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Pressure Drop Studies in a Plate Heat Exchanger

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Recent studies on heat transfer in plate heat exchangers have pointed out the need for quantitative expressions for pressure drop in this type of apparatus. This paper presents the results of experimental studies to obtain pressure drop relationships useful in plate heater design.

A plastic prototype of a commercial model (Chester-Jensen) exchanger was fabricated and used to obtain entrance, exit, crossover, ribbed section, and overall pressure drops as a function of velocity with water as the fluid. These individual losses were combined into an expression for overall pressure drop for a series plastic plate pack having any number of plates and employing any fluid.

Overall pressure drop data on a commercial Chester-Jensen Model HTF plate heater were available from several heat transfer studies reported previously. The data for series patterns were resolved by inserting overall pressure drop, physical properties of the fluid, and number of plate passes into the general form of the equation derived for plastic plates and by solving for a new set of constants. The resulting computer-derived expression was found to predict series pressure drops in the apparatus to ± 10%.

From a similar approach it was possible to correlate the overall pressure drop data for the heater for looped flow patterns.

Although plate heat exchangers have been commercially available for many years (*1*, *2*, *3*), heat transfer and pressure drop data on these units have appeared in the literature only recently. Most of the available data deals only with heat transfer characteristics, with little attention given to pressure drop. This paper presents the results of the experimental pressure drop studies performed to develop data for plate heater design.

The heat transfer design of plate heaters is discussed by Buonopane (*4*) for turbulent flow in various plate arrangements of a Chester-Jensen model HTF exchanger. Jackson (*5*) has extended this work to the region of laminar flow in the same exchanger using corn syrup solutions.

In a typical plate heater the flow pattern is too complex for the theoretical derivation of velocity profiles and pressure drops. In these units the fluid enters and leaves a rectangular flow channel at diagonal corners, inlet and outlet ports are at right angles to the plates, and the surfaces of the plates are covered with ribs or other turbulence promoters (2, 3). Friction across the ribs may be composed of numerous contraction and expansion losses as well as skin friction because of alternating variations in plate spacing. In addition, many of the usual plate pack configurations have crossover pressure drops, that is, the pressure drop of the fluid between plates in series.

Owing to these complexities, an experimental approach must be used. Previous work on flat parallel plates (*6*) is of little assistance because end effects were carefully eliminated, and no turbulence promoters were present. Watson (*7*) studied velocity profiles between one heater plate and a flat Plexiglas plate using dye injection techniques, but as Maranci (*8*) points out, this arrangement lacks the effect of interlocking ribs found in the typical plate heater.

Maranci (*8*) and Brown (*9*) constructed ribbed Plexiglas plates of various geometries similar to those found in commercial plate heaters. Using a dye injection technique and color photography they studied the effect of rib pattern, plate separation, and type of corner gasket on the velocity profiles, areas of stagnation, and clearing times. Some pressure drop data was obtained, but correlations of these data suitable for heater design were lacking.

Maranci (*8*) showed that the friction factors for flat plates were not applicable to plate heater plates, although the equation was of the same general form. He found that

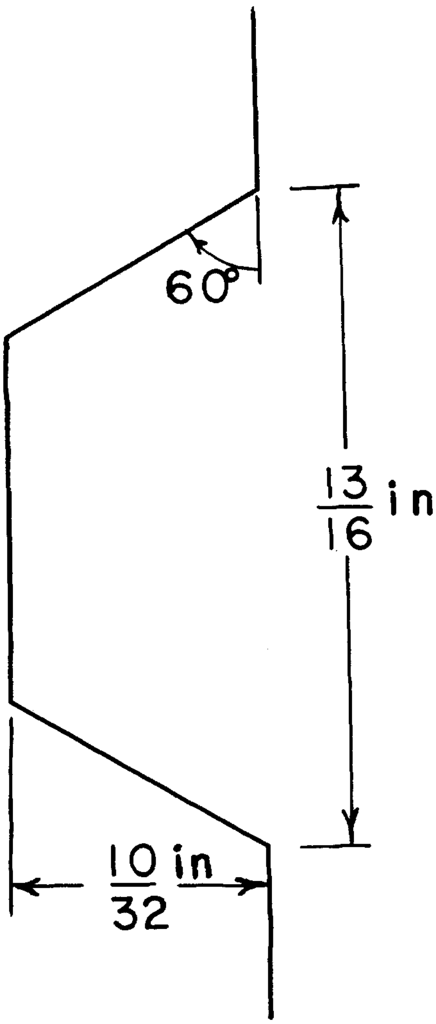

Fig. 1. Rib geometry of pattern *A* ribs.

the transition region for flow in heater plates was lower than for flow in flat plates and that the friction factor varied with rib type and increased with increasing plate separation. In his work it was found that for water in a single pass ribbed plate, pressure drop data could be correlated by

$$\Delta P = cV^s \qquad (1)$$

where the values of c and s depended on whether the pressure drop was for entrance, exit, or plate.

In all the studies reported, data was presented for a single plastic plate passage only, while for practical design purposes information is needed on large packs of metal plates in various groupings.

Direct measurement of the individual pressure drops which make up the total pressure drop of a commercial plate heater is virtually impossible owing to the nature of the construction of this apparatus. These individual losses are entrance loss, exit loss, loss across the plate face (rib section), and crossover loss between plates, if the plate pack involves more plates than the most basic configurations.

In order to solve this problem, it was decided to construct a prototype model in transparent plastic from techniques similar to those of Maranci (*8*) and Brown (*9*). This model was designed to nearly duplicate the main geometric features of an available commercial heater. A Chester-Jensen prototype was chosen because of the previous work reported on heat transfer characteristics in this model and because of the availability of overall pressure drop data on the commercial unit (*10, 11, 12*). It was felt that pressure drop data determined from the model would simplify analysis of data from the commercial unit when a mathematical expression for pressure drop was sought in the latter.

Other advantages of transparent plastic plates were visual observation of the flow patterns and the ease of making geometric changes in order to determine the effect of geometry on pressure drop.

EXPERIMENTAL

The plastic plates used in the experimentation were constructed of ½-in. Plexiglas sheets 32 in. long and 9.5 in. wide to which trapezoidal Plexiglas ribs were cemented. Two rib patterns are reported in this paper:

	Pattern *A*	Pattern *B*
Rib base	13/16 in.	13/16 in.
Rib top	7/16 in.	1/4 in.
Base angle	60 deg.	60 deg.
Rib spacing	17/32 in.	1/4 in.
Plate dimensions (inside gasket)	7.5 × 27.5 in.	7.5 × 27.5 in.

Pattern *A* simulates the Chester-Jensen plate and the rib geometry is shown in Figure 1. Additional patterns are being studied.

Neoprene rubber sheeting was cut to form the corner gaskets and the gaskets to separate the plates. The shape of these gaskets was identical to those used in the commercial apparatus. Three-quarter inch inside diameter ports were installed in diagonally opposite corners.

For study of single channel flow, two plates were assembled, separated by the gasket, and with interlocking ribs. Complete rigidity was assured by clamping the plastic assembly between two perfectly flat, parallel steel grids.

Pressure taps were located on piezometer rings at the inlet and outlet ports, also on the plates 1 in. above and below each rib section and at several points in the rib section.

Principal locations of these taps are shown in Figure 2 which also shows the flow path between plates.

Exact value of the plate separation was established by measuring the overall distance between the outer surfaces of

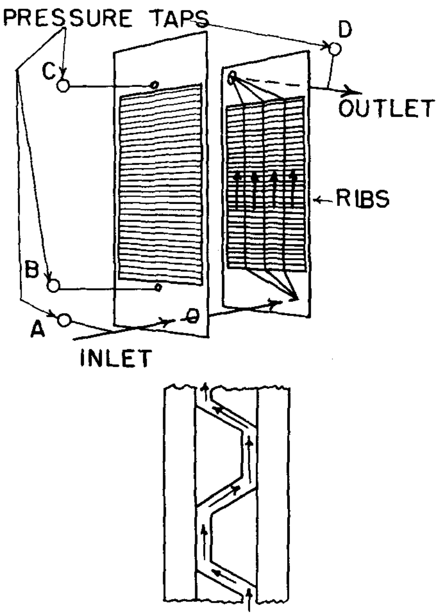

Fig. 2. Plastic plate assembly showing pressure tap locations and flow path.

the frames at eight spaced locations with and without gaskets. A depth gauge, having a vernier accurate to 0.001 in. was used, and the measurements were averaged.

Water was used as the test fluid and was pumped through the plate channel. The flow rates were accurately measured by use of rotameters which had been calibrated by collecting and weighing the effluent. Pressure drops were measured on standard manometers with either mercury or carbon tetrachloride used in accordance with the magnitude of the differential pressure. The average error in measurement was found to be 0.2%.

To measure crossover pressure drop, a third Plexiglas sheet, ⅛ in. thick and with ribs on both sides, was placed between the two plates previously described and separated from them by gaskets, Figure 3. A crossover port was cut in the corner of this middle plate. The pressure drop across this port was measured. All plastic plate data are available in reference *13*.

CORRELATION OF PLASTIC MODEL PRESSURE DROPS

For water in plastic model *A*, the pressure drop data (Figure 4) was correlated with velocity by use of Equation (1), given previously, with correlation coefficients greater than 0.99 in every case. For a single passage, the pressure drop across the ribs is 40 to 80% of the total, with entrance port pressure loss 10 to 25% and exit port pressure loss 10 to 30% of the total. Distribution between these three sources of loss varies with velocity and plate spacing. In a commercial exchanger, multiple plate arrangements are used; therefore the entrance port and exit port losses are a much smaller percentage of the total, and crossover losses must be taken into account.

Once the individual pressure drop correlations have been obtained, it should be possible to combine these to determine the overall pressure drop for any number and configuration of plates in an exchanger. The basic configurations are referred to in the literature as the series pattern and the looped pattern (*4, 5*) and will not be redefined here.

For a heater with plates arranged in the series pattern, the pressure drop may be summed up as follows:

$$\Delta P_t = \Delta P_{ent} + n\,\Delta P_{rib} + (n-1)\,\Delta P_{cr} + \Delta P_{ex} \qquad (2)$$

This summation of the individual experimental pressure drops was performed for plate packs with up to twenty plate passes in series. The results of these calculations are shown in Figure 5. The total pressure drops obtained for each value of n were correlated with velocity in Equation (1) by a least-squares method on an IBM-1620 computer. The values of c and s obtained in Equation (1) for each value of n were then correlated with n by a least-squares method on the computer. Equation (3) represents the result for water in pattern A:

$$\Delta P_t = 0.56\,nV^{1.77} \qquad (3)$$

For pattern B under similar conditions

$$\Delta P_t = 1.08\,nV^{1.66} \qquad (4)$$

The pressure drop expressions can be expanded to any fluid under any conditions by calculating friction factor or Euler number vs. Reynolds number correlations. From the experimental data on pattern A and the physical properties of water at the prevailing temperature, friction factors were calculated for the ribbed sections by

$$f_{rib} = \frac{12b\,g_c\,\Delta P}{\rho\,V^2\,L} \qquad (5)$$

and for the entrance, exit, and crossover by

$$f = \frac{2\,g_c\,\Delta P\,144}{\rho\,V^2} \qquad (6)$$

Corresponding Reynolds numbers, based on the ribbed section, were calculated by

$$N_{Re} = \frac{2\,b\,V\,\rho}{12\,\mu} \qquad (7)$$

The total pressure drop for series flow was expressed in terms of individual friction factors by summing and rearranging:

$$\Delta P_{SP} = \frac{V^2\rho}{144\,g_c}\left[\frac{f_{ent}}{2} + \frac{f_{ex}}{2} + (n-1)\,\frac{f_{cr}}{2} + \frac{12\,nf_{rib}L}{b}\right] \qquad (8)$$

Equation (8) can be rearranged to give Euler numbers $144\Delta Pg_c/V^2\rho$. For each value of n, a curve fit was obtained for Euler number vs. corresponding Reynolds number with a least-squares technique used on the computer. Finally, the exponents and coefficients obtained in this series of expressions were correlated with values of n, and the relations obtained were reinserted into the Euler number—Reynolds number equation.

The resulting expression rearranged for ΔP was

$$\Delta P_{SP} = (1.98n - 0.05)\,(V^2\rho/g_c)\,N_{Re}^{(0.04/n-0.254)} \qquad (9)$$

Average deviation of pressure drops calculated by this equation from those shown in Figure 5 is ± 4.5%.

APPLICATION TO COMMERCIAL PLATE HEATERS

Overall pressure drop data on a commercial Chester-Jensen Model HTF plate heater were available from several heat transfer studies reported previously (*10, 11, 12*). The data for series patterns were resolved by inserting overall pressure drop, physical properties of the fluid, and number of plate passes into the general form of Equation (9) and solving for a new set of constants. The equations were solved for these constants with an IBM-1620 computer. The resulting expression was

$$\Delta P_{SM} = (1.87n + 7.56)\,(V^2\rho/g_c)\,N_{Re}^{(-0.18/n-0.187)} \qquad (10)$$

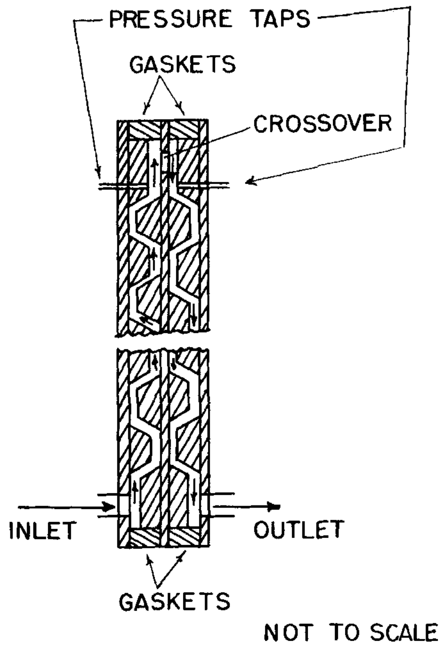

Fig. 3. Plastic plate assembly for measuring crossover pressure drop showing pressure tap locations.

Average deviation of the experimental data from this equation is ± 10%, which is approximately the average accuracy of the experimental data taken in the commercial unit.

By means of the same approach, it was possible to correlate the overall pressure drop data from the Chester-Jensen heater for the cases where a looped flow pattern was involved:

$$\Delta P_{LM} = (38.96n + 121.22)(V^2\rho/g_c)\, N_{Re}^{(-0.13/n - 0.565)} \tag{11}$$

Average deviation of the data from this equation is also ± 10%.

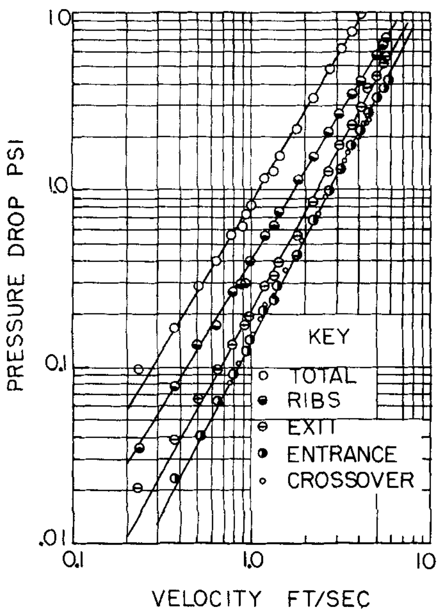

Fig. 4. Variation of pressure drops with linear velocity between plates in a single channel plastic plate (pattern *A*) and crossover pressure drop in a three-plate pack.

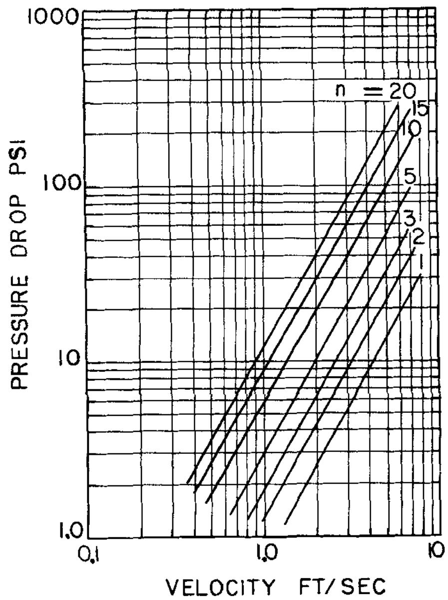

Fig. 5. Calculated variation of total pressure drop with linear velocity between plates of an *n*-pass plastic pack in series flow (pattern *A*).

It should be noted here that the metal plate pressure drop is higher than that for plastic plates. This is to be expected, since surface roughness is greater in the metal and there are some differences in geometry between the plastic and metal plates.

OTHER STUDIES

The work on pattern *B* is the beginning of a larger study on the effect of rib geometry, plate geometry, and other effects on the pressure drop and flow pattern. This study is continuing, but certain generalizations can be made at this time.

Figures 6 and 7 indicate that rib section and entrance pressure drops decrease with increased spacing for pattern *B*, while Figure 8 indicates in the case of exit pres-

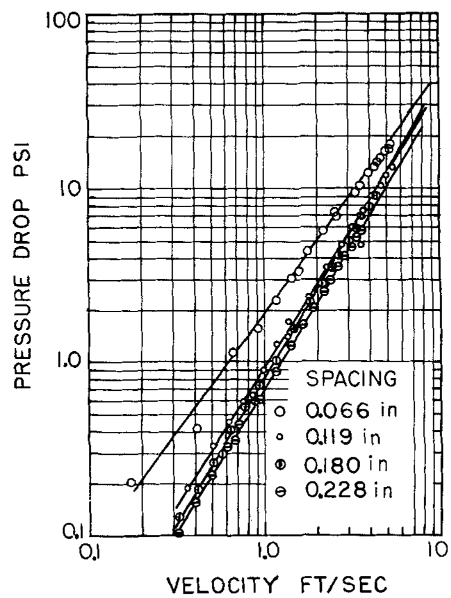

Fig. 6. Effect of plate spacing on rib section pressure drop (pattern *B*).

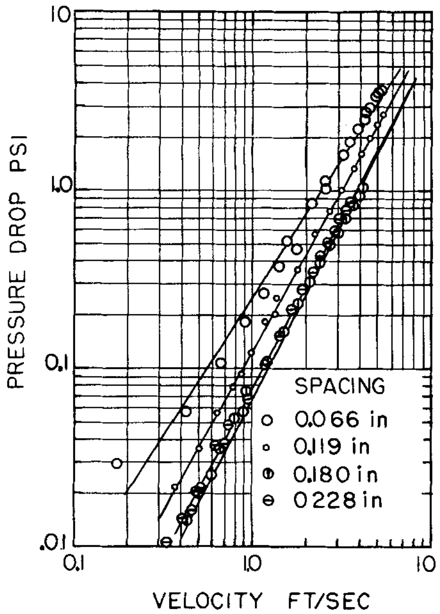

Fig. 7. Effect of plate spacing on entrance pressure drop (pattern *B*).

sure drop an increasing loss with increased spacing. The overall pressure drop, which may be obtained by summing the individual losses, also decreases with increased spacing. The range of spacing studied was 0.066 to 0.228 in. In most cases the effect of spacing is relatively small above a spacing of 0.119 in. Since commercial plate spacing is from 0.115 to 0.195 in., it can be seen that overall pressure drop is relatively insensitive to slight changes in spacing in this case.

From Equations (3) and (4) it can be seen that rib geometry is significant in determining pressure drop, indicating the importance of the projected studies on geometry.

CONCLUSIONS

As a result of the studies reported here it can be concluded that:

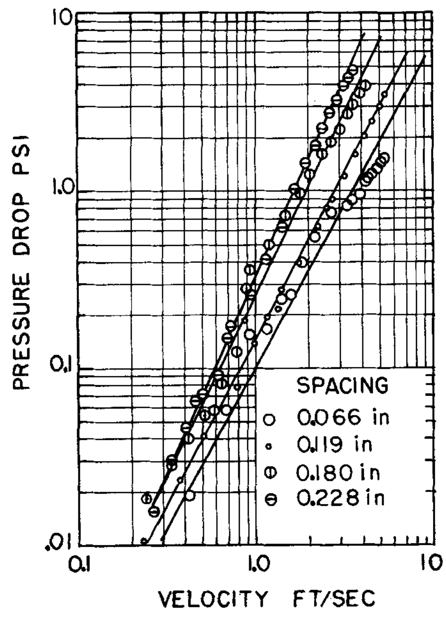

Fig. 8. Effect of plate spacing on exit pressure drop (pattern *B*).

1. It is possible to obtain an expression to predict the pressure drop in a commercial plate heater with sufficient accuracy for design purposes.
2. The equation developed here predicts the pressure drop in a Chester-Jensen exchanger with an average deviation of $\pm$ 10%.
3. Spacing between plates has little effect on pressure drop in the range utilized in most commercial plate heaters.
4. Plate and rib geometry affect pressure drop.

ACKNOWLEDGMENT

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NOTATION

b = plate spacing, in.
c = constant in Equation (1)
f = friction factor, dimensionless
g_c = gravitational constant, (ft./sec.^{2}) (lb.$_m$/lb.$_f$)
L = length of flow path, ft.
n = number of plate channels
ΔP = pressure drop, lb./sq. in.
N_{Re} = Reynolds number, dimensionless = $(2bV\rho)/(12\mu)$
s = constant in Equation (1)
V = linear velocity between plates, ft./sec.
ρ = density, lb./cu. ft.
μ = viscosity, lb./ft. sec.

Subscripts

cr = series flow crossover
ent = entrance
ex = exit
LM = looped flow with metal plates
rib = rib or plate section (thirty-three ribs in pattern A, forty-three in pattern B)
SM = series flow with metal plates
SP = series flow with plastic plates
t = total or overall

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