MASTER

PRESSURE DROP OF MULTI-ROD ELEMENTS
WITH HELICAL SPRING SPACERS

By

E. P. Quinn

June 1961

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SUMMARY

The pressure drop of a new fuel element design concept of spacing rods by means of helical wire springs was investigated experimentally and analytically. Extensive single and two phase pressure drop data at 1,000 psia was obtained for one flow geometry and helical spring spacer. Test conditions ranged from 0.7 to 1.2×10^6 lb/hr ft² in mass velocity and from 0-15% in quality. The effect of the specific spring which was tested was to increase the overall pressure drop by 70%.

A general analytical model was developed to predict the pressure drop of an element with helical spring spacers when the pressure drop without springs is known. The accuracy of the model, compared to the experimental data, is better than $\pm 22\%$. The analytical model allows determination and evaluation of an optimum helical spring spacer design, so that pressure drop will not be a serious disadvantage.

INTRODUCTION

A new concept in fuel element design is under consideration for use in the Consumers' High Power Density reactor.

Figure 1 shows a prototype assembly of this kind. The outstanding feature of this design is the spacing of individual fuel rods by means of full length helical springs as shown in Figure 2. Several advantages of the design can be listed:

- a. Simplicity
- b. Elimination of intermediate conventional spacers
- c. High permissible manufacturing tolerances
- d. Reduced element fabrication costs
- e. Positive rod positioning (to oppose thermal bow, vibration, and fretting corrosion), and
- f. Possible increase of burnout heat flux as a result of fluid spin induced by the springs.

The disadvantage of this design is that the pressure drop of an optimum spring spacer design may exceed that of a conventional spacer design for a given element geometry. In this regard, the work described herein was initiated to provide a method of predicting the pressure drop of helical spring spaced fuel elements.

Pressure drop of a prototype element with spring spacers of arbitrary wire diameter and pitch was measured at 1,000 psia and various conditions of single and two-phase flow. The pressure drop of the same element without the springs was also measured for a basis of comparison. An independent analytical model was developed to predict the pressure drop of a helical spring spaced element of any geometry with any helical springs, if the pressure drop with the springs absent is known.

FIGURE 1 VIEW OF ELEMENT

EXPERIMENTAL RESULTS

Figure 2 displays the cross section of the VBWR size Consumers' element that was pressure drop tested in the heat transfer facility in Building G. As may be seen in Figure 1, adjacent springs were coiled in opposite directions. The 100 test series was obtained with the springs absent and the 200 test series included the springs. Tests were conducted at constant conditions of about 60°F (x01), 500°F (x02), 5% quality (x03), 10% quality (x04) and 15% quality (x05). The mass rates were nearly 8.7 (A), 12.2 (B), and 15.0 (C) lb/sec. Fifteen pounds per second corresponds to a mass velocity slightly above 1.2×10^6 lb/hr ft², which is the reactor design condition. Table 1 lists the values of actual test conditions.

TABLE 1
TEST CONDITIONS

Run	Pressure Psia	Temperature °F	Quality	Mass Rate lb/sec.
101A 201A	67 61	65 63		8.64 8.56
101B 201B	73 68	70 61		12.2 12.2
101C 201C	71 65	74 62		15.0 15.0
102A 202A	1000 1000	508 487		8.60 8.64
102B 202B	1000 1000	494 490		12.4 12.2
102C 202C	1000 1000	496 494		15.0 15.1
103A 203A	1000 1000		.054 .045	8.71 8.77
103B 203B	1000 1000		.060 .051	12.3 12.4
103C 203C	999 1003		.064 .058	15.1 15.1
104A 204A	1000 1000		.108 .105	8.71 8.70
104B 204B	1000 1000		.110 .108	12.3 12.4
104C 204C	1000 1003		.110 .108	15.1 15.1
105A 205A	1000 1000		.156 .160	8.8 8.7

Table 2 shows that the increase in pressure drop is of the order of 70%. This increase is valid only for the specific geometry and spring which was tested. It would be possible to diminish this pressure drop differential, if a knowledge of the relationship between it and the spring dimensions was available for design usage.

TABLE 2

Overall Non-Hydrostatic Pressure Drop*

Run	Inches Water @ 60F	% Difference (Pressure Drop Without Springs)
101A 201A	9.44 16.5	75
101B 201B	18.1 32.2	78
101C 201C	26.8 48.4	81
102A 202A	10.4 18.1	74
102B 202B	20.2 35.1	74
102C 202C	28.1 51.8	84
103A 203A	20.8 33.5	61
103B 203B	40.0 66.1	65
103C 203C	59.2 99.1	67
104A 204A	29.9 51.2	71
104B 204B	59.8 100	67
104C 204C	89.5 153	71
105A 205A	38.8 69.9	81

*NOTE: This comparison is approximate because conditions are not common. See Table 1.

The pressure drop of the continuous springs manifests itself as an increase in the pressure gradient of the element. The overall pressure drop of an element employing helical spring spacers may be computed, if the effect of the springs on the pressure gradient can be predicted. Figure 3 shows an elevation view of the test element with pressure tap locations. No entrance nozzle was employed. Pressure gradients were computed from static pressure measurements for the lower and upper sections of the bundle, because a conventional spacer was located somewhat near the center for both the 100 and 200 test series. The exact conventional spacer location was dictated by pressure tap considerations. (When spring spacers are used, the conventional spacer would be normally employed only to separate the outer rods from the channel.) The method of calculating the pressure gradients was to find the slope of the straight line that fits the data by numerical computation employing the method of "least squared error." The gradients are based on 4 lower taps and 5 upper taps for the 100 series and on 4 lower taps and 6 upper taps for the 200 series. The pressure taps were distributed over 8 to 16 inch lengths. Details are shown in the Appendix. The pressure gradients are tabulated in Table 3.

TABLE 3

PRESSURE GRADIENTS*

Run	Gradient in. H ₂ O 60F/in.		Ratio	
	Lower	Upper	Lower	Upper
101A 201A	.091 .241	.082 .243	2.64	2.96
101B 201B	.166 .464	.143 .471	2.80	3.30
101C 201C	.237 .705	.187 .670	2.98	3.58
102A 202A	.080 .241	.115 .208	3.02	1.81
102B 202B	.158 .419	.132 .466	2.65	3.53
102C 202C	.195 .728	.167 .694	3.73	4.15
103A 203A	.195 .457	.276 .490	2.34	1.76
103B 203B	.352 .825	.326 .899	2.35	2.76
103C 203C	.503 1.174	.497 1.319	2.34	2.65
104A 204A	.291 .681	.296 .708	2.34	2.39
104B 204B	.553 1.201	.482 1.358	2.18	2.82
104C 204C	.681 1.760	.660 1.974	2.58	2.99
105A 205A	.391 .950	.378 1.008	2.43	2.66
* NOTE: The gradient pairs are at different conditions (See Table 1)				

The table lists the ratio of the pressure gradient along the multi-rod assembly with spring spacers to that without spring spacers. This ratio is of the order of almost 3. It is well to note that the ratio of overall pressure drop with springs to that without springs is only about 1.7 as indicated by Table 2. The entrance, center spacer, and exit losses make up the difference.

FIGURE 2
FLOW AND SPRING GEOMETRY

FIGURE 3

ELEVATION VIEW OF ELEMENT

ANALYTICAL MODEL

The analytical model which is developed here is a simplified attempt to approximately predict the pressure gradient for helical spring spaced elements, when the gradient is known for the same element without springs.

On the following assumptions, the model is based.

- a. The conventional definition of hydraulic diameter is invalid for the overall element.
- b. The conventional definition is approximately correct for each appropriately sub-divided flow region and increases in accuracy as the number of regions is increased.
- c. Further uncertainty of friction factor and hydraulic diameter may be taken into account by assuming the relationship $f = c/Re^b$, where "c" and "b" are open constants to be determined, for a given geometry, from single phase data in absence of any spacers (The model is somewhat sensitive to the value of "b").
- d. An average velocity can be defined for each flow region.
- e. The fluid shear at interfaces between regions does not appreciably affect the hydraulic diameters, because the average velocities from region to region are not radically different.
- f. The Martinelli pressure gradient factor for a two-phase flow condition is approximately the same for all regions.
- g. The void fraction - quality measurements of Larson (1) are applicable.
- h. The two phase geometric loss coefficient for slip flow given by Tippetts (2) is applicable.
- i. The contraction-expansion pressure loss for a helical spring occurs by the same mechanism as for an orifice.

Assumptions "a" and "b" are supported by Levy (3) and others. Diskind (4) presents an analysis, which defines an effective hydraulic diameter for an eccentric annulus. "c" is acceptable in view of the experimental work by Polomik (5), (6) and Janssen (7). Figure 5 shows this relationship applied to the present flow geometry. It is nearly the same as others currently employed. Assumption "f" is unfortunately necessary as the difficulties involved in writing the fourth equation to determine the velocities and qualities in each of two regions are adverse. This assumption makes the two phase analysis an estimate. Assumption "h" is supported by Polomik (6) and partially supported by Janssen (7). Assumption "i" is open to question. It can be pointed out, however, that Polomik and Janssen experimentally find that the contraction-expansion loss for tie plates and conventional spacers occurs by nearly the same mechanism as for an orifice.

Full details of the model are given in the appendix. The general approach and results will be outlined here. Figure 4 shows the flow cross-section divided into two regions: 1 and 2 for springs absent and 3 and 4 for springs present. Writing the momentum and continuity equations in conformance with the usual definitions and the foregoing assumptions, there obtains:

Single Phase

$$-2g \frac{\rho_R}{\rho} \frac{1}{\rho_R} \frac{dP}{dy} \bigg|_{wo} = \frac{f_1}{D_1} v_1^2 = \frac{f_2}{D_2} v_2^2$$

$$-2g \frac{\rho_R}{\rho} \frac{1}{\rho_R} \frac{dP}{dy} \bigg|_w = \left(\frac{f_3}{D_3} + k \right) v_3^2 = \frac{f_4}{D_4} v_4^2$$

$$v_1 A_1 + v_2 A_2 = \bar{V} A_T$$

$$v_3 A_3 + v_4 A_4 = \bar{V} A_T \quad (1)$$

Two Phase

$$-2g \frac{\rho_R}{\rho} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_{w_0} = \phi \frac{f_1}{D_1} V_1^2 = \phi \frac{f_2}{D_2} V_2^2$$

$$-2g \frac{\rho_R}{\rho} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_w = \left(\phi \frac{f_3}{D_3} + K_F k \right) V_3^2 = \phi \frac{f_4}{D_4} V_4^2$$

$$V_1 A_1 + V_2 A_2 = \bar{V} A_T$$

$$V_3 A_3 + V_4 A_4 = \bar{V} A_T \quad (2)$$

In terms of nomenclature defined in the Appendix, these become:

Single Phase

$$-2g \frac{\rho_R}{\rho} \frac{\gamma^{-b}}{C \bar{V}^{2-b}} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_{w_0} = \frac{\gamma^{2-b}}{D_1^{1+b}} = \frac{\sum^{2-b}}{D_2^{1+b}}$$

$$-2g \frac{\rho_R}{\rho} \frac{\gamma^{-b}}{C \bar{V}^{2-b}} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_w = \left[1 + \frac{k D_3}{C} \left(\frac{D_3 \bar{V}}{\gamma} \right)^b \beta^b \right] \frac{\gamma^{2-b}}{D_3^{1+b}} = \frac{\gamma^{2-b}}{D_4^{1+b}}$$

$$\alpha \gamma + (1-\alpha) \sum = 1$$

$$\alpha \beta + (1-\alpha) \gamma = 1 \quad (3)$$

Two Phase

$$-2g \frac{\rho_R}{\rho} \frac{\nu^{-b}}{c \bar{\nu}^{2-b}} \frac{1}{\phi} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_{wo} = \frac{\gamma^{2-b}}{D_1^{1/b}} \cdot \frac{\xi^{2-b}}{D_2^{1/b}}$$

$$-2g \frac{\rho_R}{\rho} \frac{\nu^{-b}}{c \bar{\nu}^{2-b}} \frac{1}{\phi} \frac{1}{\rho_R} \frac{dP}{dy} \Big|_w = \left[1 + \frac{K_F}{\phi} \frac{kD_3}{c} \left(\frac{D_3 \bar{\nu}}{\nu} \right)^b \beta^b \right] \frac{\beta^{2-b}}{D_3^{1/b}} = \frac{\gamma^{2-b}}{D_4^{1/b}}$$

$$\alpha \gamma + (1-\alpha) \xi = 1$$

$$\alpha \beta + (1-\alpha) \gamma = 1 \quad (4)$$

In each case, three equations are available to determine the gradient, and γ and ξ or the gradient and β and γ . For the geometry that was tested (Figure 4), values of c and b were found to be 0.116 and 0.11 from measured, single phase, no-spring, pressure gradients. (See Figure 5).

Defining

$$\Gamma = \frac{K_F}{\phi} \frac{kD_3}{c} \left(\frac{D_3 \bar{\nu}}{\nu} \right)^b, \text{ where} \quad (5)$$

$\frac{K_F}{\phi} \equiv 1$ for the single phase case or is given by figure 6 for the two phase case and

$$R_{uc} = \frac{\frac{1}{\rho_R} \frac{dP}{dy}|_w}{\frac{1}{\rho_R} \frac{dP}{dy}|_{w_0}} = \frac{\rho_w}{\rho_{w_0}} \left(\frac{\bar{V}_w}{\bar{V}_{w_0}} \right)^{2-b} \left(\frac{\bar{V}_{w_0}}{\bar{V}_w} \right)^{-b} \frac{\phi_w}{\phi_{w_0}} R_c, \quad (6)$$

where the factors of R_c are unity when conditions are common to the with spring and without spring gradients.

It remains to define "k". There is some interest in the possibility of the helical springs introducing spin to the flow between fuel rods. An examination of the magnitudes of pressure drop involved in initiating spin with an angular velocity inversely proportional to spring pitch and the pressure drop involved in successive contractions and expansions for zero spin, indicates that the pressure drop for any reasonable spring pitch is far greater for the former than for the latter. By the Principle of Least Action, the fluid will not spin. Hence, the quantity, "k", the geometric loss per unit length coefficient, is related to the contraction -- expansion mechanism such as that which occurs in orifices or discontinuities in channel cross-sections. An incremental area analysis shown in the Appendix and based on assumption "i", gives the following result.

$$k = \left(\frac{\pi \bar{D}}{a} \right)^2 \left[1 + \left(\frac{\pi \bar{D} d}{a} \right) \frac{d^2}{p} \right], \text{ valid for}$$

$$0 \leq \frac{\pi \bar{D} d}{a} < .3. \quad (7)$$

For a given geometry, freedom in spring design is allowed for d , the wire diameter, and p , the pitch. Various reasonable values of the two free dimensions can

drastically affect the value of k , which is a minimum in an optimum design.

With the analytical method just presented, it remains only to specify the geometry to obtain $R_c(\Gamma)$. All effects of flow conditions enter only through Γ . Clearly R_c is a minimum for good design. Figure 7, based on the geometry tested, shows that minimum R_c is obtained for minimum Γ and hence for minimum k . The figure indicates that $R \neq 1$ when $\Gamma = 0$ (apparently no springs). This is a consequence of the fact that $D_3 \neq D_1$, since the friction effect (10%) of the presence of the springs must be included. The "friction effect" is reduced with reduction in wire diameter and increase in pitch. Again, figure 7 is strictly valid only for the specific geometry and spring, which was tested. It indicates the variation of R_c with flow conditions only and can not be used for geometries different than that shown in figure 4. However, the spring "friction effect" is small, i.e., spring wetted perimeter $\ll$ rod wetted perimeter, hence an estimate of R_c for the geometry tested and any spring whatsoever may be attained from figure 7 for the isenthalpic case. The method employed to obtain $R_c(\Gamma)$ is detailed in the appendix.

The in-core pressure drop must account for variable conditions and fluid accelerations. Hence the overall Martinelli (8) friction and acceleration multipliers should be employed instead of the local friction multiplier used here.

<u>REGION</u>		<u>A</u>	<u>D</u>
1	NO SPRINGS	4.39	.96
2	NO SPRINGS	2.24	.40
3	WITH SPRINGS	4.36	.84
4	WITH SPRINGS	2.24	.40
		in. ²	in.

FIGURE 4
DIVISION OF FLOW GEOMETRY

FIGURE 5

FRICTION FACTOR RELATIONS

FIGURE 6

RATIO OF GEOMETRIC AND FRICTION COEFFICIENTS — QUALITY

FIGURE 7

PRESSURE GRADIENT RATIO — HELICAL SPRING PARAMETER

DISCUSSION

The prediction of the analytical model is compared to the experimental results of Table 3 in Table 4. Figure 7 is applied to the conditions of the experimental data by means of the computations of Table 6 (Appendix), which includes adjustment for differences in experimental conditions, to yield the predicted values of the gradient ratio shown. It is apparent from the table that the measured gradient ratios contain applicable error since the lower and upper ratios are measurements of almost the same physical magnitude. This error is the result of pressure measurements over insufficient lengths for establishing accurate gradients.

TABLE 4

COMPARISON OF ANALYTICAL MODEL AND EXPERIMENTAL RESULTS						
Run	R_{uc} (Model)	Measured Ratio		% Deviation from Model		% Inconsistency in Measurements
		Lower	Upper	Lower	Upper	
X01A	3.10	2.64	2.96	-15	- 5	± 10
X01B	3.20	2.80	3.30	-12	± 3	± 15
X01C	3.24	2.98	3.58	- 8	±10	± 18
X02A	3.35	3.02	1.81	-10	-46	- 36
X02B	3.38	2.65	3.53	-22	± 4	± 26
X02C	3.56	3.73	4.15	± 5	±17	± 12
X03A	2.43	2.34	1.76	- 4	-28	- 24
X03B	2.41	2.35	2.76	- 2	±15	± 17
X03C	2.49	2.34	2.65	- 6	± 6	± 12
X04A	2.38	2.34	2.39	- 2	0	± 2
X04B	2.44	2.18	2.82	-11	±16	± 27
X04C	2.45	2.58	2.99	± 5	±22	± 17
X05A	2.46	2.43	2.66	- 1	± 8	± 9

Note that except for two of the 26 comparisons, the experimental values fall within $\pm 22\%$ of the analytical model prediction. Furthermore, the last column of Table 4 shows that in eight out of 13 cases the data is more inconsistent than the data-model deviation. Hence we conclude that the model is remarkably accurate in predicting the ratio of with spring to without spring pressure gradients. The experimental ratios are markedly lower for two-phase flow than for single phase flow. The model predicts and explains this reduction by means of Figure 6 and Equation 5. This simplified model cannot be expected to be this accurate for the general geometry, but it probably will be sufficiently accurate for design purposes.

The power of the analytical model lies in its ability to permit optimization and evaluation of the helical spring spacer concept of element design. The proper use of the model in optimization of spring design should be along the line of the following. For a given multi-rod geometry, evaluate:

1. The minimum required radial spring stiffness,
2. The relationship between radial spring stiffness and the wire diameter and pitch of the spring,
3. Solve (2) for the pitch and substitute into equation 7, and
4. Find the value of "d" for the derivative of "k" with respect to "d" to vanish.

The spring dimensions for minimum "k" and minimum R_c are fixed by the above procedure without recourse to the expression for R_c . The overall pressure drop with the optimum spring spacers may be computed by the method given in the Appendix. R_c is evaluated for this spring, the flow geometry, the flow rate and exit quality. The overall pressure drop for springs present and absent are obtained from R_c , the pressure gradient without springs and the entrance,

conventional spacer, and exit pressure drops.

At this point, valid judgment may be exercised as to the adoption of the helical spring spacer concept of element design.

For example, the 20 gage wire x 7/8 inch pitch spring that was used in the geometry that was tested indicated the following for reactor design mass velocity and 10% quality.

$$\frac{\Delta P_{Tw}}{\Delta P_{Two}} = 1.71 \text{ (measured)}$$

$$R_{ac} = 2.45 \text{ (from model)}$$

$$k = 0.174 \text{ (from model)}$$

Now it is clear that the above spring far exceeded minimum radial stiffness requirements. Arbitrarily, another spring can be selected for demonstration purposes.

This spring is certainly not the optimum spring. Consider a spring of 24 gage wire and 2 inch pitch. Ignoring "friction effects", the following values are obtained in the Appendix for the geometry tested and the above reactor conditions.

$$\frac{\Delta P_{Tw}}{\Delta P_{Two}} \approx 1.21$$

$$R_{uc} \approx 1.43$$

$$k = 0.032$$

In terms of the advantages listed in the introduction, a pressure drop increase of the order of only 21% may be acceptable. Again, this is not the optimum spring for this geometry.

CONCLUSIONS

1. Helical spring spacer design may be optimized so that no serious disadvantage is incurred from the standpoint of excessive pressure drop.
2. A general analytical model has been developed to predict the pressure gradient of helical spring spacers for any spring and any flow geometry under single or two-phase conditions.
3. The analytical model compares extremely well with extensive pressure gradient measurements of a Consumer's element employing one particular spring.
4. A method of analytical spring spacer optimization and evaluation in terms of pressure drop is explicitly presented.
5. It appears that swirl effects which may be introduced to the flow by helical springs in a multi-rod element are not pronounced.

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NOMENCLATURE

- A Area of region or sector of flow unit
- a Area of flow unit
- b Exponent
- c Coefficient
- D Hydraulic diameter or spring diameter
- d Wire diameter
- f Weisbach friction factor
- g Acceleration of gravity
- K_F Two phase geometric pressure loss multiplier for slip flow
- K_L Orifice pressure loss coefficient
- k Geometric pressure loss per unit length coefficient
- m Parameter
- P Static pressure
- p Spring pitch
- R Ratio of pressure gradients
- Re Reynolds' number
- V Equivalent single phase axial velocity
- x Quality
- y Axial coordinate
- α Area ratio or void fraction
- β Velocity ratio, $V_3/\bar{V}$
- Γ Parameter
- γ Velocity ratio, $V_4/\bar{V}$
- η Velocity ratio, $V_1/\bar{V}$
- ν Kinematic viscosity
- ξ Velocity ratio, $V_2/\bar{V}$
- ρ Density
- ϕ Local Martinelli two phase pressure gradient multiplier

Subscripts

c	Common conditions or contraction
f	Friction or liquid
g	Vapor
m	Manometer
R	At 60° F
T	Total or overall
tp	Tie plates, etc.
uc	Uncommon conditions
w	With helical springs
wo	Without helical springs
($\bar{\quad}$)	Average

Method of Data Reduction

Pressure gradients were computed from the following equations.

$$-\frac{1}{\rho_R} \frac{dP}{dy} = \frac{1}{\rho_R} \left. \frac{dP}{dy} \right|_m + 1 - \frac{\bar{P}}{\rho_R}$$

$$\bar{P} = P_f - (P_f - P_g) \alpha \quad (8)$$

Here α was evaluated from quality by means of data of reference 1.

$\left. \frac{dP}{dy} \right|_m$ was evaluated from static pressure measurements by the method of "least squared error."

$$\left. \frac{dP}{dy} \right|_m = \frac{\sum P_n y_n - 1/n \sum P_n \sum y_n}{\sum y_n^2 - 1/n \sum y_n \sum y_n} \quad (9)$$

Evaluation of Isenthalpic $R_c(r)$

Equations 3, 4, 5 and 6 give:

$$R_c = \left(\frac{D_2}{D_4} \right)^{1/b} \left(\frac{\gamma}{\xi} \right)^{2-b} = \left(\frac{\gamma}{\xi} \right)^{2-b} \quad (10)$$

since $D_2 = D_4$

$$\gamma = (1 + r \beta^b)^{\frac{1}{2-b}} \cdot \left(\frac{D_2}{D_3} \right)^{\frac{1/b}{2-b}} \beta \quad (11)$$

$$\xi = \left(\frac{D_2}{D_1} \right)^{\frac{1/b}{2-b}} \eta \quad (12)$$

$$\beta = \frac{1}{\alpha} [1 - (1-\alpha)\gamma] \quad (13)$$

$$\eta = \frac{1}{\alpha} [1 - (1-\alpha)\xi] \quad (14)$$

Combining 11 with 13 and 12 with 14 yields:

$$\frac{1}{\gamma} = 1 - \alpha(1-m) \quad (15)$$

$$\Gamma = (1 - \alpha + \alpha_m)^b \left[\frac{1}{m^2} \left(\frac{D_3}{D_2} \right)^{1/b} - \frac{1}{m^6} \right] \quad (16)$$

$$\frac{1}{\xi} = 1 - \alpha \left[1 - \left(\frac{D_1}{D_2} \right)^{\frac{1+b}{2-b}} \right] \quad (17)$$

where 15 and 16 are parametric equations for $\gamma(r)$.

The solution is obtained by combining equations 10, 15 and 17.

$$R_c = \left[\frac{1 + \frac{\alpha}{1-\alpha} \left(\frac{D_1}{D_2} \right)^{\frac{1+b}{2-b}}}{1 + \frac{\alpha}{1-\alpha} m} \right]^{2-b} \quad (18)$$

By inserting various values for m in equations 16 and 18, $R_c(\Gamma)$ is obtained in the form of Figure 7, once b and c are known. It is seen that for no springs

$$m = \left(\frac{D_3}{D_2} \right)^{\frac{1+b}{2-b}} \quad \text{for } \Gamma = 0.$$

And hence $R_c = 1$, since $D_1 = D_3$ in this case.

Determination of "b" and "c"

The friction factor was assumed to be of the form

$$f = c / Re^b \quad (19)$$

for the range of Reynolds' number in question. Values of "b" and "c" are determined from the single phase experimental data in the absence of springs.

Re-arranging equations 3 and 17 gives the following:

$$\frac{-\frac{1}{\rho_R} \frac{dP}{dy} / w_o}{\frac{\rho}{\rho_R} \frac{1}{D_1} \frac{V^2}{2g}} = \frac{c \left(\frac{D_1 \bar{V}}{\nu} \right)^b}{\left[\alpha + (1 - \alpha) \left(\frac{D_2}{D_1} \right)^{\frac{1+b}{2-b}} \right]^{2-b}} \quad (20)$$

The left side and $\frac{D_1 \bar{V}}{\nu}$ of the right side were measured. A plot of these groups for runs 101 A-C and 102 A-C from Table 5 is shown in Figure 8. The values for b and c were obtained from the line shown as 0.11 and 0.116 respectively.

TABLE 5

Determination of Values for "b" and "c"

<u>Run</u>	$\frac{-\frac{1}{\rho_R} \frac{dP}{dy} / \frac{\rho}{A_R} \frac{1}{D_1} \frac{\bar{V}^2}{2g}}{w_0}$			$\left(\frac{D_1 \bar{V}}{\nu} \right)$
	<u>Lower</u>		<u>Upper</u>	
101 A	.0514		.0462	2.1×10^4
101 B	.0466		.0402	3.2×10^4
101 C	.0443		.0348	4.2×10^4
102 A	.0331		.0486	2.1×10^5
102 B	.0362		.0305	3.0×10^5
102 C	.0296		.0254	3.6×10^5

FIGURE 8

DETERMINING CONSTANTS OF
FRICTION FACTOR RELATIONSHIP

$$\frac{-\frac{1}{\rho_R} \frac{dP}{dy} \Big|_{w=0}}{\frac{\rho}{\rho_R} \frac{1}{D_1} \frac{\bar{V}^2}{2g}}$$

Evaluation of "k"

Every cross-section of unit flow area between any four rods in Region 3 (Figure 4) is disturbed by the elliptical cross-section of the spring. Consider an incremental sector of flow area, A , which includes only the central portion of the ellipse, so that the portion is rectangular. Flow in this sector passes around the geometric disturbance every pitch of length. Cross flow between sectors is insignificant; because, when it is not, transverse velocities must be of the order of $\bar{D} V/p$. This value is very large and involves a very large pressure drop for any plausible design. Assumption "i" (analytical model section) implies that the overall geometric losses are equal to the orifice type losses of any incremental sector. Ignoring further differences between the present "continuous orifice" and a series of orifices, one writes the orifice loss coefficient from reference 2.

$$K_L = \left[\frac{1}{C_c \left(\frac{A - \Delta A}{A} \right)} - 1 \right]^2 \quad (21)$$

The number of orifice losses per unit length is:

$$k = \frac{1}{p} \left[\frac{1}{C_c} + \frac{\Delta A}{A} - 1 \right]^2 \quad (22)$$

Regardless of the size of A , for A sufficiently small

$$\frac{\Delta A}{A} = \frac{\pi \bar{D} d}{a} \quad (23)$$

Within 5%, $1/C_c$ may be represented by:

$$\frac{1}{C_c (1 - \frac{\Delta A}{A})} = e^{\frac{\Delta A}{A}}$$

for

$$0 \leq \frac{\Delta A}{A} < .3 \quad (24)$$

Substituting 24 into 22 and neglecting higher terms in $\Delta A/A$ gives

$$k = \frac{1}{p} \left(\frac{\Delta A}{A} \right)^2 \left(1 + \frac{\Delta A}{A} \right) \quad (25)$$

Combining 23 and 25 gives equation 7 of the report body.

Computation of Γ

Knowledge of the value of Γ (Equation 5) is required to obtain a specific value of R_c . Γ is obtained from flow conditions, a plot like Figure 8, Figure 6 and Equation 7.

Computation of Model Predictions

Table 6 shows the computation of the model predictions for the experimental values of pressure gradient ratios where conditions are not common. Values of the various quantities are taken from Table 1 and Figures 4 and 7. Model results are compared to experimental results in Table 4 (Discussion section).

TABLE 6

Computation of Model Predictions

Run	$(D_3 \bar{V} / \nu)_w^{b*}$	$(K_F / \rho)_w^*$	Γ^{**}	R_c^{***}	R_{uc}/R_c^{****}	R_{uc}
X01A	2.94	1.0	3.70	3.10	1.00	3.10
X01B	3.04	1.0	3.83	3.14	1.02	3.20
X01C	3.11	1.0	3.92	3.18	1.02	3.24
X02A	3.78	1.0	4.76	3.43	.977	3.35
X02B	3.92	1.0	4.94	3.48	.972	3.38
X02C	4.02	1.0	5.07	3.53	1.01	3.56
X03A	3.96	.54	2.69	2.68	.906	2.43
X03B	4.10	.52	2.68	2.68	.900	2.41
X03C	4.19	.50	2.64	2.66	.937	2.49
X04A	3.96	.43	2.14	2.43	.977	2.38
X04B	4.10	.43	2.22	2.47	.985	2.44
X04C	4.19	.43	2.27	2.49	.985	2.45
X05A	3.96	.45	2.24	2.48	.990	2.46

* From Table 1 and Figures 4 and 6

** From Equation (7), $k = 0.174$, $kD_3/c = 1.26$.

*** From Figure 7

**** From Equation 6

Estimate of Pressure Drop With a Different Spring

The following computation is an estimate (less than 10% high in this case), since Figure 7 will not be recomputed to account for the change introduced in Equation 16 resulting from the alteration of D_3 . The geometry remains the same and the spring is chosen as $d = 0.023$ " (24 gage), $p = 2$ ", ($\bar{D} = 0.464$ ").

From Equation 7, $k = 0.032$. $R_c = 1.43$ is obtained from Figure 7 for f computed with this value of k and the conditions of run 104C. The new overall pressure drop ratio is obtained from combining two relations like 25 for the new spring and the spring that was tested.

$$\frac{\Delta P_{T/w}}{\Delta P_{T/w_0}} \approx 1 + \frac{\Delta P_f}{\Delta P_f + \Delta P_{tp}} (R_c - 1) \quad (26)$$

The value of $\Delta P_{T/w} / \Delta P_{T/w_0}$ for the new spring is obtained as 1.21.