The Limits of Helium Cooling in Fusion Reactor First Walls and Blankets

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THE LIMITS OF HELIUM COOLING IN FUSION REACTOR FIRST WALLS AND BLANKETS

INTRODUCTION

Helium is an attractive coolant candidate for fusion reactor first walls and blankets for several important reasons:

- Helium flow is not affected by intense, transient magnetic fields as are liquid metals.
- Since helium is chemically inert, it does not present the extreme fire hazard of liquid metals nor the corrosion problems of water.
- As an ideal gas, phase change difficulties such as water boiling and liquid metal freezing are avoided.
- Helium can potentially be used directly to power turbogenerators for higher power plant efficiency.

Arguments against helium or any gas coolant as opposed to water or liquid metals center around its lower heat capacity and surface heat transfer rate and the requirement of high operating pressure to achieve adequate heat transfer and a practical pumping power. On the balance, however, helium is certainly an adequate cooling medium as has been shown by its HTGR application and its use in several fusion reactor concepts.⁽¹⁻⁵⁾

None of these applications approach the ultimate capability of helium cooling because their operating conditions are determined by other inter-related design objectives. The shape and size of the duct may be set by neutronics or the duct length restricted by the placement of neutral beam ports to less than optimum values.

This study avoids these obstacles by exploring the practical limits of helium cooling in a simple geometry unconstrained by a particular conceptual design. Specifically, the configuration was chosen to be an externally heated straight tube considering both uniform heating and heating of half the external perimeter. Both thermal hydraulic and structural limits to the heat flux have been investigated.

Curves are presented to show the heat flux and tube length which simultaneously attain both a wall temperature and pressure drop/pumping power limit for a range of diameters from 0.05 to 8.0 inches and pressures from 50 to 5000 psia. Tube wall stress limits on heat flux are also shown for the same range of pressure and diameter. These results should serve as an aid in planning more complex concepts as well as evaluating helium cooling in this specific configuration.

SUMMARY

The results of this investigation show that under the proper conditions helium can achieve heat fluxes comparable to boiling water systems. However, thermal stresses will limit heat flux more often than the coolant, particularly when the heating is non-uniform.

Consider a practical sized section of coolant tube 20 ft long and about 1.5 inches inside diameter. At a helium pressure of 1000 psi the uniformly heated tube can support a heat flux of 320,000 Btu/hr-ft² [just over 1 MW(t)/m²] before reaching the limiting wall temperature of 1000°F and pressure drop of 50 psi. The temperature/pressure drop limited heat flux decreases approximately linearly with pressure and diameter.

If the length were reduced below 12 ft, the heat flux could be increased until it became limited by thermal stress to approximately 480,000 Btu/hr-ft² [~ 1.5 MW(t)/m²]. Assuming a sufficiently short tube to avoid the wall temperature/pressure drop limit, the stress limited heat flux decreases with decreasing pressure but increases with decreasing diameter because of the reduced wall thickness requirement.

Thermal stresses are a much more severe restriction if the tube is heated over only half its perimeter as it might be in a fusion reactor first wall application. The 20 ft, 1.5 inch I.D. tube would be stress limited to only 65,000 Btu/hr-ft² [$\sim 0.2 \text{ MW(t)}/\text{m}^2$] which is about one fifth the capacity of the uniformly heated tube. However, since only half the heat enters per unit length at a given heat flux, the non-uniformly heated tube may be twice as long as the uniformly heated tube. The non-uniformly heated tube could be lengthened to 125 ft before becoming temperature/pressure drop limited at 65000 Btu/hr-ft².

It is important to note that the thermal stress limits apply regardless of what coolant is being used. Therefore, in non-symmetrically heated situations, such as the first wall, the mechanical design of the wall is probably much more important than the choice of coolant.

METHOD OF ANALYSIS

Three independent aspects determine the maximum heat flux attainable from a convectively cooled surface:

- The maximum allowable surface or duct wall temperature.
- The maximum pumping power or pressure head available to drive the cooling fluid over the surface.
- The maximum allowable steady and cyclic thermal stresses in the duct wall.

In designing an entire cooling system, only one of these is usually limiting since the length and diameter of the coolant passage is fixed by other considerations. If the coolant duct dimensions are allowed to vary arbitrarily, however, the heat flux and dimensional combination which meets at least the first two limiting conditions simultaneously may be determined. Further increases in heat flux beyond this point require a reduction in duct length until the point of maximum heat flux is reached in one of two ways:

1. Assuming the duct wall thickness is prescribed by the internal pressure, a limiting steady or cyclic thermal stress level is reached.
2. A hypothetical "zero length" is reached at which the surface temperature at the duct inlet already exceeds the maximum and no further heat addition is possible.

The locus of points where either of these limits occurs for a range of pressure and lateral duct dimension defines the surface of maximum possible heat flux attainable through the surface under the given temperature, ΔP /power and stress limits.

The shape of the duct and its thermomechanical properties have a direct influence on the maximum heat flux. In order to reduce the amount of data to be generated, this study will be limited to circular tubes of stainless steel which is typical of many proposed state-of-the-art concepts. The limiting pressures, temperatures and stresses chosen for this configuration are listed in Table 1. Stress limits are prescribed by the ASME code.⁽⁶⁾ Both uniform internal heating and the case of 1/2 the perimeter heated will be considered.

TABLE 1. Limiting Values Chosen for an Externally Heated, Helium Cooled Stainless Steel Tube

<u>Limit</u>	<u>Value</u>
Wall temperature (outer surface at tube exit)	1000°F
Pressure head	1% operating pressure
Pumping power	5% of thermal power
Tensile stress, σ_m	15,000 psi
Steady thermal stress and tensile stress	3 σ_m
Cyclic component of thermal stress for infinite life	10,500 psi

The range of operating conditions considered is limited to approximately the state of the art also. It is assumed that the helium is driven through a secondary heat exchanger (steam generator) by a circulator of roughly HTGR configuration rather than any advanced direct cycle. The operating conditions chosen are inlet temperature, pressure, and heat flux and tube diameter. The base case and range of values considered for these operating parameters are listed in Table 2.

TABLE 2. Operating Conditions

<u>QUANTITY</u>	<u>BASE VALUE</u>	<u>RANGE</u>
Inlet temperature	300°F	not varied
Pressure	1000 psi	50-5000 psi
Heat flux	---	$6(10^3)$ - $6(10^5)$ btu/hr-ft ²
Tube diameter	1.47 in.	0.05-8.0 in.

The tube length and inlet mass flux required to meet the inside surface temperature and pumping power/pressure head limits simultaneously for a given heat flux, tube diameter and operating pressure are found by solving the one-dimensional steady-state conservation equations for mass, energy and momentum. The heat flux and inside surface temperature are adjusted to satisfy the thermal stress limiting criteria and maximum outside wall temperature. The equations used to perform these calculations are derived in appendices A and B, respectively. A computer program, HOTGAS, has been constructed to incorporate this procedure with the necessary logic to vary the diameter and operating conditions for parametric studies.

RESULTS

The quantitative results of this study consist of the tube length L , and heat flux, q'' , necessary to satisfy the given temperature and pressure drop limits. These curves of q'' vs. L are presented for a range of diameters at a constant base pressure and also for a range of pressures at a base diameter. The stress limiting surface for all diameters and pressures is also shown. Refer again to Tables 1 and 2 for the limits and range of operating conditions. Results for a uniformly heated tube will be presented first.

UNIFORMLY HEATED TUBE

Consider first the q'' vs. L curves at a constant pressure of 1000 psi with tube diameter range of 0.05 to 8 inches shown in Figure 1.

Interpreting the figure at a diameter of 1.47 in. and heat flux of $10,000 \text{ Btu/hr-ft}^2$ the maximum pressure head will drive a sufficient flow rate through a length of 3300 inches so that the outside surface temperature at the tube exit will be the limiting value of 1000°F . If the heat flux is increased holding the same diameter, the length must decrease to stay at the temperature limit. The shorter length produces less drag and the mass flux increases. Heat flux can continue to increase up to approximately $480,000 \text{ Btu/hr-ft}^2$ at this diameter before the maximum thermal stress occurs. The figure indicates that higher heat fluxes can be obtained with smaller diameters and a corresponding decrease in length.

Performance variation with pressure at a constant 1.47-inch diameter is shown in Figure 2. Note that a 1000 psi curve in Figure 2 would correspond exactly to the 1.47-inch diameter curve on Figure 1. On both figures, an increase in heat flux at a constant pressure and diameter requires a decreased length. At any given length, higher pressure will allow a significantly higher heat flux until the stress limit is reached. For a practical tube length of 8 to 20 feet, a pressure range of 650 to 1200 psi would be most efficient. Use of a higher pressure would actually lower the maximum attainable heat flux. A lower pressure on the other hand, would raise the heat flux but only if the length were much reduced.

This behavior is more evident on Figure 3 which is the stress limit surface for the entire range of pressure and diameter. The area enclosed by the dash line on the figure represents the currently practical design range of 650 to 1200 psi pressure and from 0.25 to 1.5 inches diameter. Note that the maximum heat fluxes possible are on the order of 10^6 Btu/hr-ft^2 which compares quite favorably with the highest heat fluxes attainable with boiling water.

FIGURE 1. Heat Flux versus Length with Diameter Variation for Symmetric Heating

FIGURE 2. Heat Flux versus Length with Pressure Variation for Symmetric Heating

FIGURE 3. Stress Limited Heat Flux versus Pressure for Symmetric Heating

NON-SYMMETRICALLY HEATED TUBE

In this case, one half the tube perimeter is heated as it might be in a first wall application. This places an additional bending stress in the tube wall with the heated half in compression and the unheated half in tension. The same family of curves as for the uniformly heated tube are shown in Figures 4, 5 and 6. Below the stress limits the trends are the same as for the uniformly heated tube. The only significant change is that the length may be made longer since only 1/2 the total heat is getting into the fluid. At the stress limit, however, the behavior is quite different.

By following the 0.479-inch diameter line on Figure 6 it can be seen that the maximum heat flux increases with increasing pressure up to a value of about 120,000 Btu/hr-ft² at about 2000 psi and then decreases again as the pressure further increases. This implies an "ideal pressure" for any given diameter at which the thermal stress limitation is least severe. This assumes that the tube wall is just thick enough to withstand the internal pressure. Below the "ideal pressure" it would actually be advantageous to make the tube wall thicker than necessary in order to present a larger area to the bending load produced by non-symmetric heating.

Whatever pressure is used, a comparison of the practical design range in Figures 3 and 6 shows immediately that there is a severe penalty for non-symmetric heat flux due to thermal stresses. The maximum heat flux within the design range in Figure 3 is over $2.5(10^6)$ Btu/hr-ft² (over 50 MW/m² neutron loading) while the design range for the non-symmetric case in Figure 6 is below 150,000 Btu/hr-ft² (equivalent to about 3.2 MW/m² neutron loading). Admittedly, the case analyzed is conservative but it indicates that wall design may easily limit heat transfer more often than the choice of coolant.

FIGURE 4. Heat Flux versus Length with Diameter Variation for Non-Symmetric Heating

FIGURE 5. Heat Flux versus Length with Pressure Variation for Non-Symmetric Heating

FIGURE 6. Stress Limited Heat Flux versus Pressure for Non-Symmetric Heating

CONCLUSIONS

The results of this analysis show that the use of helium as a coolant fluid does not restrict the maximum heat flux attainable through a stainless steel circular tube. If thermal stress is not a factor, helium can, under practical operating conditions, achieve heat fluxes comparable to boiling water systems. It is also evident, however, that thermal stresses must be considered carefully in any design and that this will limit the heat flux more often than the surface heat transfer. This is particularly true when the coolant passages are not heated symmetrically which is typically the case for fusion reactor first wall applications.

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APPENDIX A

FLUID TEMPERATURE AND PRESSURE DROP COMPUTATION PROCEDURE

APPENDIX A. Fluid Temperature and Pressure Drop Computation Procedure.

Consider a uniformly heated horizontal straight duct, as shown in Figure A-1, through which a cooling fluid is driven by a limited pressure head and pumping power. The following assumptions are made:

- Fluid velocity is sufficiently low that sonic effects can be neglected.
- Frictional heating is ignored as insignificant compared to wall heat transfer.
- The total pressure drop is sufficiently small compared to the exit pressure that only density change due to heating need be considered.

Under these assumptions, the one dimensional conservation equations for mass, momentum and energy in the steady state are:

mass

$$\rho v \equiv G = \text{constant} \quad \text{A-1}$$

momentum

$$g\Delta P = \frac{1}{2} \frac{fG^2}{D_h} \int_0^L \frac{1}{\rho} dx + G^2 \left(\frac{1}{\rho_L} - \frac{1}{\rho_0} \right) \quad \text{A-2}$$

energy

$$T = T_0 + \frac{q'' P_h x}{G A C_p} \quad \text{A-3}$$

where f is the wall friction factor, D_h is the hydraulic diameter, q'' is the surface heat flux, P_h is the heated perimeter and C_p is the fluid specific heat. In addition to equations A-1, A-2 and A-3 an equation of state for the fluid is necessary to relate the density to temperature and pressure

$$\rho = \rho(T, P) \quad \text{A-4}$$

FIGURE A-1. Geometry and Temperature Variation for an Externally Heated Tube

We need to devise a computation procedure that relates the mass flux, G , and duct length, L , to the given operating conditions and limits. First, an expression for G as a function of ΔP , L and q'' will be developed.

For a low pressure drop the change in density with pressure is small compared to the change due to heating. If the equation of state is linear at a constant pressure, the change in density along the duct is

$$\rho - \rho_0 = \left. \frac{\partial \rho}{\partial T} \right|_P (T - T_0)$$

or, in terms of specific volume

$$\frac{1}{\rho} - \frac{1}{\rho_0} = \left. \frac{\partial 1/\rho}{\partial T} \right|_P (T - T_0) \quad A-5$$

After the energy equation, A-3, is substituted for the temperature difference, equation A-5 is used to provide an expression for the specific volumes in the momentum equation, A-2, to yield:

$$g\Delta P = \frac{1}{2} \frac{fG^2}{D_h} \int_0^L \left(\frac{1}{\rho_0} + \left. \frac{\partial 1/\rho}{\partial T} \right|_P \frac{q''_h}{GAC_p} x \right) dx$$

$$+ G^2 \left. \frac{\partial 1/\rho}{\partial T} \right|_P \frac{q''_h L}{AGC_p} \quad A-6$$

After performing the integration and solving for G we have

$$G = \left[\frac{g\Delta P \rho_0}{\left(\frac{fL}{2D_h} + \left. \frac{\partial 1/\rho}{\partial T} \right|_P \rho_0 (T_L - T_0) \left[\frac{fL}{4D_h} + 1 \right] \right)} \right]^{1/2} \quad A-7$$

Given the temperatures, q'' and ΔP , the length, L , and friction factor, f , are functions of G . An iterative solution is therefore necessary. The solution procedure is as follows:

1. Calculate the maximum surface temperature, T_s , given the wall thickness, heat flux and maximum duct wall temperature, T_w .

$$T_s = T_w - \frac{q'' P_h}{2\pi K} \ln\left(1 + \frac{2t}{D}\right) \quad A-8$$

Note that the wall thickness and heat flux are computed by means of the expressions derived in Appendix B. The maximum surface temperature may also be specified directly. See Figure A-1.

2. Compute the friction factor and heat transfer coefficient, H , by appropriate correlation using either an initial guess for G or the last iterate value.
3. Compute the fluid exit, T_L , temperature required to make the duct surface temperature T_s

$$T_L = T_s - \frac{q''}{H} \quad A-9$$

4. Calculate the duct length required to raise the fluid temperature from T_o to T_L at the current G

$$L = (T_L - T_o) \frac{G A C_p}{q'' P_h} \quad A-10$$

If the length is negative, stop.

5. Check the given pressure head, ΔP_o , against the pumping power limit, ψ , which is defined as a fraction of the total heat input to the duct.

$$\Delta P = \min \left(\Delta P_o, \frac{\rho_o \psi q'' P_h L J}{G A} \right) \quad A-11$$

6. Calculate the new mass flux with equation A-7 and check for convergence. If the change in G is too large, return to step 1 and repeat.

APPENDIX B

THERMAL STRESS LIMITS ON HEAT FLUX

APPENDIX B. Thermal Stress Limits on Heat Flux

The wall of a coolant tube has a minimum thickness required to support the internal coolant pressure and remain below the ASME allowable design stress, σ_m .⁽⁶⁾ For thin tubes this thickness is expressed as

$$t = \frac{PD}{2\sigma_m} \quad \text{B-1}$$

where P is the internal pressure and D is the tube inside diameter.

If the tube is cooled uniformly on the inner surface, the temperature gradient across the wall gives rise to an additional tangential and axial tensile stress at the inner surface and a corresponding compressive stress on the outside, as shown in Figure B-1. The stress σ_t is computed by

$$\sigma_t = \frac{E\alpha\Delta T}{2(1-\nu)\ln(1 + \frac{2t}{D})} \left[\frac{2 \ln(1 + \frac{2t}{D})}{1 - \frac{1}{(1 + \frac{2t}{D})^2}} - 1 \right] \quad \text{B-2}$$

where E is Young's modulus, α is the linear thermal expansion coefficient and ν is Poisson's ratio.⁽⁵⁾ If the thickness is prescribed by the internal pressure via equation B-1, the wall temperature gradient and, therefore, heat flux, is limited by the maximum allowable thermal stress.

The combination of the pressure stress and thermal stress is limited to three times σ_m . Since the wall thickness is chosen to make the pressure stress exactly σ_m , the condition on thermal stress is $\sigma_t > 2\sigma_m$. Substituting equation B-1 for the ratio of thickness to diameter in equation for B-2 and solving for the temperature gradient yields:

$$\Delta T = \frac{4(1-\nu)\ln(1 + \frac{P}{\sigma_m})\sigma_m}{E\alpha \left[\frac{2\ln(1 + P/\sigma_m)}{1 - \frac{1}{(1 + P/\sigma_m)}} - 1 \right]} \quad \text{B-3}$$

FIGURE B-1. Temperatures and Stresses for a Uniformly Heated Tube

The temperature difference is related to the heat flux through

$$\Delta T = \frac{q'' P_h}{2\pi K} \ln\left(1 + \frac{2t}{D}\right) \quad \text{B-4}$$

The heat flux limit follows by substituting equation B-4 into equation B-3 and solving for q'' .

$$q'' = \frac{8\pi K \sigma_m (1 - \nu)}{E \alpha P_h \left[\frac{2 \ln(1 + P/\sigma_m)}{1 - \frac{1}{(1 + P/\sigma_m)^2}} - 1 \right]} \quad \text{B-5}$$

Equation B-5 prescribes the steady state heat flux limit due to thermal and pressure stresses. Since most proposed CTR devices operate in a cyclic mode in which the temperature gradient may change from a maximum value to near zero every few minutes, thermal fatigue is also an important consideration.

The heat flux limit for thermal fatigue may be determined by setting half the stress given by equation B-2 to maximum cyclic stress for infinite life (or some specified number of cycles), σ_f , at the average wall temperature. Temperature effects are incorporated by adjusting σ_f by the ratio of Young's moduli. Equation B-1 and B-4 are substituted for the thickness to diameter ratio and temperature difference, respectively, so that

$$q''_f = \frac{8\pi K \sigma_f (1 - \nu)}{E \alpha P_h \left[\frac{2 \ln(1 + P/\sigma_m)}{1 - \frac{1}{(1 + P/\sigma_m)^2}} - 1 \right]} \quad \text{B-6}$$

Note that since $q''_f = (\sigma_f q'')/\sigma_m$, fatigue will usually prescribe the minimum heat flux since $\sigma_f < \sigma_m$ for stainless steel.

If the coolant tube surface is non-symmetrically heated as in a first wall application, an additional thermal stress occurs from one side of the tube being hotter than the other. A conservative "worst case", where half

the tube perimeter is heated and the other half is at the coolant exit temperature, will be considered. This is illustrated in Figure B-2.

The axial thermal stress σ_a at any position can be calculated by a simple beam model constrained in bending but free axially as

$$\sigma_a = -E\alpha(T - \bar{T}) \quad \text{B-7}$$

where T is the local temperature and $\bar{T}$ is the average temperature over the cross section. Assuming that the cold half of the tube is at the fluid exit temperature, T_L , the cross section average is

$$\bar{T} = \frac{1}{2}\left(\frac{1}{2} T_W + \frac{1}{2} T_S + T_L\right) \quad \text{B-8}$$

where T_W and T_S are the hot side outer and inner surface temperatures, respectively.

The temperature difference in equation B-7 is significant at the outer and inner surfaces on the hot side and on the cold side where the tube wall temperatures are T_W , T_S , and T_L respectively. The relations between these temperatures and the heat flux in Equations A-8 and A-9 are utilized to relate the axial thermal stress to heat flux at each of the three locations. These are summarized in Table B-1.

The axial thermal stress due to the radial temperature gradient from equation B-2 and the axial internal pressure stress must also be included. The absolute value of the sum of the three stresses must be less than three times σ_m as in the previous uniformly heated case. When this condition is reached the axial pressure, which is only $1/2 \sigma_m$, is small compared to the thermal stresses which then determine the sign of the sum. In addition, if fatigue is to be considered, half the sum of the axial thermal stress components must be less than the maximum cycle stress for the infinite life (or specified number of cycles), σ_f .

The above four stress limiting conditions are solved for the heat flux in the same manner as equations B-5 and B-6 previously. The results are shown in Table B-2.

FIGURE B-2. Temperatures and Stresses for a Non-Symmetrically Heated Tube

TABLE B-1. Expressions for Axial Thermal Stress

Location	Tube Temperature, T	Stress $\sigma_a = -E\alpha(T - \bar{T})$
Outer Surface, hot side	T_W	$-\frac{E\alpha q''}{2} \left[\frac{3P_h}{4\pi K} \ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{1}{H} \right]$
Inner Surface, hot side	$T_S = T_W - \frac{q'' P_h}{2\pi K} \ln\left(1 + \frac{P}{\sigma_m}\right)$	$-\frac{E\alpha q''}{2} \left[\frac{-P_h}{4\pi K} \ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{1}{H} \right]$
Cold Side	$T_L = T_S - \frac{q''}{H}$	$+\frac{E\alpha q''}{2} \left[\frac{P_h}{4\pi K} \ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{1}{H} \right]$

TABLE B-2. Summary of Heat Flux Limits for a Non-symmetrical Heated Tube

Limit and Location	Expression
Outer surface hot side Compression	$q_{oc} < \frac{7\sigma_m}{E\alpha \left(\frac{1}{H} + \frac{P_h}{2\pi K} \frac{3}{2} \left[\ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{B}{1-\nu} \right] \right)}$
Fatigue	$q_f < \frac{4\sigma_f}{E\alpha \left(\frac{1}{H} + \frac{P_h}{2\pi K} \left[\frac{3}{2} \ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{B}{1-\nu} \right] \right)}$
Inner surface hot side tension	$q_{it} < \frac{5\sigma_m}{E\alpha \left(\frac{P_h}{2\pi K} \left[\frac{1}{2} \ln\left(1 + \frac{P}{\sigma_m}\right) + \frac{B}{1-\nu} \right] - \frac{1}{H} \right)}$
Cold side tension	$q_{ct} < \frac{5\sigma_m}{E\alpha \left[\frac{1}{H} + \frac{P_h}{4\pi K} \ln\left(1 + \frac{P}{\sigma_m}\right) \right]}$

$$B = \left[\frac{2 \ln(1 + \frac{P}{\sigma_m})}{1 - \frac{1}{\left[1 + \frac{P}{\sigma_m}\right]^2}} - 1 \right]$$

Since $q_f = [(4\sigma_f)/(7\sigma_m)]q_{oc}$ fatigue is even more likely to be limiting for the non-symmetrically heated tube. If fatigue is not considered, sensitivity analysis shows that q_{oc} will be the minimum at all but very low pressures where q_{ct} becomes limiting. q_{it} is not a significant limit because of the cancellation of compressive and tensile stresses.

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