MASTER

WFPS TME-O18
OCTOBER 1975

**Westinghouse Electric Corporation
Fusion Power Systems Department**

**COLLECTION OF PRELIMINARY THERMAL
AND STRUCTURAL CALCULATIONS
FOR TFTR COMPONENTS**

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

WFPS TME-018
October, 1975

COLLECTION OF PRELIMINARY THERMAL AND
STRUCTURAL CALCULATIONS FOR TFTR COMPONENTS

Authors: E. A. DeZubay
G. Driesen

Compiled by: A.Y. Lee

Approved by:

T. C. Varljen
Engineering Manager,
TFTR Project

Approved by:

C. K. Kim, Manager
Systems Engr. & Integration
TFTR Project

Westinghouse Electric Corporation
Fusion Power Systems Department
Post Office Box 10864
Pittsburgh, Pennsylvania 15236

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

829

ABSTRACT

Many scoping type thermal/hydraulic and structural calculations have been performed for the TFTR component conceptual designs. Five analyses are still applicable to the current October conceptual design. The analyses are for the vacuum vessel wall temperature, neutral beam calorimeter thermal response, the discharge time of the tritium line, estimate of air leakage in the TFTR Test Cell, and maximum allowable TF coil temperature. The methods of analysis and the results are presented in this report.

ACKNOWLEDGEMENT

This work was supported by U. S. Energy Research and Development Administration Contract E(11-1)-3073 and Subcontract 192 thereto. Reproduction, translation, publication, use and disposal, in whole or in part, by or for the United States Government is permitted.

LEGAL NOTICE

This report was prepared as an account of Government sponsored work. Neither the United States, nor the Administration, nor any person acting on behalf of the Administration:

- A. Makes any warranty or representation, express or implied, with respect to the accuracy, completeness, or usefulness of the information contained in this report, or that the use of any information, apparatus, method or process disclosed in this report may not infringe privately owned rights; or
- B. Assumes any liabilities with respect to the use of, or for damages resulting from the use of any information, apparatus, method, or process disclosed in this report.

APPLIED TECHNOLOGY

Any further distribution by any holder of this document, or of the data therein, to third parties representing foreign interests, foreign governments, foreign companies and foreign subsidiaries, or foreign divisions of United States companies, should be coordinated with the Director, Division of Thermonuclear Research, United States Energy Research and Development Administration.

TABLE OF CONTENTS

	<u>Page</u>
1.0 INTRODUCTION	1-1
2.0 VACUUM VESSEL WALL TEMPERATURE	2-1
2.1 SUMMARY	2-1
2.2 METHOD OF ANALYSIS	2-1
2.3 RESULTS OF ANALYSIS	2-4
3.0 DESIGN CONSIDERATIONS FOR 40 KW/cm ² NEUTRAL BEAM CALORIMETER	3-1
3.1 DISCUSSION	3-1
3.2 ANALYSIS	3-1
4.0 DISCHARGE TIME OF TRITIUM LINE TO TORUS	4-1
4.1 DISCUSSION	4-1
4.2 ANALYSIS AND RESULTS	4-1
5.0 ESTIMATE OF FLOW THROUGH CRACKS IN THE TFTR CELL	5-1
5.1 SUMMARY	5-1
5.2 ANALYSIS	5-1
5.3 ESTIMATE FOR DESIGN LEAKAGE	5-5
6.0 MAXIMUM ALLOWABLE TF COIL WINDING TEMPERATURE	6-1
6.1 DISCUSSION AND ANALYSIS	6-1
6.2 CONCLUSIONS	6-7

LIST OF FIGURES

<u>Figure No.</u>	<u>Title</u>	<u>Page</u>
5. 3-1	Air Leakage Flow Behavior in Wall Cracks at 2.3 cm H ₂ O Pressure Differential	5-6
6. 1-1	Stress Range vs. Strain Cycles and $\Delta\epsilon_d$	6-3
6. 1-2	Plastic Longitudinal Strain Range vs. Total Diametrical Strain Range	6-5
6. 1-3	Plastic Strain Range vs. Cycles to Failure	6-6

LIST OF TABLES

<u>Table No.</u>	<u>Title</u>	<u>Page</u>
2.2-1	Vacuum Vessel Wall Temperature at Different Values of Heat Deposition	2-6
3.2-1	Calorimeter Incidence Angles	3-2
4.2-1	Discharge Time of Tritium Line to Torus	4-5

1.0 INTRODUCTION

During the course of the TFTR conceptual design evolution, various preliminary thermal and hydraulic calculations were performed to provide design guidance and concepts evaluation. Five analyses were identified that are still applicable to the current October conceptual design and are documented in this report. The five subjects are:

- (1) Vacuum vessel wall temperature at various energy depositions,
- (2) Design consideration for the neutral beam calorimeter having an incident flux of 40 kW/cm^2 ,
- (3) Discharge time of the tritium line from generator to the torus,
- (4) Estimates of air flow through cracks in the TFTR test cell, and
- (5) Maximum allowable TF coil temperature.

The analysis and the results for each of the subjects are presented in the following sections.

2.0 VACUUM VESSEL WALL TEMPERATURE

by E. A. DeZubay

2.1 SUMMARY

Calculations are made to estimate the vacuum vessel temperature as a function of the number of energy pulses and the energy delivered per pulse. The vessel is assumed to convect to ambient air in the test cell, at 21°C (70°F). Table 2.2-1 lists the maximum temperature which occurs at the end of the energy deposition period and also at the end of the cycle where each 300 second cycle consists of 1/2 sec. of energy deposition, followed by 299.5 sec. of no energy deposition. The maximum equilibrium temperature of the vessel is estimated to be 93°C for 14 MJ pulses.

2.2 METHOD OF ANALYSIS

The differential equation for the vessel wall treated as a lumped system is

$$Q_{in} - Q_{out} = MC_p \frac{dT}{d\theta}$$

where

Q_{in} = energy supplied to wall/unit time

Q_{out} = convective energy dissipation to ambient air/unit time = hAT

M = mass of vessel wall

C_p = specific heat of vessel wall

T = temperature difference between wall and ambient air

h = convective coefficient

A = convection area

The energy supplied has the following history

$$Q_{in} = Q_o, \quad 0 < \theta < \theta_H$$

$$Q_{in} = 0, \quad \theta_H \leq \theta \leq \theta_p$$

for the first pulse. For the "n'th" pulse

$$Q_{in} = Q_o, \quad (n-1)\theta_p < \theta < \theta_H + (n-1)\theta_p$$

$$Q_{in} = 0, \quad \theta_H + (n-1)\theta_p \leq \theta \leq n\theta_p$$

where θ_H is the energy pulse length and θ_p is the repetition period.

Hence in the region $(n-1)\theta_p < \theta < \theta_H + (n-1)\theta_p$,

$$Q_o - hAT = MC_p \frac{dT}{d\theta}$$

or

$$-\frac{hA}{MC_p} \int_{(n-1)\theta_p}^{\theta_H + (n-1)\theta_p} d\theta = \int_{T_{n-1}}^{T_{Hn}} \frac{dT}{-\frac{Q_o}{hA} + T}$$

$$-\frac{hA}{MC_p} \theta_H = \log_e \frac{-\frac{Q_o}{hA} + T_{Hn}}{-\frac{Q_o}{hA} + T_{n-1}}$$

$$T_{Hn} = +\frac{Q_o}{hA} + e^{-\frac{hA\theta_H}{MC_p}} \left(-\frac{Q_o}{hA} + T_{n-1} \right)$$

In the region $\theta_H + (n-1)\theta_p \leq \theta \leq n\theta_p$

$$-hAT = MC_p \frac{dT}{d\theta}$$

$$-\frac{hA}{MC_p} \int_{\theta_H + (n-1)\theta_p}^{n\theta_p} d\theta = \int_{T_{Hn}}^{T_n} \frac{dT}{T}$$

$$-\frac{hA}{MC_p} (n\theta_p - \theta_H - (n-1)\theta_p) = \log_e \frac{T_n}{T_{Hn}}$$

$$T_{Hn} e^{-\frac{hA}{MC_p} (\theta_p - \theta_H)} = T_n$$

or

$$T_n = e^{-\frac{hA}{MC_p}(\theta_p - \theta_H)} \left(\frac{Q_o}{hA} + e^{-\frac{hA\theta_H}{MC_p}} \left(-\frac{Q_o}{hA} + T_{n-1} \right) \right)$$

2.3 RESULTS OF ANALYSIS

The calculations were performed with the following numerical parameters and the results are summarized in Table 2.2-1.

$$E_o = 7, 14, \text{ \& } 21 \text{ MJ}$$

$$\theta_H = 1/2 \text{ sec}$$

$$\theta_p = 300 \text{ secs.}$$

$$A = 1.15 \times 10^6 \text{ cm}^2$$

$$h = 5.67 \times 10^{-4} \text{ joules/sec-cm}^2\text{-}^\circ\text{C (1 BTU/ft}^2\text{-}^\circ\text{F-hr)}$$

$$M = 30000 \text{ kg}$$

$$C_p = .502 \text{ joules/gm } ^\circ\text{C}$$

$$\frac{hA}{MC_p} = .1564/\text{hr (4.34} \times 10^{-5}/\text{sec)}$$

$$\frac{Q_o}{hA} = \frac{E_o/\theta_r}{hA} = K \times 2.147 \times 10^4 \text{ } ^\circ\text{C; } K = 1, 2, 3$$

Using the above numerical values the following equations result:

$$T_{Hn} = 2.147 \times 10^4 \text{ K} \left(1 - e^{-2.173 \times 10^{-5}} \right) + e^{-2.173 \times 10^{-5}} T_{n-1}$$

$$T_n = e^{-1.301 \times 10^{-2}} T_{Hn}$$

Hence in any cycle

$$T_{\max} = T_{Hn} + T_{\text{ambient}}$$

$$T_{\text{cycle end}} = T_n + T_{\text{ambient}}$$

Based on this analysis a steady state vessel temperature of 93°C was used to establish the physical properties of the vessel material for the structural analysis.

TABLE 2.2-1

VACUUM VESSEL WALL TEMPERATURES AT
DIFFERENT VALUES OF HEAT DEPOSITION

Heat Deposition		7MJ		14 MJ		21 MJ		28 MJ	
<u>Time - hrs.</u>	<u>Cycles</u>	<u>T_{max}*</u>	<u>T_{end}</u>	<u>T_{max}</u>	<u>T_{end}</u>	<u>T_{max}</u>	<u>T_{end}</u>	<u>T_{max}</u>	<u>T_{end}</u>
0	0	21.11	21.11	21.11	21.11	21.11	21.11	21.11	21.11
.0833	1	21.57	21.56	22.03	22.02	22.48	22.47	22.96	22.94
.25	3	22.48	22.46	23.86	23.82	25.34	25.18	26.61	26.54
.5	6	23.80	23.77	26.50	26.44	29.21	29.11	31.92	31.78
1.0	12	26.30	26.23	31.50	31.37	36.71	36.51	41.92	41.65
2	24	30.75	30.63	40.41	40.16	50.07	49.69	59.72	59.22
5	60	40.60	40.35	60.10	59.60	79.60	78.84	99.10	98.09
20	240	55.46	55.02	89.83	88.94	124.19	122.85	158.56	156.78
50	600	57.02	56.56	92.94	92.02	128.81	127.47	164.79	162.93

* T in °C

3.0 DESIGN CONSIDERATIONS FOR 40 kW/cm² NEUTRAL BEAM CALORIMETER^{**}

by E. A. DeZubay

3.1 DISCUSSION

The following calculations were performed for a neutral beam calorimeter accepting an incident flux of 40 kW/cm². The surface temperature rise equation (1) is valid for an infinitely thick plate, but does not lead to an appreciable error when used instead of the exact solution of a cooled finite plate for short time periods. Two cases are considered, copper and tungsten, for the case of incipient surface melting. The trade-off of pulse length as a function of incidence angle is given. The incidence angle Φ becomes large for the flux considered, which means that steep cones or wedges must be used for the calorimeter.

3.2 ANALYSIS

The temperature rise at the surface of a wall exposed to an energy flux F_0 is

$$\Delta T = \frac{2F_0 \cos \Phi}{K} \sqrt{\frac{K}{\rho c} \frac{\theta}{n}} \quad \text{eq. (1)*}$$

where

- K = thermal conductivity
- ρ = density
- c = specific heat
- θ = duration of energy flux exposure
- Φ = angle of incidence (angle between beam path and normal to surface)

While this solution is specifically developed for an infinite plate, the incident surface temperature closely approximates that of a finite slab of reasonable thickness cooled on

* "Modern Operational Mathematics in Engineering," by R.V. Churchill, McGraw Hill Co., p. 107.

** 10 kW/cm² is the present estimate of the Neutral Beam energy flux.

the opposite side. *** If ΔT is defined as the temperature for incipient surface melting

$$\Delta T_m = T(M, P) - T(\text{ambient})$$

Hence

$$\Phi = \cos^{-1} \left\{ \frac{K \Delta T_m}{2F_0} \sqrt{\frac{n}{\theta} \frac{\rho c}{K}} \right\}$$

The copper and tungsten cases considered are tabulated below.

TABLE 3.2-1
CALORIMETER INCIDENCE ANGLES

	Copper	Tungsten
ΔT_m	1050°C	3340°C
ρ	8.9 gm/cm ³	19.3 gm/cm ³
K	3.53 w/cm°C-sec	1.30 w/cm°C-sec
c	.44 w/gm°C	.167 w/gm°C
$\frac{\Delta T}{2F_0} \sqrt{\pi K \rho c}$	.0865 sec ^{1/2}	.151 sec ^{1/2}

***Conduction of Heat in Solids, "Carslaw and Jaeger, Oxford Press, p. 125.

TABLE 3.2-1 (Continued)

<u>Millisecs</u>	Φ (copper) <u>Degrees</u>	Φ (tungsten) <u>Degrees</u>
0.0	0.0	0.0
7.48	0.0	0.0
8.00	14.75	0.0
10.0	30.12	0.0
20.0	52.29	0.0
22.9	55.14	0.0
23.0	55.23	2.81
30.0	60.04	29.01
50.0	67.24	47.36
100.0	74.13	61.38
150.0	77.09	66.98
200.0	78.84	70.20
300.0	80.90	73.94

From these tabulated values, it is apparent that both copper and tungsten require very deep and closely spaced saw teeth or cones for durations in the range of 200 milliseconds.

Two possible solutions can be explored, the first is pre-cooling to cryogenic temperature (LN_2 at -193°C) which would provide about 20% margin for copper, but only 6% for tungsten, or investigate the possibility of very thin cooled walls where the above approximation for the surface temperature is not valid.

4.0 DISCHARGE TIME OF TRITIUM LINE TO TORUS by E. A. DeZubay

4.1 DISCUSSION

The following calculated discharge time of the tritium supply into the torus is in the range of 10 milliseconds, with the exact numerical value dependent on the pressure in the supply line and the torus. This value is a lower bound estimate and is provided as an aid to establish the control system requirements*. The analysis is based on transient supersonic flow which would be expected in a condition where momentum forces strongly overshadowed friction forces, such as this case where stationary gas is suddenly permitted to flow into a low pressure volume. Note, that with the short gas discharge time, the valve opening time may become a constraining limit.

The tritium supply system consists of the measuring volume, the four parallel series valve shut-off system and the torus inlet valve. When the tritium supply is to be discharged, the system can be represented by the measuring volume connected through a low flow impedance (the four parallel series valves are open) to the supply line. The torus inlet valve is then opened suddenly. The system would behave as an open ended shock tube, since the pressure ratio between the gas supply and the torus is many times the critical pressure required for sonic flow. The critical pressure is

$$P_{cr} = \left(\frac{2}{\gamma + 1} \right)^{\gamma/(1-\gamma)} = 1.893$$

for a diatomic gas of $\gamma = 1.4$.

The pressure ratio across the torus valve can be estimated in the following manner:

* Another analysis with a different approach, based on an initial choked flow as the line discharge valve is opened, has been made. This model considers the flow regime change from turbulent to laminar, then to molecular flow as the pressure in the vacuum vessel approaches that of the discharge line and results in an upper bound number of approximately 250 m sec. This analysis will be reported in the near future.

$$\text{Tritium generator} = \quad = 391 \text{ cm}^3$$

$$\text{Line} = 1" \times 60' = (2.54)^2 \frac{2\pi}{4} \times 30.48 \times 60 = 3650 \text{ cm}^3$$

$$V_{\text{total}} = 4040 \text{ cm}^3$$

Assume generator pressure, $P_g = 1.2$ to 148 Torr, and

$$\text{Torus pressure, } P_t = 10^{-4} \text{ to } 10^{-8} \text{ Torr}$$

When the line is filled by unblocking the shut-off valves the system pressure will fall by a factor of approximately 10 due to the increase volume occupied, or

$$P = .12 \text{ to } 15 \text{ Torr}$$

still maintaining a pressure ratio of line to torus in the range of

$$1.2 \times 10^3 < P_r < 1.5 \times 10^9$$

well above the required range of $P_r > 2$ for sonic flow.

When the torus inlet valve is rapidly open, the system behavior is assumed to be similar to a shock tube when the diaphragm between the driver and the driven gas is broken. The contact surface or interface between the high and low pressure gas will move at velocity u into the low pressure gas behind the shock wave traveling at velocity c into the low pressure gas.

The discharge velocity is given by

$$u = a (P_s - 1) \sqrt{\frac{2/\gamma}{(\gamma + 1)P_s + \gamma - 1}}, \quad a = \sqrt{\gamma g R T}$$

where the shock strength P_s can be expressed in the terms of the initial pressure ratio P_r as

$$P_r = P_s \left[1 - \frac{(\gamma - 1)(P_s - 1)}{\sqrt{2\gamma} \sqrt{2\gamma + (\gamma + 1)(P_s - 1)}} \right]^{-\frac{2\gamma}{\gamma - 1}}$$

If the composition and temperatures between the line and torus would differ markedly the above expression can be modified to reflect such changes (Liepmann & Roshko "Elements of Gas Dynamics," p. 81, Equation 3.26).

Combining the results of the above for $\gamma = 1.4$ and a molecular weight of $M = 6$, yields

$$u = 264 \frac{\text{ft}}{\text{sec}} \left(\frac{\text{lb}}{\text{mole} \cdot ^\circ\text{R}} \right)^{1/2} \sqrt{\frac{T(^{\circ}\text{R})}{M \text{ lb/mole}}} (P_s - 1) \sqrt{\frac{.595}{P_s + .166}}$$

and

$$P_r = P_s \left[1 - \frac{.24 (P_s - 1)}{\sqrt{2.8 + 2.4 (P_s - 1)}} \right]^{-7}$$

Since the pipe is the predominant part of the system, the time to empty the 60 feet would be

$$t (\text{sec}) = \frac{60}{u}$$

In Table 4.2-1 the pressure ratio P_r across the torus valve is shown as a function of the time for $T = 21^\circ\text{C}$ and a molecular weight of 6 (T_2).

The values on Table 4.2-1 are lower bound estimates since no attempt was made to consider frictional and boundary layer effects.

TABLE 4.2-1
DISCHARGE TIME OF TRITIUM LINE TO TORUS

Supply to Torus Pressure Ratio P_r	Line Discharge Time milliseconds	Shock Strength P_s
10^9	5.3	36.89
1.6×10^8	5.5	35.0
4.5×10^6	5.9	30.0
3.2×10^5	6.6	25
3.4×10^4	7.4	20
4.4×10^3	8.7	15
1×10^3	10.3	11.53

5.0 ESTIMATE OF FLOW THROUGH CRACKS IN THE TFTR CELL

by E. A. DeZubay

5.1 SUMMARY

Calculations are made to estimate the flow which could be expected through the cracks of the TFTR cell when the cell is maintained at 2.5 cm of water negative pressure differential. For a generous allowance of fifteen cracks communicating with the outside and a wind condition of 30 mph, the leakage would be less than $3.4 \text{ m}^3/\text{min}$ (120 SCFM). This design condition leads to a tightness specification that:

The leak rate of the TFTR cell shall be less than $3.4 \text{ m}^3/\text{min}$ (120 SCFM) at a pressure differential of 2.5 cm of water (-1.0-in. H_2O).

5.2 ANALYSIS

The model uses a thin slot in a 1.8 m (6 ft) wall under a static pressure difference.

The crack has a minor dimension t , through a wall thickness L . The major dimension will be taken as a unit dimension. (All calculated values of flow will be referenced to 1 meter of crack length on the surface.)

Hence:

$$\Delta P = \left(K_1 + K_2 + 4f \frac{L}{D} \right) \frac{\rho u^2}{2g} \quad (1)$$

where:

ΔP = pressure difference across wall

K_1 = inlet coefficient
 K_2 = exit coefficient $\left. \vphantom{\begin{matrix} K_1 \\ K_2 \end{matrix}} \right\} K_1 + K_2 = K$

f = Fanning friction coefficient

L = wall thickness

ρ = mass density

u = velocity through the gap

D = equivalent diameter = $4 \times$ hydraulic radius

Since the flow is expected to be laminar (which can be checked by the Reynolds number),

$$f = \frac{16}{R_e} \quad \& \quad R_e = \frac{u \rho D}{\mu} ; \mu = \text{viscosity}$$

The equivalent diameter of a slot $t \times h$ is:

$$D = 4 r_n = 4 \frac{\text{Area}}{\text{Perimeter}} = 4 \frac{th}{2h + 2t} , \quad r_n = \text{hydraulic radius}$$

$$= \frac{2t}{1 + t/h} \text{ as } h \gg t$$

$$D \geq 2t , \text{ where } t = \text{crack width and } h = \text{crack length along wall}$$

Hence:

$$f = \frac{16 \mu}{u \rho D} = \frac{8 \mu}{u \rho t}$$

By substitution of the above quantities, Equation (1) becomes:

$$\Delta P = K \frac{\rho u^2}{2g} + 4 \frac{L}{D} \cdot \frac{8 \mu}{u \rho t} \cdot \frac{\rho u^2}{2g} \quad (2)$$

The above equation can be solved for the flow velocity through the crack as

$$u = \frac{8 \mu L}{K \rho t^2} \left(\sqrt{1 + \frac{g \Delta P K \rho t^4}{32 \mu^2 L^2}} - 1 \right) \quad (3)$$

Volumetric Flow is:

$$V = u t h ;$$

$$\text{or } V/h = u t$$

$$= \frac{8 \mu L}{K \rho t} \left\{ \left(1 + \frac{g \rho \Delta P}{32} \frac{K t^4}{\mu^2 L^2} \right)^{1/2} - 1 \right\} \quad (4)$$

Reynolds Number is:

$$R_e = \frac{u \rho 2t}{\mu} = \frac{16 L}{K t} \left\{ \left(1 + \frac{g \rho \Delta P}{32} \frac{K t^4}{\mu^2 L^2} \right)^{1/2} - 1 \right\} \quad (5)$$

For external wind at velocity N , where N is in m/sec

$$\Delta P = \Delta P_{\text{static}} + \frac{\rho N^2}{2g}$$

$$\Delta P_{\text{static}} = 2.5 \text{ cm H}_2\text{O (1-in. H}_2\text{O)} = 25.4 \text{ kg/m}^2$$

$$\Delta P = 25.4 + 0.0578 N^2$$

If N is to be in MPH,

$$\begin{aligned} \Delta P &= 25.4 + 0.0578 \left(\frac{1609}{3600} N \right)^2 \\ &= 25.4 + 0.01155 N^2 \text{ kg/m}^2 \end{aligned}$$

For the following conditions:

Std. Air Conditions

$$T = 21^\circ\text{C}$$

$$\rho = 1 \text{ atm} = 10335 \text{ kg/m}^2 (14.7 \text{ psia})$$

$$\mu = 189 \times 10^{-7} \text{ kg/m-sec} (127 \times 10^{-7} \text{ lb/ft-sec})$$

$$\rho = 1.135 \text{ kg/m}^3 (0.07085 \text{ lb/ft}^3)$$

$$L = 1.83 \text{ m (6-ft)}$$

$$K_1 = 0.5^*$$

$$K_2 = 1.0^*$$

$$K = 1.5,$$

*"Handbook of Hydraulic Resistance Coefficients of Local Resistance and of Friction,"
I. E. Idelchik, AEC-TR-6630 (1966).

the terms in Equations (4) and (5) are as follows:

$$\frac{g \rho \Delta P}{32} \times \frac{K t^4}{\mu L} = (209545.17 + 95.28 N^2) t^4$$

where:

N = wind velocity in mph and

t = crack width in m.

$$\frac{8 \mu L}{K \rho t^2} = \frac{1.625 \times 10^{-4}}{t^2} \text{ (m/sec)}$$

$$\frac{8 \mu L}{K \rho t} = \frac{9.75 \times 10^{-3}}{t} \text{ (m}^3\text{/min-m)}$$

$$\frac{16 L}{K t} = \frac{64}{t} \text{ No Dimensions}$$

The parametric flow behavior of a crack from Equation (4) is shown in Figure 5.3-1.

5.3 ESTIMATE FOR DESIGN LEAKAGE

The crack dimensions and external wind speed must be arbitrarily assumed. Fifteen cracks from the gallery floor to the cell ceiling are assumed and half of the cracks are of the width of 0.04 cm (1/64-in.) and the other 0.08 cm (1/32 in.), at a wind speed of 30 mph. Since the cell ceiling height is 10.7 m (35-ft.) above ground level, there would be a total of 160 m of crack length. The specific leakage at 30 mph for the assumed crack sizes would be:

$$V/h \text{ at } 0.04 \text{ cm} = 0.0047 \text{ m}^3/\text{min per meter of crack}$$

Figure 5.3-1 Air Leakage Flow Behavior In Wall Cracks at 2.5 cm H₂O Pressure

and V/h at 0.08 cm = 0.0381 m³/min per meter of crack

$$\text{therefore } V_{\text{total}} = (0.0047 + 0.0381) \cdot \frac{160}{2} = 3.4 \text{ m}^3/\text{min} \text{ (120 SCFM)} \\ \text{(at 30 mph)}$$

In still air, the equivalent leakage is:

$$V_{\text{total}} = (0.00334 + 0.027) \cdot \frac{160}{2} = 2.4 \text{ m}^3/\text{min} \text{ (86 SCFM)} \\ \text{(at 0 mph)}$$

For design specification purposes a value of 3.4 m³/min (120 SCFM) is chosen for an outside wind velocity up to 30 mph.

Since this analysis was completed, the test cell arrangement has changed considerably, but the 3.4 m³/min in leakage is still a reasonable design criterion.

6.0 MAXIMUM ALLOWABLE TF COIL WINDING TEMPERATURE by G. E. Driesen

6.1 DISCUSSION AND ANALYSIS

A structural evaluation of the TF coil copper (OFHC) windings was conducted in order to establish a maximum winding temperature from a structural integrity standpoint. This study was conducted during the conceptual design phase of the TFTR project so some simplifying assumptions were made. For example, the effects of a potential creep-fatigue interaction were not evaluated for the windings. Also not considered were the effects of any winding thermal gradients. Instead, the analysis model was selected on the basis of being sufficiently conservative in order to realistically evaluate a safe winding peak temperature level for reactor operation.

The analysis model utilized is one which evaluates imposed strains occurring in the winding circumferential direction. The model assumes complete restraint in the winding radial and circumferential directions, therefore, the computed strain is cyclic and completely reversed (tensile and compressive) in nature. This assumption is consistent with the nature of a winding external constraint which is in the form of interference with either the adjacent windings or the steel containment members of the TF coil. It has been assumed for the purpose of this model evaluation that no mechanical load strain exists in the winding circumferential direction. The result is that only cyclic thermal strain exists in this direction during TF coil operation, and this cyclic strain exhibits both elastic and plastic characteristics due to the completely reversed nature of the stress. This is shown schematically in the cyclic stress-strain diagram below.

where

$\Delta \sigma$ is the circumferential stress range per cycle

$\Delta \epsilon$ is the total circumferential strain range (elastic + plastic) per cycle

The plastic strain range, $\Delta \epsilon_p$, is represented by the width of the hysteresis loop in the figure.

A knowledge of cyclic stress-strain data and some simple manipulations are required in order for the allowable cyclic total strain (elastic plus plastic) to be determined which in turn can be used to evaluate the maximum allowable temperature level of the winding. The cyclic stress-strain data for copper which is presented in reference 1 (see page 6-3) is based on diametral type strain. These strains are measured as the change in the cross sectional area of a uniaxial test specimen. With the aid of the appendix in reference 1, diametral strain was related to longitudinal plastic strain (strain in the direction of loading) for use in the present analysis. A review of this method of conversion follows.

The plastic longitudinal strain range per cycle is related to the diametral strain range and the stress range per cycle as follows:

$$\Delta \epsilon_p = 2 \Delta \epsilon_d - \frac{2 \nu \Delta \sigma}{E} \quad (1)$$

where

$\Delta \epsilon_d$ = total diametral strain range (elastic + plastic)

ν = elastic Poisson's ratio = 0.3

E = elastic Young's modulus = 17.8×10^6 psi

Figure 15 of reference 1 indicates the relationship between $\Delta\sigma$ and $\Delta\epsilon_d$ at the point of failure in OFHC. This graph is reproduced in Figure 6.1.1 below.

Figure 6.1.1 Stress Range vs. Strain Cycles and $\Delta\epsilon_d$

1. Coffin, L.F. Jr. and Tavernelli, J. F., "The Cyclic Straining and Fatigue of Metals," Transactions of the Metallurgical Society of AIME, Vol. 215, October 1959, p. 794.

This data combined with equation (1) was used to show the relationship between $\Delta\epsilon_p$ and $\Delta\epsilon_d$ at OFHC failure. This relationship has been plotted in Figure 6.1.2. Since the stress limit of proportionality for annealed copper is very low ($\sim 2,000$ psi), the maximum total diametral strain range which could exist when the longitudinal plastic strain range is 0.0 in/in/cycle was found from equation (1) to be

$$\begin{aligned}\Delta\epsilon_d &= \frac{\nu\Delta\sigma}{E} \\ &= \frac{(0.3)(4,000)}{17.8 \times 10^6} \\ \Delta\epsilon_d &= 6.74 \times 10^{-5} \text{ in/in/cycle}\end{aligned}$$

This information yielded an additional data point for Figure 6.1.2 which is located approximately at the origin ($0, 6.74 \times 10^{-5}$). The dashed line portion of the curve in Figure 6.1.2 represents the extrapolation of data points to this computed data point.

The total longitudinal strain per cycle is equal to the sum of its elastic and plastic components. Therefore,

$$\Delta\epsilon = \Delta\epsilon_p + \Delta\epsilon_e$$

Since the elastic strain change per cycle can be approximated by,

$$\Delta\epsilon_e = \frac{\Delta\sigma}{E}$$

it can be shown from equation (1) that the total longitudinal strain per cycle is equal to

$$\Delta\epsilon = 2 \Delta\epsilon_d + \frac{\Delta\sigma}{E} (1 - 2\nu) \quad (2)$$

Experimental data which relates longitudinal plastic strain range and cycles to failure for annealed copper has been generated by reference 1. Figure 19 of reference 1 is reproduced in Figure 6.1.3.

Figure 6.1.2 Plastic Longitudinal Strain Range vs. Total Diametral Strain Range

Figure 6.1.3: Plastic Strain Range vs. Cycles to Failure

An examination of the experimental data curve for annealed copper in Figure 6.1.3 indicates that for a design life of 1×10^5 cycles of operation, the maximum allowable longitudinal plastic strain range per cycle is 0.001 in/in. Figure 6.1.2 implies that this is equivalent to a diametral strain range of 0.00074 in/in at annealed OFHC failure. The corresponding *stress* stress range was computed using equation (1) to be

$$\begin{aligned} \Delta \sigma &= (\Delta \epsilon_p - 2 \Delta \epsilon_d) \frac{E}{2\nu} \\ &= (0.001 - 2 \times 0.00074) \left(\frac{17.8 \times 10^6}{2 \times 0.3} \right) \\ \Delta \sigma &= 14,200 \text{ psi} \end{aligned}$$

The total permissible thermal strain range per cycle was computed using equation (2) as

$$\Delta \epsilon = (2) (0.00074) + \frac{14,200}{17.8 \times 10^6} \left[1 - 2(.3) \right]$$

$$\Delta \epsilon = 0.0018 \text{ in/in/cycle}$$

This value is equivalent to an allowable winding mean temperature change of

$$\Delta T_m = \frac{\Delta \epsilon}{\alpha}$$

$$= \frac{0.0018}{9.8 \times 10^{-6}}$$

$$\Delta T_m = 184^{\circ}\text{F} (102^{\circ}\text{C}) \quad \text{from room temperature, } 70^{\circ}\text{F} (21^{\circ}\text{C})$$

where T_m is the mean temperature for an entire winding turn of copper. This would imply a maximum allowable winding mean temperature of $254^{\circ}\text{F} (123^{\circ}\text{C})$. To be conservative, it is felt that the winding peak temperature should not exceed about $250^{\circ}\text{F} (121^{\circ}\text{C})$.

6.2 CONCLUSIONS

Although the allowable copper temperature of $250^{\circ}\text{F} (121^{\circ}\text{C})$ was computed based on a conservative method of analysis, several qualitative observations concerning OFHC material characteristics may be made which tend to support this computation. For example, hardened copper begins to anneal at temperatures above $300^{\circ}\text{F} (149^{\circ}\text{C})$. A copper winding that possesses any cold working (strain hardening) due to winding fabrication will have a greater resistance to creep than one which is fully annealed. Another adverse effect of this annealing would be that an additional amount of plastic strain would be encountered in the winding circumferential direction for each thermal cycle of operation. Such a phenomenon might result in a decrease in the safe operating life of the windings.