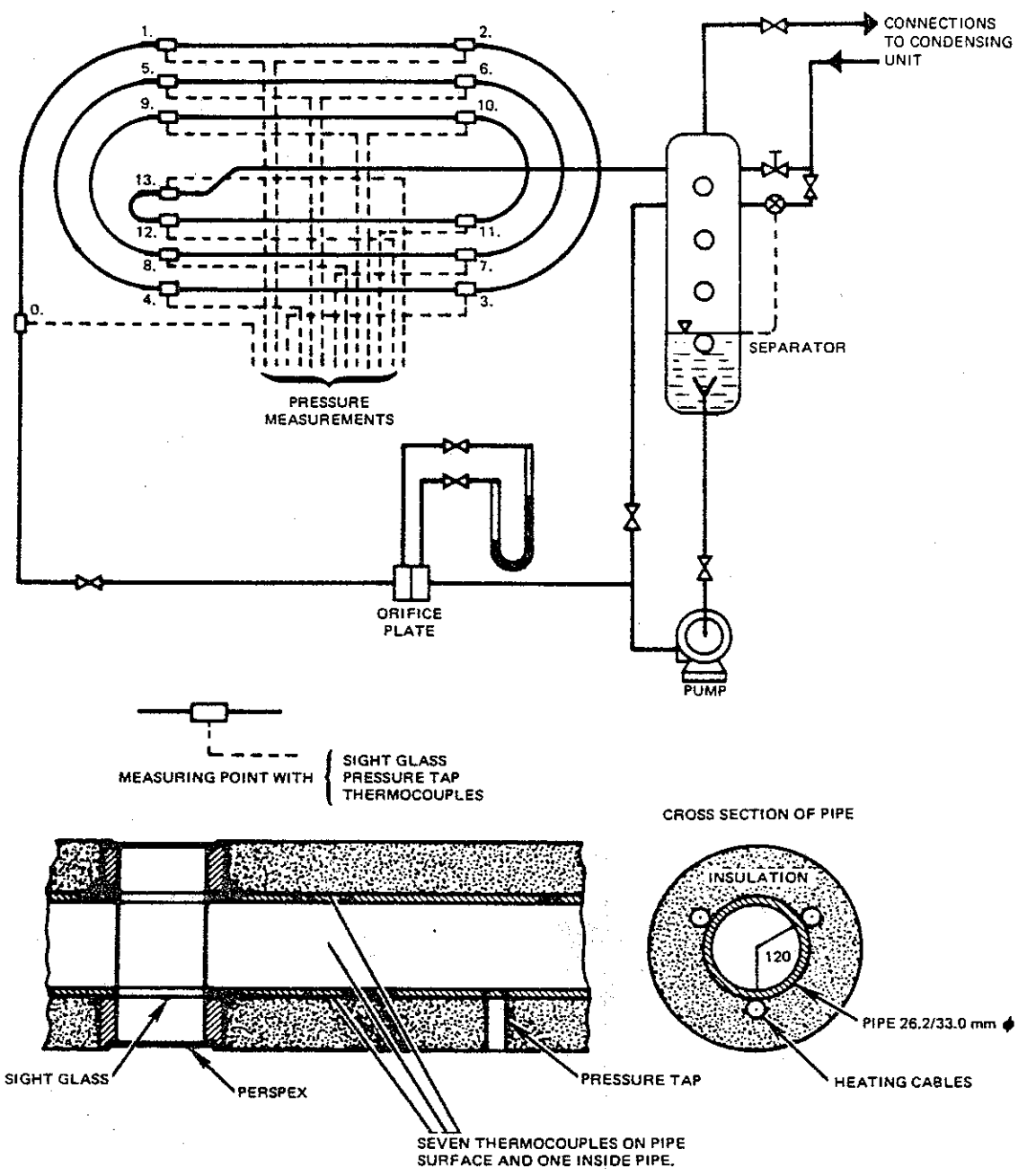


Fig. 1 Schematic arrangement and some design details of test evaporator

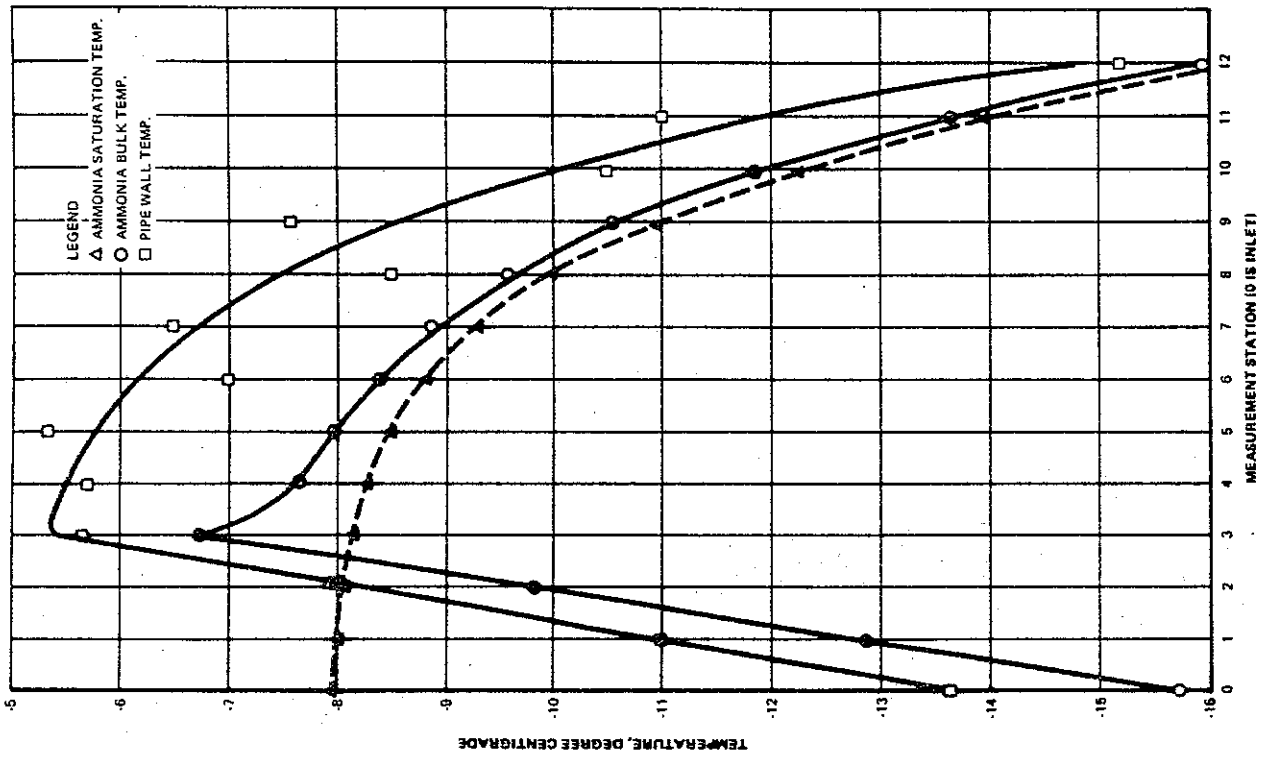


Fig. 2 Temperature measurements in a typical run

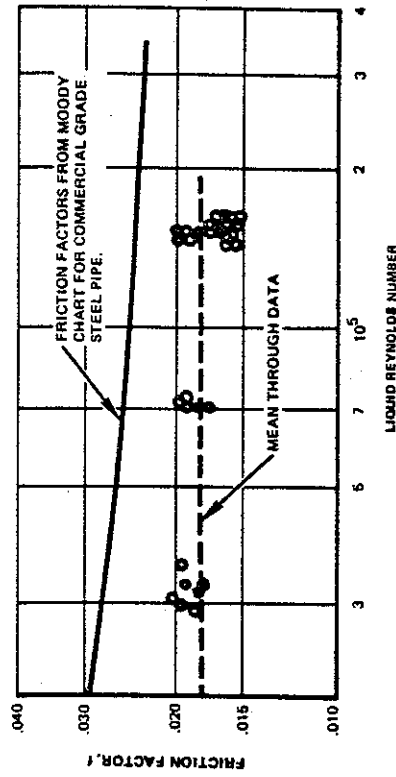


Fig. 3 Calculated Darcy friction factors. Ammonia temperature 00 to -150C

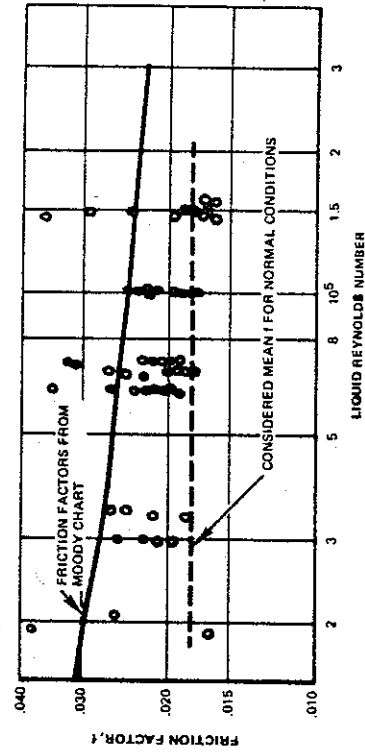


Fig. 4 Calculated Darcy friction factors. Ammonia temperature -220 to -370C

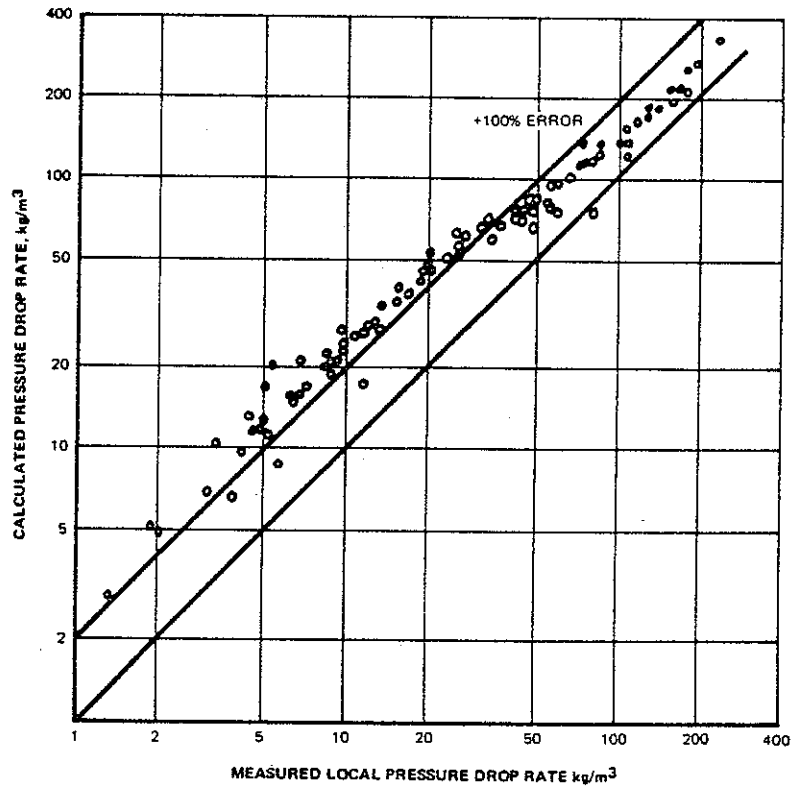


Fig. 7 Local pressure drop rates by martinelli method. Friction factors from Moody chart, turbulent-turbulent region

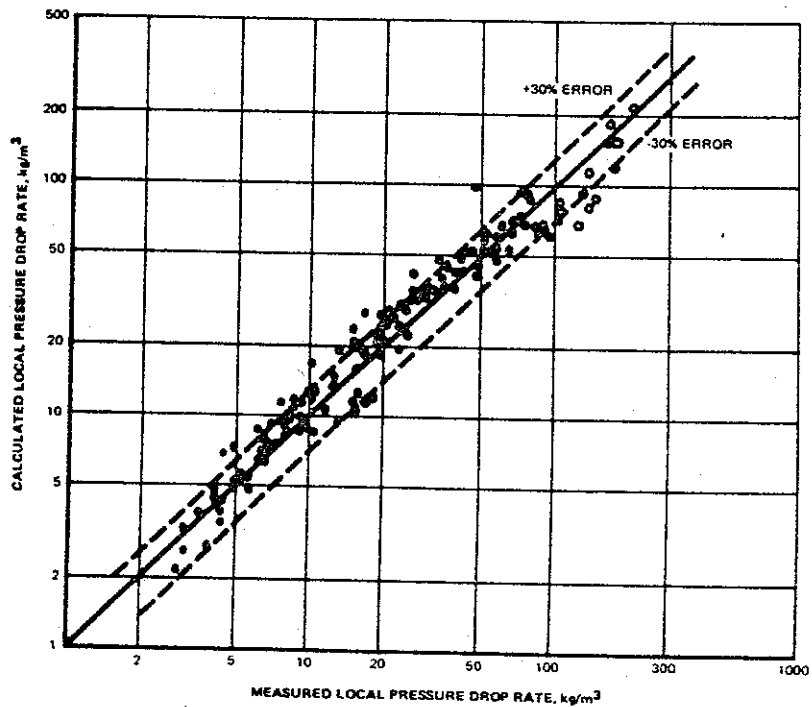


Fig. 8 Calculation of local two phase pressure drop by Martinelli method. Constant friction factor .018 turbulent-turbulent region

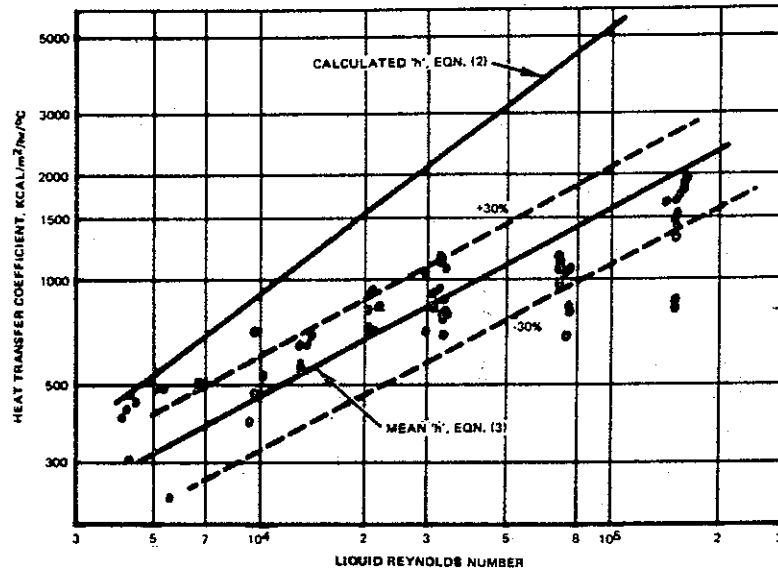


Fig. 5 Measured and calculated single phase heat transfer coefficients

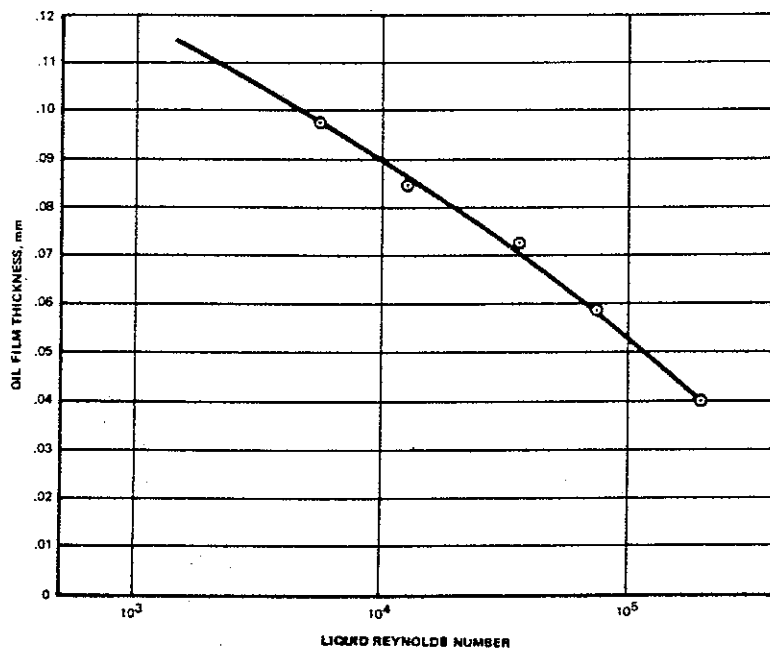


Fig. 6 Thickness of oil film required to explain low single phase heat transfer coefficients

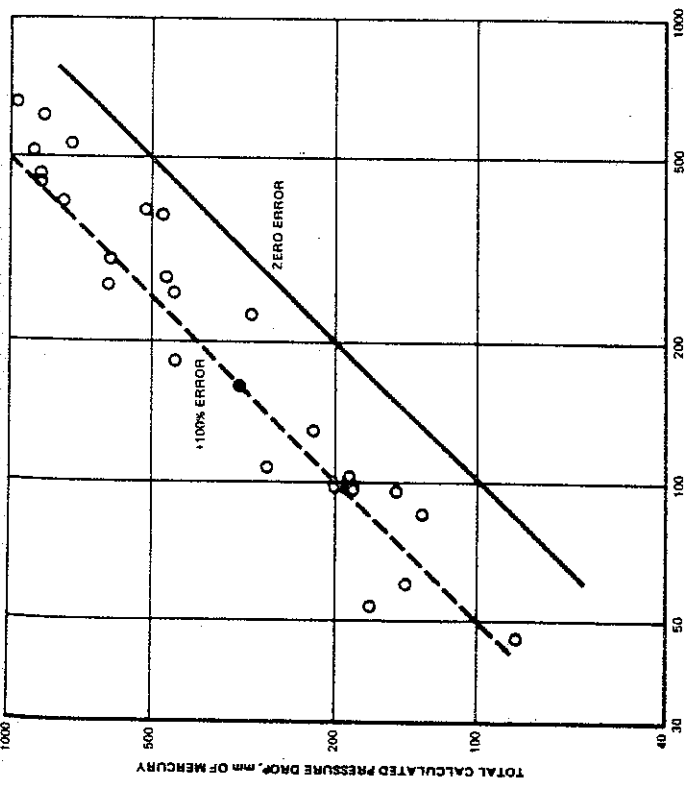


Fig. 9 Total pressure drop for the evaporator. Standard friction factors from Moody chart

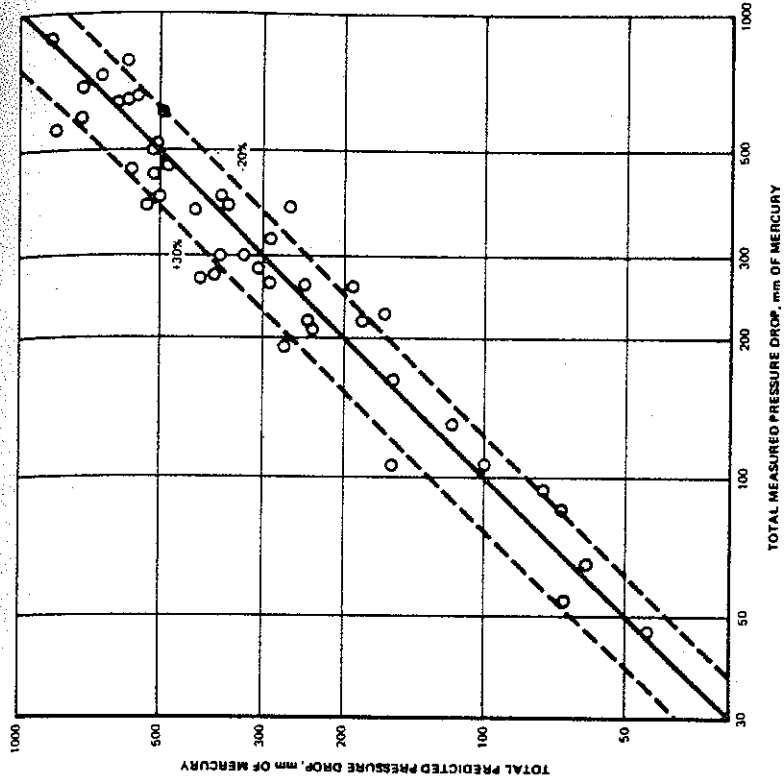


Fig. 10 Total pressure drop for the evaporator using $f = .018$

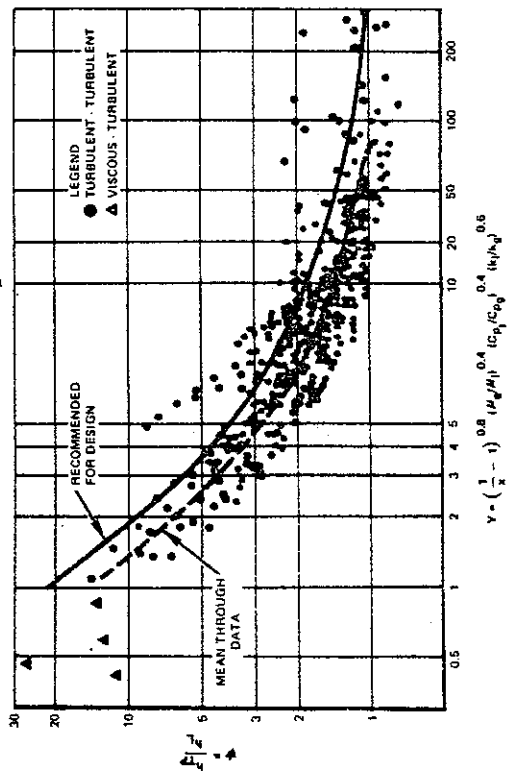


Fig. 11 Relation between parameters Y & ψ with author's data on ammonia evaporator

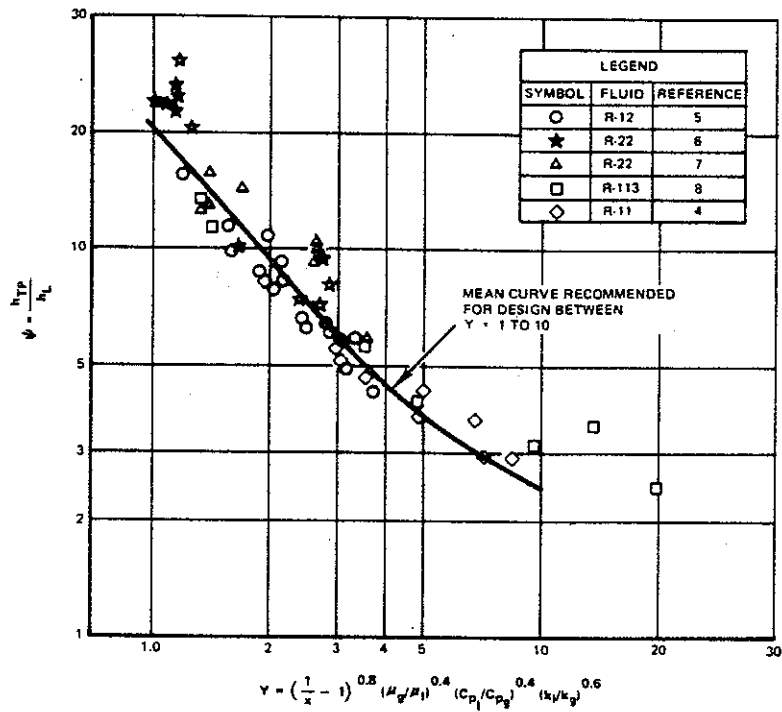


Fig. 12 Relation between Y and ψ with data on several refrigerants

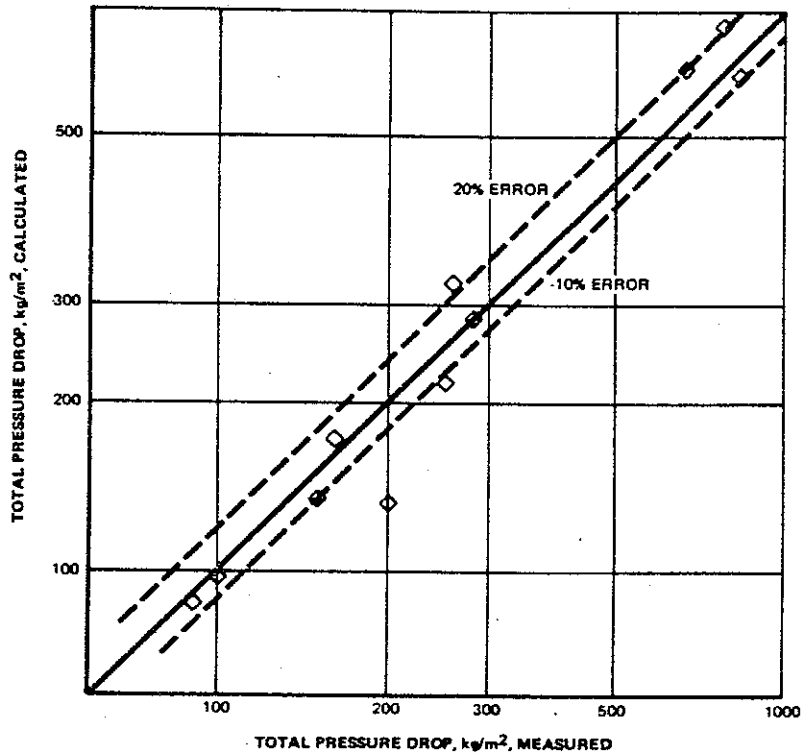


Fig. 13 Analysis of data from Grasso evaporator. Constant friction factor .018 used

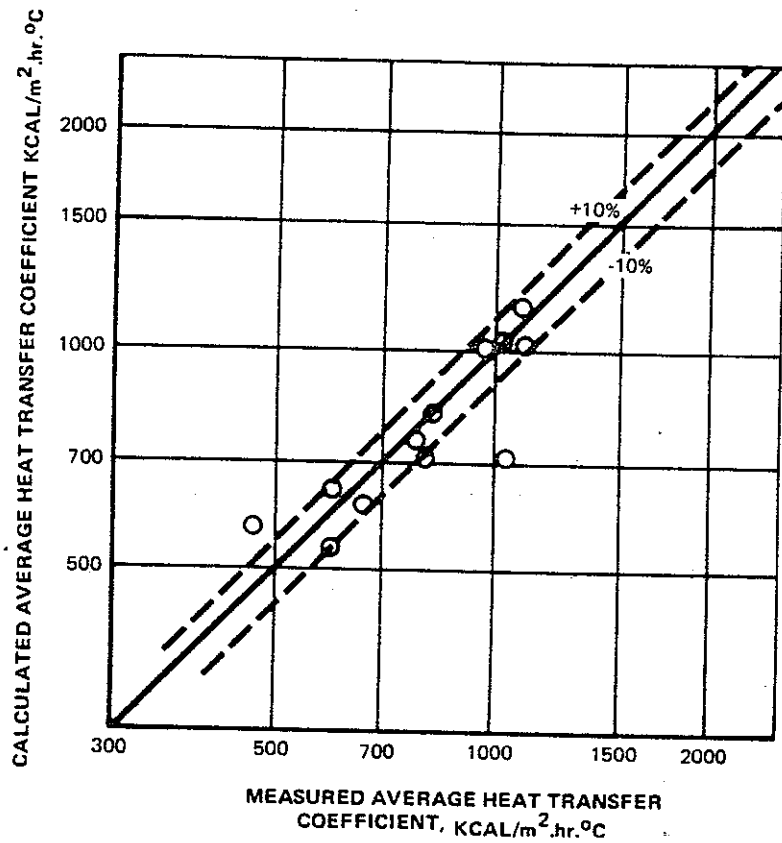


Fig. 14 Analysis of data from Grasso evaporator. Calculations using equations 3 & Fig. 11