THERMAL STRESS ANALYSIS AND LIFE PREDICTION OF MSRE FREEZE FLANGES

C. H. Gabbard

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C. H. Gabbard

Molten-Salt Reactor Program

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C. H. Gabbard

I. ABSTRACT

Freeze flanges were included in the fuel and coolant piping systems of the Molten-Salt Reactor Experiment to provide the capability of remote removal and replacement of major components within the reactor cell. These freeze flanges were subjected to high cyclic thermal stresses during the various reactor operations because of their high thermal inertia and because of the high radial thermal gradient inherent in the design.

The suitability of the freeze flange design was ensured both by a comprehensive thermal stress analysis covering all phases of the reactor operation and by thermal-cycle testing a full scale prototype flange assembly. The thermal stress analysis indicated high thermal stresses in the flange bore, but the life prediction by low-cycle fatigue analysis showed that the life would be adequate for the intended service.

The prototype flange assembly was still serviceable when testing was discontinued after 540 cycles in the Thermal Cycle Facility. However, cracking and porosity were observed in the flange bore during the dye-penetrant inspection following 268 thermal cycles. The surface porosity type failure had been predicted at 300 cycles by the thermal stress analysis. The 540 thermal cycles of the prototype flange assembly without mechanical failure or leakage would permit the calculated allowable life of the MSRE flanges to be increased by a factor of 1.95.

The MSRE flanges completed up to 26,100 hours of high temperature salt exposure without indication of failure.

Keywords: *freeze flange, *MSRE, *stress, *thermal stress, *thermal-cycle testing, fused salts, reactors.

II. INTRODUCTION

The Molten Salt Reactor Experiment, which has been more fully described in Refs. 1 and 2, was successfully operated from June 1965 to December 1969. This was a high temperature fluid fueled reactor that

operated at a reactor outlet temperature of 1210°F. The reactor system was designed in accordance with the procedures of the ASME Unfired Pressure Vessel Code, the applicable ASME Code Case Interpretations, and American Standard Code for Pressure Piping ASA B31.1. However, at that time these codes did not adequately cover the design of a reactor system operating at temperatures in the creep range. Therefore, prior to high temperature operation, the system design was reviewed to determine if there were any areas having excessively high cyclic thermal stresses where failures might occur.

This report covers the thermal stress analysis and the prediction of the thermal cycle life of the freeze flanges which were used to provide remote maintenance capability for the major items of equipment in the reactor containment vessel. The coolant piping anchor sleeves where the coolant piping penetrated the secondary containment vessel also required additional analysis. The anchor sleeves were modified to reduce the excessively high thermal stresses that were indicated by the analysis. The results of the anchor sleeve analysis and the subsequent design modifications were presented in an ORNL internal report.³

The freeze flange design had been tested to 103 cycles in a thermal cycle test facility prior to its acceptance as an MSRE component, and this facility was reactivated when it became apparent that the operating life of the MSRE would approach the predicted safe life of the flanges. The original pair of test flanges were subjected to a total of 540 thermal cycles, and surface cracking and porosity did occur in the bore of one of the flanges as predicted by the analysis. However, the cracking and porosity were not deep and the flanges were still serviceable at the conclusion of the test. This pair of freeze flanges is a relatively rare example of an incipient failure by high temperature thermal cycle fatigue under controlled and known conditions. At the time this analysis was completed, the various ASME code rules and analytical procedures in use today for creep range design either did not exist or were not generally accepted practices. We attempted to make a realistic but appropriately conservative analysis using elastic shell theory and the thermal fatigue data that was available at that time. This report covers the method of analysis that was used for these flanges, the method of life prediction,

and a brief comparison with the procedures currently specified by Section III of the ASME Boiler and Pressure Vessel Code and by code case 1331-5 covering nuclear vessels at high temperature.

III. DESCRIPTION OF THE FREEZE FLANGE DESIGN AND OPERATION

Freeze flanges were used in the MSRE to provide the capability of remotely disconnecting the major items of equipment within the secondary containment vessel of the reactor. A total of five freeze flange assemblies were used; two on the coolant piping and three on the fuel piping. Figure 1 is a cross sectional drawing of a freeze flange. A ring of frozen salt provided an absolute barrier to salt leakage and also prevented salt from contacting the sealing surfaces of the ring gasket. The leakage of radioactive gas was prevented by buffering the modified ring gasket with argon or helium at greater than salt system pressure. The buffer gas pressure also provided a convenient means of monitoring the flanges for an improperly seated or leaking gasket which would cause buffer gas leakage. A more detailed description of the flange assembly is presented on pages 178-189 of Ref. 2.

The flanges were fabricated of Hastelloy "N" which was the material of construction for all the salt containing parts of the reactor system. The nominal composition and the more important physical and mechanical properties are listed in Table 1. The compositions and physical properties of the two molten salts used in the reactor system are shown in Table 2.

The requirement of a ring of frozen salt inboard of the ring gasket caused a steep radial temperature gradient in the flanges. The temperature must decrease from the operating salt temperature of 1210°F at the bore to below the salt freezing point (850°F) at a radius of about 5 5/8 in. The exterior flange surfaces were uninsulated, and the heat losses to the containment vessel atmosphere provide the thermal gradient necessary to form the ring of frozen salt.

This steep radial thermal gradient plus the high thermal inertia of the flanges caused high thermal stresses and large stress cycles during the various operations of the reactor. Three phases of reactor operation

FIGURE 1. CROSS SECTION OF 5-IN. FREEZE FLANGE USED IN THE MSRE

Table 1. Composition and Physical Properties
of Hastelloy "N"

Chemical Composition		<u>%</u>	
Nickel		66-71	
Molybdenum		15-18	
Chromium		6-8	
Iron		5	
Modulus of Elasticity			
Temperature, °F		<u>E(psi)</u>	
60		31.7 x 10 ⁶	
1070		26.3	
1170		26.2	
1290		24.8	
Coefficient of Thermal Expansion			
Temperature, °F		<u>$\frac{\Delta L}{L-\text{°F}}$</u>	
70-400		6.45 x 10 ⁻⁶	
70-800		7.09	
70-1000		7.43	
70-1200		7.81	
Mechanical Properties			
<u>Temp., °F</u>	<u>Tensile Stress, psi</u>	<u>Yield Stress psi</u>	<u>Design Stress for Primary Stresses</u>
100	100,000	40,000	25,000
600	90,800	30,000	20,000
1000	82,000	26,500	17,000
1200	68,400	25,200	6,000
1300	60,000	24,600	3,500

Table 2. Composition and Physical Properties of MSRE
Fuel and Coolant Salts at Operating Temperature

Composition (mole %)	Fuel	Coolant
LiF	65	66
BeF ₂	29.1	34
ZrF ₄	5	-
UF ₄	0.9	-
Density, (lb/ft ³)	141.2	123.1
Specific Heat, [Btu/(lb-°F)]	0.47	0.58
Viscosity, [lb/(ft-hr)]	18.7	23.6
Thermal Conductivity, [Btu/(hr-ft-°F)]	0.83	0.66

were identified which would cause temperature and stress cycling of the freeze flanges, and since these operations could occur in different sequences, they were analyzed separately. The entire salt system was heated to nominally 1200°F prior to filling to avoid freezing the salt on any cold surfaces. However, the temperature in the flange bore reached only 550 to 600°F because of the intentional heat losses from the flange. The salt temperature during filling was also maintained at about 1200°F to avoid thermal shock to the system. However, the freeze flange bore surfaces did receive a thermal shock during each fill because of the temperature depression when the system was empty. Two of the freeze flanges on the coolant system piping were also subject to temperature and stress cycling during power changes of the reactor as the coolant system temperature was decreased for heat removal. The three stress cycles used in the analysis were defined as follows:

1. Heating Cycle - Heating the reactor system from room temperature to 1200°F and returning to room temperature.
2. Fill Cycle - Filling the heated system with salt, establishing normal circulation at 1225°F, and draining the salt back to the drain tank with the system piping at 1200°F.

3. Power Cycle - Increasing the reactor power from zero to maximum and returning to zero. This was equivalent to lowering the cold leg salt temperature 200°F for the coolant system and about 50°F for the fuel system and returning.

The analysis assumed equilibrium temperatures and stresses were reached at each end point condition for each cycle.

The final design of the freeze flanges evolved over a period of several years and the thermal cycling tests that were performed were aimed at demonstrating the mechanical integrity, the leak tightness, and the ability to remotely open and reseal the flanges. The flanges were also checked for dimensional changes and cracking after thermal cycling. A stress analysis was completed to ensure the flanges were adequate for the forces and moments that would be applied by the 5-in. piping. A thermal stress analysis had also been completed previously which indicated stresses in the plastic region but not of sufficient degree to question the life of the flange. However, this previous analysis which required the solution of several simultaneous equations was completed without the benefit of a computer, and the compatibility conditions between the various sections of the flange were probably not sufficiently accurate to give realistic results. Therefore, the analysis described in this report was undertaken to more accurately evaluate the stress cycling conditions of the flanges.

IV. CALCULATION PROCEDURES

The procedure for estimating the life of a component subjected to cyclic thermal stresses requires several steps as outlined in Fig. 2. The end result of the analysis is the number of cycles of each type which can be safely tolerated without danger of failure. To estimate this number, the total plastic strain for each cycle was evaluated and compared to a curve of plastic strain range vs allowable cycles. The first step in making an analysis of this type is the determination of the transient temperature distributions of the component for the various operating cycles.

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FIGURE 2. DIAGRAM OF CALCULATION PROCEDURE FOR THERMAL STRESS AND FATIGUE LIFE ANALYSIS

IV-1. Transient Temperature Analysis

The transient temperature analysis for each of the three types of operational cycle varied somewhat because of the nature of the cycle. The heating cycle of the reactor was relatively slow and a steady state temperature distribution in the heated condition was adequate to define the cycle. During this cycle the reactor system was heated from room temperature to 1200°F with helium at 5 psig circulating in the fuel and coolant systems. When the system was heated in this manner the bore temperatures of the freeze flanges were lower than 1200°F because of the heat losses from the flange. The pipe line temperatures dropped from 1200°F at the nearest heater to about 550°F at the midplane of the flange.

When the reactor was filled, salt at 1200°F was pushed into the reactor system by gas pressure in the drain tanks. The freeze flange bore received a thermal shock when contacted with the 1200°F salt. Subsequent analysis showed that the bore surface did not reach full salt temperature until salt circulation with the salt pump was started and that a less severe stress cycle would occur if the forced circulation of salt was delayed until a portion of the temperature and stress transient had been completed. The fill transient consisted of the salt fill at 1200°F, starting salt circulation after an appropriate hold time, and then heating the system to 1225°F. The reactor operating procedures were later modified to limit the maximum normal system temperature to 1210°F, but since this was a conservative change the temperature and stress analyses were not revised.

The reactor power change cycle was somewhat more difficult because the temperature transients within the salts had to be established for the various modes of operation prior to the transient analysis of the freeze flange. The temperature transients in the salt were analyzed by an Analog computer model of the reactor system.

The steady state and transient temperature distributions of the flange were computer calculated using the ASTRA HEATING code.⁴ This code is a thermal conduction code with provision for coupling regions with different thermal properties and for assigning various boundary conditions. Figure 3 shows the flange layout used for these calculations. The outside surfaces

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FIGURE 3. FREEZE FLANGE GEOMETRY USED IN HEATING CODE STEADY STATE AND TRANSIENT TEMPERATURE ANALYSIS

of the flange were coupled to the reactor cell ambient by convective and radiative heat transfer coefficients. These coefficients were first estimated using conventional procedures and were then adjusted until the calculated temperature distribution was the same as that observed on the prototype flange in the thermal cycle facility. The various transients were introduced into the freeze flange by changing the heat transfer coefficient at the flange bore or by assigning a surface temperature to the bore that could be varied as a function of time.

A typical set of curves for the transient temperature distribution following a fill are shown in Fig. 4. These curves show the rapid increase in bore temperature followed more slowly by the temperatures at the outer diameter of the flange. The curves also show the effect of starting salt circulation and the effect of the hold time. The flange bore would effectively see a temperature increase about 90°F higher if circulation were started immediately.

The various radial temperature distributions obtained from the HEATING code were reduced to polynomial equations to simplify their use in a stress program that was being written. Previous work with somewhat similar geometry had shown that an equation of the form below would properly represent the data for radial heat conduction in a cylindrical fin.

$$T(r) = A + B/r + Cr + Dr^2 + Er^3$$

All the temperature distributions from the HEATING code were fitted to this form of equation by a least squares curve fitting program. The circled points shown on two of the curves on Fig. 4 were calculated by equations of this form and show the excellent representation obtained by these equations. The solid lines are the temperature distributions calculated by the HEATING program.

IV-2. Stress Analysis

A computer program was written to calculate the thermal and pressure stresses as a function of position in the flange and its neck. The analysis included the effects of the radial temperature gradient, the internal

FIGURE 4. TRANSIENT RADIAL TEMPERATURE DISTRIBUTIONS OF MSRE FREEZE FLANGE DURING A SALT FILL

pressure and the various internal forces and moments associated with coupling the neck to the flat disk. The analysis ignored the external forces and moments exerted on the flange from the piping system because these had been considered previously and produced negligible stresses at the locations of interest in this analysis.

The radial and tangential thermal stresses in a flat disk with a radial non-linear temperature gradient are given by the equations⁵:

$$S_{tr}(r) = -\frac{\alpha E}{r^2} \int_a^r r T(r) dr + \frac{E}{1-\nu^2} \left[C' (1+\nu) - C'' (1-\nu) \frac{1}{r^2} \right]$$

$$S_{t\theta}(r) = \frac{\alpha E}{r^2} \int_a^r r T(r) dr - \alpha E T(r) + \frac{E}{1-\nu^2} \left[C' (1+\nu) + C'' (1-\nu) \frac{1}{r^2} \right].$$

The radial deflection is given by the equation:

$$U_f(r) = \frac{(1-\nu)\alpha}{r} \int_a^r r T(r) dr + r C' + \frac{C''}{r}$$

The constants C' and C'' can be solved by using the boundary conditions that $S_{tr} = 0$ at the inside and outside radii of the flange. When the solutions for C' and C'' are substituted, the above equations reduce to:

$$S_{tr}(r) = -\frac{\alpha E}{r^2} \int_a^r r T(r) dr + \frac{E\alpha}{b^2-a^2} \left[1 - \frac{a^2}{r^2} \right] \int_a^b r T(r) dr$$

$$S_{t\theta}(r) = \frac{\alpha E}{r^2} \int_a^r r T(r) dr - \alpha E T(r) + \frac{E\alpha}{b^2-a^2} \left[1 + \frac{a^2}{r^2} \right] \int_a^b r T(r) dr$$

$$U_f(r) = -\frac{(1+\nu)\alpha}{r} \int_a^r r T(r) dr - \frac{r^2\alpha(1-\nu) + a^2\alpha(1+\nu)}{r(b^2-a^2)} \int_a^b r T(r) dr$$

Additional stresses are produced by internal pressure and by the forces and bending moments which occur because of the difference in radial thermal expansion between the pipe and the flange bore. These

stresses were calculated using elastic shell theory. Figure 5 shows the flange geometry and sign convention used in calculating these additional stresses. A total of seven equations describing the boundary and compatibility conditions of the pipe and flange were required to calculate the stresses. The tapered flange neck was assumed to be an infinitely long pipe with a wall thickness of 0.875 in. and the pipe was assumed to be at uniform temperature. Actually a thermal gradient existed on the flange neck when the system was heated and empty, but the stress program was already written at the time this was recognized and a major revision would have been required to include this thermal gradient. A conservative assumption was made that the flange neck was at the nominal piping system temperature. The geometry of the actual flanges provided a generous fillet at the pipe-to-flange connection and the stress concentration factor was assumed to be 1.

The equations describing the compatibility and boundary conditions for the two members are given below.

A. At Pipe-to-Flange Joint

1. $\Sigma \text{ Moments} = 0$ EQ-1
2. $\Sigma \text{ Radial Forces} = 0$ EQ-2
3. $U_c - u_c = U_f + u_f$ EQ-3
4. $du_c/dx = dw/dr$ EQ-4

B. At the Flange OD

1. The Axial Displacement = 0 EQ-5
2. The Radial Force = 0 EQ-6
3. $dw/dr = 0$ EQ-7

U_c and U_f = radial displacements of pipe and flange due to thermal expansion. $U_c = a' \alpha T_{\text{sys}}$.

u_c and u_f = radial displacements of pipe and flange caused by the mechanical joining of the two members.

du_c/dx = angular displacement of pipe wall.

dw/dr = angular displacement of flange face.

The equations for the forces and moments and for the radial and angular deflections which were required for the solution of the above equations were obtained from Refs. 6 and 7.

FIGURE 5. GEOMETRY AND SIGN CONVENTION USED IN STRESS ANALYSIS
OF MSRE FREEZE FLANGES

For Cylinder:

$$M_x = \frac{1}{4a'\beta^2} \left[C_1 (-2e^{-y} \cos y) + C_2 (2e^{-y} \sin y) \right]$$

$$Q_r = \frac{1}{4a'\beta} \left[C_1 \left(-2e^{-y} (\cos y + \sin y) \right) + C_2 \left(2e^{-y} (\sin y - \cos y) \right) \right]$$

$$u_c = \frac{a'}{Et} \left[C_1 e^{-y} \sin y + C_2 e^{-y} \cos y - a'p + N_x v \right]$$

$$du/dx = \frac{a'\beta}{Et} \left[C_1 \left(e^{-y} (\cos y - \sin y) \right) + C_2 \left(-e^{-y} (\cos y + \sin y) \right) \right]$$

Where:

$$\beta = \left[\frac{3(1-\nu)^2}{a'^2 t^3} \right]^{1/4},$$

$$y = \beta x.$$

For Flange:

$$M_r = \frac{D}{b^2} \left[K (2.6 \ln R + 3.3) + \frac{3.3R^2 \rho}{4} + 0.65 C_3 - \frac{0.7C_4}{R^2} \right]$$

$$Q_r = \frac{Eh}{0.91b} \left[1.3 C_5 - \frac{0.7 C_6}{R^2} \right]$$

$$u_f = C_5 R + \frac{C_6}{R}$$

$$w = K (R^2 \ln R) + \frac{\rho R^4}{16} + \frac{C_5 R^2}{4} + C_6 (\ln R) + C_7$$

$$dw/dr = \frac{1}{b} \left[K (2R \ln R + R) + \frac{\rho R^3}{4} + \frac{C_5 R}{2} + \frac{C_6}{R} \right]$$

Where:

$$K = \text{axial edge load parameter} = - \left[\frac{ab^2}{4D} \right] \left(N_x - \frac{ap}{2} \right),$$

$$D = \text{flexural rigidity of plate} = Eh^3/10.92,$$

$$\rho = \text{internal pressure load parameter} = - \left[\frac{\rho b^4}{4D} \right].$$

The solution of these equations produced seven constants C_1 to C_7 which were then used in the appropriate force and moment equations to calculate the various bending and membrane stresses. Additional equations which were not necessary for the solution of the constants in the boundary and compatibility equations were required to compute the total stresses. The equations, given below, were also obtained from Refs. 6 and 7.

For Cylinder:

$$M_{\theta} = \nu M_r$$

$$N_{\theta} = - \left[C_1 e^{-y} \sin y + C_2 e^{-y} \cos y \right] + a'p$$

$$N_x = \frac{\pi a^2 p}{2\pi a'} = \frac{a^2 p}{2a'}$$

For Flange:

$$M_{\theta} = \frac{D}{b^2} \left[K (2.6 \ln R + 1.9) + \frac{1.9 R^2 \rho}{4} + 0.65 C_3 + \frac{0.7 C_4}{R^2} \right]$$

$$Q_{\theta} = \frac{Eh}{0.91 b} \left[1.3 C_5 + \frac{0.7 C_6}{R^2} \right]$$

The principal stresses were obtained by adding the thermal stresses in the flange disk to the membrane and bending stresses due to the mechanical coupling of the two members.

The principal stresses are:

For Cylinder

$$S_x = \frac{N_x}{t} \pm \frac{6 M_x}{t^2}$$

$$S_{\theta} = \frac{N_{\theta}}{t} \pm \frac{6 M_{\theta}}{t^2}$$

$$S_r = 0$$

For Flange

$$S_x = 0$$

$$S_{\theta} = \frac{Q_{\theta}}{h} \pm \frac{6 M_{\theta}}{h^2} + S_{t\theta}$$

$$S_r = \frac{Q_r}{h} \pm \frac{6 M_r}{h^2} + S_{tr}$$

When a biaxial or triaxial stress condition exists, the usual procedure is to use the stress intensity with the normal design fatigue curves. The stress intensity is the largest of the three quantities:

$$S_x - S_\theta$$

$$S_\theta - S_r$$

$$S_x - S_r$$

The computer program calculated the radial and circumferential bending and membrane stresses at various locations in the flange and then combined these into the principal stresses and into the stress intensity at each location. A total of twelve transient conditions for the MSRE reactor system plus the transient for the thermal cycle facility were investigated with about 11 to 12 time increments per transient.

Table 3 shows a typical set of computer output from the stress program. These stresses were calculated for the steady-state condition at zero power with salt circulating at 1225°F.

IV-3. Freeze Flange Life Prediction

The thermal stresses that were calculated for the freeze flanges were considerably greater than those usually encountered. Fortunately these were thermal stresses which would be relieved by yielding and relaxation of the Hastelloy "N" without actual failure of the flange. The cyclic nature of these stresses during the various operations of the reactor required that the allowable life of the freeze flange be determined by a low-cycle fatigue analysis. The primary stresses were sufficiently low to avoid any stress rupture or creep problems.

The various design and fabrication codes and standards which were available at the time of the analysis were not adequate to establish a design procedure which would define the allowable life. The following procedure was used as an effort to obtain a realistic but appropriately conservative life estimate. The complete thermal stress history of the freeze flange was calculated for each operational cycle as outlined previously in this report. The point of maximum stress cycling was found to be the corner of the inside flange face and the bore.

A typical stress-strain history of these points during a fill-drain cycle is shown in Fig. 6. Table 4 is a summary of the plastic strain range per cycle for the cycles of interest in the life prediction. The

Table 3. Typical Output from Thermal Stress Program Case 2-1 Steady State Operating Temperature

MSRE FREEZE FLANGE TRANSIENT THERMAL STRESS							
E	ALPHA	MU	IN RAD	OUT RAD	FL THK	CY THK	PRESS
0.2900E 08	0.7810E-05	0.3000E-00	0.2525E 01	0.1156E 02	0.1500E 01	0.8750E 00	0.8000E 02
CASE DIST THER CISP							
2	1	-0.9367E-02					
TEMP COEF	0.1742E 04	0.2298E 03	-0.2955E 03	0.2325E 02	-0.6253E 00		
EQ CCNST	0.1094E 04	C.	0.9392E-02	-0.1450E-04	-0.2491E-02	0.	-0.9962E-02
INT CCNST	0.3244E 05	0.6342E 05	-0.3954E-01	0.9808E-02	0.2650E-03	0.4921E-03	0.7395E-02
CYLINDER STRESSES							
Y	X	X BEND	CIR BEND	X MEMB	C MEMB		
0.	0.	-0.6732E 05	-0.2020E 05	0.9838E 02	-0.7220E 05		
0.1000E 01	0.1253E 01	0.2735E 05	0.8206E 04	0.9838E 02	-0.2561E 05		
0.2000E 01	0.2505E 01	0.1998E 05	0.5995E 04	0.9838E 02	-0.2102E 03		
0.3000E 01	0.3758E 01	0.4243E 04	0.1273E 04	0.9838E 02	0.3583E 04		
0.4000E 01	0.5010E 01	-0.1018E 04	-0.3054E 03	0.9838E 02	0.1652E 04		
0.5000E 01	0.6263E 01	-0.9789E 03	-0.2937E 03	0.9838E 02	0.3719E 03		
FLANGE STRESSES		RAC	RA THER	CIR THER			
		0.2525E 01	-0.	-0.1348E 06			
		0.3428E 01	-0.2572E 05	-0.6805E 05			
		0.4332E 01	-0.3036E 05	-0.3089E 05			
		0.5235E 01	-0.2826E 05	-0.7142E 04			
		0.6139E 01	-0.2391E 05	0.8771E 04			
		0.7042E 01	-0.1898E 05	0.1940E 05			
		0.7946E 01	-0.1420E 05	0.2622E 05			
		0.8849E 01	-0.9851E 04	0.3020E 05			
		0.9753E 01	-0.6037E 04	0.3215E 05			
		0.1066E 02	-0.2768E 04	0.3271E 05			
		0.1156E 02	-0.9766E-03	0.3250E 05			
DIM RAD	RAC	RA BEND	CIR BEND	RA MEMB	CIR MEMB		
0.2184E-00	0.2525E 01	-0.3005E 05	0.2130E 05	-0.1895E 05	0.2085E 05		
0.2966E-00	0.3428E 01	-0.1804E 05	0.9661E 04	-0.9846E 04	0.1175E 05		
0.3747E-00	0.4332E 01	-0.1252E 05	0.4623E 04	-0.5813E 04	0.7712E 04		
0.4529E-00	0.5235E 01	-0.9377E 04	0.2084E 04	-0.3680E 04	0.5579E 04		
0.5311E 00	0.6139E 01	-0.7293E 04	0.7121E 03	-0.2418E 04	0.4317E 04		
0.6092E 00	0.7042E 01	-0.5723E 04	-0.3176E 02	-0.1609E 04	0.3508E 04		
0.6874E 00	0.7946E 01	-0.4418E 04	-0.3983E 03	-0.1060E 04	0.2959E 04		
0.7655E 00	0.8849E 01	-0.3246E 04	-0.5175E 03	-0.6708E 03	0.2570E 04		
0.8437E 00	0.9753E 01	-0.2137E 04	-0.4619E 03	-0.3845E 03	0.2284E 04		
0.9218E 00	0.1066E 02	-0.1045E 04	-0.2749E 03	-0.1679E 03	0.2067E 04		
1.0000E 00	0.1156E 02	0.5514E 02	0.1654E 02	-0.3009E-04	0.1899E 04		

Table 3 Contd.

CYLINDER STRESS INTENSITY

Y	X	PR X IS	PR CIR IS	ST INT IS	PR X OS	PR CIR OS	ST INT OS
0.	0.	0.6742E C5	-0.5201E 05	0.1194E 06	-0.6722E 05	-0.9240E 05	0.2518E 05
0.1000E 01	0.1253E 01	-0.2726E 05	-0.3382E 05	0.6564E 04	0.2745E 05	-0.1741E 05	0.4486E 05
0.2000E 01	0.2505E 01	-0.1589E 05	-0.6206E 04	-0.1368E 05	0.2038E 05	0.5785E 04	0.1430E 05
0.3000E 01	0.3758E 01	-0.4144E C4	0.2310E 04	-0.6454E 04	0.4341E 04	0.4855E 04	-0.5143E 03
0.4000E 01	0.5010E 01	0.1116E C4	0.1958E 04	-0.8415E 03	-0.9177E 03	0.1347E 04	-0.2267E 04
0.5000E 01	0.6263E 01	0.1077E C4	0.6656E 03	0.4117E 03	-0.8835E 03	0.7824E 02	-0.9587E 03

FLANGE STRESS INTENSITY

MM RAD	RAD	PR RA IS	PR CIR IS	ST INT IS	PR RAD OS	PR CIR OS	ST INT OS
0.2184E-00	0.2525E 01	0.1110E 05	-0.1352E 06	0.1463E 06	-0.4931E 05	-0.9263E 05	0.4362E 05
0.2966E-00	0.3428E 01	-0.1753E 05	-0.6597E 05	0.4844E 05	-0.5351E 05	-0.4665E 05	-0.6963E 04
0.3747E-00	0.4332E 01	-0.2366E C5	-0.2780E 05	0.4141E 04	-0.4869E 05	-0.1855E 05	-0.3013E 05
0.4529E-00	0.5235E 01	-0.2256E 05	-0.3647E 04	-0.1892E 05	-0.4132E 05	0.5217E 03	-0.4184E 05
0.5311E 00	0.6139E 01	-0.1903E C5	0.1238E 05	-0.3141E 05	-0.3352E 05	0.1380E 05	-0.4742E 05
0.6092E 00	0.7042E 01	-0.1487E C5	0.2294E 05	-0.3781E 05	-0.2632E 05	0.2288E 05	-0.4920E 05
0.6874E 00	0.7946E 01	-0.1084E 05	0.2957E 05	-0.4042E 05	-0.1968E 05	0.2878E 05	-0.4846E 05
0.7655E 00	0.8849E 01	-0.7275E C4	0.3329E 05	-0.4057E 05	-0.1377E 05	0.3226E 05	-0.4602E 05
0.8437E 00	0.9753E 01	-0.4285E C4	0.3489E 05	-0.3918E 05	-0.8558E 04	0.3397E 05	-0.4253E 05
0.9218E 00	0.1066E 02	-0.1891E C4	0.3505E 05	-0.3694E 05	-0.3931E 04	0.3450E 05	-0.3848E 05
1.0000E 00	0.1156E 02	-0.5514E C2	0.3438E 05	-0.3444E 05	0.5514E 02	0.3442E 05	-0.3436E 05

NOTE: The columns labeled ST INT IS or ST INT OS are the algebraic differences of the previous two columns.

The actual stress intensity at a particular location is the largest absolute value of the three columns.

FIGURE 6. STRESS-STRAIN HISTORY FOR TYPICAL FILL-DRAIN CYCLE
OF MSRE FREEZE FLANGE FOR CYCLE NUMBER ≥ 2

Table 4. Evaluation of Allowable Cycle Life of Freeze Flanges Subjected to Various Thermal Cycle Histories

Cycle Type and System Condition	Max. Stress Intensity	Δ -Stress	Total Strain (10^{-3})	Elastic Strain (10^{-3})	Plastic Strain (10^{-3})	Residual Stress	$\frac{1}{2} \frac{\text{Plastic Strain}}{\text{Cycle}}$	Allowable Cycles
<u>Thermal Cycle Facility N$\geq$2</u>								
1. Cold	0	-	-	-	-	+35,000		
2. Hot with helium	-74,670	-74,670	2.58	2.338	0.342	-30,000		
3. Salt fill transient	-159,700	-85,000	3.24	0	3.24	-25,000		
4. Steady state with salt	-125,400	+34,300	1.234	1.234	0	+9,300		
5. Cold	0	+125,400	4.51	0.924	3.566	+35,000	3.58×10^{-3}	30
<u>MSRE Heat Cycle to 1200°F N$\geq$2</u>								
1. Cold	0	-	-	-	-	+35,000		
2. Hot with circulating helium	-53,700	-53,700	1.932	1.932	0	-18,700		
3. Hot w/circulating He after drain	-53,700	-	-	-	-	+26,150		
4. Cold	0	+53,700	1.932	0.318	1.614	+35,000	0.807×10^{-3}	160
<u>MSRE Fill-Drain Cycle 2.6-hr Hold Time N$\geq$2</u>								
1. Hot w/circulating helium	-53,700	-	-	-	-	+26,150		
2. Salt fill transient	-158,500	-104,800	3.770	1.84	1.93	-25,000		
3. Filled with static salt	-133,800	+24,700	0.388	0.888	0	-300		
4. Salt flow transient	-154,300	-20,500	0.737	0.737	0	-20,800		
5. Steady state filled and circ.	-146,300	+8,000	0.288	0.288	0	-12,800		
6. Steady state after power cycle	-146,300	-	-	-	-	-11,800		
7. Salt flow stoppage transient	-128,300	+18,000	0.647	0.647	0	+6,200		
8. Filled with static salt	-133,800	-5,500	0.198	0.198	0	+700		
9. Salt drain transient	-49,850	+83,950	3.020	1.054	1.966	+30,000		
10. Hot w/circulating helium	-53,700	-3,850	0.138	0.138	0	+26,150	1.948×10^{-3}	58
<u>MSRE Power Cycle N$\geq$2</u>								
1. Steady state at zero power	-146,300	-	-	-	-	-11,800		
2. Transient to full power	-201,800	+44,500	1.601	1.324	0.277	+25,000		
3. Steady state at full power	-117,500	-15,700	0.565	0.555	0	+9,300		
4. Transient to zero power	-159,500	-42,000	1.511	1.234	0.277	-25,000		
5. Steady state at zero power	-146,300	+13,200	0.475	0.475	0	-11,800	0.277×10^{-3}	550

total change in stress intensity was evaluated for each cycle and was converted to total strain. The plastic strain per cycle was found by subtracting the strain that occurred in the elastic range below the yield stress. The plastic strain range per cycle, ΔE_p , which was used in conjunction with the thermal fatigue curve, was one half of the total plastic strain per cycle.

In cases where the flange bore remained above creep range temperatures, the residual steady state stresses were sufficiently low that the additional plastic strain due to relaxation was expected to be small. In the case of a drain or system cooldown where the residual stress was at the yield point, the flange temperature was well below the creep range where no additional relaxation would occur.

The allowable number of each type of cycle was determined from the thermal fatigue data shown in Fig. 7.

The safe life of a component is determined by accumulating the effects of the various types of cycles. The cycle ratio is the fraction of life consumed by a particular type of cycle (i.e., the number of cycles accrued divided by the allowable number). The safe life of the freeze flange would then be exhausted when the summation of the cycle ratios for the heating, fill, and power cycles was equal to 1.0.

Table 4 includes the allowable number of each type cycle that could be safely tolerated by the MSRE freeze flanges and by the prototype freeze flange in the Thermal Cycle Test Facility. These predictions of allowable cycle life were based on surface cracking in the flange bore because the maximum strain-range occurred there and decreased rapidly with radius. This surface cracking would not necessarily establish the point of a gross failure of the flange. A substantial number of additional cycles would be required to propagate the cracking through the lower stressed regions of the flange to the extent that leakage or rupture would be eminent.

V. DEVELOPMENT TESTING OF THE FREEZE FLANGE

The development testing of the freeze flange design included a thermal cycle test which was intended to demonstrate the mechanical integrity and reliability of the flange assembly under service conditions.

FIGURE 7. THERMAL FATIGUE DIAGRAM FOR HASTELLOY-N FOR TEMPERATURES TO 1300°F

The test was also intended to verify the performance of the flange in regard to the leakage rate of buffer gas during thermal cycling and to verify that the flanges could be opened and resealed to the prescribed buffer gas leakage specifications after thermal cycling.

The thermal cycle test was discontinued after 103 cycles in late 1962 after the objectives of the original test program were satisfied. The inspection of the flanges after the thermal cycle tests showed no indication of surface cracking. The thermal cycle tests were resumed in late 1967 when the projected MSRE operation showed the flanges would approach their allowable cycle life as predicted by the calculations earlier in this report. These tests continued until mid 1970 for a total of 540 cycles.

Brief descriptions of the thermal cycle test facility and its operation were presented in Ref. 9. The temperature history of the bore and ring gasket during a typical thermal cycle are shown on Fig. 8. The stress calculations and life prediction of a cycle of this type indicated that surface cracking could be expected in the flange bore after 300 cycles.

The first complete inspection after testing resumed was made at a total of 268 cycles. The dye penetrant inspection revealed a crack and a band of porosity in the flange bore as shown in Fig. 9. The crack was located about 1 1/4 in. from the end of the alignment stub and ran circumferentially from the top centerline about 90-110° clockwise. A band of porosity immediately behind the crack and about 1/4 in. wide followed the length of the crack and also extended counterclockwise past the top centerline about 120-130° of arc.

The flanges were reassembled and thermal cycling was resumed. Complete inspections were made after 321, 400, and 470 cycles, but no final inspection was made after the shutdown at 540 cycles. The cracking and porosity had increased at both the 321 and 400 cycle inspections. After 400 cycles the crack had propagated clockwise from the upper centerline to about 135° and extended intermittently to about 150° counterclockwise. A second intermittent parallel crack had appeared about 1/8 in. in front of the first. The porosity band had also increased to about 2 in. in width beginning at the original crack. The two parallel cracks outlined the root of the weld attaching the alignment stub to the flange face.

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FIGURE 8. TEMPERATURE HISTORY OF FREEZE FLANGES
IN THERMAL CYCLE TEST FACILITY DURING TYPICAL THERMAL CYCLE

PHOTO 75990A

FIGURE 9. PHOTOGRAPH OF FREEZE FLANGE
SHOWING DYE-PENETRANT CRACK INDICATIONS AFTER 268 THERMAL CYCLES

Figure 10 is a photograph of the freeze flange after 400 cycles showing the fluorescent dye-penetrant indications of cracking and porosity. A clearer view of the location of this cracking and porosity is shown on the cross section drawing of the flange assembly in Fig. 11.

The last inspection at 470 cycles was made using a different procedure to clean the surfaces that were exposed to salt. These surfaces were mechanically cleaned by grinding with alumina oxide cloth on a pneumatic hand grinder. Although the cracking and porosity covered the same general area, the inspection indicated a lesser degree of cracking and porosity than had been previously observed at 400 cycles. The abrasive cleaning could have either sealed the pores and prevented a proper inspection or some of the porosity and cracking could have been removed by the removal of metal from the surface.

The cracking and porosity were observed only on the male flange which had the alignment stub. There was no evidence of cracking or porosity on the female flange. The thermal stress analysis had not included the effects of the alignment stub, but the calculations did show the highest stress cycling to be at the location of the original crack and had predicted surface cracking after 300 cycles. The temperature cycling would have produced an additional bending moment at the base of the alignment stub where the crack was located, but the magnitude and importance of this bending moment have not been estimated. The effects of the weld attaching the alignment stub are also unclear, but the two parallel cracks outlining the weld root indicate some degree of change in metal properties or high residual stresses at the weld. The location of the porosity band extending deeper into the bore from the inner crack and the absence of porosity outboard of the second intermittent crack are indications that the porosity was caused more by the high stress levels in the flange bore surface than by the heat affected zone of the weld.

The female flange did not include surfaces equivalent to those containing porosity in the male flange. As shown on Fig. 11 the alignment stub acted as a heat shield for the surfaces that would have been most highly stressed. The slower and therefore less severe temperature transients reduced the surface stresses sufficiently to avoid the surface porosity.

FIGURE 10. PHOTOGRAPH OF FREEZE FLANGE
SHOWING FLUORESCENT DYE-PENETRANT INDICATIONS
OF CRACKING AND POROSITY AFTER 400 CYCLES

FIGURE 11. CROSS SECTION OF FREEZE FLANGE
SHOWING LOCATION OF POROSITY AND CRACKING THAT OCCURRED
DURING THERMAL CYCLE TESTING

Although cracking and porosity did occur during the test program, the flange pair was still serviceable at the conclusion of the 540 thermal cycles and there were no indications of leaks. The inspection of the exterior surfaces showed no evidence that the cracks or porosity had propagated through the flange thickness.

VI. PREDICTION OF ALLOWABLE LIFE FROM THERMAL CYCLE TESTS

The ASME Code Section III, Appendix II -1520 (Ref. 10) allows thermal cycle testing to establish the allowable life of a reactor component. A set of rules are also presented to establish the allowable life. Code case 1331-5,¹¹ which covers the design of reactor vessels at elevated temperatures in the creep range, also permits the allowable life to be established by a fatigue test, but no rules are established and the designer must justify the rules and procedures he uses. For the purpose of the fatigue test, the above section of the code defines a failure as "the propagation of a crack through the entire thickness such as would produce a measurable leak." For our test the flange assembly survived 540 thermal cycles without failure and probably could have survived considerably more had the testing continued.

The allowable cyclic life of the freeze flanges can be obtained from the number of test cycles using the procedures included in Section III of the code. Since the test flange and test cycle would be identical to the actual application, the allowable number of thermal cycles identical to those of the thermal cycle test would be obtained by dividing the number of test cycles without failure by a factor of 4.6 which accounts for the statistical variation of the component or test procedure when only one sample is tested. This would permit an increase in the allowable cyclic life of the freeze flanges for this one particular type cycle from 30 as calculated earlier in this report to 117 based on the results of the thermal cycle tests. The problem in applying the above procedures comes in attempting to translate the results of the test cycles of the thermal cycle facility to the various operational cycles at the MSRE. ASME Code Case 1331-5 states that the effect of hold time and strain rate must be considered, but the designer is left with the responsibility of justifying the procedures he uses.

The operational cycles of the MSRE were considerably different from those of the thermal cycle facility, the hold times at each steady-state condition were longer and the overall cyclic rate was much slower. Therefore some additional margin of safety would be required to cover the effects of any additional plastic strain that might occur because of these differences. A review of the calculated residual steady state stresses with temperatures in the creep range in Table 4 show that these stresses are relatively low in each case, and the additional relaxation would be expected to be small. One series of calculations were run with complete relaxation of the steady state thermal stresses when the metal temperature was in the creep range. The calculated reduction in cycle life varied from about a factor of 2 for the power change cycles to about a factor of 1.3 for the fill and drain cycles. An additional factor of safety of 2 should then be sufficient to cover the effects of any additional relaxation in the MSRE flanges.

The allowable number of cycles calculated from the stress analysis earlier in this report are actually a measure of the allowable accumulated plastic strain. The thermal cycle facility has shown that for one particular type cycle the flanges could safely survive 117 cycles or a factor of $117/30 = 3.9$ greater strain than was permitted by the stress analysis. The allowable cyclic life of the MSRE flanges based on the results of the thermal cycle tests could then be increased by the above ratio of 3.9 and then reduced by the additional safety factor of 2 giving a net increase in allowable life of a factor of 1.95.

VII. PERFORMANCE OF THE MSRE FLANGES

In the MSRE, three flange assemblies in the fuel system and two in the coolant system were exposed to elevated temperatures for 30,800 and 27,400 hours, respectively. Of this total high temperature exposure, the fuel and coolant systems contained circulating salt 21,800 and 26,100 hours. A detailed thermal cycle history of the freeze flanges was maintained during the operation of the reactor and the total thermal cycles are shown in Table 5.

Table 5. Thermal Cycle History of MSRE Freeze Flanges

	Heat/Cool	Fill/Drain	Power
Fuel System	13	51	101
Coolant System	12	18	97

Based on the original thermal stress analysis, the fuel and coolant freeze flanges had used 99 and 57.7% of their allowable life, respectively. However, the allowable life could have been extended by a factor of 1.95 based on the results of the thermal cycle tests.

All the freeze flange assemblies in the MSRE maintained their integrity through the life of the reactor system. There was no evidence of leakage of salt or radioactive gas from the freeze flanges at any time. However, one of the coolant system flanges did have a higher than normal leakage rate of buffer gas when the system was empty and cold but this did not present any operational difficulties with the reactor. None of the flanges were opened after high temperature operation of the reactor system began.

VIII. CONCLUSIONS

This analysis was completed in the fall of 1964 prior to high temperature operation of the MSRE. The question at that time was whether the freeze flanges could safely tolerate the strain cycling that would occur over the expected life of the reactor or whether some design modification or removal of the flanges would be necessary to ensure the integrity of the reactor piping system. The procedure used to complete the analysis was similar to that used earlier to analyze the fuel and coolant pump tanks^{1,2} with some modification to utilize the more recent thermal fatigue data that had become available for Hastelloy "N".

Although the thermal stresses were found to be very high, the analysis indicated that the flanges could safely tolerate the cycle history anticipated for the life of the reactor. In addition to the results of the

thermal stress analysis, the test flanges in the thermal cycle facility had at that time accumulated 103 test cycles with no indication of failure. The thermal stress analysis predicted an allowable safe life of 30 test cycles (or possible failure after 300 cycles). The results of the thermal stress analysis and the thermal cycle test led us to the following conclusions:

1. The freeze flanges would be satisfactory for the intended service in the MSRE.
2. The life of the freeze flanges could be increased by providing a 2.6-hr hold time between filling the system and starting circulation.
3. Additional operation of the thermal cycle test facility would be required to demonstrate the adequacy of the calculations and to provide an additional margin of safety in the event the MSRE incurred a greater number of cycles than anticipated.
4. A detailed and up-to-date record would be required of the thermal cycle history of the MSRE to ensure that the accumulated fatigue damage for all modes of operation did not exceed the safe limit.

The thermal stress analysis and life prediction of the MSRE freeze flanges covered by this report were less conservative than would be required if the analysis were done at the present time. The rules and guidelines established by Section III of the ASME code and by code case 1331-5 would require a plastic analysis by finite element technique to account for the local yielding and changes in metal properties with temperature. In addition the design fatigue curves for determining the allowable life in ASME code case 1331-5 are based on total strain and have a factor of safety of 20 applied to the number of cycles (for our area of interest). The design fatigue curve used in this report was based on only the plastic strain and had a factor of safety of 10 applied to the number of cycles. A stress analysis and life prediction calculated by the current analytical procedures of ASME Code Section III and code case 1331-5 would probably determine that the design was unacceptable for the intended service.

The planned operation of the MSRE was concluded at the time the fuel freeze flanges had reached their safe life as determined by the thermal stress analysis. The results of the thermal cycle test could have been

used to extend the safe life had continued operation been desirable. This was a case in which a reactor component had a marginal calculated cyclic life because of high thermal stresses. However, the calculated allowable life could have been extended by a factor of 1.95 or possibly greater by the results of the thermal cycle test. Thermal cycle tests of this type are expensive and time consuming, but there may be cases where the time and expense would be justified in order to qualify a component for a given service. However, a considerable amount of judgment may be required in the interpretation of results from a thermal cycle test. In the case of the freeze flange test, for example, surface porosity and cracking indications were first observed after 268 cycles as compared to the calculated number of 300 cycles. However, testing continued and the flanges were still serviceable after 540 cycles. The porosity and cracking were in the area of highest calculated stresses but were apparently relatively shallow and were in a thick section where the mechanical integrity of the flange would not be threatened unless the cracking or porosity increased grossly. If the cracking and porosity had occurred in an area at high primary stress or in a thin section where external piping forces and moments in addition to the internal thermal stresses might propagate the crack through the thickness, an extension of the allowable life could not have been justified even if permitted by the ASME code rules.

NOMENCLATURE

C_1, C_2	Integration constants for cylinder
C_3, C_4, C_5, C_6, C_7	Integration constants for flange
E	Modulus of elasticity
M_r, M_x, M_θ	Radial, axial, or circumferential bending moment
N_x	Axial membrane force on cylinder and on inside radius of flange = $\pi a^2 p$
N_θ	Circumferential membrane force in cylinder
Q_r	Radial force on cylinder or flange
Q_θ	Circumferential membrane force of flange
R	Dimensionless coordinate of flange = $\frac{r}{b}$

$S_{tr}, S_{t\theta}$	Radial and circumferential thermal stress in flange
S_r, S_x, S_θ	Radial, axial, and circumferential principal stresses
$T(r)$	Temperature at radius, r
U_c, U_f	Free thermal expansion of cylinder or flange at radius a'
a	Inside radius of flange
a'	Mean radius of cylinder
b	Outside radius of flange
h	Thickness of flange
p	Internal pressure
r	Radial position on flange
t	Thickness of cylinder
u	Radial displacement of flange or cylinder due to the mechanical coupling of the two members
w	Axial displacement of flange
x	Axial coordinate of cylinder
α	Coefficient of thermal expansion
ν	Poissons ratio

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