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TWO-PHASE PRESSURE DROP WITH TWISTED-TAPE SWIRL GENERATORS

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Abstract—An experimental study has been conducted to determine the effect of twisted-tape swirl generators on adiabatic and diabatic two-phase flow pressure drops in vertical straight tubes. Tape-twist ratios (length for 180° twist/inside tube diameter) of 3.94, 8.94, and 13.92 were tested with R-113 over a range of pressures, mass velocities, qualities, and heat fluxes. Empty tube reference data were successfully predicted with a correlation from the literature. The twisted tape data were successfully correlated by using the hydraulic diameter and a single-phase swirl flow friction factor in the empty tube correlation. Data from the literature also were predicted well with this correlation.

INTRODUCTION

Twisted-tape swirl generators have been shown to be effective in increasing single-phase and dispersed flow film boiling convective heat transfer coefficients and in suppressing the critical heat flux condition in two-phase forced-convection boiling (e.g. Lopina & Bergles 1969, Bergles *et al.* 1971, Matzner *et al.* 1965, Gambill *et al.* 1961, Gambill & Bundy 1962). However, with the demonstrated benefits of using swirl generators comes the penalty of increased pressure losses (e.g. Gambill *et al.* 1961, Gambill & Bundy 1962, Lopina & Bergles 1969, Smithberg & Landis 1969, Blatt & Adt 1963, Agrawal *et al.* 1982).

The single-phase friction losses associated with the use of full-length twisted tapes have been determined in several studies (Lopina & Bergles 1969, Smithberg & Landis 1964) as has the overall pressure drop associated with two-phase flows (Blatt & Adt 1963, Agrawal *et al.* 1982). The studies involving two-phase flows generally have shown that the total pressure drop with the twisted-tape inserts (ΔP_{tt}) over a range of tape-twist ratios and flow conditions can be as much as two to four times the pressure drop (ΔP_s) in an empty tube. None of these studies have presented any adiabatic two-phase pressure drop data. Both Blatt & Adt (1963) and Agrawal *et al.* (1982) correlated total diabatic two-phase pressure drop data (acceleration and friction components of pressure drop) with a power function: $(\Delta P_{tt}/\Delta P_s) = ay^b$. This relationship implies that both the acceleration and frictional pressure drops are affected by the insertion of the twisted tape. While these correlations may adequately predict the data from these studies for the particular flow conditions studied, they lack generality and a physical basis for their construction. Stone (1970) developed a correlation to predict the two-phase pressure drop when using helical-wire inserts which does address each pressure drop component (hydrostatic, acceleration, and friction) individually but the correlation is not applicable to full-length twisted tapes. Rosuel and Sourioux (1961) using water at 1 atm correlated a very limited number of full-length twisted tape pressure drop data using the hydraulic diameter. No adiabatic pressure drop data were taken. By evaluating each pressure drop component individually, reasonable agreement was obtained between the predictions and the experimental data.

To effectively evaluate the performance of passive augmentation devices (such as twisted tapes) which are used in forced convection situations, both the heat transfer and pressure drop characteristics of the enhancement technique must be known (Bergles *et al.* 1974, Webb 1981). While the various single-phase friction factor correlations developed for swirl flows are general enough to use in performance comparisons, the methods presented for calculating two-phase pressure drops with twisted-tape inserts are not adequate for use in a general performance evaluation in a design procedure. Hence, the present study was undertaken to develop a more general correlation for predicting the two-phase pressure drop in evaporating flows in straight tubes with full-length twisted-tape swirl generators over a large range of flow conditions.

EXPERIMENTAL APPARATUS

To accomplish the objective set forth in the preceding section, a test section was designed for insertion into a closed flow loop which uses refrigerant R-113 as the working fluid (see figure 1). The fluid flowrate to the vertical test section was controlled by bypassing fluid around the test section and by adjusting valves upstream of the test section. Flow direction through the test section was vertically upward. Inlet fluid conditions were set by adjusting the power to variable-power electric preheaters. Test-section pressure was controlled with valves located just downstream of the test section. Valves just upstream of the test section were used to keep a high pressure (1.5 MPa) in the preheaters so that when a large pressure drop was taken across those valves the liquid would flash into a liquid/vapor mixture. This large pressure drop also prevented thermalhydraulic instabilities in the test section. A four-thermocouple thermopile was located just upstream of these flashing valves to measure the liquid temperature prior to flashing. After passing through the test section, the two-phase mixture flowed through pressure control valves and a water-cooled condenser; the liquid R-113 then flowed through one of three calibrated rotameters.

A high accuracy (± 0.69 kPa) pressure gauge was used to measure the test-section inlet pressure. A thermocouple located near the pressure tap was used to measure the fluid temperature at that point and served as a check on the saturation temperature obtained from the pressure measurement. Pressure drops were measured either with a mercury (SG = 13.6) or with a Meriam red fluid (SG = 2.95) manometer. D.C. power was supplied to the test section by a low ripple (<1%) power supply. Test-section current was obtained from a calibrated shunt. Test-section voltage drop, shunt voltage, and all thermocouple voltages were measured using a high accuracy (± 1 μV) digital voltmeter, which was part of a Hewlett–Packard data acquisition system.

The vertical test section was constructed from a 8.10/9.53 mm diameter 304 stainless

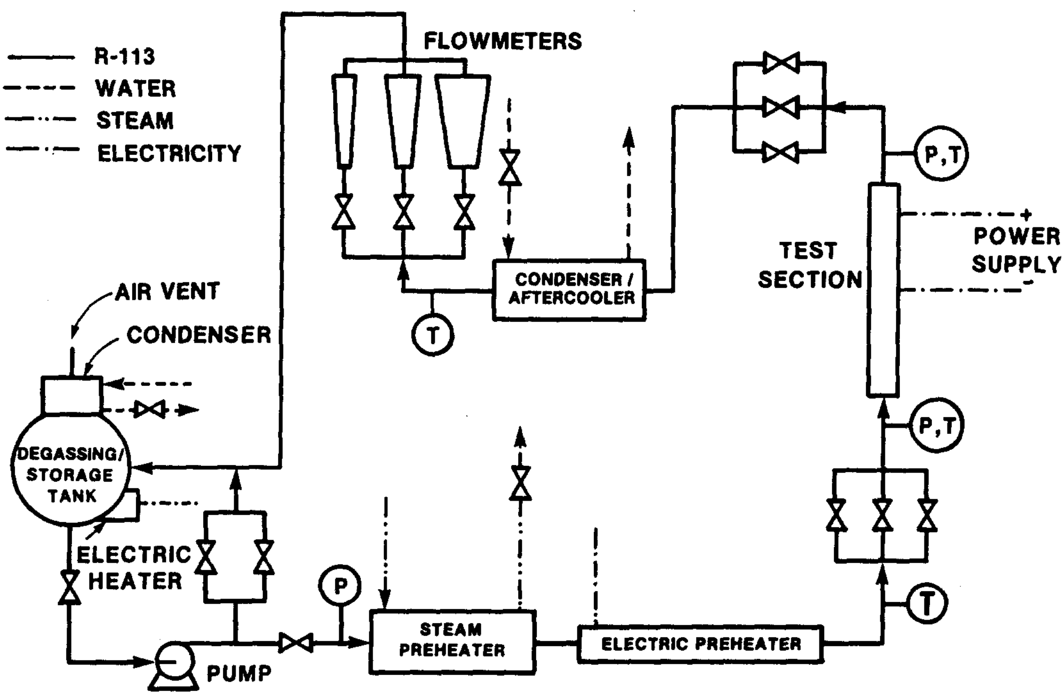

Figure 1. Schematic of test loop.

steel tubing. The heated length was 1200 mm; there was an unheated entrance length of 527 mm and an unheated exit length of 76 mm. Brass bus bars were silver soldered to the tube and pressure taps were drilled through the bus bars. Another pressure tap was located midway between the two bus bars. A fourth pressure tap was located 100 mm from the entrance to the tube. The pressure tap holes (1.77 mm in diameter) were deburred. The test section was electrically insulated from the rest of the loop by using insulated packing glands at each end. Twenty thermocouples were spaced along the outside of the test section.

Three twisted tapes were tested. The tape-twist ratios y (length for 180° twist/inside tube diameter) were $y = 3.94$, 8.94, and 13.92. The tapes were made from 0.254 mm thick 304 stainless steel. Their lengths were approximately 1835 mm so that the tape went from the beginning of the unheated entrance length to the end of the unheated exit length. The average tape width was 7.85 mm. The tape was held firmly in place at its upper end so that it could not move during a test.

A guard heater was used to eliminate heat losses from the test section. Fiberglass insulation about 25 mm thick was held tightly in place around the test section with a 50 mm I.D. copper pipe. Insulated Nichrome wire was attached to the inside of the copper pipe as were three thermocouples. Another 25 mm layer of fiberglass insulation was then used to completely enclose the copper pipe. This guard heater covered the heated portion of the test section. The guard heater power was adjusted until its temperature was approximately that of the test section. A second guard heater with an attached thermocouple was used to eliminate heat losses from the piping between the flashing valves and the inlet bus bar. The power to this guard heater was adjusted so that its temperature was approximately the same as the inlet fluid temperature. These two guard heaters had separate power supplies so that the temperature of each could be independently set.

EXPERIMENTAL PROCEDURE

To ensure the accuracy of the data, the R-113 was thoroughly degassed prior to any testing. This degassing was accomplished by boiling the refrigerant in a storage/degassing tank for several hours after any refrigerant was added to the system. The R-113 vapor was condensed in a water-cooled condenser and returned to the tank while liberated gases were vented to the atmosphere. After degassing, the tank was sealed to prevent any air from going back into solution. At this point the experiment was ready to begin.

The flow rate to the test section was set at predetermined levels by adjusting bypass valves. The guard heaters were adjusted to the approximate temperature levels. The back pressure held on the preheaters was fixed by setting the flashing valves; the test-section pressure was then maintained by adjusting the valves downstream of the test section. Both the test-section flow rate and pressure were steady. The inlet test-section quality was determined by an enthalpy balance across the flashing valves. The temperature of the liquid R-113 upstream of these valves was changed by varying power to the electric preheaters. This temperature was fixed such that a specified inlet test-section quality would be obtained when the pressure of the liquid R-113 was reduced to the test-section pressure. The quality change across the test section was calculated from an enthalpy balance between the enthalpy gain of the fluid and the electric power dissipation in the test section. Actual measured temperatures and pressures were used in all calculations. Specified heat fluxes were set by adjusting the voltage drop across the test section.

Before any data were taken once the flow conditions were set, the flow loop was allowed to come to steady state; this took approximately one and a half hours. For diabatic test runs the heat flux was then set, the test section was allowed to come to thermal equilibrium (about 10 min), and all data then were taken; this procedure continued until data at all heat fluxes were obtained. Flow conditions were then changed to the next desired set point, and the complete procedure was repeated.

Both adiabatic and diabatic pressure drop tests were run with the empty tube and with the three twisted-tape inserts. Using the adiabatic pressure drop measurements and examining the pressure gradients between the four pressure taps, there did not appear to be a significant entrance length. Thus, the data being reported represents fully developed pressure drops. The nominal two-phase experimental conditions covered in this study are as follows:

heat flux	0 to 50,000 W/m^2
inlet quality	0% to 61%
outlet quality	0% to 78%
inlet pressure	276, 551, and 827 kPa
pressure drop	4.8 to 145 kPa
mass velocity	120 to 1600 kg/m^2 s
tape-twist ratio	3.94, 8.94, and 13.92.

In addition, several single-phase pressure drop tests were run. In all, 45 adiabatic and 81 diabatic tests were run with the empty tube and 123 adiabatic and 232 diabatic tests were run with the twisted tapes. Uncertainties in the experimental data, as estimated through a propagation-of-error analysis, are estimated to be: pressure P, ± 1.5 kPa; pressure drop ΔP, ± 0.35 kPa; mass velocity G, $\pm 2\%$; quality x, $\pm 6\%$; heat flux q'', $\pm 1\%$.

RESULTS AND DISCUSSION

The empty tube adiabatic and diabatic pressure drop data were obtained to provide a reference against which the swirl flow data could be compared and to ensure that the empty tube data obtained from this test loop agreed with predictions obtained from correlations in the literature. To make the empty tube predictions of the total two-phase pressure drop (ΔP_{tot}) [1], the hydrostatic (ΔP_h), acceleration (ΔP_a), and friction pressure drops (ΔP_f) had to be calculated:

$$\Delta P_{\text{tot}} = \Delta P_f + \Delta P_a + \Delta P_h. \qquad [1]$$

The acceleration pressure drop was calculated from

$$\Delta P_a = G^2\left[\left(\frac{x^2}{\epsilon\rho_G} + \frac{(1-x)^2}{(1-\epsilon)\rho_L}\right)_{\text{outlet}} - \left(\frac{x^2}{\epsilon\rho_G} + \frac{(1-x)^2}{(1-\epsilon)\rho_L}\right)_{\text{inlet}}\right], \qquad [2]$$

where ρ_G is the vapor density, ρ_L is the liquid density, and ϵ is the void fraction. For the adiabatic flow, ΔP_a was negligible. The hydrostatic pressure drop was calculated from

$$\Delta P_h = \int_{z_1}^{z_2} (\epsilon\rho_G + (1-\epsilon)\rho_L)\mathrm{d}z, \qquad [3]$$

where z is the distance along the test section. This equation was numerically integrated over the length of the test section.

The friction pressure drop was calculated from

$$\Delta P_f = \frac{2fG^2L}{\rho_L D}\phi_{f0}^2, \qquad [4]$$

where D is the tube diameter, L the test-section length, and f the friction factor. Various correlations exist for the two-phase friction multiplier ϕ_{f0}^2 which represents the ratio of the frictional pressure gradient in two–phase flow to the frictional pressure gradient if the total flow was considered to be liquid. In a recent report, Reddy *et al.* (1983) have tabulated a large number of adiabatic and diabatic empty tube pressure drop data from the literature (for water) and have tested many of the well-known two-phase pressure drop correlations (homogeneous, Dukler, Martinelli and Nelson, Thom, Baroczy, Chisholm) against this data. Five void fraction correlations were used (homogeneous, Thom, Bankhoff, Armand, Van Glahn).

Because the resulting predictions were not adequate for their needs, Reddy *et al.* (1983) developed a new correlation for ϕ_{f0}^2 from the adiabatic data. This new correlation accounts for the variation in ϕ_{f0}^2 with pressure, quality, and mass velocity. Using this correlation [5a],

$$\phi_{f0}^2 = 1 + x\left(\frac{\rho_L}{\rho_G} - 1\right)C, \qquad [5a]$$

where

$$C = 1.17x^{-0.175}G^{-0.45} \quad \text{for } P/P_{cr} > 0.187 \qquad [5b]$$

and

$$C = 0.41\,(1 + 10P/P_{cr})x^{-0.175}G^{-0.45} \quad \text{for } 0.094 \le P/P_{cr} \le 0.187, \qquad [5c]$$

where P_{cr} is the critical pressure, both the adiabatic and diabatic data were predicted much better than the other correlations. In this correlation the mass velocity G must have the units kg/m^2 s. In evaluating the acceleration and hydrostatic pressure drops, Reddy *et al.* (1983) found that the homogeneous void fraction ϵ gave as good a prediction as other models from the literature, so it was recommended. For the single-phase friction factor, the Blasius correlation ($f = 0.079/\mathrm{Re}^{0.25}$) was recommended. The Reynolds number is $\mathrm{Re} = GD/\mu_L$ where μ_L is liquid viscosity.

Using this correlation, the adiabatic two-phase pressure drop data (45 points) obtained with the empty tube were predicted with an average deviation between the calculated and the experimental pressure drop of $\pm 17.3\%$. The average ratio of the calculated pressure drop to the experimentally measured pressure drop was $\bar{R} = 0.91$. Much of the deviation is attributable to underprediction at the lower mass velocities and pressure. For the higher mass velocity data, the average deviation was $\pm 8.8\%$. It should be noted that the lowest pressure used in this study was slightly under the lower limit of the pressures used by Reddy *et al.* (1983); the lowest two mass velocities also were considerably lower than the lowest used by them. Because of the relatively poorer prediction at the lower mass velocities and pressure, the present correlation was not used. Therefore, for the lower mass velocities and pressures, the following correlation was fit to the present data:

$$C = 0.61\,(1 + 10P/P_{cr})x^{-1.2}\,G^{-1.74} \quad \text{for } P/P_{cr} \le 0.187 \quad \text{and} \quad G < 475\ \mathrm{kg/m^2\,s}. \qquad [6]$$

This correlation fits the lower mass velocity data with an average deviation of $\pm 15.1\%$ with $\bar{R} = 0.98$. Thus, using [5a], [5b], and [6] to represent ϕ_{f0}^2, all of the adiabatic pressure drop data were predicted with an average deviation of $\pm 11.8\%$ with $\bar{R} = 1.02$. The scatter in the data is shown in figure 2.

To predict the diabatic pressure drop, the two-phase friction multiplier in [4] needs to be integrated over the quality change in the test section:

$$\overline{\phi_{f0}^2} = \frac{1}{(x_2 - x_1)} \int_{x_1}^{x_2} \phi_{f0}^2\,\mathrm{d}x, \qquad [7]$$

where $\overline{\phi_{f0}^2}$ is the average two-phase friction multiplier. Because of the form of the expression for ϕ_{f0}^2, this integral can be evaluated analytically. Using [5a], [5b], and [6] to describe ϕ_{f0}^2,

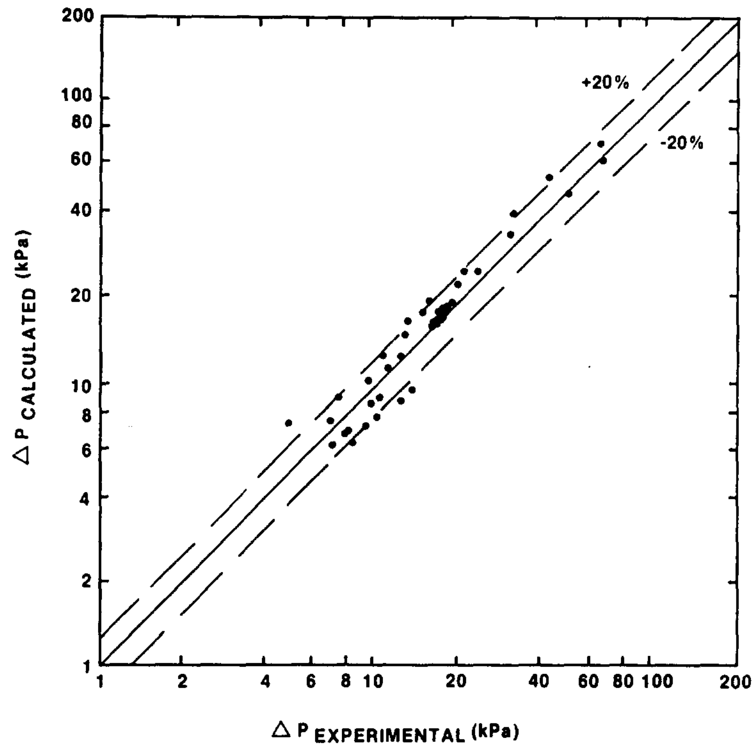

Figure 2. Comparison between the experimental and predicted adiabatic straight tube pressure drop.

the diabatic pressure drop data (81 points) were predicted with an average deviation of $\pm 13.3\%$ with $\bar{R} = 1.06$. The scatter in the data is shown in figure 3. For both the adiabatic and diabatic data, the scatter appeared to be random with no discernable pattern associated with the mass velocity or pressure. With an adequate reference for the empty tube pressure drop data, the data using the twisted tape inserts were then correlated.

The adiabatic two-phase pressure drops obtained with the twisted-tape swirl generators were larger than the empty tube pressure drops. The pressure drops increased as the tape-twist ratio decreased (tighter twist) and were in the range of about 1.2 to 3.5 times the empty tube data at the same flow conditions. Generally, when compared to the empty tube data, the increased pressure loss with swirl generators appeared to depend on the tape-twist ratio and mass velocity, not on the quality, or pressure level.

To correlate the twisted-tape pressure drop the hydraulic diameter D_h [8],

$$D_h = \frac{4(\pi D^2/4 - \delta D)}{\pi D + 2D}, \qquad [8]$$

where δ is the tape thickness, was used in the empty tube correlation [1], [5a], [5b], and [6] along with a single-phase friction factor developed for swirl flow. The mass velocity used was based on the actual flow area. Lopina & Bergles (1969) correlated their twisted tape single-phase pressure drop data with the friction factor

$$(f_{tt}/f_s)_h = 2.75y^{-0.406} \qquad [9]$$

described in [9] with the hydraulic diameter being used. They used the McAdams correlation ($f_s = 0.046/\mathrm{Re}^{0.2}$) to calculate the empty tube single-phase friction factor. For large tape-twist ratios this correlation [9] predicts swirl flow friction factors which are less than the straight tube friction factors. Therefore, the Gambill et al. (1960) correlation [10]

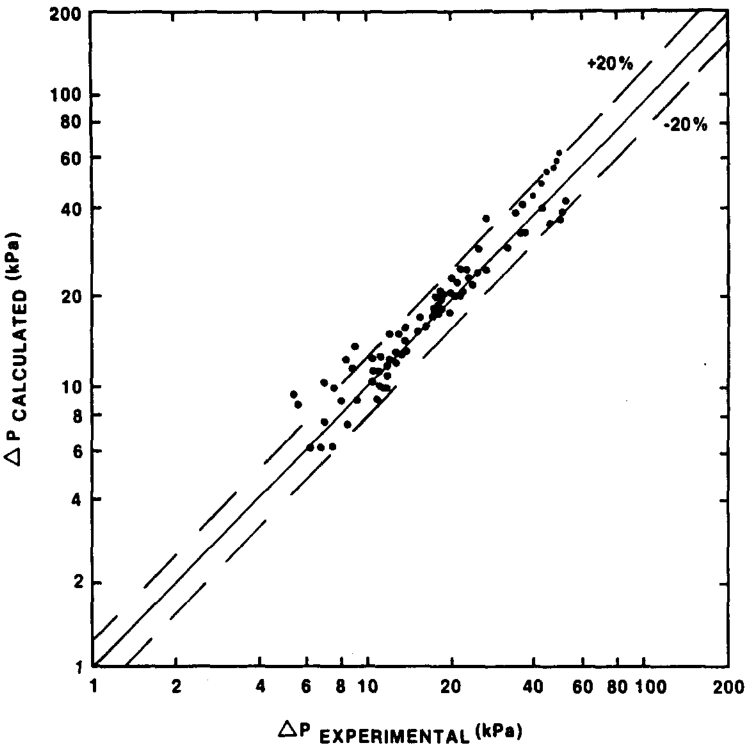

Figure 3. Comparison between the experimental and predicted diabatic straight tube pressure drop.

was used at large y's with the transition point between

$$(f_{tt}/f_s)_h = (4y^2 + \pi^2)^{3/2}/8y^3, \qquad [10]$$

[9] and [10] being their intersection at $y = 11.25$. Both the McAdams and Blasius straight tube single phase correlations were used for predicting the swirl flow data from this study. Negligible differences were obtained. Therefore, since Reddy *et al.* (1983) used the Blasius equation, it was decided that the Blasius equation would be used with both [9] and [10].

Using [1], [5a], [5b], [6], [8], [9], and [10], the adiabatic, swirl flow two-phase pressure drop data were predicted well at the higher mass velocities ($G > 475$ kg/m^2 s) but were overpredicted at the lower mass velocities and pressure. These trends in the predictions were the same for all three tape-twist ratios. Since the Reddy correlation [5c] generally underpredicted the data at the lower mass velocities, it was also used with [1], [5a], [5b], [8], [9], and [10], to predict the swirl flow pressure drop. Generally, good agreement with the experimental data was obtained, although there still was a tendency for the low mass velocity data to be overpredicted. The average deviation for the 123 adiabatic swirl flow data points was $\pm 18.8\%$ with $\overline{R} = 1.05$. The scatter in the data is shown in figure 4.

The diabatic swirl flow data were predicted with the Reddy correlation as described above. Equations [1], [5a], [5b], [5c], [7], [8], [9], and [10] were used. Good agreement between the predicted and the experimental data was obtained. The average deviation was $\pm 14.2\%$ with $\overline{R} = 0.97$. The scatter in the data is shown in figure 5. As can be seen in both figures 4 and 5, there is a tendency for the data to be underpredicted as y increases. The set of data at the high pressure drop which is underpredicted corresponds to the highest mass velocity and lowest pressure. Why it is so underpredicted is uknown.

Twisted-tape data from the literature were obtained and compared to the correlation developed here. Staub (1969) presented nine diabatic pressure drop data with R-22 at approximately 1300 kPa for a tape twist ratio of 16.25. Mass velocities ranged from about

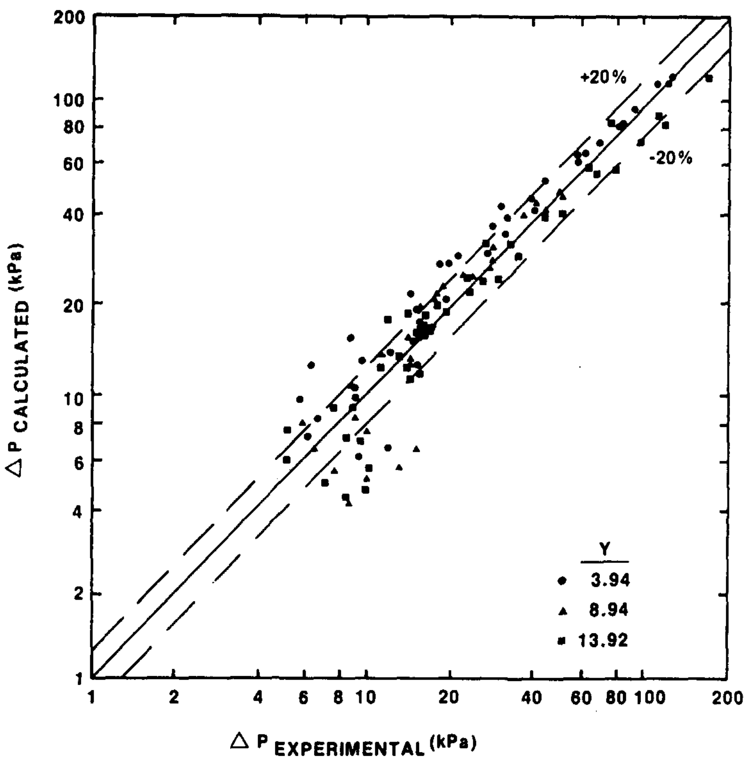

Figure 4. Comparison between the experimental and predicted adiabatic swirl flow pressure drop.

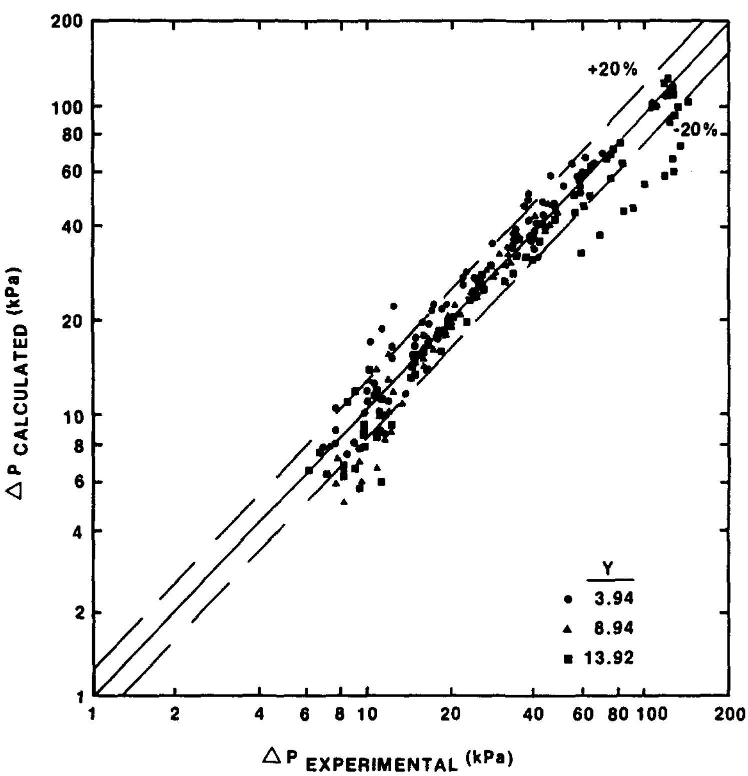

Figure 5. Comparison between the experimental and predicted diabatic swirl flow pressure drop.

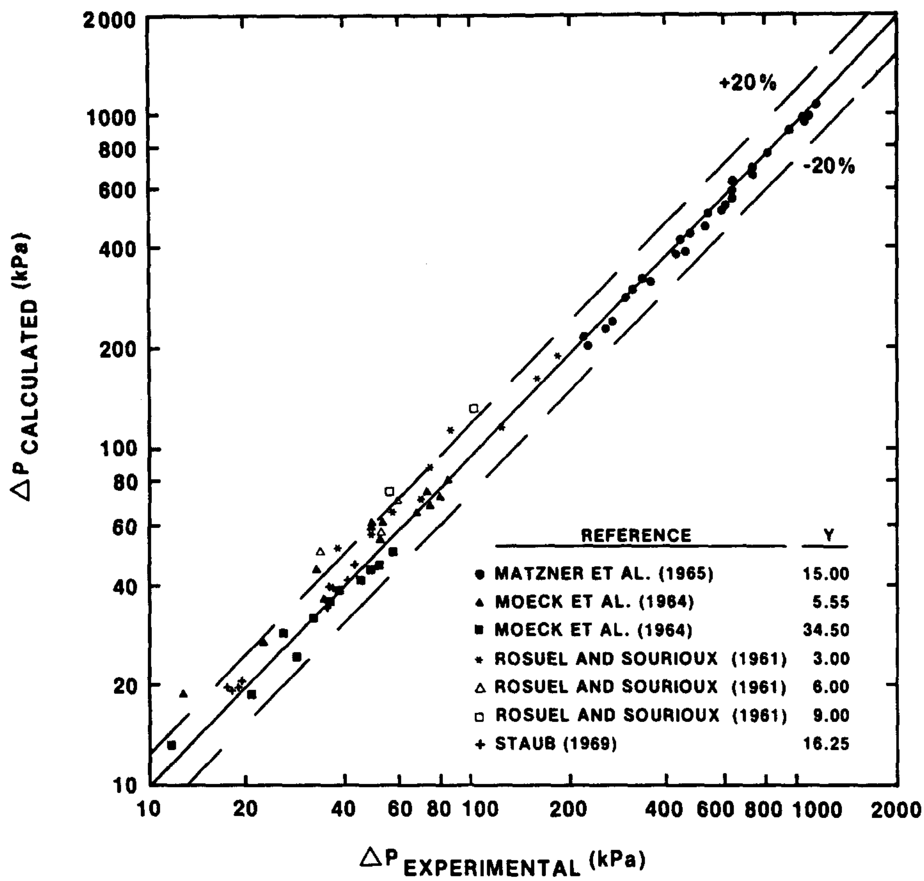

Figure 6. Comparison between the experimental and predicted diabatic swirl flow pressure drop for data from the literature.

510 to 1275 kg/m^2 s. Outlet qualities up to 89% were obtained. Pressure drops ranged from 18 to 44 kPa. These data were predicted with an average deviation of $\pm 4.3\%$ with $\bar{R} = 1.03$. Moeck *et al.* (1964) obtained diabatic data for 16 runs with $y = 5.55$ and for 10 runs with $y = 34.5$. All tests were performed at 7000 kPa with water. Inlet qualities as high as 85% were used; outlet qualities up to 95% were obtained. Mass velocities ranged from 390 to 1140 kg/m^2 s. Pressure drops ranged from 12 to 86 kPa. The data for $y = 5.55$ were predicted with an average deviation of $\pm 11.9\%$ with $\bar{R} = 1.07$. For $y = 34.5$, the average deviation was $\pm 9.5\%$ with $\bar{R} = 0.94$.

Matzner *et al.* (1965) also obtained diabatic data for water at 7000 kPa. The tape-twist ratio used was 15.0. The outlet quality ranged up to 88%. Mass velocities ranged from 1260 to 4600 kg/m^2 s. Pressure drops ranged from 228 to 1150 kPa. The 27 data points were predicted with an average deviation of -9.9% with $\bar{R} = 0.90$. These data were consistently underpredicted. Rosuel & Sourioux (1961) presented diabatic pressure drop data for $y = 3$, 6, and 9 for water at one atmosphere. Outlet quality ranged up to 99%. Mass velocities ranged from 120 to 520 kg/m^2 s. The 13 data points were predicted with an average deviation of $\pm 18.1\%$ with $\bar{R} = 1.17$. The scatter in the data from these three investigations is shown on figure 6.

CONCLUSIONS

An experimental study has been conducted to determine the effects of twisted-tape swirl generators on adiabatic and diabatic two-phase pressure drop in vertical straight tubes. The following conclusions can be drawn from this investigation.

(1) The two-phase friction multiplier correlation [5] developed by Reddy *et al.* (1983) when used with [1], [5a], [5b], and [5c] predicts the empty tube adiabatic and diabatic two-phase pressure drop well for mass velocities greater than 475 kg/m^2 s. For lower mass velocities and the lower pressures, [6] was developed and gives better predictions than [5c].

(2) The two-phase pressure drop with twisted-tape swirl generators increases when the tape-twist ratio decreases. Depending on the flow conditions and tape-twist ratio, the swirl flow pressure drop can range from about 1.2 to 3.5 times the empty tube pressure drop.

(3) The adiabatic and diabatic two-phase pressure drop with twisted-tape swirl generators was successfully correlated by using the hydraulic diameter in [4], the Reddy *et al.* (1983) two-phase friction multiplier [5], and a friction factor [9] and [10] which were developed for single-phase swirl flow.

Two additional points should be made. First, the proposed correlation should be applicable to horizontal flows if the flow regime is annular. If the horizontal flow is stratified, then the applicability of the proposed correlation is unknown.

Second, for the data obtained with $G > 475$ kg/m^2 s, the empty tube and swirl flow predictions were better than the predictions for the data with $G \leq 475$ kg/m^2 s. This may be due to poor correlation of the empty tube data in this low mass velocity region. The relatively few adiabatic data obtained in this study at the low mass velocities and pressure are not sufficient to adequately establish a correlation for the two-phase friction multiplier. A study, such as was done by Reddy *et al.* (1983), should be performed so that an accurate two-phase friction multiplier (which is a function of pressure, quality, and mass velocity) can be developed for lower mass velocities and pressures. With a better empty tube prediction, the twisted-tape predictions at the lower mass velocities and pressures should be improved.

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NOTATION

- D tube diameter (m)
- f friction factor
- G mass velocity (kg/m^2 s)
- L test section length (m)
- P pressure (kPa)
- ΔP pressure drop (kPa)
- q'' heat flux (W/m^2)
- R ΔP calculated/ΔP experimental
- Re Reynolds number GD/μ_L
- x quality
- y tape-twist ratio (length for 180° twist/inside tube diameter)
- z distance along test section (m)

Greek letters

- ϵ homogeneous void fraction $(x/\rho_G)/(x/\rho_G + (1 - x)/\rho_L)$
- δ tape thickness (m)
- μ dynamic viscosity (N s/m^2)
- ρ density (kg/m^3)
- ϕ_{f0}^2 two-phase friction multiplier
- $\overline{\phi_{f0}^2}$ average two-phase friction multiplier

Subscripts

a acceleration
cr critical
f friction
G vapor
h hydrostatic, hydraulic
L liquid
s straight or empty tube
tot total
tt twisted tape

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