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DESIGN REPORT
TURBINE POWER CONTROL VALVE
FOR USE AS
DILUENT CONTROL VALVE
IN
CRYOGENIC SERVICE

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TABLE OF CONTENT

I	ABSTRACT
II	INTRODUCTION
III	OBJECTIVE
IV	SUMMARY AND CONCLUSIONS
V	DESCRIPTION AND FUNCTION OF COMPONENT
VI	DESIGN CRITERIA
VII	ANALYSIS RESULTS

ENCLOSURE:

1. Photograph #1446 "DISASSEMBLED VIEW OF THE TPCV"
2. Sketch-1.2.2-65-S-023 "PARTIAL VIEW - TPCV COMPONENTS"
3. STRESS ANALYSIS (11 Pages)
4. Sketch-1.2.2-65-S-024 "TPCV TRANSIENT TEMPERATURE MAP"
5. Sketch-1.2.2-65-S-025 "TPCV FLOW LOADS"

ABSTRACT

A design analysis of the Turbine Power Control Valve was conducted to investigate its suitability for use as a Diluent Control Valve. Factors of safety used in the analysis were 1.25 for yield strength and 1.50 for ultimate strength. All margins of safety calculated for the components were positive. The results of this design analysis indicate the valve is suitable for use in cryogenic service.

II INTRODUCTION

A design analysis was conducted to determine the suitability of the Turbine Power Control Valve (TPCV) to qualify for use as a Diluent Control Valve (DCV). The TPCV, designed for hot gas service, was analyzed from an engineering view point and considered satisfactory for further analysis, development and acceptance testing in order to prove its suitability as the DCV in the NRX/EST System. This design analysis report is the initial effort in the endeavor.

III OBJECTIVE

The objective of this design analysis was to determine the effects of cryogenic service and thermal and pressure stresses on the TPCV as an initial effort to qualify it for use as the DCV in the NRX/EST system.

IV SUMMARY AND CONCLUSIONS

The following areas were investigated:

- A. Steady state thermal and pressure stresses in the valve body.
- B. Steady state thermal and pressure stresses in the valve body adjacent to the end flange.
- C. Stress in the roller bearing outer race.
- D. Stress in the housing around the bearing.
- E. Stress in the long axle bolt.
- F. Bending stress in the valve gate.
- G. Bending stress in the axle. (adverse dimensional tolerance condition)
- H. Gate shift direction and distance under cryogenic service.
- I. Cryogenic mechanical properties of Kenametal K162B titanium carbide.

12 April 1965

Page 2

An operating pressure of 700 psi, and a temperature of -320°F were used in the analysis. Factors of safety used were 1.25 for yield strength and 1.50 for ultimate strength. All margins of safety calculated were positive. The roller bearing outer radius contracts 0.0009 inches minimum to 0.0013 inches maximum due to the housing thermally contracting more than the roller bearing. The housing thermal contraction forces are transmitted to the outer bearing race and imposes a stress in the outer race, however the internal bearing diametral clearance (0.0028 inches min. to 0.0049 inches max. reduced from manufacturers drawing) is not adversely affected. Thermal contraction @ -320°F operating temperature and deflection due to a pressure differential of 150 psi across the gate including pressure loading of the axle, bolt and lower roller bearing was calculated to result in 0.0009 inches downward gate shift, which is acceptable for operation.

An investigation was made of the cryogenic mechanical properties of Kennametal "Kentanium K162B", a titanium carbide base material containing nickel and molybdenum binders. Kennametal Corporation indicates that no cryogenic data was available. However, determinations were conducted at both Aerojet-General (Liquid Operations Cryogenics Laboratory) and Kennametal to determine some of the more important cryogenic properties. The data obtained from testing are shown in Table I.

Since the intended use of this material is for the races and rollers of a bearing to be used at -320°F , the bearing and impact properties were examined. It will be noted from Table I, Section VII, that the material has relatively low impact strength at room temperature, but that the impact strength at cryogenic temperature (-320°F) does not change significantly. The very low

coefficient of linear thermal expansion indicates that there will be little or no problem associated with seating the bearing in Inconel X-705 (AISI)688). Thermal cycling between room temperature and -320°F produced no effect on the material.

V DESCRIPTION AND FUNCTION OF COMPONENT

The TPCV, P/N 278100-69, is a 4-inch diameter butterfly valve and is a composite of eight major parts: body, gate shaft, axle, bolt, thrust washer bearings, and gate piston rings. The gate, shaft, axle and bolt are fabricated from Inconel X; the body is Inconel 718, bearings Kintanium 162-B, thrust washer carbon CDJ-83 and piston rings iron-base alloy A286. A photograph of the disassembled TPCV is shown in Figure 1.

The function of the TPCV in the proposed application as a DCV is to control the supply of diluent fluid to the hot bleed port which in turn controls the turbine inlet gas temperature.

VI DESIGN CRITERIA

The following design criteria and data was used in the analysis:

A. Valve material - Inconel X (annealed)

Symbol

α_x coefficient of expansion = 5.6×10^{-6} inches/inch/ $^{\circ}\text{F}$.

E Modulus of elasticity = 31×10^6 psi

F_{Tu} Ultimate strength = 185,000 psi @ -320°F

F_{Ty} Yield strength = 110,000 psi @ -320°F

B. Bearing material - Kentanium 162 B

Symbol

α_e coefficient of expansion = 2.1×10^{-6} inches/inch/ $^{\circ}$ F.

E Modulus of elasticity = 59×10^6 psi

F_{cy} Yield strength = 200,000 psi @ ambient (conservative)

Poisson ratio = 0.236

C. Bolt material A-286

Symbol

α_b coefficient of expansion = 9.0×10^{-6} inches/inch/ $^{\circ}$ F.

E Modulus of elasticity = 31×10^6 psi

F_{ty} Yield strength = 110,000 psi @ -320° F

F_{tu} Ultimate strength = 185,000 psi @ -320° F

D. Gate and Axle material Inconel X (annealed) loading:

Operating pressure upstream 700 psi maximum, downstream 550 psi minimum.

Gas temperature -320° F (simulates DCV operational temperature).

E. Thrust washer - carbon CDJ-83

Symbol

α_i coefficient of expansion = 15×10^{-7} inch/inch/ $^{\circ}$ F

E Modulus of elasticity = 3.5×10^6 psi

F. Stress - Formulas

Thick wall, open end, cylinder

Rigid-flange: Reference F. J. Roark

VII ANALYSIS RESULTS

The detail stress and thermal analysis is presented in Enclosure (3).

The analysis results are tabulated in the table below:

Investigation Description	Enclosure 3 Ref. Page #	Margin of Safety	
		Yield	Ultimate
1. Steady state thermal & pressure stresses in the valve body		+ 5.6	+ 8.1
2. Steady state thermal & pressure stresses in the valve body adjacent to the end flanges		+ 2.62	<u>+ 4.07</u>
3. Stress in the roller bearing outer race		+ 1.58	*
4. Stress in the housing around the bearing		+ 2.03	<u>+ 3.25</u>
5. Stress in the long axle bolt		+ 0.72	+ 1.42
6. Bending stress in the valve gate		+ 2.86	+ 4.38
7. Bending stress in the axle (adverse tolerance conditions)		+ 1.42	<u>+ 2.41</u>

Additional Information:

A. The outer radius of the roller bearing contracts 0.0009 to 0.0013 inch due to low temperatures; pressure on the outer race is induced by the housing tending to contract more than the bearing.

B. Thermal contraction @ -320°F operating temperature and deflection due to a pressure differential of 150 psi across the gate results in 0.0009 inches downward gate shift.

* Compressive strength of the bearing is not well defined, $f_{yt} = 200,000$ psi is a minimum yield strength valve.

TABLE I

PROPERTIES OF KENNAMETAL "KENTANIUM K162B" AT CRYOGENIC TEMPERATURES

A. THERMAL EXPANSION PROPERTIES AS DETERMINED BY LRO-AGC

<u>T(°F)</u>	<u>L/L₀ (10⁻⁵in/in)</u>	<u>(10⁻⁶in/in-°F)</u>
71	0.0	2.86
50	- 6.1	2.85
0	-20.0	2.82
- 50	-33.7	2.78
-100	-47.0	2.73
-150	-59.5	2.67
-200	-70.2	2.56
-250	-78.8	2.43
-300	-85.5	2.29
-320	-87.5	2.23

THERMAL EXPANSION PROPERTIES AS DETERMINED BY KENNAMETAL

<u>T(°F)</u>	<u>L/L₀ (10⁻⁵in/in)</u>	<u>(10⁻⁶in/in-°F)</u>
32	0.0	2.86
-18.5	-14.0	2.78
-123	-39.0	2.52
-173	-50.0	2.44
-198	-55.0	2.39
-231	-61.0	2.32
-320	-74.0	2.10

TABLE I (Continued)

IMPACT PROPERTIES (KENNAMETAL) - AVERAGE OF FIVE SPECIMENS

<u>TEMPERATURE</u>	<u>DROP HEIGHT FOR FRACTURE (INCHES)</u>
Room	10.6"
-320°F	8.5"

TRANSVERSE MODULUS OF RUPTURE - KENNAMETAL - FOUR POINT LOADING

<u>TEMPERATURE (°F)</u>	<u>MOD. RUPTURE (KSI)</u>
Room	216 (avg. 4 Tests)
-320	269 (avg. 5 Tests)

EFFECTS OF THERMAL CYCLING

- (1) AGC performed thermal cycling tests between room temperature and -320°F 106 times. No changes of any kind were noted.
- (2) Kennametal performed thermal cycling tests between room temperature and -320°F 70 times. No changes of any kind were noted.

Disassembled Turbine Power Control
Valve - Fig 1 (u)

PARTIAL VIEW - TPCV COMPONENTS

FIGURE 2

NERVA PROGRAM

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REPORT NO.

PAGE 1 OF 11

DATE

2-4-65

WORK ORDER

DATE

AGC 2-844

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES -
-320°F AND PRESSURE - 700 psi

BY

E. Lueders

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ANALYSIS

1. Thermal Stresses in the Valve Body (Inconel X)

From the preliminary thermal analysis the temperatures at two typical nodes (temp. points) at the I.D. and O.D. of the valve body are:

Node 34 = 291°R } after 100 sec H₂ flow at 700 psig and -320°F (steady state)
118 = 248°R }

REF: ENCLOSURE (4)

The maximum steady temperature gradient.

$$TP_{34} - TP_{118} = 291 - 248 = 43^\circ$$

The elastic axial (∇_z) and hoop (∇_θ) thermal stresses

$$\nabla_{z_t} = \nabla_{\theta_t} = \frac{E \alpha}{2(1-\nu)} (\Delta T) = \frac{31 \times 10^6 \times 5.6 \times 10^{-6}}{2(1-.3)} (43)$$

$$\nabla_{z_t} = \nabla_{\theta_t} = \underline{\underline{5,340 \text{ psi}}}$$

(Assume this is the thermal stress throughout the valve body)

Pressure stresses in the valve body (REFERENCE B)

$$\text{Hoop } \nabla_{\theta p} = \frac{pR}{t} = \frac{700 \times 2.160}{.185} = \underline{\underline{8180 \text{ psi}}}$$

Where: R = Valve Body Radius, t = Thickness = .185

$$\text{Axial } \nabla_{z p} = \frac{pR}{2t} = \underline{\underline{4090 \text{ psi}}}$$

Maximum bending stress at the flange edge is:

$$\nabla_b = 1.82 \frac{pR}{s} = \underline{\underline{14,900 \text{ psi}}} \quad \left(\begin{array}{l} \text{Assuming rigid flange:} \\ \text{REFERENCE: R. J. Roark} \end{array} \right)$$

AGC 2-844

REPORT NO.

PAGE 2 OF 11

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES -
-320°F AND PRESSURE - 700 psi

DATE

2-4-65

WORK ORDER

BY

E. Lueders

CHK. BY

DATE

Total stresses in the valve body

Total hoop stress in valve body

$$\sigma_{\theta Tot} = \sigma_{\theta t} + \sigma_{\theta p} = 5340 + 8180 = \underline{\underline{13,520 \text{ psi}}}$$

$$M.S. = \frac{F_{ty}}{S.F. (\sigma_{\theta Tot})} - 1 \quad (F_{ty} = 110,000 \text{ psi at } -320^\circ\text{F})$$

$$M.S._y = \frac{110,000}{1.25 (13,520)} - 1 \quad M.S._u = \frac{185,000}{1.50 (13,520)} - 1 = \underline{\underline{+8.1}}$$

$$M.S._y = \underline{\underline{+5.6}}$$

2. Total axial stress in the valve body at the point adjacent to the end flanges.

$$\sigma_{z Tot} = \sigma_{z t} + \sigma_{z p} + \sigma_b$$

$$= 5340 + 4090 + 14,900$$

$$\sigma_{z Tot} = 24,330 \text{ psi}$$

$$\text{Yield } M.S._y = \frac{F_{ty}}{S.F. (\sigma_{z Tot})} - 1 = \frac{110,000}{1.25 (24,330)} - 1$$

$$\underline{\underline{M.S._y = +2.62}}$$

$$\text{Ultimate } M.S._u = \frac{F_{ty}}{S.F. (\sigma_{z Tot})} - 1 = \frac{185,000}{1.5 (24,330)} - 1 = \underline{\underline{+4.07}}$$

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ENCLOSURE (3)

AGC 2 844

REPORT NO.

PAGE 3 OF 11

DATE

2-4-65

WORK ORDER

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES -
-320°F AND PRESSURE * 700 psi

BY

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DATE

3. Roller Bearing Stresses

Housing

Bearing

The minimum clearance between the bearing and housing is 0.000. Since the housing coefficient of expansion is greater than that of the bearing, a contact pressure (P_o) is developed as the two parts contract. This is determined as follows:

$$\Delta h = \Delta b$$

Where, Δh = Radial Displacement of the Housing

Δb = Radial Displacement of the bearing

NOTE: First assume the temperature is -320°F., and the bearing is solid. Next, assume the bearing is composed of only the outer race.

AGC 2-844

REPORT NO.

PAGE 4 OF 11

DATE

2-4-65

WORK ORDER

DATE

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC
TEMPERATURES - -320°F AND PRESSURE - 700 psi

BY

CHK. BY

$$(1) \Delta_h = \alpha_h r_2 \Delta T - p_c \frac{r_2}{E_h} \left(\frac{r_1^2 + r_2^2}{r_1^2 - r_2^2} + \nu \right)$$

$$(2) \Delta_b = \alpha_b r_2 \Delta T + p_c \frac{r_2}{E_b} \left(\frac{r_2^2 + r_3^2}{r_2^2 - r_3^2} - \nu_b \right)$$

$$r_1 = 1.25$$

$$r_2 = .750$$

$$r_3 = \frac{.756}{2} = .378$$

$$(1) \stackrel{\text{Thus}}{=} (2) \alpha_h r_2 \Delta T - p_c \frac{r_2}{E_h} \left(\frac{r_1^2 + r_2^2}{r_1^2 - r_2^2} + \nu \right) = \alpha_b r_2 \Delta T + p_c \frac{r_2}{E_b} \left(\frac{r_2^2 + r_3^2}{r_2^2 - r_3^2} - \nu_b \right)$$

$$\Delta T (\alpha_h - \alpha_b) = p_c \left[\left(\frac{r_1^2 + r_2^2}{r_1^2 - r_2^2} + \nu \right) \frac{1}{E_h} + \left(\frac{r_2^2 + r_3^2}{r_2^2 - r_3^2} - \nu \right) \frac{1}{E_b} \right]$$

$$(-320 - 80)(5.6 - 2.1) = p_c \left[\left(\frac{(1.25)^2 + (.75)^2}{(1.25)^2 - (.75)^2} + .3 \right) \frac{1}{31} + \left(\frac{.75^2 + .378^2}{.75^2 - .378^2} - .24 \right) \frac{1}{59} \right]$$

$$p_c = \frac{1420}{.1037} = 13,640 \text{ psi}$$

Bearing Stresses

$$\nabla_{\theta} = p_c \left(\frac{2r_1^2}{r_2^2 - r_3^2} \right) = 13,640 \times 2 \left(\frac{.75^2}{(.75)^2 - (.378)^2} \right)$$

$$\nabla_{\theta} = 36,600 \text{ psi Compression}$$

$$10^6 \Delta_b = 2.1 \times .750 \times 400 + 13,640 \times \frac{.75}{.59} (1.68 - .24)$$

$$= 630 + 250$$

$$\text{Min } \Delta_b = .0009 \text{ in}$$

$$\text{Bearing M.S. } \frac{F_{yc}}{S.F. \times \nabla_{\theta}} - 1 = \frac{200,000}{1.25 \times 36,600} - 1$$

$$M.S.y = +3.38 \text{ (See page 6 for Min M.S.)}$$

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AGC 2-844

REPORT NO

PAGE 5 OF 11

DATE

2-4-65

WORK ORDER

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES
-320°F AND PRESSURE 700 psi

BY

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CHK. BY

DATE

4. Housing Stress around the bearing:

$$\nabla_{\theta h} = P_c \left(\frac{r_1^2 + r_2^2}{r_1^2 - r_2^2} \right) = 13,640 (2.124)$$

$$\nabla_{\theta h} = \underline{\underline{29,000 \text{ psi}}}$$

$$M.S.y = \frac{Fty}{S.F. (\nabla_{\theta h})} - 1 = \frac{110,000}{1.25(29,000)} - 1 = \underline{\underline{+ 2.03}}$$

$$M.S.u = \frac{Fty}{S.F. (\nabla_{\theta h})} - 1 = \frac{185,000}{1.50 (29,000)} - 1 = \underline{\underline{+ 3.25}}$$

2-4-65

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES
-320°F AND PRESSURE 700 psi

BY

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If the outer race is considered to be the only portion of the bearing which resists the contact pressure r_3 Page 4, becomes .673 in.

$$(1) + (2) \text{ Thus } 1400 = P_c \left(.0783 + \left(\frac{.75^2 + .673^2}{.75^2 - .673^2} - .24 \right) 1/59 \right)$$

$$P_c = 6100 \text{ psi}$$

Bearing Stress

$$\nabla_{\theta_b} = P_c \left(\frac{2r_2^2}{r_2^2 - r_3^2} \right) = 6100 \left(\frac{2(.75)^2}{.75^2 - .633^2} \right) = 62,100 \text{ psi}$$

at 0.0 of bearing

Housing at the inner radius

$$\nabla_{\theta_h} = P_c \left(\frac{r_1^2 + r_2^2}{r_1^2 - r_2^2} \right) = 6,100 \times 2.124 = 12,900 \text{ psi}$$

$$(2) \quad 10^6 \Delta_b = \alpha_b r_2 \Delta T + P_c \frac{r_2}{E h} \left(\frac{r_2^2 + r_3^2}{r_2^2 - r_3^2} - \nu_b \right)$$

$$= 2.1 \times .75 \times 400 + 6100 \times \frac{.75}{59} (9.18 - .24)$$

$$= 630 + 692$$

$$\text{Max } \Delta_b = .0013$$

Margin of Safety in the bearing outer race

$$\text{Min M.S.}_y = \frac{F_{cy}}{S.F. (\nabla_{\theta_b})} - 1 = \frac{200,000}{1.25 (62,100)} - 1 = +1.58$$

AGC 2-844

REPORT NO.

PAGE 7 OF 11

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC
TEMPERATURES -320°F AND PRESSURE 700 psi

DATE

2-4-65

WORK ORDER

BY

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DATE

5. Bolt Stresses

Bolt Material: CRES A286 3/8 - 24

$$d = 9.0 \times 10^{-6}$$

$$E = 31 \times 10^6$$

$$\left. \begin{array}{l} F_{ty} = 110,000 \\ F_{ty} = 184,000 \end{array} \right\} \text{ at } -320^\circ\text{F}$$

Axle Material: Inconel X

$$d = 6.4 \times 10^{-6} \quad E = 31 \times 10^6$$

As the bolt tends to contract more than the axle, tensile stresses are set up in the bolt. The load in the bolt is determined as follows:

$$\Delta_{\text{bolt}} = \Delta_{\text{axle}}$$

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REPORT NO.

AGC 2-844

PAGE 8 OF 11

DATE

2-4-65

WORK ORDER

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-500 FOR CRYOGENIC TEMPERATURES
-320°F and PRESSURE - 700 psi

BY

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$$\Delta_b = \alpha_b L_b \Delta T - \frac{PL_b}{A_b E_b}$$

$$A_b = \frac{\pi}{4} (.375)^2 = .1102$$

$$\Delta_a = \alpha_a L_a \Delta T + \frac{PL_a}{A_a E_a}$$

$$A_a = \frac{\pi}{4} (.75^2 - .39^2) = .322$$

$$L_a = 3.10$$

$$L_b = 3.63$$

$$\Delta_b = \Delta_a$$

$$\alpha_b L_b \Delta T - \frac{PL_b}{A_b E_b} = \alpha_a L_a \Delta T + \frac{PL_a}{A_a E_a}$$

$$\Delta T (\alpha_b L_b - \alpha_a L_a) = P \left(\frac{L_a}{A_a E_a} + \frac{L_b}{A_b E_b} \right)$$

$$(-320 - 80)(9.0 \times 3.63 - 5.6 \times 3.10) = P \left(\frac{3.10}{.322 \times 31} + \frac{3.63}{.1102 \times 31} \right)$$

$$6120 = P (.3105 + 1.062)$$

$$P = 4460 \text{ lb.}$$

The stress area for the $\frac{3}{8}$ -24 bolt is:

$$A_{b-s} = .0876 \text{ in.}^2$$

The tensile stress in the bolt is,

$$\text{Max. } \sigma_{b-t} = \frac{P}{A_b} = \frac{4,460}{.0876} = \underline{\underline{51,000 \text{ psi}}}$$

The Margin of Safety in the bolt,

$$\text{Yield M.S.}_y = \frac{F_{ty}}{S.F. \sigma_{b-t}} - 1 = \frac{110,000}{1.25 (51,000)} - 1 = \underline{\underline{+0.72}}$$

$$\text{Ultimate M.S.}_u = \frac{F_{tu}}{S.F. (\sigma_{b-t})} - 1 = \frac{185,000}{1.50 (51,000)} - 1 = \underline{\underline{+1.42}}$$

SUBJECT

STRESS ANALYSIS OF TPCV P/W 278100-59 FOR CRYOGENIC TEMPERATURES
-320°F AND PRESSURE - 700 psi

DATE

2-4-65

WORK ORDER

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6. Analysis of the valve gate and bearing assembly.

Bending stress in the valve gate.

On Page 17 of Reference (b) this stress was 18,000 psi, based on 550 psi

The stress for pressure - 700 psi is:

$$\nabla_{bb} = 18,000 \times \frac{700}{550} = 22,900 \text{ psi}$$

$$M.S._{by} = \frac{F+y}{S.F.(\nabla_{bb})} - 1 = \frac{110,000}{1.25(22,900)} - 1$$

$$\underline{M.S._{by} = 2.86}$$

$$M.S._{bu} = \frac{F+y}{S.F.(\nabla_{bb})} - 1 = \frac{185,000}{1.5(22,900)} - 1 = \underline{+4.38}$$

AGC 2-843

REPORT NO.

PAGE 10 OF 11

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES
-320°F AND PRESSURE - 700 psi

DATE

2-4-65

WORK ORDER

BY

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CHK. BY

DATE

7. Bending stress in the axle.

On Page 24 of Reference (b) the bending stress in the axle is calculated to be 35,500 psi for an internal pressure of 550 psi.

The stress for pressure - 700 psi is:

$$\nabla_{b-2} = 35,500 \pm \frac{700}{550} = 45,200 \text{ psi}$$

The margins of safety in the Inconel X axle considering plastic bending are:

$$\text{Yield M.S.}_y = \frac{K(Fty)}{S.F.(\nabla_{b-2})} - 1 = \frac{1.25(110,000)}{1.25(45,200)} - 1 = +1.42$$

$$\text{Ultimate M.S.}_u = \frac{1.25(Fty)}{S.F.(\nabla)} - 1 = \frac{1.25(185,000)}{1.50(45,200)} - 1 = +2.41$$

NOTE: This stress was calculated for adverse tolerance conditions.

AGC 2 843

SUBJECT

STRESS ANALYSIS OF TPCV P/N 278100-59 FOR CRYOGENIC TEMPERATURES
-320°F AND PRESSURE - 700 psi

DATE

2-4-65

WORK ORDER

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CHK BY

DATE

8. Contraction in bolt and axle (Δ_{ab})

$$\Delta_{ab} = \Delta_b = \alpha_b L_b \Delta T_{em} - \frac{PL_b}{A_b E_b} \quad (\text{See Page 8, P} = 4460 \text{ lb})$$

$$\Delta_{ab} = (9 \times 10^{-6} \times 3.63) (400) - \frac{4460(3.63)}{0.1102(31 \times 10^6)}$$

$$\Delta_{ab} = 0.008 \text{ Inches}$$

Bearing Contraction

$$\Delta_{be} = \alpha_b L_b \Delta T = (2.1 \times 10^{-6}) (1.0) (400) = 0.0008 \text{ Inches}$$

Gate Shift Downward, Δ_{Total}

$$\Delta_T = \Delta_{ab} - \Delta_{be} = 0.0080 - 0.0008 = 0.0072 \text{ Inches}$$

Gate shift due to operational loading (flow forces) is upward as shown in Figure 5.

$$\text{Gate Load, } F_2 = \Delta P A_b \sin \phi$$

Pressure differential across gate, $\Delta P = 150 \text{ psi}$

$$F_2 = 150 (11.7) (\sin 20^\circ) = 600 \text{ lbs.}$$

$$\text{Bolt deflection to gate load } \Delta_b = \frac{F_a L_b}{A_b E_b} = \frac{600(3.63)}{0.1102(31 \times 10^6)} = 0.0063''$$

$$\text{Bearing deflection inches compression } \Delta_{be} = \frac{F_a L_b}{A_b E_b} = \frac{600(1.0)}{\pi/4 (1.5^2 - 1.35^2) (59 \times 10^6)}$$

$$\text{Total gate shift} = \Delta_T - \Delta_b = 0.0072 - 0.0063 = 0.0009''$$

TPCV NODE LOCATION MAP

Y

NODE LOCATIONS
MAP FOR DCOTTING TRANSIENT
TEMPERATURE COMPUTER RUN
PREDICTIONS.

NERVA PROGRAM

£ X 10

TPCV FLOW LOADS

F_1 - RESULTANT GATE LOAD
DUE TO FLOW FORCES.

F_2 - GATE LOAD DUE TO GATE CANT.

F_3 - LOAD ON BEARING.