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DESIGN STUDY -- HRT PRESSURIZER LEVEL FLOAT

The attached study is submitted per the request of R. L. Moore, July 31, 1956⁽¹⁾. It was requested that a design study be made to determine the float weight versus length, for a displacement of approximately 6-in³. In considering longer float lengths, the float weight must be taken into consideration, since the float weight has a large effect on damping and zero shift characteristics, and since, for linearity purposes, it is desirable to operate the springs below the proportional limit. The present float weight is approximately 0.8 lbs. It was also requested that the feasibility of using titanium for the float be investigated.

Table 1 shows some typical float dimensions based on Figures 1, 2, and 3 presented herein for a Type 347 stainless steel construction.

Table 1

<u>L</u> <u>in.</u>	<u>D</u> <u>in.</u>	<u>t</u> <u>in.</u>	<u>F</u> <u>in³</u>	<u>P</u> <u>lb/in³</u>	<u>W</u> <u>lbs</u>
5	1.250	0.092	6	0.285	0.8
10	0.880	0.065	6	0.285	0.8
15	0.720	0.053	6	0.285	0.8

Stainless Steel Float (Type 347)

If it is assumed that the allowable pressure, as predicted from Fig. UHA-28.2 V (1952 ASME Code, Sect. VIII), is based on a factor of safety of 3 on the actual collapsing pressure, then for 2500 psi design pressure the D/t ratio required is 13.5 from Fig. 2.

(1) See Appendix

Table 2

Wall Thickness vs Outside Diameter at Collapse (2500 psi)

<u>D</u> <u>in.</u>	<u>t</u> <u>in.</u>
0.500	0.0371
0.750	0.0556
1.000	0.0740
1.250	0.0926
1.500	0.111

The wall-thickness required for various float diameters is listed in Table 2 and plotted on Fig. 3.

Table 3

External Design Pressure for Type 347 Stainless
Steel From Fig. UHA-28.1 vs t/D

<u>D/t</u>	<u>t/D</u>	<u>B</u>	<u>Pa</u>	<u>FS=2</u> <u>P₂</u>	<u>FS=3</u> <u>P₃</u>	<u>FS=4</u> <u>P₄</u>
10	0.100	12,000	1200	2400	3600	4800
15	0.067	11,000	738	1476	2214	2960
20	0.050	10,000	500	1000	1500	2000
25	0.040	9,000	360	720	1080	1440
30	0.033	8,000	267	534	801	1068

Table 4
Stress in Tube Wall at Collapse

<u>D/t</u>	<u>t/D</u>	<u>S(psi)</u>
10	0.100	18,000
15	0.067	16,600
20	0.050	15,000
25	0.040	13,500
30	0.033	12,000

Weight of Float -- For a float designed to resist collapse at a given external pressure, the ratio of diameter to wall-thickness is determined. The following analysis shows that for a constant displacement and a fixed ratio of diameter to wall thickness, the weight of the float is substantially constant except for small variations in the end-closures.

Let:

- W = weight of float, lbs
- L = length of float, in.
- D = outside diameter of float, in.
- d = inside diameter of float, in.
- ρ = density of material, lb/in³
- F = float displacement, in³
- w = weight of end-caps, lbs.

$$W = \pi \left(\frac{D + d}{2} \right) \cdot t \cdot \rho \cdot L + w$$

$$t = D/K$$

$$d = D - 2t = D - \frac{2D}{K}$$

$$W = \pi \left(\frac{D}{2} + \frac{D}{2} - \frac{D}{K} \right) \left(\frac{D}{K} \right) \rho L + w$$

$$W = \pi \left(D - \frac{D}{K} \right) \left(\frac{D}{K} \right) \rho L + w$$

$$\text{or } W = \pi \rho L \frac{D^2}{K} \left(1 - \frac{1}{K} \right) + w$$

$$\text{Since } L = \frac{4F}{\pi D^2}$$

$$W = 4F\rho \frac{1}{K} \left(1 - \frac{1}{K} \right) + w$$

Then, if w remains substantially constant, the weight of the float remains constant. For stainless steel to resist 2500 psi external pressure with $F = 6 \text{ in}^3$, $K = 13.5$ (From Fig. 2) and

$$W = (4)(6)(0.285) \frac{1}{13.5} \left(1 - \frac{1}{13.5} \right) + w$$

$$W = 0.470 + w = 0.8 \text{ lb}$$

Feasibility of using titanium for floats

- (1) Material: MST 6Al-4V (Mallory-Sharon Titanium Corp.)

This alloy can be readily welded and heat-treated. It is primarily produced as bar or forging.

- (2) Material Properties:

Density	0.160 lbs/in ³
Thermal expansion	(90-932°F) - 5.2×10^{-6} in/in-°F (932-1652 F) - 5.8×10^{-6} in/in-°F
Ultimate strength	150,000 psi (room temp.) 118,000 psi (600°F)
Y.S. (0.2% offset)	140,000 psi (room temp.) 109,000 psi (600°F)

Modulus of Elasticity

R.T.	16.9×10^6
200°F	16.6×10^6
400°F	16.2×10^6
600°F	14.7×10^6
800°F	13.2×10^6
1000°F	10.1×10^6

(3) The collapse of a thin-walled tube is based on the equation:

$$P = 2.2E \left(\frac{t}{D} \right)^3$$

where P = collapsing pressure, psi

E = modulus of elasticity

t = thickness of shell, in.

D = outside diam. of shell, in.

Using such an equation in combination with the hoop-stress formula:

$$S = \frac{P}{2 \left(\frac{t}{D} \right)}$$

and substituting the tangent modulus from true-stress true-strain curves, design charts such as are in the ASME Code may be constructed. Such design charts have not been made for titanium.

For a $D/t = 23.5$

$$W = \pi L p t (D - t) + w$$

Since $t = D/23.5$

$$W = \pi L p \frac{D}{23.5} \left(D - \frac{D}{23.5} \right) + w$$

$$W = \pi L \rho D^2 \frac{1}{23.5} \left(1 - \frac{1}{23.5}\right) + w$$

$$W = \pi L \rho D^2 (0.0408) + w$$

$$\rho = 0.160 \text{ lb/in}^3$$

$$W = (0.0408)(0.160)(\pi) LD^2 + w$$

$$W = 0.0205 LD^2 + w$$

$$L = \frac{6}{\frac{\pi}{4} D^2} = \frac{7.64}{D^2}$$

Substituting $W = (0.0205)(7.64) = 0.157 + w \text{ lbs}$

Table 5 shows some typical float dimensions for a titanium float fabrication.

Table 5
Titanium Float Dimensions

<u>L</u> <u>in.</u>	<u>D</u> <u>in.</u>	<u>t</u> <u>in.</u>	<u>F</u> <u>in</u> ³	<u>ρ</u> <u>lb/in</u> ³	<u>W</u> <u>lb</u>
5	1.250	0.0530	6	0.160	0.267
10	0.880	0.0374	6	0.160	"
15	0.720	0.0307	6	0.160	"

Conclusions:

- 1) For a constant displacement of 6-in³ the corresponding outside diameter of the required float may be determined from Figure 1.
 - 2) The allowable external pressure for an infinitely long cylinder of Type 347 stainless steel for any given D/t ratio may be determined from Fig. UHA-28.2 in the 1952 edition of the ASME Code, Sect. VIII, Unfired Pressure Vessels.
- By assuming the allowable pressure to be some fraction of the collapsing

pressure, as shown in Fig. 2, the ratio of D/t for an assumed collapsing pressure may, conversely, be determined. At a design pressure of 2500 psi the required D/t ratio for a type 347 stainless steel cylinder of infinite length is 13.5.

- 3) The minimum wall thickness of a Type 347 stainless steel cylinder may be read directly from Fig. 3 (for an external pressure of 2500 psi).
- 4) The weight of the float remains constant at 0.8 lbs. for stainless steel or 0.267 lb. for titanium.
- 5) The dimensions of the titanium float were determined from theoretical considerations and should be verified by experiments.

Chas. L. Segaser
C. L. Segaser

APPENDIX

Experience with the displacement float level controllers on the HRT mock-up indicates that, while good steady state control has been obtained with the 5-in. float length, a longer float length would give better recovery to large level disturbances.

The 5-in. range was originally specified on the basis that the level was to be controlled in the pressurizer drum. Since the level is now being controlled in the legs, the 5-in range restriction no longer applies, and the range (float length) could be increased insofar as control range requirements are concerned. However, in considering longer float lengths, the float weight must be taken into consideration, since the float weight has a large effect on damping and zero shift characteristics, and since, for linearity purpose, it is desirable to operate the springs below the proportional limit. The present float weight is approximately 0.8 lbs.

As the operation of the present level controllers would be improved by making the float longer or lighter, and as future applications of this instrument may require wider ranges, it is requested that a design study be made to determine the float weight versus length, for a displacement of approximately 6-in³. It is also requested that the feasibility of using titanium for the float be investigated.

R. L. Moore

FIG. 1 FLOAT DIAM. VS. LENGTH FOR CONSTANT
DISPLACEMENT OF 6 IN.³

1000.0

EXTERNAL PRESSURE-PSI

ALLOWABLE PRESSURE

$D/t = 13.5$

RATIO - $\frac{D}{t}$

FS-4

FS-3

FS-2

FIG. 2 COLLAPSING PRESSURE - TYPE 347 STAINLESS STEEL
TUBES VS. D/t FOR ASSUMED SAFETY FACTORS BASED
ON ALLOWABLE PRESSURE

FIG. 3 WALL THICKNESS VS. FLOAT DIAM.

100,000

9

8

7

6

5

4

3

2

1

0

9

8

7

6

5

4

3

2

1

0

9

8

7

6

5

4

3

2

1

0

9

8

7

6

5

4

3

2

1

0

9

8

7

6

5

4

3

2

1

0

COLLAPSING PRESSURE PSI

$$P = 2.2 E \left(\frac{t}{D} \right)^3 \text{ PSI}$$

$$E = 14.7 \times 10^6 \text{ PSI @ } 600^\circ \text{F}$$

23.5

RATIO D/t

FIG. 4 THEORETICAL COLLAPSING PRESSURE
TITANIUM FLOAT

EUGENE DIETZEN CO.
PRINTED IN U.S.A.

NO. 340-L310 DIETZEN GRAPH PAPER
SEMI-LOGARITHMIC 3 CYCLES X 10 DIVISIONS

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