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SUBJECT: Flow Characteristics for HRT Pressure-Balancing Control Valve

FROM:

C. ^{Carlyle} Michelson

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Summary:

This report presents the basic criteria and calculations leading to the specification of a pressure-balancing control valve for the HRT (Specifications No. HRT-306). Included is a suggested layout for connecting the valve into the present piping system.

Basic Criteria:

The basic purpose of the pressure-balancing control valve in the HRT is to provide relief for large pressure differentials between the core and blanket pressurizers, thus protecting the core vessel against possible rupture or buckling. The valve is fail safe in that it is an air to close device. In addition, a rupture disc assembly parallels the valve thereby providing a fail safe path for uncontrollable pressure surges, and a bypass in case of valve control failure.

The situation requiring the largest mass flow through the valve between core and blanket pressurizers occurs when one dump valve is opened instantaneously while the other remains closed (or lags severely), and the system has a steam overpressure. Under such conditions the flows might reach:⁽¹⁾

	<u>Instantaneous Dump from 2000 psia Total Pressure</u>
Core to blanket	60 liters/sec
Blanket to core	20 liters/sec

These figures represent an ultra-conservative estimate of the initial instantaneous volumetric dump rate from the opened side of the system. The control valve between the pressurizers cannot compensate for this initial flow surge because of its short duration as compared to the response time of the control system. Also, the control function is not initiated until after a preset pressure

differential is reached. By the time control is then established the initial flow surge is past and dump rate considerably reduced. In view of these considerations, and the fail safe features of the system, a maximum vapor transfer capability between the pressurizers equal to one-half the instantaneous dump rate from the opened side of the system is considered adequate to protect the system⁽²⁾.

For instantaneous dump from pressures (P_x) other than 2000 psia, but with a steam overpressure, the instantaneous volumetric dump rate (Q_x) from the opened side of the system can be approximated by⁽³⁾:

$$Q_x = Q_{2000} \sqrt{\frac{P_x}{2000}}$$

Q_{2000} = dump rate from 2000 psia, liters/sec

The maximum allowable pressure drop across the control valve is determined by the core vessel strength. Conservative code-type calculations indicate a maximum internal pressure difference of 400 psi, and external pressure difference of 150 psi are allowable⁽⁴⁾. However, this does not provide for radiation and corrosion effects. These effects are not accurately known at present. Selection of the pressure-balancing control valve characteristics for the HRT is therefore conservatively based on the following maximum allowable pressure differentials:

Core to blanket	200 psi
Blanket to core	50 psi

Specification of Flow Characteristics

The physical characteristics of light water vapor are used in this specification except for a constant density correction of 1.11 to approximate the density difference between light and heavy water vapor. The estimated density of heavy water vapor as a function of saturation pressure is shown in Figure 1. The flow of steam from 2000 psia through a 10% maximum pressure drop is treated as noncompressible fluid flow. Work reported in the literature⁽⁵⁾ indicates this to be a valid assumption for the steam flows being considered.

Table I specifies maximum mass flow of heavy water vapor from core to blanket required to limit the pressure differential between the systems to 200 psi when the blanket dump valve is opened instantaneously and the core dump valve remains closed. Table II specifies the estimated maximum mass flow from blanket to core required to limit the pressure differential to 50 psi when the core dump valve is opened instantaneously and the blanket dump valve remains closed.

The pressure balancing valve between the core and blanket pressurizers of the HRT must allow these mass flows to pass with the total pressure drops indicated. The allowable pressure drop across the valve itself is determined by the piping layout selected. The proposed piping layout for use in the HRT is shown in Figure 2. The maximum flow velocities and velocity heads for this piping system are indicated in Table III.

An estimate of the maximum total pressure drop due to steam flow in the pipes and fittings is given in Table IV. The resistance coefficients for the pipe and pipe fittings are developed in the Appendix. As indicated in the Appendix, these pressure drop values are high because they are based on experimentally measured values for low Reynolds numbers and standard steel pipe. It is estimated that for high Reynolds numbers and stainless steel seamless pipe the pressure drop

through the fittings may be lower by a factor of two. The pressure drop in the straight runs of pipe is based on data for high Reynolds numbers and drawn tubing; any contribution to the error in total pressure drop should therefore be small. If full advantage is taken of the estimated error in pressure drop through the fittings, the maximum total pressure drop for instantaneous core dump becomes 4.0 psi; and 37.7 psi for instantaneous blanket dump.

On the basis of criteria established earlier for maximum allowable pressure drop across the entire system, the pressure drop across the pressure balancing valve must lie in the range 43.5—46.0 psi for instantaneous core dump and 145.4—162.3 psi for instantaneous blanket dump. These ranges of allowable pressure drop may be further extended by streamlining the 45° branch, which is one of the largest contributors to the total pressure drop in the fittings. Work of Petermann⁽⁶⁾ indicates this loss might be reduced by another factor of two if the branch is streamlined by going to a conical reducer section. The maximum allowable pressure drop across the valve may thus be extended to 170.5 psi.

Specifications for the pressure-balancing control valve to be used in the HRT⁽⁷⁾ call for a maximum allowable pressure drop of 175 psi across the valve for a light water vapor mass flow rate through the valve equal to the heavy water vapor mass flow rate predicted in Table I. Within the limits of this study, the pressure drop across the valve for a constant mass flow rate through the valve is inversely proportional to the density of the fluid flowing⁽⁸⁾. Therefore, the pressure drop across the specified valve is 158 psi for a heavy water vapor mass flow rate equal to the light water vapor mass flow rate specified. This pressure drop is near the upper limit of the estimated allowable pressure drop (for high Reynolds numbers and smooth pipe). It is recommended that a

conical streamlined reducer be used in the branch connection to reduce the total pressure drop for this condition.

For reverse flows through the valve the specification calls for a maximum pressure drop of 50 psi with 38% of the specified forward mass flow rate of light water vapor. This is in excess of the required flow, but gives some allowance for possible lack of bilateral valve characteristics. Theoretically, the valve should pass 53.5% of maximum mass flow rate with 28.6% of maximum pressure drop. The pressure drop across this valve for equal heavy water mass flow rates will be 45 psi. Again the use of a streamlined entrance region will make this pressure drop fully acceptable.

Characteristics of Commercially Available Valves

A study was made of the mass flow rate that commercially available single-seated valves will pass as a function of valve inlet pressure and pressure drop across the valve. From this study some idea is obtained of the relative size of the pressure-balancing valve specified for the HRT.

The manufacturers rating of a valve is based on a valve flow coefficient, C_v , as defined by the following equation:

$$C_v = 57.2 \frac{W}{\sqrt{p \Delta P}}$$

$$\text{or } W = 0.0176 C_v \sqrt{p \Delta P}$$

where W = mass flow rate, lbs/sec

p = density of steam at valve inlet, lbs/ft³

ΔP = pressure drop across valve, lbs/in²

For steam flow, the pressure drop across the valve cannot exceed one-half the valve inlet pressure (critical flow limit).

The C_v values for some commercially available single-seated valves are indicated below⁽⁸⁾:

<u>Valve Size</u>	<u>C_v</u>
1/2"	4
1"	10
1-1/4"	16
1-1/2"	25

A commercially available valve which passes 5.9 lbs/sec saturated light water vapor at 2000 psia with a pressure drop of 175 psi has a C_v of 11, which is approximately a 1" valve. The characteristics of such a valve are shown in Figure 3. Also shown are the specified characteristics for the HRT pressure-balancing control valve. The dotted portions of the specified characteristics represent the values as given in Tables I and II. These values have been arbitrarily increased to reflect known inaccuracies in the square root relationship of pressure to flow for very low pressure conditions. It is apparent from the figure, however, that a commercial valve which meets the maximum flow requirement should more than satisfy the specification for flows at lower inlet pressures. Some inherent inaccuracy in the square root relationship is therefore not important.

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Figure 3

Table I
Pressure-Balancing Flow for Blanket Dump

Blanket Pressure P_b , psia	Blanket Dump Rate $30\sqrt{\frac{P_b}{2000}}$ liters/sec	Core Pressure ($P_b + 200$) psia	Maximum Flow Core to Blanket lbs/sec*
200	9.5	400	0.32
600	16.4	800	1.13
1000	21.2	1200	2.30
1400	25.1	1600	3.84
1800	28.4	2000	5.94
2000	30.0		

Table II
Pressure-Balancing Flow for Core Dump

Core Pressure P_c , psia	Core Dump Rate $10\sqrt{\frac{P_c}{2000}}$ liters/sec	Blanket Pressure ($P_c + 50$) psia	Maximum Flow Blanket to Core lbs/sec**
350	4.18	400	0.14
750	6.11	800	0.42
1150	7.58	1200	0.83
1550	8.80	1600	1.36
1950	9.86	2000	2.06
2000	10.0		

* Heavy water vapor saturated at core pressure.

** Heavy water vapor saturated at blanket pressure.

Table III
Maximum Flow Velocities and Velocity Heads

<u>Instantaneous Core Dump</u>	<u>Velocity ft/sec</u>	<u>Velocity Head psi</u>
1"-Sch 80 valve inlet (~ 2000 psia)	68	2.9
valve discharge (~ 1950 psia)	70	3.0
2"-(.312 wall) (~ 1950 psia)	21	0.27
<u>Instantaneous Blanket Dump</u>		
2"-(.312 wall) (~ 2000 psia)	60	2.3
1"-Sch 80 valve inlet (~ 2000 psia)	200	25.5
valve discharge (~ 1800 psia)	232	29.6

Table IV
Maximum Total Pressure Drop in Piping System

<u>Instantaneous Core Dump</u> (Max. flow B-C 2 lbs/sec)		<u>Instantaneous Blanket Dump</u> (Max. flow C-B 5.9 lbs/sec)	
	<u>Pressure Drop psi</u>		<u>Pressure Drop psi</u>
45° Branch	2.4	Standard tee	2.6
45° elbow	0.5	Long Sweep bend	0.7
90° elbow	1.1	2" pipes ($\sim 7-1/2'$)	1.8
Reducer	0.5	Reducer	1.0
1" pipes ($\sim 2-1/2'$)	1.4	90° elbow	9.2
Long Sweep bend	0.1	45° elbow	5.3
Standard tee	0.3	45° Branch	21.0
2" pipes ($\sim 7-1/2'$)	0.2	1" pipes ($\sim 2-1/2'$)	13.0
Total	6.5	Total	54.6

APPENDIX

Pressure Drop Through Pipes and Fittings

The head loss in a fitting due to change in the direction of flow and/or disturbance of the flow pattern can be expressed as a whole or fractional part of a velocity head of the fluid flowing through the fitting.

$$H_b = K_b \frac{v^2}{2g}$$

H_b = head loss due to bending, ft

K_b = resistance coefficient for bending

v = mean velocity of fluid, ft/sec

g = gravitational constant, 32.2 ft/sec²

The head loss in any straight or curved run of pipe due to friction alone can also be expressed in terms of velocity heads.

$$H_f = K_f \frac{v^2}{2g}$$

H_f = head loss due to friction, ft

K_f = resistance coefficient for friction

v, g = as defined previously

The total head loss, H_t , in a pipe fitting due to combined turning and friction effects can be expressed as:

$$H_t = (K_b + K_f) \frac{v^2}{2g}$$

Experimental values for K_b and $K_b + K_f$ are available. Where K_f values are required the Darcy equation applies.

$$K_f = f \left(\frac{1}{d} \right)$$

f = friction factor

l = pipe length, ft

d = inside pipe diameter, ft

Correlations of experimental data developed by Pigott⁽⁹⁾ indicate the resistance coefficient for bends and elbows (K_b) is a function of friction factor (Reynolds number) as well as r/d . The Reynolds numbers in the pressure balancing system vary over the range $(0.7 \rightarrow 2.1) \times 10^6$ in the 2" line and $(1.3 \rightarrow 3.9) \times 10^6$ in the 1" line. Most of the K_b and $K_b + K_f$ values in the literature have been determined for low Reynolds numbers, and for materials somewhat rougher than stainless steel seamless pipe. If these experimentally determined values are used in calculating pressure drops for high Reynolds numbers and for pipes whose roughness more closely approximates drawn tubes, a considerable overestimate of the pressure drop may result. However, since K_b and $K_b + K_f$ values have not been measured for the material and flow conditions in the pressure balancing system, it is necessary to use the data which is available and estimate the error for high Reynolds numbers and smooth pipe conditions. The resistance coefficients for pipe fittings used in this study are given in Table V.

The resistance coefficient for flow of steam in straight runs of pipe is estimated on the basis of a constant friction factor of 0.015 over the range of $0.7 \times 10^6 < R_e < 4 \times 10^6$. This appears to be a conservative but fair estimate based on the work of Pigott⁽⁹⁾. This friction factor gives a K_f value (per foot of pipe) of 0.188 for 1"-Sch 80 and 0.103 for the 2"-(.312 wall).

Table V
Resistance Coefficients for Pipe Fittings

Type	Flow Directions	K_b	$K_b + K_f$	Reference
Standard Welding Tee (no thru run)	C - B		1.15	(10)
	B - C		1.0	(10)
Welding Branch (45°) (no thru run)	C - B		0.71*	(10)
	B - C		0.81*	(10)
Welding Reducer (4"/g. - streamlined)	C - B		0.04*	(11)
	B - C		0.18*	(11)
Long Sweep bend (r/d ~ 7)		0.3	**	(12)
Long radius welding elbow (45°)		0.18	**	(12)
Long radius welding elbow (90°)		0.36	**	(12)

C - core

B - blanket

* Use velocity head of 1" pipe

** Include length of bend or elbow in total piping length

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