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# TWO-PHASE PRESSURE LOSSES

# SEVENTH QUARTERLY PROGRESS REPORT

E. JANSSEN
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U.S. ATOMIC ENERGY COMMISSION
CONTRACT AT(04-3)-189
PROJECT AGREEMENT 27



ATOMIC POWER EQUIPMENT DEPARTMENT
GENERAL ELECTRIC
SAN JOSE, CALIFORNIA

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August 12, 1963 - November 11, 1963

E. Janssen and J. A. Kervinen

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# TABLE OF CONTENTS

## SUMMARY

Pressure drop was measured along an annular channel with dimensions $D_1 = 0.375$ inch; $D_2 = 0.875$ inch, $L = 70$ inches. Flow was vertical and upward, and only the internal surface was heated. Subcooled conditions existed at the inlet, with two-phase conditions at the exit. Groups of three radial spacer pins on 18-inch centers along the channel, held the inner surface concentric with the outer surface. The single phase loss coefficient for each spacer group is $K_s = 0.21$. The single phase friction factor for the annular channel is given by

$$f = 0.16\ N_R^{-0.16}$$

The two-phase pressure drop increases as the quality increases for $\frac{G}{10^6} = 1.68$ and 1.0 lbs/hr ft$^2$, but decreases as the quality increases (up to qualities of 21 percent) for $\frac{G}{10^6} = 0.5$ ;b/hr ft$^2$. The effect of heat flux on the pressure drop is very slight over the range of fluxes tested $(0.55 \leq \frac{\varphi}{10^6} \leq 0.8)$. The two-phase pressure drop gradient in the same annulus, with no heat addition is qualitatively the same as for a 1/4-inch by 1-3/4 inches rectangular channel but is quantitatively greater than for the rectangular channel.

## INTRODUCTION

This is the seventh quarterly report of work performed under Contract AT(04-3)-189, Project Agreement No. 27, and covers the period of August 12 to November 11, 1963.

The objectives for the program are to:

1. Determine, for a range of geometries, pressures, flows, and qualities, the two-phase pressure loss in channels with relatively sudden changes in cross-sectional area.
2. Determine the two-phase pressure loss in straight channels, also for a range of pressures and flows, and particularly at high qualities.
3. Determine the two-phase pressure loss in straight channels with heat addition.

Two basic channel geometries are being used:

1. A rectangular (two-dimensional) channel.
2. A circular (three-dimensional) channel.

Channel orientations are with flow up, flow down, and flow horizontal.

During the quarter ending November 11, pressure drop was measured in a vertical annular channel ($D_1$ = 0.375 inch; $D_2$ = 0.875 inch) with boiling at the inner surface, under the conditions listed in Table 1. The essential results of these measurements are contained in this report.

## EQUIPMENT

The facility for two-phase pressure drop work has been described in an earlier report.[1] The test section for annular two-phase flow with heat addition (the "single rod" test section) is shown in Figure 1. This test section is fitted with a liner having the desired ID (0.875 inch ID in this case). There is a circumferential groove on the outside of the liner at each pressure tap position. Four static pressure ports were drilled through the liner wall at each groove, and the burrs were carefully removed after the drilling operation. When the liner is in place, each groove serves to connect its four pressure ports to an external pressure connection, in the same manner as a piezometer ring. The reference pressure tap is in the bottom plenum, on the same elevation as the test section inlet which is taken as the datum.

TABLE 1

PRESSURE DROP RUNS

Annular Channel ($D_1$ = 0.375 inch; $D_2$ = 0.875 inch; $L_{heated}$ = 70 inches); flow vertical upward; boiling on inner surface

| Pressure (psia) | Flow Rate (lb/hr ft$^2$) | Single Phase (No heat addition) T( F) | Heat Flux (Btu/hr ft$^2$) | Two Phase with Heat Addition* Exit Quality (percent) | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | 2.5 | 5 | 7.5 | 10 | 12.5 | 15 | 20 | 22 |
| 1000 | 0.5 | 75 | 0.55 | | | | | | x | x | |
| | | | 0.7 | | | | x | | x | $\underline{x}$ | $\underline{x}$ |
| | 1.0 | 75 | 0.55 | | | x | | | | | |
| | | | 0.7 | | x | x | $\underline{x}$ | $\underline{x}$ | | | |
| | 1.68 | | 0.55 | | | x | | | | | |
| | | | 0.7 | x | x | $\underline{x}$ | | | | | |
| | | | 0.8 | | x | | | | | | |
| | 2.0 | 75 | | | | | | | | | |
| | 3.0 | 75 | | | | | | | | | |
| 600 | 1.0 | | 0.7 | | x | x | x | x | | | |
| 1400 | 1.0 | | 0.7 | | x | x | x | x | | | |

*Runs for which the pressure drop in the unheated top three feet of the test section was measured, are denoted by underlining.

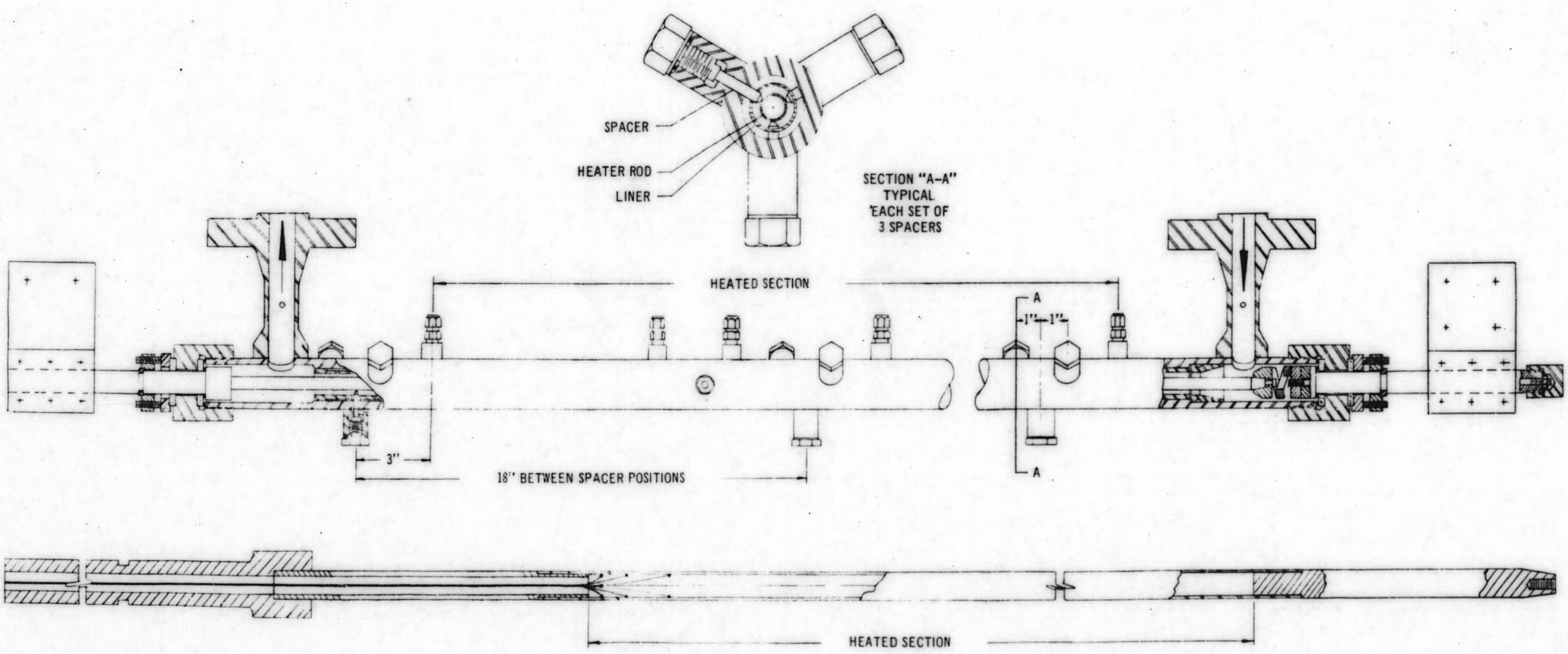


Figure 1. New Single Rod Test Section

The inner surface of the test channel is a rod with the desired OD (0.375 inch OD in this case). It is held concentric by means of spacer pins which also insulate it electrically from the rest of the test section. Each spacer pin is 0.226 inch OD. The pins are arranged in groups of three, the pins in each group being spaced 120 degrees circumferentially and one inch axially (see detail, Figure 1). The groups, in turn, are located on 18 inch centers along the axis of the channel. The rod may be heated electrically over 70 inches of its length.

The elevations of the various pressure taps, spacer pin groups, and ends of the heated length are listed below:

| Item | Elevation (inches) |
|---|---|
| Spacer Pin Group No. 7 | 116 |
| Pressure Tap No. 13 | 113 |
| Pressure Tap No. 12 | 104 |
| Spacer Pin Group No. 6 | 98 |
| Pressure Tap No. 11 | 95 |
| Pressure Tap No. 10 | 86 |
| Spacer Pin Group No. 5 | 80 |
| Pressure Tap No. 9 | 77 |
| End of Heated Length | 73.75 |
| Pressure Tap No. 8 | 71 |
| Pressure Tap No. 7 | 65 |
| Spacer Pin Group No. 4 | 62 |
| Pressure Tap No. 6 | 59 |
| Pressure Tap No. 5 | 53 |
| Pressure Tap No. 4 | 47 |
| Spacer Pin Group No. 3 | 44 |
| Pressure Tap No. 3 | 41 |
| Spacer Pin Group No. 2 | 26 |
| Pressure Tap No. 2 | 23 |
| Spacer Pin Group No. 1 | 8 |
| Beginning of Heated Length | 3.75 |
| Beginning of Annular Channel | 2 |
| Pressure Tap No. 1 (Reference Tap) | 0 |

The first several runs were made with an "after heater" in the top three feet of the test section. A minimum rate of steam is necessary at the exit of the test section to maintain system pressure, and the after heater permits operation in a lower quality range. The last few runs, however, were made with the annular channel extending to the top of the test section, the top three feet in this case being unheated. These runs are identified in Table 1.

## EQUATIONS

The single phase pressure loss (no heat addition; hence constant density) is made up of three parts:

$\Delta P_e$ Includes all entrance losses, up to the beginning of the heated lengths

$n\Delta P_s$ $\Delta P_s$ is the net loss across one spacer group, n is the number of spacer groups.

$\Delta P_f$ Channel friction loss.

The total pressure drop is then given by

$$\Delta P_t = \int_0^{\Delta z_e + \Delta z_n} w\, dz + \Delta P_e + n\Delta P_s + \Delta P_f \tag{1}$$

The net pressure drop, for both single and two-phase flow is defined as

$$\Delta P_n = \Delta P_t - \left\{\Delta P_e + \Delta z_e\, w_e\right\} \tag{2}$$

The entrance losses may be given by a single coefficient

$$\Delta P_e = K_e \frac{1}{w} \left\{\frac{(G/3600)^2}{2g}\right\} \tag{3}$$

Similarly, the single-phase spacer and channel friction losses may be given by

$$n\Delta P_s = nK_s \frac{1}{w} \left\{\frac{(G/3600)^2}{2g}\right\} \tag{4}$$

$$\Delta P_f = \frac{fL}{D_h} \frac{1}{w} \left\{\frac{(G/3600)^2}{2g}\right\} \tag{5}$$

where the friction factor f is a function of Reynolds number $N_R$.

$$N_R = \frac{GD_h}{\mu} \tag{6}$$

One of the early steps in reducing the data is to determine value of $\Delta P_t$, $\Delta P_n$, and $\Delta P_s$ for the single phase runs from plots of $\Delta P$ vs z. From these the coefficients $K_e$, $K_s$ and f may be determined.

Combining equation (2) and (3),

$$\Delta P_n = \Delta P_t - \left[ \frac{K_e}{w_e} \left\{ \frac{(G\ 3600)^2}{2g} \right\} + \frac{\Delta z_e\ w_e}{1728} \right] \tag{7}$$

Equation (7) applies to both the single and two-phase runs. Note that $\Delta P_t$ and G are measured directly, and $w_e$ may be determined from system pressure and inlet temperature.

## RESULTS AND DISCUSSION

A plot of $\Delta P$ vs z for a typical single phase run (Run No. 587) is shown in Figure 2. $K_e$ and $K_s$ were evaluated for all the single phase runs. There was apparently a flow effect at low Reynolds numbers but this effect was not well defined, and there was some indication that both $K_e$ and $K_s$ tended to be constant at higher Reynolds numbers. For the purposes of this report, $K_e$ was taken to be 1.42 for runs 584 to 611, and 1.20 for runs 612 to 620. The value for $K_s$ was taken to be 0.21, which is close to the average determined from the single phase plots, and agrees with the previously determined value (Reference 2 and 3).

The friction factor f was evaluated using Equations (1), (3), (4), and (5). The value of f was found to be in close agreement with the value determined in References 2 and 3, which is given by

$$f = 0.16\ N_R^{-0.16} \tag{8}$$

an expression used for commercial pipe.

The net pressure drop $\Delta P_n$ between the beginning of the heated length and pressure tap no. 9 was calculated for all the two-phase runs using Equation (7). $\Delta P_n$ for the 1000 psia runs is plotted versus exit quality in Figure 3 for $\frac{G}{10^6}$ = 0.5, 1.0, and 1.68 lb/hr ft$^2$, and $\frac{\phi}{10^6}$ = 0.55, 0.7, and 0.8 Btu/hr ft$^2$. Also superposed is the calculated net pressure drop for saturated liquid at 1000 psia, the same three flows, and $\phi = 0$. Figure 3 shows that:

1. The pressure drop increases as the quality increases, except at the lowest flow. This means that, for the two higher flows, the pressure loss increases more than the hydrostatic pressure difference decreases, with increasing quality. Measurements at the lowest flow show a reverse in this trend. It is interesting to note that as the exit quality approaches zero the drop approaches that calculated for saturated liquid.
2. The pressure drop increases as the flow increases, for a given value of exit quality and heat flux. This is to be expected.

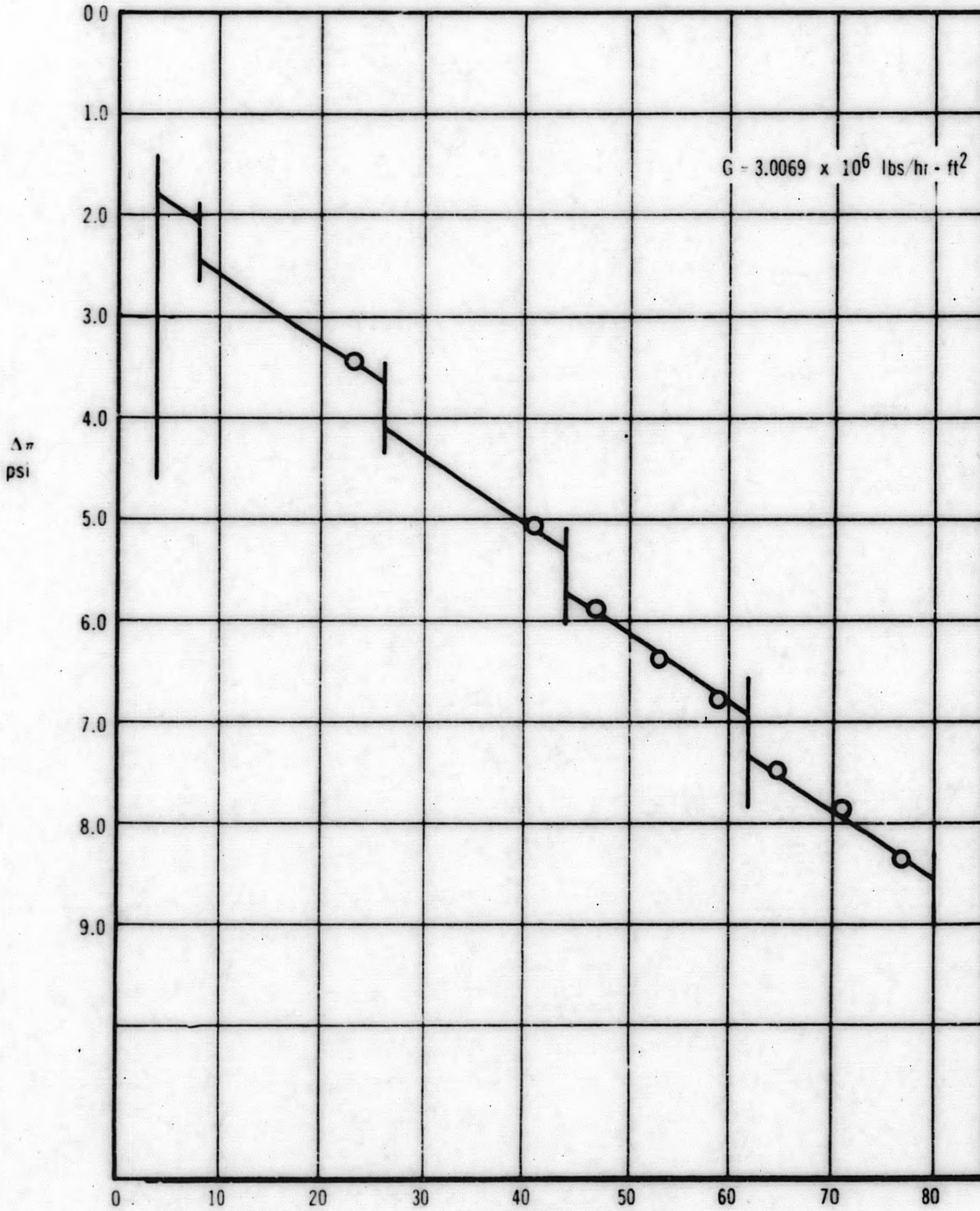


Figure 2. $\Delta\pi$ vs. z. for annular channel typical single-phase, no heat addition (Run No. 587, T = 70 F. $\frac{G}{10^6}$ = 3.0 lb/hr ft$^2$). Note: Hydrostatic term has been subtracted from measured pressure drop. For single phase. $\Delta\pi = \Delta P_{Loss}$.

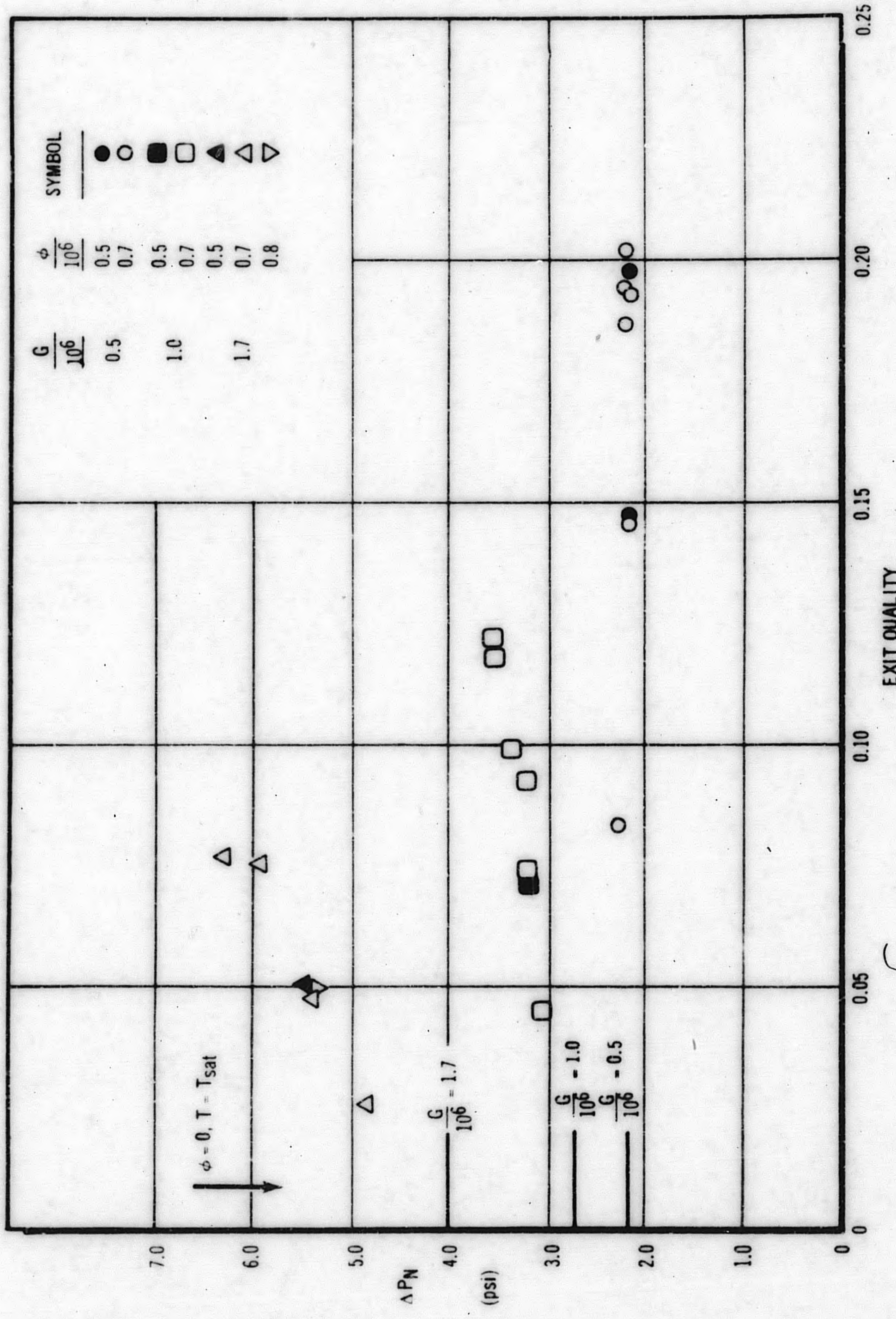


Figure 3. $\Delta P_N$ vs. $X_e$, for annular channel, flow up, 1000 psia, various flows and heat fluxes (note limiting values corresponding to $X = 0$, $\Phi = 0$, $T = T_{sat}$).

3. The effect of heat flux on the pressure drop is very slight over the range of fluxes tested.

A plot of $\Delta P$ vs z along the unheated annulus, for one of the few two-phase runs for which no afterheater was used (Run No. 617), is shown in Figure 4. No transition region is apparent at the inlet end of the unheated length. This is typical for all the runs with the unheated annulus. The pressure drop across spacer group numbers 5 and 6 and the straight channel pressure drop gradient may be determined from Figure 4. The two-phase pressure drop gradient* for the unheated annulus is plotted versus quality in Figure 5 for $\frac{G}{10^6}$ = 1.0 and 1.68 lb/hr ft$^2$. The best-fit curves of pressure drop gradient for the 1/4 inch by 1-3/4 inches channel of Reference 5 are superposed for comparison. The hydraulic diameters are nearly the same for the two channels.

| | $D_n$ (inches) |
|---|---|
| Annulus | 0.500 |
| 1/4 × 1-3/4 channel | 0.438 |

The friction factors, however, are quite different, the factor for the annulus being as much as 80 percent greater than the factor for the rectangular channel. It is not surprising that at the same flow and quality, the pressure drop gradient for the annulus is greater than the pressure drop gradient for the rectangular channel.

Footnote:

* The pressure drop across the spacer group has been subtracted, as explained above. The pressure drop gradient still includes both the channel friction and hydrostatic terms.

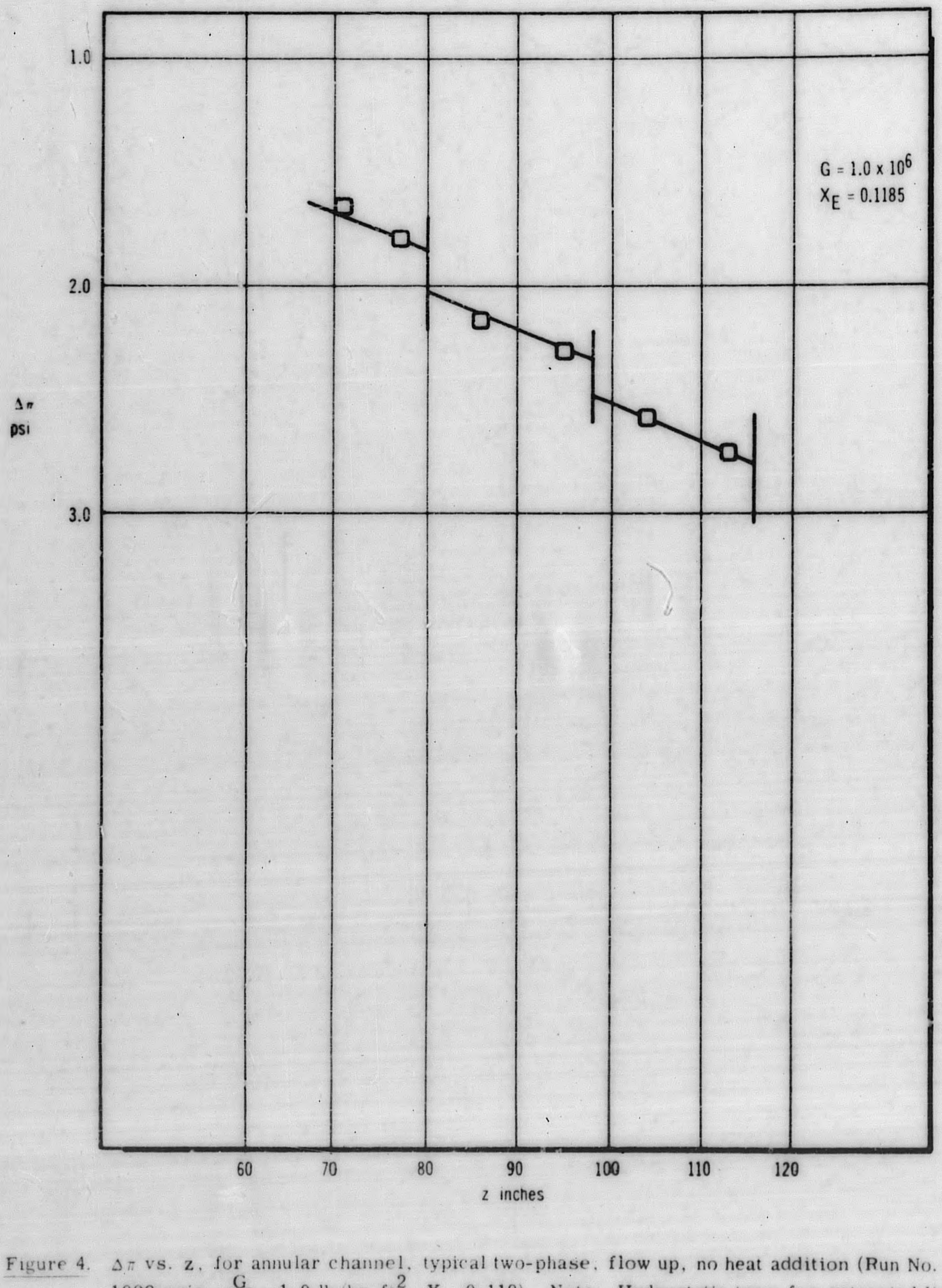


Figure 4. $\Delta\pi$ vs. z, for annular channel, typical two-phase, flow up, no heat addition (Run No. 617, 1000 psia, $\frac{G}{10^6}$ = 1.0 lb/hr ft$^2$, X = 0.119). Note: Hydrostatic term for saturated liquid has been subtracted from measured pressure drop. For two-phase, $\Delta\pi$ does not equal $\Delta P_{Loss}$.

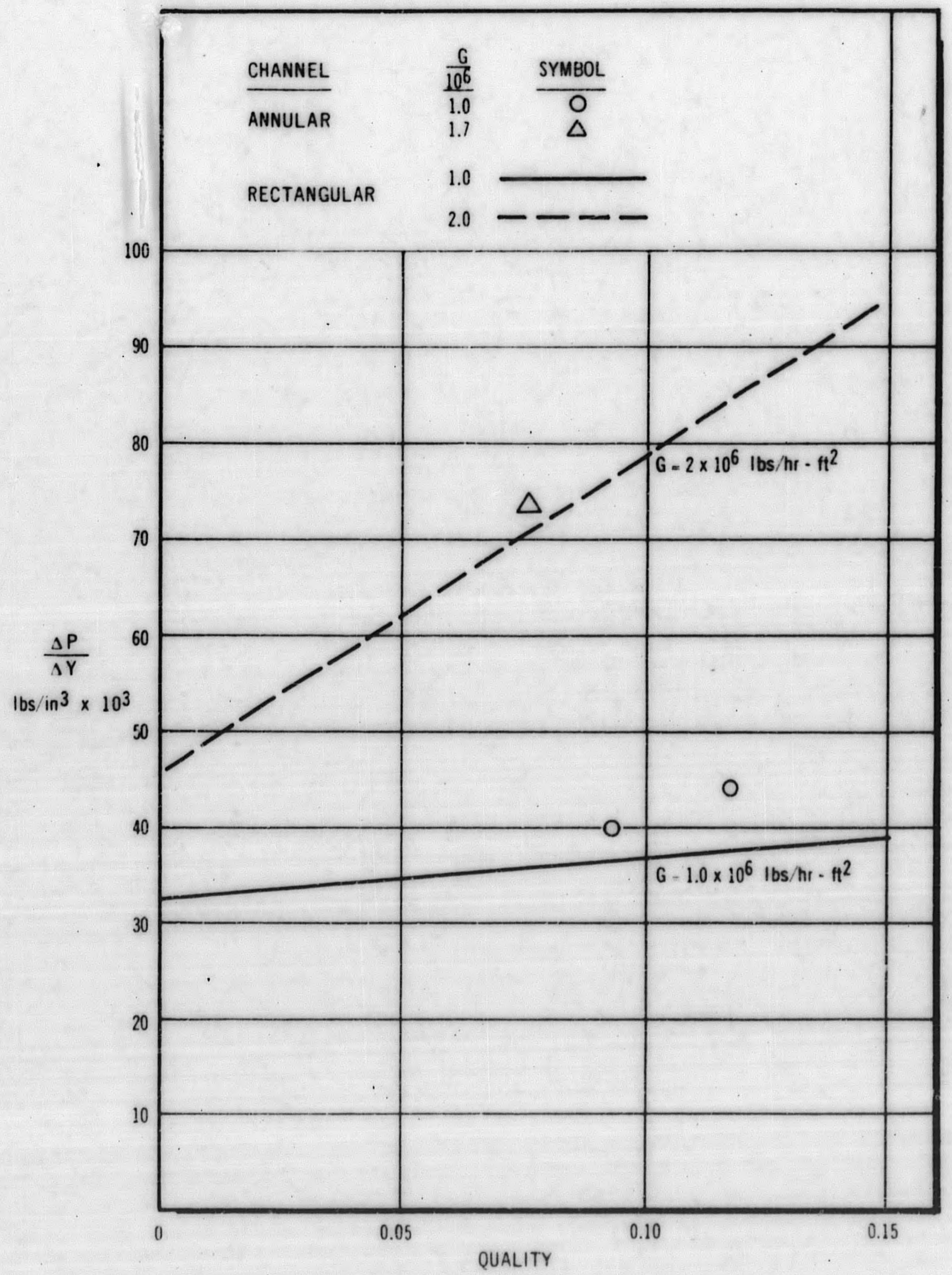


Figure 5. $\frac{\Delta P}{\Delta z}$ vs. X, for annular channel, flow up, two-phase, no heat addition, 1000 psia (best-fit curves for 1/4-inch by 1-3/4-inch rectangular channel superposed).

## NOTATION

| | |
|---|---|
| $D_h$ | hydraulic diameter, inches |
| f | pipe friction factor |
| g | 32.174 ft/sec$^2$ |
| G | mass velocity, lb/hr ft$^2$ |
| $K_e$ | loss coefficient, 1st pressure tap to beginning of heated length |
| $K_s$ | loss coefficient, one group of 3 spacer pins |
| L | hydraulic length, inches |
| n | number of spacer groups |
| $N_R$ | Reynolds number |
| $\Delta P_e$ | pressure loss, 1st pressure tap to beginning of heated length, psi |
| $\Delta P_f$ | friction pressure loss in channel, psi |
| $\Delta P_n$ | pressure drop, beginning of heated length to 9th pressure tap, psi |
| $\Delta P_s$ | pressure loss across one spacer group, psi |
| $\Delta P_t$ | total pressure drop, 1st pressure tap to 9th pressure tap, psi |
| w | density of fluid, lb/ft$^3$ |
| $w_e$ | density of liquid ahead of heated length, lb/ft$^3$ |
| $\Delta z_e$ | elevation difference, 1st pressure tap to beginning of heated length, inches |
| $\Delta z_n$ | elevation difference, beginning of heated length to 9th pressure tap, inches |
| $\mu$ | viscosity, lb/hr ft |
| $\Phi$ | heat flux, Btu/hr ft$^2$ |

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