EVALUATION OF SENSORS
FOR SIMULATOR ANNULUS
AND RELATED MEASUREMENTS
IN THE PLENUM-FILL EXPERIMENT
SECTION I - PRELIMINARY EVALUATION
(7-1-74 TO 8-20-74)

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INTRODUCTION

This report constitutes the first section of what is expected to be a series of reports on the status and findings of a program of testing, evaluation, and - where necessary - modification of special instrumentation. This instrumentation has been obtained for the determination of liquid level, flow direction, phase distribution, and void fraction in the plenum, annulus, and return legs for each of the various conformations of the reactor simulator vessels employed in PFE.

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SUMMARY

Inspection and static testing have been performed on samples of three types of sensors and their ancillary instrumentation to determine general operability, range, output, and mechanical compatibility with the intended applications, and preliminary findings are reported herein.

Dynamic testing, using air and water flows to simulate two-phase steam/water conditions, will be initiated in early September. An onsite-fabricated test chamber is now on hand and is being installed, plumbed and instrumented at the test location (Building 314). This chamber will be used to test the momentum vector and hot wire instruments for transient response and for their ability to survive mechanical shock, at ambient temperatures and at low pressures. A facility for testing the float/switch and heated-thermocouple level sensors is being fabricated offsite, and will be available in October. Following the preliminary air/water testing at this site, high pressure controlled-quality steam/water testing in the Westinghouse Canada test facilities at Hamilton (Ontario) or in the Combustion Engineering Company facility at Windsor (Connecticut) is under consideration.

This report presents the results obtained to date from the "bench" tests on those sensors now on hand, with comments and suggestions relative to applicability, necessary changes, and proposed modifications. The tested instruments include:

1. Float/switch liquid-level probe (PEECO)
2. Momentum vector (drag rod) flow direction probe (PEECO)
3. Discrete (single element) hot-wire phase sensor (Datametrics)

The multi-element hot-wire "regime" sensors (Datametrics) have been inspected for physical condition, but have not been tested. They are expected to be essentially identical to the single-element probes in their response.

The heated-thermocouple liquid-level assemblies (ARI) and the void-fraction gamma absorptiometer (DD Electronics) are not yet available for testing.

FLOAT/SWITCH LIQUID-LEVEL SENSOR (PEECO)

This device consists of a vertical stainless steel tube containing an array of magnetically operated reed switches which connect a common lead to the junctions of 100 equal (100 ohm) series resistors, constituting a 100-step potentiometer. The magnetic reed switches are spaced at 1/2-inch intervals, and are actuated by a magnet within a free-moving toroidal float which encircles the tube. As the float moves up or down from one switch to the next, the next switch closes before the first switch opens, "make before break," to prevent the possibility of an open-circuit "wiper."

During the interval in which two adjacent switches are closed, the series resistor string is reduced from 100 units (10 Kohm) to 99 units (9.9 Kohm). For a constant applied voltage, this produces a current increase during the "overlap" interval, resulting in a variation from linearity dependent upon the float position. For example, for 10.00 volts applied voltage across the string (as used in our tests), the voltage on the "wiper" is $10 (n/100)$ when one switch is closed, and $10 (n/99)$ when two switches are closed. This produces a marked variation in the position of the major voltage step relative to the equal-spaced switch positions.

For example:

Switches			Output Voltage						
			E	n = 10		n = 50		n = 90	
S_{n-1}	S_n	S_{n+1}		E	ΔE	E	ΔE	E	ΔE
x			$10(n-1)/100$	0.900	--	4.900	--	8.900	--
x	x		$10(n-1)/99$	0.909	0.009	4.949	0.049	8.990	<u>0.090</u>
	x		$10\ n/100$	1.000	<u>0.091</u>	5.000	0.051	9.000	0.010
	x	x	$10\ n/99$	1.010	0.010	5.051	0.051	9.091	<u>0.091</u>
		x	$10(n+1)/100$	1.100	0.090	5.100	0.049	9.100	0.009

This effect, while quite noticeable, contributes a maximum deviation from linearity of only about 1/4-inch, if the system is calibrated to the highest major-step position, and it can be eliminated by employing a constant current supply.

One sample of these probes, supplied by Power Engineering and Equipment Company, Inc., was tested by measuring the float position and readout voltage at each observable voltage step, indicating a switching action (Figures 1 and 2). The measured voltages were found to conform very closely with the calculated values (proving switch operation and resistor precision), and the measured intervals were found to conform within about $\pm 1/8$ -inch of the nominal intervals for "make" and "break," for float travel in either direction. There was, however, a variable offset in switch operation in the two opposite directions. For equal output voltages, indicating identical switching conditions, the float position was found to vary as much as 3/8-inch, with a standard deviation (for a 30-point test) of ± 0.21 inches. This variation is in addition to the nonlinearity imposed by constant-voltage operation, but is judged acceptable relative to the inherent resolution limit due to the 1/2-inch switch spacing.

A circuit has been devised (Figure 3) which will provide float-height readout directly in inches on a local digital panel meter, and also will provide a 4-20 ma signal to the control panel. The circuit also supplies constant current to the resistor string, eliminating ambiguous and non-linear voltage steps.

FIGURE 1. Float-Switch Level Sensor - Switching Nonlinearity

FIGURE 2. Float/Switch Liquid Level Probe Calibration

FIGURE 3. Float/Switch Level Sensor Circuit

MOMENTUM VECTOR SENSORS (PEECO)

The momentum vector ("drag rod") sensor, as received, consists of a cylindrical rod (11/16 inch diameter) extending about 20 inches from a pivot gland and up to nearly 6 inches into the fluid-flow region, depending upon the application. An actuator extension of this rod, on the opposite side of the pivot gland, has a ball surface in contact with the ends of the spring-loaded cores of two linear variable differential transformers (LVDTs) which are set normal to the rod axis and at 90° to each other. The relative displacement (from null) of the two cores indicates the angle of displacement of the actuator ($\text{arc tan } x/y$), and thus the angle of flow of the fluid. Within the limits of the elastic restoring forces centering the rod, for symmetrical radial forces the magnitude of displacement of two cores is a measure of the velocity of flow ($\sqrt{x^2 + y^2}$) in the indicated vector direction.

The MV probes obtained for flow-direction indication in the PFE were supplied by Power Engineering and Equipment Company, Inc., and employ LVDTs and signal-conditioning and transmission equipment manufactured by Schaevitz Engineering. One of these units (S/N 59122-12; LVDTs nos. 181 and 182) has been tested for linearity, range (displacement and force) and symmetry; and all of the units have been visually inspected for mechanical operability. All have been found to have minor deficiencies and maladjustments, and to have unnecessary operational limitations, as discussed below.

Bi-directional Flow

The probes, as received, have been found to be responsive only to drag rod motions which displace the LVDT cores into the coils--by direct pressure from the actuator rod extension. Displacement of the cores out of the coils depends upon pressure from a follower spring to hold the cores in contact with the "retreating" actuator. In a few units the springs are misformed so that they do not hold the core-end in contact with the actuator, even in the null position. In all cases the mechanical travel of the spring extension is stopped by its striking the mounting plate well short of the full "out" range of the core, so that the cores do not maintain contact with

the actuator ball for the entire range (approximately 0.03 inches), and some cases only a few mils negative travel is available.

These difficulties can be overcome by reforming the leaf spring, where necessary, and by inserting a spacer washer (about 0.025 inches thick) between the spring extension and the shoulder of the core. Corrections must be made on both axes of all units. Too thin a spacer will not provide adequate correction, and too thick a spacer will result in limiting travel in the "in" direction, so individual attention is required for each axis of each unit.

Force Response

Studies of output versus force on the drag bar were made on one unit (S/N 59122-12)* presumed to be typical, after correction of the mechanical limitations detailed above (B-D flow). These measurements were made with the rod axis vertical in order to obtain null output at the center of the mechanical range of the device. Null reading was obtained by first balancing the demodulator at zero transducer input (zero gain), then zeroing the LVDT mechanically--as nearly as possible--at high gain, then making final null adjustment electrically at full gain. The full-gain adjustment (made alternately with the final zero step until all interactions are compensated) was made to 10.00 volts at the maximum "in" position of the LVDT core--a reproducible condition.

For an LVDT thus adjusted to ± 10.00 volts full range, the maximum linear range via the drag bar was about ± 4.5 volts at 200 g force (Figure 4). Beyond the movement yielding that output, mechanical restraints impede further response, even though the LVDTs would respond to over twice as much displacement.

Force was applied to the sensing end of the drag bar by means of a string and pulley, using brass balance weights. The force was applied at an arbitrarily chosen point--3 inches from the free end--approximating the

* S/N 59122-12. Only one LVDT (#182) was fully evaluated; but less extensive tests on the other LVDT (#181) of the same unit indicate that these devices are quite uniform and are interchangeable.

FIGURE 4. Momentum Vector Sensor Response Versus Force

effective centroid of force for immersion into a 6-inch annulus. This involves a lever arm length of approximately 14 inches, cf. 5 inches from the pivot to the actuator ball. The equivalent forces to yield corresponding displacement and output at other level lengths are easily calculated.

Successive readings were taken for both increasing and decreasing loads, at both 0° (into coil) and 180° (out of coil) orientations. Significant hysteresis was evident, especially following loads above the linear response range.

It is evident that without any restoring force other than the semi-elastic pivot gland (bellows), even approximately-linear response is limited to about 200 g maximum force. This is well below the calculated forces expected to exist under operating conditions (Appendix A).

Displacement Response and Limits

The response of the LVDT systems to deflection of the drag rod was determined (on the same unit) by applying force near the sensing end and by measuring both the displacement and the resultant electrical signal--relative to 10.00 volts for maximum displacement of the core into the coil (Figure 5). [The arrangement used was the same as that for force versus response measurements (above), but separate measurements were necessary because of the counterforce imposed by the dial indicator used to measure displacement.] Displacement was measured at 2 inches from the free end of the drag rod, or about 15 inches from the pivot. This is about 3:1 leverage ratio to the LVDT actuator. Thus the linear range of ± 0.200 inches corresponds to ± 0.067 at the actuator; and this confirms conformance with the claimed linear range of ± 0.050 inches for the LVDTs.

Gravity displacement of the rod in the horizontal position, without applied restoration force, was found to be approximately 0.050 inches, corresponding to about 0.008 inches at the actuator end, or about 1/3 the

FIGURE 5. Momentum Vector Sensor Response Versus Displacement

specified range. Thus, some mechanism to restore the rod to the null position when horizontally mounted will be necessary. This has been contemplated from the outset, and is discussed in some detail under "Proposed Modifications."

Vector Determination

With radial elastic restoration force, or with low-friction circular fixed restraint, the displacement of the drag bar will be in the direction of fluid flow and, within the linear response range, will produce x- and y-axis outputs from the "x" and "y" measuring circuits from which the vector angle can be determined $\phi = \arctan E_x/E_y$.

A brief study was made to establish the validity of this relationship in terms of measured output. No absolute angle measurements were made, but by data correlation the angular resolution was indicated as having a maximum error of less than 2%.

The rod was aligned horizontally, and the unit was rotated manually to obtain various positions from $x = -90^\circ$ (horizontal) and $y = 0^\circ$ (vertical) to $x = 0^\circ$ and $y = +90^\circ$. A fixed weight was suspended from the rod at an assigned rod length, and the amplifier Gain and Zero controls were adjusted to yield readings of 5.00 volts maximum to 0.00 volts minimum for the vertical and horizontal LVDT outputs, respectively, at the two limit rotations. The unit was then rotated in unmeasured increments, the values E_x and E_y recorded, and the values of ϕ calculated.

If the values of E_x and E_y are in fact true vector components of the angle ϕ , their values will be $k \sin \phi$ and $k \cos \phi$, respectively, at all angles (in this case $k = 5.00$). Therefore, any difference between the sum ($E_x + E_y$) for the experimental data and $5 (\sin \phi + \cos \phi)$, calculated for the ϕ values determined from E_x/E_y , is a measure of the "tracking" error of the probe. This error will include any nonlinearity in the two LVDT responses, plus (in this case) any circular nonuniformity in the radial restoration force, or (in the case of circular fixed mechanical limits) any frictional angular restraint. Such a correlation has been

plotted (Figure 6) and the error found to maximize at about 1.75% at 46° to 55°--for this selected (in-in) quadrant of this particular probe. In view of the other data obtained, and of the uniform construction of the probes (after coil centering and follower-spring correction), it probably is safe to assume that (within the linear displacement limit) the vector flow angle derived from the LVDT outputs can be held within about $\pm 2\%$ (tangent values) or $\pm 0.6^\circ$ (see below).

Angle:	1°	10°	30°	45°	60°	80°	89°
Error*:	0.02°	0.20°	0.50°	0.58°	0.50°	0.20°	0.02°
*due to 2% error in tangent value							

However, with circular mechanical limitation of drag rod movement (as by the interior orifice of the pipe housing) it will be essential that the displacement-limiting contact surface is smooth, truly circular, and concentric to the rod at the rest condition. This situation does not obtain for a horizontally mounted unit, for which the gravity displacement from center was found to be about 0.05 inches at 2 inches from the rod end, for the test unit. For a Schedule 40 3/4-inch pipe (ID = 0.824 cf. rod OD = 0.688), the concentric annular space is only 0.068 inches, and a rod droop of 0.050 inches would impose about 75% eccentricity and a severe angle-resolution error.

Velocity Measurement

Although the momentum vector probe has the capability of providing velocity as well as flow-direction data, the dynamic range of velocity information is quite limited. For a given installation, reliable force measurements can be made over about a 1:100 range, but this represents only a 1:10 range of velocity. In contrast, the anticipated range of velocities may exceed 1:100 for any given phase (Appendix A). Thus, flow measurements of adequate dynamic range cannot be obtained with linear restoration force, but would require forces which increase exponentially with displacement from null. Moreover, for any given stream velocity, the drag force varies directly with density, and density varies by a factor of

FIGURE 6. MVP Output Vector Data

about 600 from 40 psia steam to liquid water. Thus, vector amplitude cannot yield unambiguous velocity data unless considerable information as to phase composition and distribution also is available.

It nevertheless appears advisable to retain as much amplitude capability in these devices as reasonably possible. Fortunately, the needed changes to assure reliable vector direction information tend also toward this end (see below).

Proposed Modifications

Because of hysteresis in the "elastic" bellows which constitutes the seal and pivot of the MV unit, as well as because of the low restoration-to-null-position force provided by the bellows, additional restoration force is needed in this unit. Maximum accuracy in determining the vector angle dictates that the added force be large enough to overcome both hysteresis forces and gravity, to reliably return the horizontal bar to a null position accurately concentric to the deflection limits in the absence of flow pressures. Ideally, the restoration force should be exponential with displacement, so that the rod always operates within its elastic range yet remains sensitive to low flow forces. (This also would provide wide-range velocity data.) Such forces are not easily obtained and, although nonlinear forces by means of variable-pitch springs or magnetic devices are possible, they would distort vector direction measurements if applied in a rectilinear array. It may be possible to obtain the desired effect by means of a toroidal magnetic centering device which would exert true polar rather than rectilinear forces, and this possibility is being investigated.

Proportional measurement in a limited range and reliable centering to provide vector direction accuracy at and beyond full range (by means of a circular stop) can be obtained by a rectilinear arrangement of linear force (constant pitch) springs. A simple "restorer" of this type has been devised which can easily be adapted to the MV units (Appendix B). Calculations have been made showing that a range of restoration forces adequate for all anticipated applications can be obtained by means of springs

fabricated from standard spring wire stock. Initial installation of low-force springs adequate to provide gravity compensation and reliable centering yet sensitive to low flows is recommended. These should provide flow direction information at high flow rates at deflection limit positions under most flow conditions but, if hammering or damages due to extreme force and turbulence occur, heavier springs can be installed.

The restorer unit will require only 2 inches or less along the rod axis, but even this space is unavailable if a "spool section" double-union adapter is used to provide the different immersion lengths for the 2 to 6 inch annulus gaps, as originally planned. For this reason, as well as to permit reliable null-center pre-adjustment, the restorer will be mounted uniformly on the MV unit, and interchangeable target bars will be installed to accommodate the different annuli. The modified unit is depicted in Appendix B.

HOT-WIRE PHASE SENSORS

The 18 five-element "regime" sensors (Datametrix) have been received, and all have been visually inspected for integrity and continuity of the filaments. These will be installed with the array aligned radially into the annulus, to determine the phase condition at three points on the radius--near each wall and in the center of the annulus. The five-element array permits the same probe to be used interchangeably in 2-inch, 3 1/4-inch or 6-inch annuli. These units have not yet been tested operationally, but are presumed to respond similar to a single element.

The single-element probes also have been received, and one of these has been tested for general response to phase change. No controlled-flow tests have been made, but the probe was subjected to varying velocity air jets (from a laboratory line-air spigot) and to both static and flowing hot and cold water (from the laboratory faucets). For the instrumental conditions of the test, all readings in air were highly sensitive to flow and were below 10 on the digital readout; and all readings in water, hot

or cold, still or flowing, were 14.28 or 14.29. Thus phase determination in an air-water system appears to be positive for distinct phases. It has not yet been determined what the response will be to "fog" or to larger droplets of suspended water in air, or to steam or steam/water mixtures at various temperatures. These studies will be undertaken, and the air/water studies will be started before October 1.

APPENDIX A

DRAG FORCE CALCULATIONS

APPENDIX A
TO "PRELIMINARY EVALUATION OF SENSORS FOR THE PLENUM-FILL EXPERIMENT"
SECTION I

DRAG FORCE CALCULATIONS

The following calculations and derived data and figures present approximate net-average-flow velocity, flow pressure and drag-bar force values for various single-phase flow conditions in the annuli of the large and small simulators--without thermal shield, for a range of mass flow rates.

The calculated data are for the average net vertical flow in the annulus, based on mass flow times specific volume for the phase stipulated and for the annular cross-sectional area involved, and thus represent an unreal velocity situation. In any real case, flow must of necessity be bi-directional--or multi-directional, and thus the available cross-sectional area will be shared by downflow (chiefly water), upflow (chiefly steam), and perhaps zones of no flow or cross flow or turbulence with little or no net vertical vector.

Thus these data serve chiefly as indications of the velocities and forces likely to exist, and of their relative values at different mass flows and under different pressures, for the four different vessel configurations, and are not directly applicable to any real two-phase situation. The data on water flow can be particularly misleading, since the full-annulus vertical flow condition to which they apply would relate only to the uniform rate of rise of water in a closed-bottom annulus, and not to water cascading downward, by gravity, in a partially filled annulus. However, the forces present in a two-phase system will be the algebraic (vector) sum of the single-phase forces for the appropriate partial mass flows; and thus an estimate of these forces can be made on the basis of reasonable assumptions (see Sample Calculation, p. A.5).

CALCULATION BASES

Because of the uncertainty and wide range of possible phase ratios, local flow regimes, velocities, and pressures, no attempt to calculate Reynolds numbers or exact drag coefficients was considered feasible; and the data presented are estimates based on an average drag coefficient value ($C_D = 1.2$) for averaged flow values. The data extend, however, beyond velocities for which this coefficient is fully valid.

Only one temperature-pressure condition was considered in the calculations for liquid water, viz. 100 psia (328°F). The density and derived velocity values for water remain within about $\pm 10\%$ of the values at 100 psia over the entire range from 15 to 450 psia (212-457°F), so this one condition adequately represents the liquid state. However, water velocities in a two-phase system may greatly exceed those due simply to mass flow.

Steam pressure-temperature conditions of 15, 40, 100 and 500 psia (212-467°F) were used in the tabulated calculations, representing a range from the "atmospheric" condition at the point of rupture to near-maximum steady-state pressure. Intermediate-pressure interpolations were calculated ($k \cdot m^2$) in preparing the graphical representations of flow pressure (P_f , psi) for a cylindrical body (Figures A.1 to A.4).

Drag forces (F_d , lbf) are given for "typical" drag-bar target areas of $\sim 1 \text{ in.}^2$ and $\sim 3\text{-}1/2 \text{ in.}^2$ (1-1/2 and 5-1/2 in. insertions of the 11/16 in. diameter drag bar of the MV units, as received). Smaller target areas may be used but for practical reasons (strength, stiffness, clearance) the effective "target" area in the flow region must be limited to between about 1/4 and about 4 in.^2 . Thus the applied force will be about 1/4 to 4 times the P_f (psi) values. The latter can be expected to range from <0.01 to >5 psi, as shown in the figures, and quite possibly will be much higher for channeled flow in local areas and for ballistically projected liquid water.

DRAG FORCE VERSUS WATER AND STEAM FLOWS

GIVEN - Annulus Cross-Sectional Areas:

Large-Scale Simulator w/6 in. Annulus - 16.5 ft^2
Large-Scale Simulator w/2 in. Annulus - 5.7 ft^2
Small-Scale Simulator w/6 in. Annulus - 4.7 ft^2
Small-Scale Simulator w/2 in. Annulus - 1.75 ft^2

FIND drag forces on 5-1/2 in. long rod for 6 in. annulus and on 1-1/2 in. long rod for 2 in. annulus for total mass flows of 50, 200, and 500 lbm/sec (500 lbm/sec = 1.8×10^6 lbm/hr, approximate design maximum).

Mass Flow lbm/sec	Mass Flow/Unit Area, $\dot{m}/A$ (lbm/ft ² sec)			
	<u>LSS/6</u>	<u>LSS/2</u>	<u>SSS/6</u>	<u>SSS/2</u>
500	30.3	88	106	286
200	12.1	35	42.5	114
50	3.03	8.8	10.6	28.6

For these flow rates and for the densities and viscosities involved, the Re values are estimated to fall within the range 5000 to 3×10^5 for which the drag coefficient, $C_D = 1.1 \pm 0.1$. Assume $C_D = 1.2$ in all cases.

$$\text{Flow pressure, } P_f = C_D \rho V^2 / \text{age lbm/ft}^2 = 1.29 \times 10^{-4} \rho V^2 \text{ psi}$$

$$\text{Velocity, } V = (\dot{m}/A)(1/\rho)$$

$$\therefore, P_f = 1.29 \times 10^{-4} (\dot{m}/A)^2 (1/\rho) \text{ psi (directional)}$$

$$\text{Neglecting end effects, drag force, } F_d = P_f A$$

$$\begin{aligned} \text{For 11/16 dia rod, at 5-1/2 in. insertion, } F_d &= 3.78 P_f \\ &\text{at 1-1/2 in. insertion, } F_d = 1.03 P_f \end{aligned}$$

SAMPLE CALCULATION - CASE 3

Small-Scale Simulator/6 in. Annulus (4.7 ft²)

	$\frac{1}{\rho}$ ft ³ /lbm	V ft/sec	P_f psi	$F_d(11/16 \times 5-1/2)$ lbf	g
<u>500 lbm/sec</u>					
$\dot{m}/A = 106$					
$P_f = 1.46 (1/\rho)$					
Water at 100 psia	0.0177	1.88	0.026	0.10	44
Steam at 15 psia	26.3	(2788)	(38.4)	--	--
Steam at 40 psia	10.5	1113	15.3	<u>57.9</u>	<u>2.6×10^4</u>
Steam at 100 psia	4.4	466	6.42	<u>24.3</u>	<u>1.1×10^4</u>
Steam at 600 psia	0.77	82	1.12	<u>4.25</u>	<u>1929</u>
<u>200 lbm/sec</u>					
$\dot{m}/A = 42.5$					
$P_f = 0.233 (1/\rho)$					
Water at 100 psia	0.0177	0.75	4.1×10^{-3}	1.6×10^{-2}	7
Steam at 15 psia	26.3	1118	6.13	<u>23.16</u>	<u>1.05×10^4</u>
Steam at 40 psia	10.5	446	2.45	<u>9.25</u>	<u>4198</u>
Steam at 100 psia	4.4	187	1.03	<u>3.88</u>	<u>1759</u>
Steam at 600 psia	0.77	33	0.18	<u>0.68</u>	<u>307</u>
<u>50 lbm/sec</u>					
$\dot{m}/A = 10.6$					
$P_f = 0.0146 (1/\rho)$					
Water at 100 psia	0.0177	0.188	2.6×10^{-4}	0.0010	0.44
Steam at 15 psia	26.3	279	0.384	<u>1.45</u>	<u>660</u>
Steam at 40 psia	10.5	111	0.153	<u>0.58</u>	<u>263</u>
Steam at 100 psia	4.4	46.6	0.064	0.24	110
Steam at 600 psia	0.77	8.2	0.011	0.043	19

SAMPLE CALCULATION OF TWO-PHASE FORCES

ASSUMED - For a total mass flow in the small simulator with 2 in. annulus of 400 lbm/sec, assume 90% of that flow is downward as liquid water, and 10% of that water is converted to steam and flows upward.

FIND - Water and steam flow pressures at 5 feet below return legs.

CALCULATION

Water downflow

Assume downflow at 1/2 free fall velocity due to upflow pressures:

$$V = 1/2 (2gs)^{1/2} = 1/2 (2.32, 2.5)^{1/2} = \underline{9.0} \text{ ft/sec}$$

$$X\text{-sectional flow area} = 0.9 (400) (0.017)/9.0 = 0.68 \text{ ft}^2$$

$$P_f = C_D \rho V^2 / \text{age} = (1.2/0.017) 81/2(32.2) 144 = \underline{0.62} \text{ psi}$$

Steam upflow

$$\text{Available annulus x-section} = 1.75 - 0.68 = 1.07 \text{ ft}^2$$

$$\text{Velocity} = 0.1 (0.9) 400 (4.4)/1.07 = \underline{148} \text{ ft/sec}$$

$$P_f (\text{full annulus, from Figure A.4}) \text{ at } 36 \text{ lb/sec} = 0.24 \text{ psi}$$

$$P_f = 0.24 (1.75/1.07)^2 = \underline{0.64} \text{ psi}$$

$$\text{or } P_f = (1.2/4.4) (148)^2 / 64.4 (144) = \underline{0.64} \text{ psi}$$

Thus, quite by chance, the assumed flow conditions of this example are such that a drag target exposed 50% to the water downflow and 50% to the steam upflow would show about zero net flow. On the average, for multiple sensors, the flow force would be $-(0.68)(0.62) + (1.07)(0.64)/1.75 = + 0.15 \text{ psi (up)}$.

FIGURE A.1. Large Scale Simulator with 6 in. Annulus

FIGURE A.2. Large Scale Simulator with 2 in. Annulus

FIGURE A.3. Small Scale Simulator with 6 in. Annulus

FIGURE A.4. Small Scale Simulator with 2 in. Annulus

APPENDIX B

MOMENTUM VECTOR PROBE MODIFICATIONS

APPENDIX B
TO "PRELIMINARY EVALUATION OF SENSORS FOR THE PLENUM-FILL EXPERIMENT"

SECTION I

MOMENTUM VECTOR PROBE MODIFICATIONS

Modification of the PEECO momentum vector (drag bar) probe for determining flow direction appears necessary in order to achieve the desired accuracy and reliability, particularly in assuring that the vector displacement of the drag bar is in the true direction of fluid flow. Since the displacement range of the bar is small compared to the drag bar diameter, and thus the annulus between the bar and the displacement-limiting circular stop (and mounting pipe ID) is quite small compared to the stop ring diameter, any displacement at null (zero flow force) from true concentricity will result in error in the angular displacement of the rod, i.e., it will tend to seek the widest section of an eccentric ring if displacement is in the same general direction. Thus restoration forces symmetrical to the true center of the limiting stop ring are important.

A device to accomplish this aim is shown in Figure B.1. The vertical upward force will be adjusted to compensate for gravity (for an indexed rotation of the restorer and the x and y-axis LVDTs) by shimming the bottom spring. Otherwise the springs will be adjusted to exert equal forces by grinding to equal length springs cut from the same coil, and by preloading equally by precision nonadjustable followers. A range of null-restoration forces appropriate for this application can be obtained by means of simple constant-pitch springs wound from standard gage wire. In the heavier springs, if needed, little or no shimming will be required.

The restorer will be mounted on the measuring head assembly in a fixed position such that the applied forces will be at a standard lever arm distance from the nominal pivot point relative to the distance from the pivot to the plane of the LVDTs; and displacement limitation in the restorer will be equivalent to ± 0.025 inches at the actuator. In order to obtain equal

displacement clearance at the most distant end of the mounting pipe (nom. ID = 0.824), the drag bar diameter must be reduced. For a drag bar section between the restorer and inner surface of the vessel wall of 1/2 in. diameter, full displacement and full-range LVDT output can be achieved.

For fixed positions of the MV probe pivot, actuator and restorer relative to the vessel wall inner surface, variable penetration into the annulus by the drag bar target section (for 2 in. to 6 in. annulus configurations) requires that the drag bar length be made variable. (This is in contrast to the alternative of maintaining constant bar length and varying the mounting position. The addition of the restorer mechanism make that approach impractical). By employing removable and interchangeable drag bar extensions, not only different lengths but different diameter target sections can be employed, to provide a selection of applied-force (lbf) to flow-force (psi) ratios. By selecting diameters to compensate for lever length changes with target length, targets can be selected to yield whole-number ratios relating flow force to restoration force, and thus to LVDT displacement and readout (Figure B.2).

Various target shapes also can be used if desired. A tapered target can be employed to obtain constant moment across the annulus gap, or the target could be shaped to accentuate response in selected zones.

FIGURE B.1. Null Restorer and Universal Stub Assemblies

TARGET BAR FOR 2" ANNULI (ALL VESSELS)

TARGET BAR FOR 6" ANNULI (TYPICAL)

* [1bf @ 10" = 4xPSIF
1bf @ 10" = 6xPSIF
1bf @ 10" = 9xPSIF]

FIGURE B.2. Target Bars

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