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ANALYSIS OF THERMAL STRESSES IN BERYLLIUM ASSEMBLIES FOR THE HIGH FLUX REACTOR

BY

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Final Report

Problem Assignment No. TX13-5

ANALYSIS OF THERMAL STRESSES IN BERYLLIUM ASSEMBLIES FOR THE HIGH FLUX REACTOR

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1.0 ABSTRACT

This report covers an experimental investigation of the stability of the beryllium reflector assemblies to thermal stresses expected in the high flux pile.

Assemblies were subjected to thermal stresses which would result from heat fluxes 4.5 times greater than anticipated in the reactor during operation. Fifty cycles of alternate heating to 4.5 times the maximum expected reactor level and sudden cooling produced no cracking or failure.

It is concluded that the beryllium reflector assemblies are adequately stable to thermal stresses which would result from heat fluxes 4.5 times greater than will occur in the high flux reactor as presently designed.

On this basis, it is believed that the cooling requirements per assembly may be safely reduced by at least one half.

2.0 INTRODUCTION

It was the purpose of this investigation to establish the stability of the beryllium reflector assemblies to thermal conditions in the high flux pile during operation. It was proposed, further, to determine how severe thermal conditions the assemblies would withstand before failure. The first objective was carried to completion. The second objective was not reached in that the assemblies were not ruptured. However, experimental results obtained indicate that the assemblies are thoroughly adequate since thermal fluxes and stresses several times more severe than expected in the pile were produced.

3.0 THEORETICAL

It was proposed in the design of the beryllium reflector for the high flux pile that the heat generated from neutron collisions and gamma ray absorption within the assemblies be removed by water flowing through holes which were distributed uniformly through the lateral cross section. Four

cylindrical holes were located in the interior of each assembly and the equivalent of five additional holes in the form of slots were placed around the periphery of the assembly. For purposes of calculating temperature and stress distributions through the assemblies, it was assumed that each cross section was composed of a number of cylindrical heat drainage areas, each area surrounding a cooling hole.[1]

The mathmetical case of an infinitely long cylinder heated uniformly throughout with heat removed from the surface of an axial hole reasonably fits the assumed conditions.

3.1 Temperature Patterns

The temperature gradient across a cylinder wall in the above case is defined as[2]:

$$(1) \qquad T = \frac{Q}{2K} \left[R^2 \ln R/a - \left(\frac{R^2 - a^2}{2} \right) \right]$$

where Q = heat production rate per unit volume BTU/(ft^3)(hr).

K = thermal conductivity - BTU/(hr) (ft^2)($^\circ$F)/(ft)

R = heat drainage radius = ft

a = radius of cooling hole - ft

In the current design case the maximum value of Q in the beryllium as stated by Weinberg (Mon P-272) is 4 cal/(sec)(cm^3). The maximum value of Q occurs at the center of the longitudinal beryllium-reactor interface. The heat production rate drops rapidly with distance into the beryllium. At a distance of one inch from the interface where the first 1/8 inch diameter cooling holes are located the production rate is 2.5 cal/(sec)(cm^3) and is 1.8 cal/(sec)(cm^3) at a distance 2 inches from the interface as calculated by J. A. Lane. It may be concluded that an average rate value of about 2.5 cal/(sec)(cm)^3 would more nearly represent the heat production in the reflector; however, for a basis of the experiment the most severe case was assumed.

When Q is 4 cal/(sec)(cm^3) [162 x 10^4 BTU/(hr)(ft^3)] ;

K = 0.4 cal/(cm^2)(sec)($^\circ$C)/cm [97 BTU/(hr)(ft^2)($^\circ$F)/ft] ;

R = 1.49 cm (0.049 ft);

a = 0.15 cm (0.0049 ft).

T is calculated to be 36°F (20°C).

It was not feasible to experimentally obtain the value of T by measurement, but rather to determine the cooling water rate and temperature rise of the water. The heat flux at the cooling hole surface becomes:

$$H = \frac{Q\pi\ (R^2 - a^2)}{2\pi\ a} = \frac{Q}{2a}\ (R^2 - a^2)$$

$$H = 392{,}000\ \text{BTU/(hr)(ft}^2\text{)}\ \text{or}\ 29.5\ \text{cal/(sec)(cm}^2\text{)}$$

The equivalent total heat generation in two experimental cases becomes 90 BTU/min and 143 BTU/min respectively for 5.1 inch and 8 inch long cylinders with cooling holes 0.125 inches in diameter, and is calculated as follows:

$$q = 392 \times \frac{10^3\ \text{BTU/(hr)(ft}^2\text{)} \times 13.8 \times 10^{-3}\ \text{ft}^2}{60}$$

$$q = 90\ \text{BTU/min}$$

A factor, q', is now calculated as the ratio of the experimental value of heat removed, q_E, to the design value of heat to be removed, q.

$$q' = \frac{q_E}{q} = \frac{45^\circ\ \text{F temp. rise} \times 9\ \#/\text{min}}{90\ \text{BRU/min.}} = \frac{405}{90} = 4.5$$

3.2 Stress Patterns:

The case of normal stresses in long hollow cylinders has been considered by Timoshenko(4). His general equations were fitted to the specific case of uniform heating throughout with cooling at the surface of an axial hole, by Briggs(5), the equations arrived at being:

(4) $\sigma_r = 0$

(5) $\sigma_t = \frac{E\alpha\ Q\ a^2}{8K\ (1-\nu)}\ (\ \Phi\ P_1)$

(6) $\sigma_E = \sigma_t$

where $\sigma_r, \sigma_t, \sigma_E$ are radial, tangential, and longitudinal stresses respectively in lbs./in^2.

E = Youngs modulus, lbs/in^2

α = linear coefficient of thermal expansion per $^\circ$F

ν = Poisson's Ratio.

At the inner surface of the cylinder, stresses attain a maximum and are the only onces considered in this paper. The function, ΦP_1, is defined at this location as:

$$\frac{4P^4 \ln P - 3P^4 + 4P^2 - 1}{P^2 - 1}$$

where $P = R/a = 1.49 \text{ cm}/.15 \text{ cm} = 10$ for the design case.

Experimental cylinders used had values of P equal 9.4 and 13.

The experimental cylinders were heated uniformly on the outside surface and cooled at the surface of an axial hole. For the case of a cylinder the same general stress equations apply regardless of the method of heating provided the proper temperature function is used. For a cylinder heated on the outer surface with heat removed from the surface of an axial hole the function becomes:

(7) $$T = \frac{H}{2\pi k} \ln \frac{R}{a} = \frac{Q(P^2 - 1)a^2}{2k} \ln \frac{R}{a}$$

where H = heat transferred, BTU/(hr)(ft. of length)

and the stresses at the surface of the axial hole are:

(8) $$\sigma_r = 0$$

(9) $$\sigma_t = \frac{E \alpha Q a^2}{8K(1 - \nu)} (\Phi P_2)$$

(10) $$\sigma_E = \sigma_t$$

where $\Phi P_2 = 4P^2 \ln P - 2P^2 + 2$

A sample calculation provides the method of obtaining a stress factor, σ actual/ σ normal.

$$\frac{\sigma \text{ actual}}{\sigma \text{ normal}} = \frac{\dfrac{E \alpha (4.7Q)}{8K(1-\nu)} a^2 (\Phi P_2)}{\dfrac{E \alpha Q a^2}{8K(1-\nu)} (\Phi P_1)}$$

ΦP_1 is calculated as 1500 when P is 13, as for the test piece and $\Phi P_2 = 590$ when P is 10 as in the design case.

$$\frac{\sigma \text{ actual}}{\sigma \text{ normal}} = \frac{(C)(4.7Q)(1500)}{(C)(Q)(590)} = 12$$

4.0 EXPERIMENTAL

Test work was conducted on two beryllium cylinders; the first being 8 inches long, 1.17 inches in outside diameter and with a 0.125 inch diameter axial hole. The second cylinder was 5 inches long, 1.62 inches outside diameter and 0.125 inches inside diameter. The cylinders were heated at the outside surfaces by induction. The frequency of the induced field was of the order of 100 KC; hence, heat was supplied only to a thin surface layer of the cylinder. The time required to bring the cylinders to maximum temperature was about 1 minute. The axial hole surface was water cooled. The water flow and temperature rise of the water were recorded for calculation of total heat transfer across the cylinder walls.

In the theoretical cylinder no heat flows along the longitudinal axis; however, for a practical case of a cylinder of finite length some heat flow in this direction cannot be eliminated. The test pieces were end mounted and insulated to reduce end effects in the temperature pattern to a minimum. Further, the induction coils extended beyond the cylinder ends.

Water flow was measured by a calibrated rotameter with a range of 2 - 12 lb/min. The cooling water flow was 9 lbs/min. Temperatures were indicated by standard mercury thermometers which were located well outside of the induced field. (See Figure 1).

5.0 RESULTS

Tabulated below are the q' factor and stress factor results from two test runs.

	Number Thermal Cycles	Heat Flux $BTU/ft^2 x10^{-3}$	Average q' Factor	P = R/a	Average Stress Factor
Design	- -	392	1.0	10.0	1.0
Test A	100	1180	3.0	9.4	3.6
Test B	50	1840	4.7	13.0	12.0

Drawing # 5948

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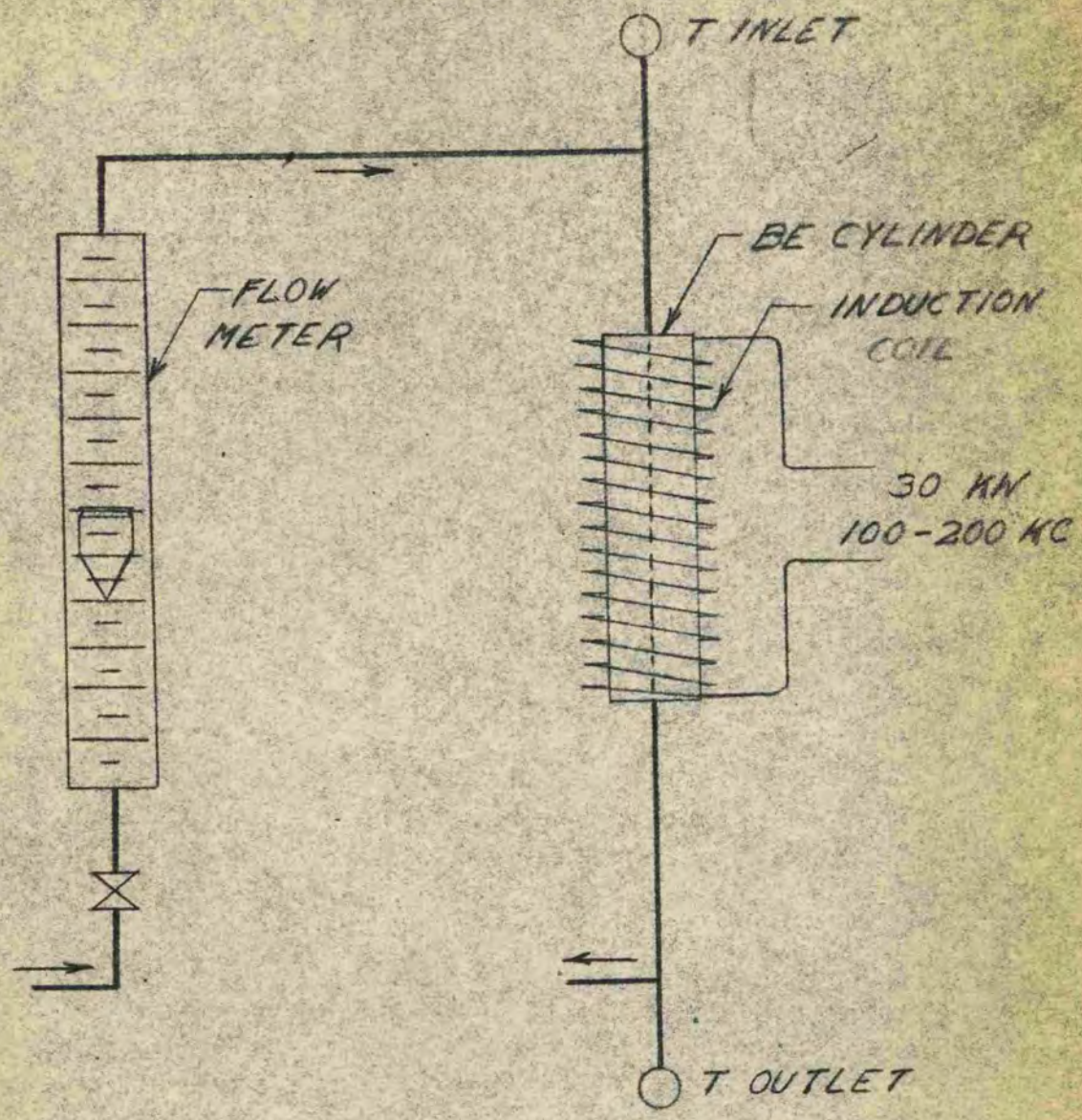

FIGURE 1 - APPARATUS FOR BERYLLIUM CYLINDER STRESS MEASUREMENT

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Table 1

Values of stress are calculated using a constant value of Youngs Modulus. It is doubtful that the modulus does remain constant in light of the known value of ultimate strength of beryllium which is about 30,000 psi. Calculated stresses from data in test B are of the order of 90,000 psi, and about 30,000 psi for test A. These data would predict failure of test pieces. Such has obviously not happened. For this reason, specific values of stress have not been reported but rather a stress factor for comparative purposes only. It may be seen that stresses 12 times larger are calculated for test B than for design thermal conditions.

The point of greatest emphasis is that heat fluxes which were respectively 3.0 and 4.7 times greater than expected to exist in the reflector assemblies closest to the reactor were induced into the test cylinders, and no failure or cracking occured.

6.0 CONCLUSIONS AND RECOMMENDATION

It is concluded that the beryllium reflector assemblies are adequately stable to thermal stresses which would result from heat fluxes 4.5 times greater than presently designed into the high flux reactor.

On this basis it is believed that the cooling requirements per assembly may be reduced by at least one half.

7.0 ACKNOWLEDGMENTS

Considerable credit is due J. E. Cunningham for his aid in the experimental work.

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