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Fusion Energy Division

LIQUID NITROGEN COOLING CONSIDERATIONS OF THE COMPACT IGNITION TOKAMAK

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NOMENCLATURE

A	wetted area
A_c	coolant cross-sectional area, cm^2
A_f	coil cross section, cm^2
c_p	material specific heat, J/g K
D	diameter, cm
D_e	hydraulic diameter, cm
f	friction factor
f_c	coolant channel cross-section area fraction
f_v	void fraction
g	acceleration due to gravity, cm s^{-2}
h	heat transfer coefficient, $\text{W/cm}^2 \text{ K}$
h_{fg}	heat of vaporization, J/g
K	coolant thermal conductivity, W/cm K
L	channel length, cm
m	material mass
Q	total energy to be removed, J
q	heat rate, W
P	coolant pressure
P_w	wetted perimeter, cm
R	fractional pipe volume filled by liquid or vapor
T	temperature, K
t	cooling time to 80 K
x	vapor quality
V	coolant velocity, cm s^{-1}
μ	coolant viscosity, g/cm s

NOMENCLATURE (continued)

ρ	density, g/cm ³
W	coolant mass flow rate, g s ⁻¹

Subscripts

c	material
c_i	material, initial
e	exit condition
l	liquid
v	vapor
tpf	two-phase flow

Dimensionless Groups

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

ABSTRACT

A simple model was developed to estimate the cooldown time between pulses of toroidal field (TF) coils of the Compact Ignition Tokamak (CIT) using liquid nitrogen. Good agreement was obtained between the analysis results and those measured in the early fusion experimental devices. A cooldown time of about 1 h would reduce the TF coil temperature to about 80 K. An R&D experimental program is required to determine the actual cooldown time between pulses, an issue in the conceptual design of the CIT.

1. INTRODUCTION

During the past year, preconceptual design studies have been made on four different concepts for Compact Ignition Tokamak (CIT) devices to be built in the United States. All of these concepts utilize copper magnets for toroidal field (TF) and poloidal field (PF) coils. An Ignitor-type device has been chosen as the baseline for further conceptual design studies.¹ This machine is proposed to operate at around liquid nitrogen temperature to minimize the power requirements for the magnets. The temperature of the magnet systems is allowed to increase during the operating pulse, and cooldown is initiated at the end of the pulse. One of the design requirements is for cooldown time between full-power pulses to be less than 1 h in order to provide the capability for a reasonable number of operating pulses per day. A number of devices such as Alcator-A and -C² have operated around liquid nitrogen temperature, but these machines have much shorter pulses and produce no neutrons. Therefore, the temperature rise during each pulse is modest, and the cooldown is less of a problem than in an ignited device with burn times of several seconds.

The purpose of this work is to develop a simple model to estimate the cooldown rate of a magnet with various cooling surface configurations. This will help in proposing an optimum configuration that would minimize the cooling time between pulses. Since it is anticipated that cooling of the TF and ohmic heating (OH) coils would take longer than other PF coils due to higher current density, the analysis will be performed around the TF and OH coils. This report is concentrated around TF coil cooling, and OH coil cooling will be reported in the future. The material of the inner leg of the TF coil is composed of 25% Inconel 625, 65% copper alloy (CDA-102), and 10% insulation. The outer leg material is copper CDA-102 with the necessary insulation. The CIT horizontal and vertical build is shown in Fig. 1. The pulse length is about 9.75 s with a 3.75-s flat top. Based on the Kaye-Goldston (L-mode) confinement model, the average neutron wall loading is about 7.8 MW/m^2 during operations. Table 1 shows the temperatures and energy deposition in the various sections of the TF and

(ALL DIMENSIONS IN mm)

Fig. 1. CIT horizontal and vertical build.

Table 1. Temperatures and energy deposition in the TF and OH coils at the end of pulse
(pulse length = 9.74 s; flat top = 3.75 s)^a

TF coil			OH coil		
Region	Temperature (K)	Energy deposition (MJ)	Region	Temperature (K) (Divertor case)	Energy deposition (MJ)
Inboard	372 ^b -342 ^c	750	OH2 ^d	317	460
Outboard	100-125	530	OH1 ^d	222	146
Crown	~170				

^a Neutron wall loading is 7.8 MW/m².

^b Midplane, OH side.

^c Midplane, plasma side.

^d See Fig. G-1 of Conceptual Design Report (ref. 1).

the OH coils at the end of the pulse.³ Figure 2 shows the temperature distribution in the TF coil at the end of pulse.³ The initial temperature in this figure is 80 K. The highest temperature occurs in the coil inner leg, as shown in Table 1. Therefore, the study has been concentrated around the inner leg to minimize the cooldown time between pulses. The coil inner leg dimensions are shown in Fig. 3.

Four cooling configuration options have been chosen for cooling the inner leg:

1. Cooling channel at the plasma side of the TF coil inner leg.
2. Cooling channel at the OH side of the TF coil inner leg.
3. Cooling channel inside the TF coil. (Only 5% of available cross-section area is chosen for coolant passage so as to not significantly degrade the load-carrying capability of the inner leg.)
4. Cooling only through conduction to the rest of the TF coil.

For the first three options, it is proposed to introduce liquid nitrogen from the bottom of the TF coil to produce a vertical upward flow of the gas.

The cooling method proposed for the outer leg uses channels either inside or on the sides of the TF coils, with coolant flow from the bottom to the top of the TF coil.

2. METHOD OF ANALYSIS

2.1 HEAT TRANSFER

Despite the large number of experimental and analytical investigations conducted in the area of forced-convection vaporization, it is not yet possible to predict all of the characteristics of this process quantitatively. This is due to the many variables upon which this process depends, in addition to the complexity of the two-phase flow as the quality of the liquid-vapor mixture increases during the vaporization. According to Fig. 2, the temperature at the bottom of the TF coil is less than 100 K right after the pulse termination and gradually increases. This will allow liquid nitrogen entering the cooling channel to go through bubbly flow and slug-flow regimes as the flow quality

Fig. 2. TF coil temperature distribution after the pulse.

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SEPARATION BOUNDARY BETWEEN
INNER LEG AND THE CROWN FOR
COOLDOWN TIME CALCULATION

ALL DIMENSIONS IN cm

Fig. 3. Schematic of TF coil inner leg.

increases. These regimes occur only at very low qualities, and the main heat transfer mechanism is the boiling process.⁴ Depending upon the temperature difference between the surface and the coolant, the process could be either nucleate or film boiling. As the coolant flow progresses, the mixture quality increases and the coolant enters into the annular-flow regime. In this regime, the wall of the tube is covered by a thin liquid film, and heat is transferred through this liquid film. This regime continues up to a quality where the coolant enters a mist-flow regime.⁴ This regime continues until the quality reaches 100%. In this regime, the heat transfer from the surface is through the vapor flowing near the surface, and it is then transferred to the liquid droplets existing in the mist flow. After the mist-flow regime, the flow enters into the evaporated regime, where the heat transfer can be predicted by forced convection of a vapor in the coolant channel.

The heat transfer treatment, as a function of the mixture quality, requires knowledge of the time-dependent two-dimensional temperature distribution of the TF coil, which is beyond the purpose of this study. This type of analysis will be pursued in the future, but a simple zero-dimensional heat transfer model will be discussed here to estimate the cooldown time of the TF coil based on the following assumptions:

1. The temperature distribution along the length of the channel is uniform at any given time. The initial temperature of the coil is assumed to be a temperature that would give the same stored energy as would be obtained from the real temperature distribution.
2. The temperature distribution across the TF inner leg is uniform at any given time. This seems to be a reasonable assumption since the cooldown time is longer than the thermal diffusion time across the TF inner leg.
3. Nucleate and film boiling will be the only heat transfer mechanism along the coolant channel. This may not be an oversimplified assumption since (a) the temperature from the bottom of the TF coil up to about 40 cm is less than 100 K after the pulse termination (see Fig. 2) and gradually increases, thus leading to boiling at the

coolant entrance; (b) although the quality may well be above what is needed for film boiling as flow progresses into the channel, calculations show that the heat transfer rate to flowing vapor is of the same order of magnitude as film boiling at the mass flow rates and temperatures relevant to our situation; (c) due to unavailability of boiling heat transfer in a vertical upward forced-flow channel, another correlation based on nitrogen nucleate and film pool boiling was utilized.⁵ This is shown in Fig. 4, where the heat transfer rate is plotted vs the temperature difference between the wall and saturated nitrogen (ΔT) for various coolant pressures and tube diameters. However, the critical heat flux in vertical upward forced-flow boiling was calculated,⁶ and the results indicate that this value is about 8 times higher than the critical heat flux for nucleate pool boiling at atmospheric pressure. This suggests that the pool boiling approach may be a conservative approximation of the real flow condition. The actual cooldown time may, therefore, be shorter than that obtained from this analysis.

2.1.1 Pool Boiling Correlations

The following equations were obtained as a best fit for the correlation shown in Fig. 4, for the conditions of interest (coolant pressure = 1 atm, tube diameter > 0.2 cm). For the nucleate boiling regime,

$$\frac{q}{A} = 0.034 \Delta T^{2.58} \quad (\text{W/cm}^2) \quad 1 < \Delta T < 13 \text{ K} , \quad (1)$$

and for the film boiling regime,

$$\frac{q}{A} = 0.0143 \Delta T \quad (\text{W/cm}^2) \quad \Delta T > 53 \text{ K} . \quad (2)$$

The critical heat flux occurs at $\Delta T = 13 \text{ K}$. Heat fluxes greater than this maximum will occur only in the film boiling region at relatively high values of ΔT . Since there is no correlation in the transition region, two approaches were selected to obtain the heat transfer rate in this region. The first approach, which is called the "average

Fig. 4. Nucleate and film pool boiling correlations for nitrogen.

approach," is to interpolate the heat transfer rate linearly between the two regimes. The second approach, the "pessimistic approach," is to extrapolate the film pool boiling regime down to $\Delta T = 13$ K. Equations (1) and (2) could be written in general form as

$$\frac{q}{A} = h(T - T_f)^k \quad (3)$$

The analysis was performed for a coolant channel with or without insulation at the cooling surfaces. The energy equation for a coolant channel without insulation is

$$q = -mc_p \frac{dT_c}{dt} \quad (4)$$

Substituting q from Eq. (3) into Eq. (4) yields

$$Ah(T_c - T_f)^k = -mc_p \frac{dT_c}{dt} \quad (5)$$

This equation holds everywhere except at $13 < \Delta T < 53$ K in the "average approach" method. This region will be dealt with separately.

Integrating Eq. (5),

$$\int_{T_{c1}}^{T_c} (T_c - T_f)^{-k} dT_c = \frac{-Ah}{m} \int_0^t \frac{dt}{c_p} \quad (6)$$

This equation shows that the cooldown time is proportional to the mass/surface area ratio. Therefore, a general cooldown curve vs cooldown time could be constructed if initial magnet temperature is known. Such a generalized curve is constructed for a pure copper magnet at the initial magnet temperature of 300 K and is shown in Fig. 5 for both average and pessimistic approaches. (This figure could also be applied to the TF coil made of the components mentioned in the Introduction because c_p of the TF coil composite is not very much different than that of pure copper.) The cooldown time for the pessimistic approach is 50% higher than for the average approach. Schultz⁷ performed a similar analysis, and his result is also shown in Fig. 5. The comparison shows an excellent agreement between the results. It is interesting to note

Fig. 5. Generalized cooldown time for initial temperature of 300 K.

that rapid cooldown is feasible when initial surface temperature is 110 K or less, which corresponds to nucleate boiling. At surface temperatures above 110 K, not only does the heat transfer coefficient drop, but also the specific heat of copper increases, which both tend to increase the cooling-down time.

The analysis was extended for a cooling surface with 1-mm glass insulation with thermal conductivity as shown in Fig. 6. The result is also shown in Fig. 5, where the cooldown time is about 50% higher than the case when the surface is not insulated. The insulation causes the nucleate boiling regime, which corresponds to rapid cooldown, to occur at higher temperatures depending on the insulation thickness and its thermal conductivity. On the other hand, as Fig. 5 shows, the cooldown from about 100 to 80 K is relatively slow due to very small temperature differences and thus a very small heat transfer rate between the wall and the coolant (across the insulation).

The result of this simple model was compared with some available experimental data on Alcator-A and -C.² The comparison is shown in Table 2. The results indicate that the agreement between the model and the experimental data is fairly good in view of the simplicity of the model.

Figure 7 presents the universal cooldown period, TAM (time - area/mass), for various initial cooldown temperatures. The final cooldown temperature is 80 K.

2.1.2 Effective Heat Transfer Rate

An effective heat transfer rate from any initial temperature, q_{eff} , is defined as the average heat transfer rate from any initial temperature, T_{ci} , down to 80 K, and is obtained by

$$q_{eff} = \frac{-m \int_{80}^{T_{ci}} c_p dT}{At} = \frac{-\int_{80}^{T_{ci}} c_p dT}{At/m}, \quad (7)$$

where t is the cooling time from any initial temperature to 80 K. The ratio At/m can be obtained from Eq. (6). The effective heat transfer

Fig. 6. Thermal conductivity of insulators.

Table 2. Comparison of model with experimental data on cooling time with liquid nitrogen

	Mass (tonne)	Cooling surface area (m ²)	Cooling time (h)	
			Model ^a	Experiment
Alcator-A	5.5	1.73	3.4	2-3
Alcator-C ^b	18	3.3	5.8	5.2

^aThe model is for a bare copper surface without insulation.

^bActual liquid nitrogen consumption is 6.2 tonne for cooling down to 80 K. The model predicts 6.6 tonne, assuming only heat of vaporization contributes to the cooling.

Fig. 7. Universal cooldown period vs initial cooldown temperature.

rate is calculated vs initial cooldown temperature for both pessimistic and average approaches, and 1-mm glass insulation. The result is shown in Fig. 8. This figure is helpful in determining the total energy removal from the magnet, Q , from the following equation:

$$Q = q_{\text{eff}} A t \quad , \quad (8)$$

where t is given by Fig. 7.

Sydoriak and Roberts⁸ have studied the boiling of liquid nitrogen in short and narrow channels. Their model, which is based on flow cross section rather than wetted area which we so far have used, fits reasonably well with experiments in calculating the critical heat flux. They have proposed the following expression for natural convection boiling:

$$q = A_c h_{fg} \rho_l \left(\frac{f_2^2 g}{f_2(\beta_2 - 1) + 1/2} \right) (L - L_m)^{1/2} \quad , \quad (9)$$

where f_2 is the quality or mass fraction of vapor leaving the channel, β_2 is the liquid vapor mass ratio of the coolant (~ 220), L is the height of the coolant channel in centimeters, and L_m is the head of liquid equivalent to the total head of vapor liquid mixture and is given by

$$L_m = \frac{1}{\rho_l} \int_0^L \rho_m dL \quad , \quad (10)$$

where ρ_m is the density of the liquid/vapor mixture. For values of $f_2 = 1$ and $\beta_2 \gg 1$, L_m is small.

The maximum q will be obtained as a function of cross-section area and channel height from Eq. (9) by assuming $f_2 = 1$:

$$\frac{q_{\text{max}}}{A_c} = 338 L^{1/2} \quad (W/cm^2) \quad . \quad (11)$$

Since the model has been compared with experiments with short tubes (5-7 cm in length), Eq. (11) is not to be used for long tubes. A conservative estimate will be to assume a constant q_{max}/A_c (independent

Fig. 8. Effective heat flux down to 80 K for LN_2 boiling.

of L) based on $L = 10$ cm, or

$$\frac{q_{\max}}{A_c} = 1000 \text{ W/cm}^2 . \quad (12)$$

It would be interesting to compare the maximum heat fluxes, which were obtained based upon wetted area and circular cross-section area of a coolant channel. The ratio of the heat fluxes is

$$\frac{q \text{ (based on cross-section area)}}{q \text{ (based on wetted area)}} = \frac{\pi D^2/4 \times 1000}{\pi DL \times 2.5} = 100 \frac{D}{L} , \quad (13)$$

where 2.5 represents an average effective heat transfer rate plotted in Fig. 8. Equation (13) states that for an L/D value of around 100, the second method predicts the same heat transfer rate. The correlation, based upon wetted area, is used in this study since it predicts a lower heat flux for the TF coil geometry.

2.1.3 Convective Heat Transfer

It is important to estimate how much heat the superheated nitrogen gas ($T > 77$ K) could absorb before exiting the cooling channel. The convective heat transfer coefficient, h , could be obtained from⁹

$$h = \frac{KNu}{D_e} = \frac{0.023 Re^{0.8} Pr^{0.4} K}{D_e} . \quad (14)$$

This coefficient is about $21 \times 10^{-3} \text{ W/cm}^2 \text{ K}$ at a vapor temperature of 200 K, vapor velocity of 40 m/s, and hydraulic diameter of 2 cm. It drops to $17 \times 10^{-3} \text{ W/cm}^2 \text{ K}$ at a vapor temperature of 100 K, keeping the mass flow rate constant. It is interesting to note that the heat transfer coefficient for film boiling at 200 K from Fig. 4 is about $12 \times 10^{-3} \text{ W/cm}^2 \text{ K}$, which partially justifies the third assumption in Sect. 2.1.

2.1.4 Conductive Heat Transfer

A simplified method, described in Appendix A, is used to estimate the cooldown time for the case when the inner leg will be cooled only through conduction to the rest of the TF coil. This method is also used to estimate the effect of heat conduction from the inner leg to the outer leg. This cooldown time depends strongly on how close the end of the outer coolant channel is to the inner leg.

2.2 PRESSURE DROP

The primary purpose of this section is to estimate the pressure drop along the coolant channel in the inner leg of the TF coil. The pressure drop calculation in the coolant circuit system is a separate issue and should be done after the total cooling system is designed.

The total pressure drop in a flowing fluid is caused by the effects of viscous friction, the influence of body forces (such as gravity), and the momentum changes in the flows. The single-phase pressure drop per unit length, $\Delta P/\Delta L$, could be obtained from the following equation:⁹

$$\frac{\Delta P}{\Delta L} = 4 \frac{f}{D_e} \rho \frac{V^2}{2} \quad (15)$$

The friction factor for laminar flow is

$$f = \frac{64}{Re} \quad (16)$$

For turbulent flow

$$f = \frac{0.046}{Re^{0.25}} \quad (17)$$

Equation (15) could be used to estimate both the liquid coolant pressure drop in the feedlines and vapor pressure drop in the coolant channel and the vent.

The frictional pressure drop per unit length for a turbulent two-phase flow, $(\Delta P/\Delta L)_{tpf}$, could be estimated from the following equation:⁹

$$\left(\frac{\Delta P}{\Delta L}\right)_{\text{tpf}} = \left(\frac{\Delta P}{\Delta L}\right)_l \phi_l^2, \quad (18)$$

where $(\Delta P/\Delta L)_l$ is the pressure drop per unit length if liquid only flows in the pipe at the same mass flow rate as the two-phase flow. The function ϕ_l is given in Fig. 9 as a function of X , defined as:

$$X^2 = \frac{(\Delta P/\Delta L)_l}{(\Delta P/\Delta L)_v}, \quad (19)$$

where $(\Delta P/\Delta L)_v$ is the frictional pressure drop per unit length if gas only flows in the pipe at the same mass flow rate as the two-phase flow.

The fractional pipe volume filled by liquid, R_l , and that of the vapor, R_v , are also given in Fig. 9.

The momentum pressure drop is computed from⁹

$$\Delta P_M = \frac{(W_l + W_v)^2}{\rho_l A_c^2 g} \left[\frac{(1 - x_e)^2}{R_{le}} + \frac{x_e^2 \rho}{R_{ve} \rho_v} - 1 \right], \quad (20)$$

where R_{le} and R_{ve} correspond to the value of x evaluated at $x = x_e$. (x_e corresponds to the vapor quality at the end of the channel.) The maximum value of ΔP_M occurs when x_e is equal to 1. The value of X corresponding to x_e is obtained from:

$$X = \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_v}\right)^{0.125} \left(\frac{1}{x_e} - 1\right)^{0.875}. \quad (21)$$

The total pressure drop would then be the summation of Eqs. (18) and (20).

3. RESULTS

3.1 COOLDOWN TIME

The shortest cooling time corresponds to the first cooling option where the coolant channel is at the plasma side of the TF coil inner leg. This is due to the larger available cooling surface area. However, this cooling method is not recommended because the design of

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Fig. 9. Two-phase flow pressure drop function.

the inlet and outlet of the coolant channel makes it difficult to seal the coolant from the plasma.

The cooldown time for the fourth option is estimated to be about 6 h, assuming the temperature of the outboard leg drops instantly to 80 K. This cooldown time could be reduced to about 2 h if the outboard leg could be cooled to 20 K. In this case, other coolants such as helium or neon should be considered.

The cooldown time results down to 80 K for the second and third options are shown in Table 3. These results are for a 1-atm pressure liquid nitrogen flowing from the bottom of the TF coil and are based on an inner leg initial average temperature of 185 K along the channel. This average temperature corresponds to the total energy deposited in the inner leg at the end of the pulse. (Appendix B presents the analysis for the third option.) The results indicate that the second option cooldown time is slightly shorter than that of the third option. At the same time, the second cooling option is simpler and less expensive to build. This option also provides a means to cool the OH coil by inserting a solid insulator in the middle of the channel to insulate the TF and OH coils from each other. The appropriate channel width is estimated to be 2 cm, which could cool both TF inner leg and OH coils. The above results correspond to the assumption that only the heat of vaporization of nitrogen is responsible for removal of energy.

An analysis has been made to estimate the cooldown time for an initial temperature of 90 K at the start of the pulse instead of 80 K. Due to higher energy deposition (higher electrical resistivity) in the coil, the cooldown time to 90 K remains approximately the same.

3.2 PRESSURE DROP

Table 4 shows the pressure drops in the cooling channel associated with various coolant flow conditions. The calculation is based on the cooldown time of 0.5 h to 80 K, coolant channel length of 3 m, and coolant width of 1 cm. The cooldown time of 0.5 h was chosen in order to calculate the maximum possible flow pressure drop, since it might be possible to approach this level of cooldown time (see Sect. 4). The results of Table 4 indicate that the maximum total pressure drop is

Table 3. Cooldown time for cooling options 2 and 3
of the TF coil inner leg

	Cooling option 2	Cooling option 3
Cooling surface area, m^2 (based on 1 m height)	3.35	3.1
Cooling time, ^a h (no conduction to outboard leg)	0.9	1.0
Cooling time, h (with conduction to outer leg)	0.8	0.9

^aThese results are for bare copper. The values will increase by 50% if a surface insulation with 1 mm thickness is used.

Table 4. Coolant pressure drop in the inner leg of the TF coil^a

	V (cm/s)	ΔP (atm)
Liquid	6.6	0.5×10^{-4}
Vapor (200 K)	3000	0.015
Two phase (friction)	NA	0.09
Two phase (momentum change)	NA	0.017

^aBased on cooldown time of 0.5 h to 80 K, coolant channel length of 3 m, and coolant channel width of 1 cm.

about 0.1 atm, which is easily manageable. It should be noted that the static head in the coolant channel is about 0.24 atm when the channel is full of liquid nitrogen.

4. DISCUSSION OF RESULTS

The cooldown time could be reduced by starting the cooldown process with the coolant pressure higher than 1 atm. The cooldown continues at the pressure level up to the point where the critical heat flux limit is reached. The coolant pressure is then reduced in such a way that cooldown would occur at critical heat flux paths (maximum heat flux) down to 1 atm pressure (see Fig. 4). For example, the cooldown process could start with 10-atm coolant pressure and continue until the surface temperature drops to 110 K, which is the critical heat flux point corresponding to 10-atm coolant pressure. At this time, the coolant pressure should drop from 10 to 1 atm in a way that the cooldown would occur at the maximum heat fluxes. The surface temperature would drop from 110 K to 90 K in this process. The cooldown from 90 K down to 80 K would be at 1-atm coolant pressure. This discussion is based on the assumption that during this process the mixture quality is poor enough that nucleate boiling would be the dominant heat transfer mechanism. An estimate of the cooldown time for this process has been made. The result indicates that a cooldown time reduction of 17% is feasible compared with the case when the cooldown pressure occurs at a constant 1-atm coolant pressure. One drawback of this method is that the TF coil may not be able to accommodate higher coolant pressure; this should be investigated further. Another disadvantage is that the coolant pressure must be reduced at specific surface temperatures, which may involve some kind of controlling technique.

Another way to reduce the cooldown time is by increasing the coolant surface area by making machine-cut slits in the nose and in the vertical direction of the TF coil. The centerline slits could be placed 4 mm apart with 2-mm depth. This configuration increases the cooling surface area by a factor of 2. The slits may be treated as fins. The fin

effectiveness depends on the temperature difference between the cooling surface and the coolant. This could reduce the cooldown time to approximately 0.5 h at 1-atm coolant pressure operation. The nose of each TF coil could be undercut at the edges so that no wedging forces act on the fins. Electromagnetic forces could not deform the fins during machine operation since there is no bucking.

5. CONCLUSIONS

Preliminary results from a simplified model indicate that a cooldown time of about 1 h is feasible provided that the cooldown time is not limited by OH cooldown time. This will limit the number of pulses to around eight per day. A shorter cooldown time is feasible if the cooling surface area is increased by either making machine-cut fins or increasing the inner leg radial dimension. The latter method will directly impact the capital cost of the device and may not be as effective as the former method. An experimental R&D program is required to measure the actual cooldown time, since this may be a critical issue in the CIT design. A two-dimensional analysis will be performed to obtain a more realistic estimate of the cooldown time.

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APPENDIX A

SIMPLIFIED METHOD TO ESTIMATE COOLDOWN TIME BASED ON CONDUCTION ONLY

One possible method of cooling the inner leg of the TF coil would be by heat conduction to the outer leg, which is constantly being cooled. A simplified analysis has been used to estimate the cooldown time of the inner leg. This analysis is based on one-dimensional unsteady-state heat conduction in a slab of finite thickness. The temperature profiles for such heat conduction are shown in Fig. A.1 where the slab occupying the space between $y = -b$ and $y = +b$ is initially at temperature T_0 .¹ At time $t = 0$, the surfaces at $y = \pm b$ are suddenly lowered to T_1 and maintained there. The inner leg has been considered a slab where $y = 0$ is in the midplane. In Fig. A.1, at/b^2 is dimensionless and α is defined as

$$\alpha = \frac{K_c}{\rho_c c_p} \quad (\text{cm}^2/\text{s}) . \quad (\text{A-1})$$

For copper, $K_c = 4 \text{ W/cm K}$ and $\rho_c = 8.9 \text{ g/cm}^3$. The c_p varies from 0.2 at 80 K to 0.4 J/g K at 400 K and remains approximately constant at temperatures higher than 400 K. Assuming an average value for $c_p = 0.3 \text{ J/g K}$,

$$\alpha = \frac{4}{8.9 \times 0.3} = 1.5 \text{ cm}^2/\text{s} . \quad (\text{A-2})$$

The length of the inner leg from the midplane to the top of the TF coil is about 1.3 m (see Fig. 1). Therefore,

$$\tau = \frac{at}{b^2} = \frac{1.5t}{130^2} = 8.9 \times 10^{-5} t . \quad (\text{A-3})$$

The value of τ (Fig. A.1) depends on how close the temperature of the inner leg at the midplane ($y = 0$) should get to T_1 . For example, for $T_0 = 350 \text{ K}$, $T_1 = 80 \text{ K}$, and requiring that the temperature at the midplane should reach 85 K, we get

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Fig. A.1. Temperature profiles for unsteady-state heat conduction in a slab of finite thickness.

$$\frac{T - T_0}{T_1 - T_0} = \frac{85 - 350}{80 - 350} = 0.98 \quad . \quad (A-4)$$

For this ratio, τ should be around 2, or

$$t = \frac{2}{8.9 \times 10^{-5}} = 2.24 \times 10^4 \text{ s} = 6.2 \text{ h} \quad . \quad (A-5)$$

Equation (A-4) shows that this cooldown time is not very sensitive to T_0 values between 300 and 700 K if the final temperature requirement is 85 K. However, the cooldown time is more sensitive to the final midplane temperature. For example, increasing T from 85 to 100 K would result in

$$\frac{T - T_0}{T_1 - T_0} = \frac{100 - 350}{80 - 350} = 0.926 \quad , \quad (A-6)$$

which corresponds to $\tau \approx 1.3$ (Fig. A.1) with a cooldown time reduction of about 35%.

REFERENCE FOR APPENDIX A

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APPENDIX B
MAGNET COIL COOLING BY INTERIOR COOLANT PASSAGES

One method for cooling the inner leg of the TF coil would be to insert coolant passages inside the coil. The coolant passage area in the TF coil should be a small fraction of the coil area to preserve mechanical strengths. The cooling time is defined as

$$t = \frac{m \cdot \text{TAM}(T_{ci})}{A} , \quad (\text{B-1})$$

where $\text{TAM}(T_{ci})$ = time - area/mass, which is a function of the initial temperature of the coil, T_{ci} , before cooldown starts. This value could be obtained from Fig. 7 if the final cooldown temperature is 80 K.

The m/A could be rewritten as

$$\frac{m}{A} = \frac{\rho_c A_f (1 - f_v) \times L}{P_\omega L} = \frac{\rho_c (1 - f_v) A_f}{P_\omega} . \quad (\text{B-2})$$

For a circular coolant channel,

$$P_\omega = \pi D . \quad (\text{B-3})$$

If f_c represents the coolant channel cross-section area fraction,

$$f_c = \frac{\pi D^2}{4 A_f} . \quad (\text{B-4})$$

Substituting Eqs. (B-3) and (B-4) into Eq. (B-2), one obtains

$$\frac{m}{A} = \frac{\rho_c D (1 - f_v)}{4 f_c} . \quad (\text{B-5})$$

For copper and for $f_v = 0.1$ (for insulation and coolant passage) and $f_c = 0.05$, the cooldown time to 80 K is obtained from

$$t = 40D \cdot \text{TAM (min)} \quad , \quad (\text{B-6})$$

where D is the coolant passage diameter in meters. The cross section of one TF coil is about 380 cm^2 . The corresponding coolant channel diameter is 4.9 cm. The cooldown time is, therefore, equal to 1 h for initial average temperature of 185 K.

The coolant velocity is calculated here in order to estimate the coolant pressure drop:

$$V = \frac{4W}{\pi D^2 \rho_v} \quad . \quad (\text{B-7})$$

If we assume a constant heat removal rate,

$$W = \frac{Q}{th_{fg}} \quad . \quad (\text{B-8})$$

Equation (B-8) is based on the assumption that only heat of vaporization contributes to the heat removal.

Substituting Eqs. (8) and (B-8) into Eq. (B-7), one gets

$$V = \frac{4q_{eff}L}{D\rho_v h_{fg}} = 5.0 \frac{q_{eff}L}{D} \quad . \quad (\text{B-9})$$

The V will be in centimeters per second if q_{eff} is in watts per square centimeter. The $q_{eff} = 1.6 \text{ W/cm}^2$ (Fig. 8) for an initial cooldown temperature of 185 K, and L/D is about 61. The coolant velocity is found to be around 5 m/s, which does not create appreciable pressure drop in the channel (see Sect. 3 for pressure drop calculation).

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