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**Westinghouse Electric Corporation
Fusion Power Systems Department**

MASTER

**THERMAL AND HYDRAULIC ANALYSES
OF TFTR COOLING WATER SYSTEM
AND MAGNETIC FIELD COILS**

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THERMAL AND HYDRAULIC ANALYSES OF TFTR COOLING
WATER SYSTEM AND MAGNETIC FIELD COILS

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ABSTRACT

The TFTR toroidal field coils, ohmic heating, hybrid and equilibrium field coils are cooled by water from the machine area cooling water system. The system has the following major equipment and capacities: flow rate of 3600 gpm; ballast tank volume of 5500 gal; pumps of 70.4 m head; chiller refrigeration rating of 3300 tons and connecting pipe of 45.7 cm I.D. The performance of the closed loop system was analyzed and found to be adequate for the thermal loads. The field coils were analyzed with detailed thermal and hydraulic models, including a simulation of the complete water cooling loop. Under the nominal operating mode of one second of toroidal field flat top time and 300 seconds of pulse cycle time, the maximum temperature for the TF coils is 53°C; for the OH coils 46°C and for the EF coils 39°C, which are well below the coil design limit of 120°C. The maximum TF coil coolant temperature is 33°C which is below the coolant design limit of 100°C. The overall pressure loss of the system is below 6.89×10^5 Pa (100 psi). With the given chiller refrigeration capacity, the TF coils can be operated to yield up to 4 seconds of flat top time. The TF coils can be operated on a steady state basis at up to 20% of the pulsed duty design current rating of 73.2kA/coil (36.6kA/conductor).

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1.0 INTRODUCTION

This topical report presents the updated thermal and hydraulic analyses performed for the TFTR cooling water system, the toroidal field coils, ohmic heating, hybrid and equilibrium field coils, based on the June, 1975 conceptual design. ⁽¹⁾

The cooling water system has since been more completely defined and preliminary layout drawings prepared, so that a better hydraulic model for the closed loop system was used for analysis. In this design of the toroidal field (TF) coil, two pairs of parallel conductors are enclosed by a split box beam. The design basis toroidal field current is 73.3 kA per coil or 36.6 kA per conductor winding. The single cooling hole in the conductor is 0.953 cm (3/8") in diameter. The ohmic heating and the hybrid coil turns are 3.81 cm (1.5") square copper with a 0.953 cm (3/8") diameter cooling hole in the center of the cross section. The equilibrium field coil turns are rectangular with a cross section of 5.08 x 5.7 cm (2.0 x 2.25") and also a 0.953 cm diameter cooling hole in the center. The ohmic heating and hybrid coils carry a current of 40 kA and the EF coils carry a current of 80 kA for a 2.5 MA plasma.

The design objective of the cooling water system is to remove the heat generated in the field coils and the electrical power equipment and reject the heat through water chillers. The refrigeration capacity should be sufficient to conduct up to 96 pulsing cycles a day. The nominal TF power flat top time is 1 second and the cycle time is 300 seconds. The water flow rate should be sufficient to keep the peak temperature of the field coil materials below structural design limits under cyclic loading.

In this report the analysis methods and models are described and the results of the analysis presented.

2.0 SUMMARY

In this conceptual design the chiller capacity specified is 11.3 MW (3300 tons of refrigeration) at a flow rate of 3600 gpm, while the total thermal load is 9.9 MW. The system is adequate to allow design basis system operation (up to 96 full power pulses per day, with a five minute repetition period).

For normal operations of one second of TF flat top power and 300 seconds of cycle time, the maximum toroidal field coil temperature is 53°C and the maximum coolant temperature is 33°C. The maximum ohmic heating and hybrid coil temperature is 46°C and that of the equilibrium field coil 39°C. These temperatures are well below the respective design temperature limits of 120°C for the coil material and 100°C for the coolant. With the specified cooling capacity, the field coils can be operated with a flat top time of up to 4 seconds.

Under a faulted condition, such as toroidal field coolant channel flow blockage, a TF coil can be operated with a reduced flow rate to 1/3 of its nominal design value with one second of flat top time. With a loss of chiller refrigeration, tests can be conducted with up to 6 pulses starting from ambient conditions, because of the heat storage capacity of the ballast tank.

If the discharge cleaning method is used to clean the vacuum vessel wall, the toroidal field coils can be operated for steady state operation with up to 20% of the maximum field current of 36.6 kA/conductor. The constraint in this case is the chiller refrigeration capacity, not the coil material temperature, which is well below the design limit of 120°C.

3.0 COOLING WATER SYSTEM PERFORMANCE

3.1 INTRODUCTION

The water cooling system circulates water to cool the toroidal field (TF), ohmic heating (OH) and the equilibrium field (EF) coils and to remove indirectly the heat generated in the rectifiers in the power conversion systems. The system consists of the following major components:

- (1) Field coils to be cooled
- (2) Ballast tank
- (3) Pumps
- (4) Chillers, and
- (5) System piping

For normal operations (1 second of flat top time in a pulse), the thermal energy generated includes 2645 MJ in the TF coils; 45 MJ in the OH coils, 52 MJ in the EF coils and 220 MJ in the rectifiers. The water cooling system is designed to remove this quantity of heat in one pulse cycle time (300 seconds). To perform thermal and hydraulic analysis of the field coils, the entire closed loop system was modeled using the TAP-B code.⁽²⁾ With this model, single and multiple pulse operations can be analyzed. This section describes the performance of the cooling system in cooling the field coils. The detailed description of the equipment and arrangement of the system is given in Volume II of the final TFTR Conceptual Report.

3.2 COOLING WATER SYSTEM MODEL

The water cooling system is designed to completely remove the heat generated in the field coils and the rectifiers during a pulse cycle. The arrangement of the major components of the cooling system for the current conceptual design is shown schemetically in Figure I.

In the thermal modeling of the cooling system, since the TF coil performance is of primary concern and the heat generated in the OH and EF coils are small compared to that from the TF coils, only the TF coils were modeled in the system. A chiller capacity of 3000 tons of refrigeration, out of a total of 3300 tons available, was assumed to be used to cool the TF coils, leaving the remaining 300 tons to cool the rectifiers, OH and EF coils. The heat exchanger for the rectifiers was therefore omitted in the model. The ballast tank volume in the model was the portion of the total volume allotted to the TF coils. The system was then represented by a 3 channels and 4 plena model as shown schematically in Figure 2. Channel 1 is the coil to be analyzed. Channel 2 is the chiller water passage (tube side) and Channel 3 the connecting pipe. The diameter of the pipe was sized to meet a water velocity requirement of 1.52 m/sec (5 ft/sec). Plenum 1 is the coil inlet manifold which was assumed to have a large volume. Plenum 2 is the ballast tank which has a finite volume specified. Plena 3 and 4 are pseudo plena with zero volume. The pump was simulated by a negative flow resistance at the inlet to the chiller water passage. The ballast tank was assumed to be pressurized and maintained at 1.04×10^5 Pa (15 psig). The overall pressure losses of the loop is designed to be below 6.89×10^5 Pa (100 psi).

3.3 WATER CHILLER PERFORMANCE

The water chillers are specified to deliver chilled water at 10°C. At a flow rate of 3600 gpm and 3000 tons of refrigeration, the chiller water inlet temperature is 23.6°C. When the water inlet temperature is below 23.6°C, the chillers operate at partial capacity so as to maintain the outlet water temperature at 10°C. When the water inlet temperature is above 23.6°C, the chillers operate at full capacity producing the maximum temperature drop of 13.6°C. In this way the water outlet temperature would be higher than the desired temperature of 10°C by the amount the inlet temperature exceeds 23.6°C. This operating characteristic of the chiller was programmed into the TAP-B code to simulate the chiller heat removal.

Before the detailed analysis of the TF coil was performed, the chiller location relative to the ballast tank in the flow loop was investigated. It was determined that for the most effective usage of the chillers, the units should be located downstream of the ballast tank. The comparison of the TF coil exit temperature and the chiller water inlet and outlet temperatures for the two different chiller locations (downstream vs upstream) of the ballast tank is shown in Figure 3.

In this comparison the TF coil winding was assumed to be cooled one complete turn per channel, even though the current cooling scheme is two complete turns per channel. The conclusion obtained from the results shown in Figure 3 should also apply to the current scheme. It is seen from the comparison that the chiller inlet temperature exceeds the chiller operating temperature range for a much longer time, by a greater amount, with the chillers upstream of the ballast tank than with the chillers located downstream of the ballast tank. In order to minimize the chiller water inlet temperature oscillation during each cycle, the chillers are to be located downstream of the ballast tank. This cooling system model was used to perform the detailed TF coil thermal analysis presented in the next section.

The system design summary is shown in Table 1. It is seen from the table that the thermal load during one pulse (with one second of flat top time) is 9.9 MW while the chiller capacity specified is 11.3 MW (3300 tons of refrigeration). The system is therefore, adequate to meet the nominal cooling requirement.

3.4 ESTIMATED PRESSURE LOSSES OF THE COOLING SYSTEM

The pressure losses along the cooling system based on a flow rate of 3600 gpm was estimated and summarized in Table 2. The sum of the losses is about 5.89×10^5 Pa (85.3 psi). A design value of 6.89×10^5 Pa (100 psi) was allowed to include any additional losses from connectors, coupling, valves, elbows, etc. which are not well defined at this time for the conceptual design. It appears that the 6.89×10^5 Pa (100 psi) pump head specified for the pumps is adequate.

4.0 TOROIDAL FIELD COIL ANALYSIS

4.1 INTRODUCTION

The conceptual design of the TF coil consists of two separate parallel conductors, insulated from each other electrically. Two such pairs in series are inside a box beam to form a complete coil. Each conductor is wound into 22 turns and is cooled by water in a single channel. In one pair of winding, one conductor has a constant cross sectional area throughout, while the other has variable areas at the shaved region. The minimum area occurs at the midsection of the shaved region. Since the resistance heating of the material is inversely proportional to the square of the cross sectional area, it is expected that the peak coil temperature would occur at the location where the area is smallest and the distance from the location to the cooling hole surface is longest. Analysis was performed therefore, for the worst turn of the TF coils. In cooling the coils two schemes were investigated. One is cooling by one turn per channel and the other by two turns per channel. Analyses were carried out to determine the following:

- (1) the maximum TF coil and coolant temperature in multiple pulse operation for the two different ways of cooling
- (2) the effect of flat top time on TF coil peak temperature
- (3) TF coil capability at certain faulted conditions.
- (4) steady state TF coil current limit

4.2 THERMAL DESIGN CRITERIA

The coil temperature design criterion is obtained from a structural integrity consideration. A cyclic plastic strain calculation, assuming the copper is fully constrained, indicated that the maximum allowable copper temperature is 120°C for 10^5 cycles of operations⁽³⁾. To be

on the conservative side this mean temperature limit was then assumed to be the allowable peak temperature in the winding. For the coolant temperature design limit, from convenience and safety considerations, it is determined that no flashing should occur in the event of the rupture of a pressurized connection, hence a normal operating water temperature limit of 100°C was imposed. The conductor insulating material temperature limit is about 150°C , which is above the copper material temperature limit of 120°C discussed above, hence it is not a governing criterion. The pressure drop for each coil channel was taken to be no more than 5.2×10^5 Pa (75 psi) for practical purposes, and the pressure drop for the overall water system assumed to be less than 6.9×10^5 Pa (100 psi). The piping and the related equipment are to be designed, however, for a pressure rating of 10.3×10^5 Pa (150 psi).

4.3 METHOD AND MODEL OF ANALYSIS

The analyses were carried out using the TAP-B code and models of the worst turn of the TF coil and the water cooling system. The cross section of the coil at the midsection of the shaved region for the current conceptual design is shown in Figure 4 and the net areas for the 44 turns of the winding are tabulated as shown in Table 3. The table shown that turn no. 4 has the smallest cross sectional area at the midsection of the shaved region. It is expected that the peak coil temperature would occur at this turn, which was thus modeled and analyzed. The scheme of cooling the coils by one turn per channel or by two turns per channel was first investigated. In cooling two turns per channel the water enters turn no. 1 and exits before cooling turn no. 3. Another parallel channel enters turn no. 3 and exits before cooling turn no. 5 and so on. The model for turns no. 3 and 4 is shown in Figure 5. The shaded areas shown on the sectional view at the top of the figure are the portion of conductors modeled, because the hot spot would be on this farther side of the section away from the cooling hole. The internal heat generation rate at any location of the coil was calculated, however, based on the net coil area of the entire cross section at that location. For the one

turn per channel model, turn no. 4 alone was included in the model as shown by the lower part of the isometric sketch in the figure (with internal nodes from 151 to 300). For the two turn per channel model, both turns no. 3 and 4 were included (with internal nodes from 1 to 300). The coil heat conduction model is hence, 3-dimensional with 5 nodes on a transverse plane (normal to the direction of coolant flow).

The internal heat generation rates in the coil due to resistance heating was calculated by inputting the field current as function of time and the copper electrical resistivity as a function of temperature. The TF coil current profile during a pulse is shown in Figure 6 in which the OH and EF coil current profiles are also shown. The analysis of the OH and EF coils are presented in the next section. The copper resistivity curve is shown in Figure 7. It is noted that the TF current profile assumed a 14 seconds build-up time and a 7 seconds decay time. These timings are on the rather conservative side. The longer times result in a higher integrated energy generation. The coil temperature calculated on this basis will be on the high (conservative) side.

The overall water system incorporates the TF model(s) described above together with the chiller and ballast tank to form a closed loop analysis model. To start the calculation the initial coil and water temperatures were taken to be about 10°C . This initial condition can be obtained by chilling the entire system from ambient condition by circulating water in the loop with the chillers on and without field current. The chill down would take about 400 seconds.

In modeling the water system for cooling the TF coils by two turns per channel, and keeping the coolant velocity in the channel the same at 3.3 m/sec, as in the 1 turn per channel cooling method, the water system flow rate was reduced from 7200 gpm to 3600 gpm and the ballast tank volume from 11000 gal. to 5500 gal. The former numbers are in the design to cool the TF coils by one turn per channel. The reduction in hardware is quite significant.

4.4 RESULTS OF ANALYSIS

(a) Cooling by one turn per channel vs by two turns per channel

The calculations considered the nominal operating conditions of one second of flat top time and 300 seconds of cycle time. Starting from the first pulse, the peak coil temperature of the hottest coil increases slightly after each pulse until an equilibrium condition is reached in which the heat generated can be removed by the water during the cooldown period of the cycle. The comparison of the calculated coil and coolant temperatures of the worst turn (turn no. 4) for the two cooling schemes at the 7th pulse in which equilibrium condition was first reached, are shown in Figure 8 and 9. These figures show that there is little difference (about 1°C) between the two schemes during the heat up period. The appreciable differences are in the cooldown period. The reason can be readily seen from Figures 10 and 11. These two figures show the fluid temperature and the cooling channel wall temperature axial distributions along the cooling channel at the time the coil hot spot temperature is maximum (19 seconds from initiation of the pulse) for the two cases. It should be noted that because of the larger cross sectional area of turn no. 3 (see Table 3), the heat generation rate in turn no. 3 is about half of that in turn no. 4. The heat released to the coolant from turn no. 3 raises the coolant temperature to 21°C (from the inlet value of 10°C), when the coolant enters turn no. 4. The higher coolant temperature at turn no. 4 reduces the heat flux from the coil to the fluid in turn no. 4. Since the difference between the channel wall temperature and the fluid temperature is proportional to the heat flux, it is seen from these figures that the heat flux in turn no. 4 for the one turn per channel case is higher than that for the two turns per channel case. During the heating up, part of the heat generated in turn no. 4 is stored in the material and is later removed by the coolant during cooldown. It is seen in Figure 8 that the peak and the average temperatures of turn no. 4 for the two turns per channel case, remained higher and for a longer time during the cooldown period. In both cases, the total heat generated

was entirely removed by the end of the cycle.

Certainly, at steady state the coolant temperature rise in a two turn channel would be higher than in a one turn channel because the total amount of heat released to the fluid would be higher for the same coolant flow rate. Based on the results of comparison presented above, the scheme of cooling the TF coils by two turns per channel was used in the current conceptual design.

The maximum and the average coil temperatures of the worst turn as shown in Figure 8 are about 53°C and 34°C respectively. These temperatures are below the coil design temperature limit of 120°C . The maximum coolant temperature as shown in Figure 9 is 33°C and is below the desired water temperature limit of 100°C . The coolant channel pressure loss is $3.72 \times 10^5 \text{ Pa}$ (54 psi) for the two turns per channel case and $1.65 \times 10^5 \text{ Pa}$ (24 psi) for the one turn per channel case (for the same coolant velocity of 3.3 m/sec). The losses are within the desired overall pressure loss of $6.89 \times 10^5 \text{ Pa}$ (100 psi). A design summary of the TF coils is shown in Table 4.

(b) Effect of Flat top time on TF coil temperature

With the model of cooling the TF coils by two turns per channel, the effect of extending the flat top time longer than one second on the maximum coil temperature was analyzed. The results are shown in Figure 12 and 13. The solid line AB in Figure 12 shows that the TF coils can be operated with a flat top time up to 4 seconds. The constraint is the cooling system refrigeration capacity. With a four seconds of flat top time, the maximum coil temperature is only about 75°C . The maximum transverse temperature difference is 14.5°C and the maximum coolant temperature is 46°C . But the cooling load requirement is already slightly above 3000 tons which is the available chiller capacity for the TF coils. Beyond four seconds of flat top time, the peak coil temperature continues to increase after each pulse as shown by the dashed lines in Figure 12. It is interesting to note that the TF coils can still be operated for 12 pulses with a flat top time of 7 seconds, or 8 pulses with a flat top time of 8 seconds and so on.

(c) Flow blockage and loss of chiller refrigeration

Two abnormal or faulted operating conditions were investigated: (1) coolant flow channel blockage and (2) loss of chiller refrigeration. The analyses were carried out using the model with cooling two turns per channel. For channel flow blockage it is conceivable that some particles or dirt in the water system could partially block a channel flow so that coolant flow in that channel is substantially reduced. The case with a one second flat top and a coolant flow rate reduction to 1/3 of its normal rate was analyzed. The calculated coil peak and average temperatures and the maximum coolant temperature of the worst turn (turn no. 4) are shown in Figure 14. The peak coil temperature increases to 65°C and the maximum coolant temperature to 49°C. These temperatures are still below the design limits.

For the case of loss of chiller refrigeration, normally, this is to be considered a system breakdown. However, with a ballast tank of 5500 gallons of water at ambient condition, tests could still be conducted for a few pulses. Because of the heat storage capacity of the water in the ballast tank, the TF coils can still be operated for 6 pulses before the coil and the coolant temperatures reach their respective limits as shown in Figure 15.

(d) Steady state operation of TF coils

A steady state capability to operate the TF coils may be needed if the plasma discharge method is used for cleaning the vacuum vessel wall. Steady state analysis was performed and the results shown in Figure 16. With 3000 tons of chiller refrigeration capacity, the maximum permissible steady state TF coil power is about 4% of full power, or about 20% of the maximum coil current of 36.6 kA/conductor. The coil and coolant temperatures are well below the design limits. The constraint is therefore the available cooling capacity of the water system.

5.0 OHMIC HEATING, HYBRID AND EQUILIBRIUM FIELD COILS ANALYSIS

5.1 INTRODUCTION

The ohmic heating (OH), hybrid (HY) and the equilibrium field (EF) coils are located outside and surround the toroidal coils at different horizontal planes. For one pulse with one second of flat top time, the heat generated in the OH coil is 42 MJ or about 1.7% of that from the TF coils and that in the EF coils 52MJ or about 2% of that from the TF coils. The OH, HY and EF coils are also cooled by water from the machine area water cooling system. Preliminary thermal and hydraulic analysis was performed to determine the coolant flow rate, maximum coil and coolant temperatures. The same mechanical design temperature limits as those for the TF coils are applied to these coils to judge the design adequacy.

5.2 METHOD OF ANALYSIS

The design of the OH, HY and EF coils analyzed and reported herein, is the version having the following dimensions: 3.81×3.81 cm (1.5×1.5 " square) for the OH and HY coils and 5.08×5.72 cm (2.0×2.25 " rectangular) for the EF coils. The single cooling hole per conductor is 0.95 cm ($3/8$ ") in diameter. The total length of the coils was calculated to be about 2245 m. One half of this total is located at the top half of the torus and the other half at the bottom half of the torus. A numerical break-down of the total length of the coils is shown in Table 5.

Within the overall pressure drop and the available coolant from the cooling water system, 40 parallel cooling channels and a flow rate of 0.091 kg/sec (0.2 lb/sec) per channel were selected for analysis. The length of each channel is then 56.1 m. For the purpose of a preliminary study of this conceptual design, the total length of the coils was equally divided into 40 channels, disregarding where the break points of the channels would be relative to the actual mechanical arrangement of the turns and the water line connections.

This detail will be available only when the detailed mechanical design of the water lines and connections are defined.

The analysis models were simple one channel models for each of the OH and the EF coils in which the channel inlet water temperature was assumed to be constant at 10°C . The closed loop water system model was not needed, because the heat generated and the quantity of water used to cool the coils are small compared to those for the TF coils. The effect of the OH or the EF coils on the performance of the ballast tank and chillers are minimal. Also, since the OH and EF coils cross section are small and rather regular and the copper thermal conductivity is very large, the transverse temperature gradient at a cross section would be small. There is no need to divide a cross section into several internal nodes. A channel was therefore, divided into 80 uniform increments for practical purposes. Each increment is one material node. The field current profiles for the OH and EF coils were shown previously in Figure 6. Multiple pulse calculations were carried out to determine the maximum coil temperature and coolant pressure loss in the channel.

5.3 RESULTS OF ANALYSIS

The results of analysis are summarized as shown in Table 6. It is seen from the table that under normal operating conditions the maximum temperatures of the OH and EF coils are below the copper peak temperature design limit of 120°C for flat top time up to four seconds, which is the upper limit of flat top time for the TF coils discussed previously. Under faulted conditions when the equilibrium field is disrupted, the full amount of EF current would be imposed on the HY coils. The maximum hybrid coil temperature becomes 75°C with one second of flat top time and over 150°C with four seconds of flat top time. In order to have four seconds of flat top capability for all the coils, additional coolant or a shorter channel length for the HY coils would be required. Additional cooling water is available from the cooling water system. In fact, the pressure drop in the OH and EF coils channels with a flow rate of

0.091 kg/sec (0.2 lb/sec), is 1.89×10^5 Pa (20 psi), while the pressure drop in the TF coils is 3.72×10^5 Pa (54 psi, see Table 2). The water flow rate for the hybrid coil could be increased to about 0.15 kg/sec (0.33 lb/sec). A flow balance of the water cooling system for the conceptual design is shown in Table 7. It is seen that the water system flow rate of 3600 gpm and overall system pressure loss of 6.89×10^5 Pa (100 psi) are adequate to meet the thermal loads cooling requirements

TABLE 1
COOLING WATER SYSTEM DESIGN SUMMARY

Water Chiller capacity	11.3 MW (3300 Tons)
Main Circulating Pump Capacity 2 at 1800 gpm and $6.9 \times 10^5 \text{ N/m}^2$ (100 psi)	3600 gpm
Pump Motor Capacity 2 at 100 HP each	200HP
Ballast Tank	11000 GAL *
Distribution Line Size	45.72 cm (18 in.)
TF, OH, EF & SCR Thermal Loads (1 second flat top)	9.9 MW
TF = 8.84 MW	
OH = 0.15 MW	
EF = 0.17 MW	
SCR = 0.74 MW	
Estimated System Pressure Loss	$5.89 \times 10^5 \text{ Pa}$

* Ballast tank volume of 5500 gallons was used in analysis.

TABLE 2
ESTIMATED PRESSURE LOSSES IN WATER COOLING SYSTEM LOOP AT 3600 GPM

<u>Item</u>	<u>Pressure Loss</u>	
	(10 ⁵ Pa)	(psi)
1. Inlet Manifold to TF coil inlet	0.32	4.6
2. In TF coil, 2 turns/channel	3.72	54.0
3. Coil exit to outlet manifold	0.36	5.2
4. Ballast tank to chiller outlet	0.52	7.5
5. Filter	0.14	2.0
6. SCR heat exchanger	0.14	≈ 2.0
7. Miscellaneous piping system, connectors, valves, etc.	<u>0.69</u>	≈ <u>10.0</u>
Total	5.89	85.3

Design system overall pressure losses of 6.89×10^5 Pa (100 psi)

TABLE 3
CROSS SECTIONAL AREAS OF TF COIL
CONDUCTORS

<u>Turn No.</u>	<u>Minimum Area at Shaved Region</u>		<u>Area at Unshaved Region</u>	
	(cm ²)	(in ²)	(cm ²)	(in ²)
1	13.55	2.10	24.19	3.75
2	14.52	2.25	24.19	3.75
3	15.48	2.40	24.19	3.75
4	11.55	1.79 *	17.35	2.69
5	12.06	1.87	17.35	2.69
6	12.65	1.96	17.35	2.69
7	13.16	2.04	17.35	2.69
8	13.68	2.12	17.35	2.69
9	11.61	1.80	14.26	2.21
10	11.99	1.86	14.26	2.21
11	12.32	1.91	14.26	2.21
12	12.71	1.97	14.26	2.21
13	13.23	2.05	14.26	2.21
14	13.42	2.08	14.26	2.21
15	13.81	2.14	14.26	2.21
16	14.13	2.19	14.26	2.21
17 - 22	14.26	2.21	14.26	2.21
23 - 44	13.74	2.13	13.74	2.13

* Minimum area at turn No. 4

TABLE 4
TF COILS THERMAL DESIGN SUMMARY,
ONE SECOND FLAT TOP TIME & 300 SECONDS PULSE CYCLE TIME

Number of Coolant Channel Per Conductor	1
Conductor Turns Per Coil	88
Number of Parallel Cooling Channel Per Coil (2 turns/channel)	44
Coolant Velocity in Coil, m/sec	3.3
Coolant Hole Diameter, cm	0.953
Maximum Coil Temperature, °C	53.0 *
Maximum Coil Average Temperature, °C (at turn no. 4)	34.0
Maximum Transverse Temperature Difference, °C	12.0
Maximum Coolant Temperature, °C	33.0
Coil Channel Fluid Pressure Loss, Pa	3.72×10^5
Heat Transfer Coefficient, $W/cm^2 - ^\circ C$	1.17
Maximum Heat Flux, kW/cm^2	0.019

* Based on initial coil temperature of 10°C

TABLE 5
SUMMARY OF OH, HY & EF COILS LENGTH

	<u>Coil</u>	<u>No. Of Times</u>	<u>Length, m</u>
(1) Inboard of Vacuum Vessel	OH	164	1121
	HY	34	218
	EF	6	110
(2) Outboard of Vacuum Vessel	OH	4	112
	HY	2	62
	EF	22	622
Total		232	2245

One half of the total length is located above and the other half below the equatorial plane of the vacuum vessel symmetrically.

TABLE 6

SUMMARY OF OH, HY & EF COIL COOLING DESIGNS

Coil	Dimensions	Peak Field Current (KA)	Flat Top Time (Sec.)	Max. Coil Temp. (°C)	Max. Axial ΔT (°C)	Max. Radial ΔT (°C)	Channel $10^5 \Delta P$ Pa
OH	3.81 x 3.81 cm	40	1	46	28	1.0	1.38 ⁽²⁾
			4	88	60	2.0	1.38
Hy	3.81 x 3.81 cm	(80) ⁽¹⁾	1	75	52	3.0	1.38
			4	>150	-	-	-
EF	5.08 x 5.72 cm	80	1	39	24	2.0	1.38
			4	114	87	3.5	1.38

Operating Conditions: (a) No. of parallel channels = 40
 (b) Length of Channel = 56.1
 (c) Flowrate per channel = 0.091 kg/sec (0.2 lb/sec)

Notes: (1) - Hybrid coils under faulted condition at which the EF field current is applied to the coils, resulting in a higher coil temperature than if the nominal current is applied to the hybrid coils.

(2) - The 1.38×10^5 Pa (20 psi) is the momentum and frictional pressure losses in 56.1 m (2209 in) of straight through channel. Losses of hardware connections within the channel should be added to obtain the overall pressure loss.

TABLE 7
SUMMARY OF FLOW BALANCE OF WATER COOLING SYSTEM

	<u>TF</u>	<u>OH, HY & EF</u>	<u>Total</u>
Number of Coils	20	2245 ⁽¹⁾	
Channels per Coil	44	40 ⁽¹⁾	
Flow per Channel, kg/sec	0.235 ⁽²⁾	0.091 ⁽³⁾	
Total Flow in Coils, kg/sec	206.9	3.63	210.5
Total Water System Flow at 3600 gpm, kg/sec			226.8

Note: (1) Total length of coils is 2245 m divided equally into 40 parallel channels

(2) Channel coolant velocity = 3.3 m/sec

(3) Channel coolant velocity = 1.27 m/sec

6.0 REFERENCES

- (1) "Two Component Torus Joint Conceptual Design Study, Status Briefing", Plasma Physics Laboratory, Princeton, New Jersey and Westinghouse Electric Corporation, Pittsburgh, Pennsylvania, June 18, 1975
- (2) "TAP-B, A General Code For Coupled Hydraulic and Heat Conduction Analysis", A. Y. Lee, WANL DRM NO. 54 186, June 18, 1971.
- (3) "Collection of Preliminary Thermal & Mechanical Calculations for TFTR Components", E. A. De Zubay, G. Driesen, WFPS-TME 018, October, 1975

* OH and EF Coils are Omitted in TF Coil Analysis

** Heat Exchanger was Omitted in TF Coil Analysis but Heat Load was Subtracted From Chillers

Figure 1. TFTR Conceptual Design Cooling Water System Schematic Diagram

Figure 2. TF Coil Cooling System TAP-B Model

Figure 3. Comparison of TF Coil Coolant Temperatures with Two Chiller Locations

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Figure 5. TAP-B Analysis Model of TF Coil Turns

Figure 6. TF, OH & Ef Field Current Profiles For Thermal Analysis

Figure 7. Copper Electrical Resistivity Used In Thermal Analyses

Figure 8. Comparison of Coil Temperatures With Two Cooling Schemes

Figure 9. Comparison of Coolant Temperatures With Two Cooling Schemes

Figure 10. Axial Temperature Distribution of Turn No. 4 in Cooling By One Turn Per Channel At Time of Peak Coil Temperature

Figure 11. Axial Temperature Distribution of Turns No. 3 & 4 In Cooling Two Turns Per Channel At Time of Peak Coil Temperature

Figure 12. Effect of Flat Top Time on TF Coil Max Temperature

Figure 13. Effect of Flat Top Time on TF Coil Coolant Temperature

Figure 14. TF Coil Temperatures with Channel Flow Blockage to 1/3 Nominal Flow Rate

TF COIL COOLANT VELOCITY = 3.3 M/SEC

INITIAL TEMPERATURES = 20°C

BALLAST TANK VOLUME = 5500 GAL

Figure 15. TF Coil Temperatures With Loss of Chiller Refrigeration

Figure 16. Steady State TF Coil Operating Limits