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BOILING WATER COOLING OF A HYPOTHETICAL LARGE SOLID ROCKET NOZZLE LINED WITH A POROUS WALL OR A MELTING INSULATING COATING

by William A. Olsen, Jr.

Lewis Research Center
Cleveland, Ohio

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16. Abstract

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SUMMARY

A preliminary analytical investigation was made of two thermal protection methods that use a boiling liquid to cool a large solid rocket nozzle. One method involves an insulating coating, where cooling is accomplished by boiling water in tubes behind the coating. The other method involves lining the nozzle with a porous wall through which water flows and evaporates within the pores or at the hot-gas porous-wall surface.

The first thermal protection method requires the coolant tubes to be covered with an insulating coating in order to reduce the heat flux and the resulting problem of boiling burnout in the tubes. Conventional insulating coatings, which are of relatively low cost (e.g., aluminum oxide base), typically have low melting temperatures and can therefore melt under the thermal conditions in the nozzle. In fact, an initially thick insulating coating will melt rapidly to a steady-state thickness because little heat is absorbed by the melting process. In the region of the throat, where the heat flux is highest, conventional coolant tube materials (i.e., low cost materials) will fail either by simple overheating or because of the overheating that occurs when the heat flux exceeds the boiling burnout flux. The overheating and burnout problems required that the coolant be only partly vaporized in the throat region, which necessitates a high coolant flow. More complete vaporization and lower coolant flow are possible in the other regions of the nozzle.

By passing the water coolant through a porous wall, where evaporation occurs at the hot surface, it is possible to avoid the burnout and overheating problems associated with the coolant boiling in the tubes. The performance of the porous wall is not greatly affected by the porous wall parameters (e.g., porosity, thickness, and conductivity); therefore, the porous wall might be made of ablative material as insurance in case of local coolant failure. The overall coolant flows required in order to have adequate thermal protection from either of these two methods are approximately comparable. In either method, where conventional, inexpensive, low-temperature materials are required, water is a better coolant than either a subcritical cryogen or a liquid metal.

INTRODUCTION

As part of the continuing effort at the NASA Lewis Research Center in rocket nozzle cooling research, some thermal protection methods for a large solid rocket nozzle were investigated. In the preliminary analysis reported herein the following design constraints were imposed: inexpensive construction materials and methods, and a nozzle that would be reuseable after each firing with only minor repairs.

There have been many studies of thermal protection methods which would result in reliable one-shot rocket nozzles. In references 1 to 3, many prospective methods to achieve this goal are compared.

There are many thermal protection methods that might achieve the desired goal of the first paragraph. This report considers only thermal protection methods that utilize a liquid coolant. Very high heat fluxes typically occur in a large, solid rocket nozzle. In order to reduce the coolant weight penalty, it is necessary that the heat flux be absorbed by boiling the coolant to take advantage of its heat of vaporization. The coolant could be passed through and boiled within tubes that form the nozzle wall. The coolant may also be passed through a porous wall, that lines the nozzle walls, where it is boiled to absorb the heat.

For the situation where a boiling liquid flows through tubes it is necessary to reduce the heat flux and tube wall temperatures that occur by shielding the tubes with an insulating material. For practical thicknesses, insulating tiles will become so hot that only high-temperature refractory materials could be used. High-temperature refractory materials are generally expensive so that the tile method is not considered here further. Present state-of-the-art insulating materials of reasonable cost can melt under the thermal conditions existing in a large, solid rocket nozzle. One such material is an aluminum oxide base insulating material. The coolant tubes can be readily coated and recoated with this material if necessary after each test. Another example of a potential practical coating material is zirconium oxide; however, this material may unfavorably react chemically with the aluminum compounds in the exhaust.

Two thermal protection methods that use a liquid coolant are discussed in this report. An aluminum oxide coating, where the heat transferred is absorbed by boiling a coolant in tubes behind the coating, is the first thermal protection method considered. The second thermal method involves fabricating the nozzle walls of some porous material. The coolant flows through the porous wall and evaporates within the pores or at the hot surface. Conceptually, this method avoids the major problems of the first case (i.e., boiling burnout and overheating) so long as there is enough coolant flow. The same thermal conditions, which describe conditions in a representative large, solid rocket nozzle of about 3×10^6 pounds (13×10^6 N) thrust, are used in the calculations and comparison of both of these thermal protection methods.

COOLING BY BOILING IN TUBES

Description

This section deals with cooling a solid rocket nozzle by boiling a coolant liquid that flows through hollow tubes which form the nozzle walls. Cooling a rocket nozzle involves exceptionally high nonuniform heat fluxes; the coolant tubes are necessarily bent and of varying cross sections in order to form the nozzle walls. The boiling process has not been adequately described quantitatively, even for uniformly heated straight tubes of constant cross section for which there is considerable experimental data. Therefore, the analysis that follows is such that its results do not depend heavily on quantitative boiling information. Even so, some description of the boiling process would be helpful in the discussion to follow. The qualitative description in reference 4 is therefore summarized here. The coolant liquid entering the tube begins to boil a short distance down the tube, where the wall temperature exceeds the saturation temperature by a small amount (i.e., slightly superheated). This subcooled boiling (bubbly flow) region continues until the bulk liquid reaches the saturation temperature. Beyond that point, an annular film flow region exists, where the vapor quality increases down the tube until the liquid film evaporates away. Somewhere near the film dry-out location, high quality burnout can occur (i.e., the heat-transfer coefficient drops greatly which could result in a significant increase in the wall temperature, possibly beyond the wall's capability, such that it overheats and fails). Figure 1, which was taken from reference 5, indicates the burnout heat flux for uniformly heated constant cross section tubes, with and without twisted tape inserts, as a function of exit quality and flow rate. This figure shows that the burnout heat flux increases as the exit quality decreases. The effect of flow rate on the burnout flux is smaller. However, in order to absorb a given heat flux that is near the burnout flux, there must be a low exit quality and a correspondingly high coolant flow rate. A swirl device greatly increases the burnout flux for a given exit quality. The coolant-side heat-transfer coefficient changes appreciably in each boiling region. This coefficient is considerably higher in the boiling regions than in the entrance liquid region or beyond the burnout location, where essentially a vapor flow exists. Swirl devices may increase the coefficient somewhat in all regions.

An insulating coating is necessary because its additional thermal resistance is needed to reduce the heat flux below the burnout heat flux and also to reduce the tube wall temperatures so that conventional (inexpensive) tube materials (e.g., stainless steel) can be used. At typical nozzle gas temperature (e.g., about 5500° F (3300 K)) and heat fluxes, conventional insulating coating materials can melt. One available high-temperature coating material that has been used as a nozzle coating is largely composed of aluminum oxide (Al_2O_3), which is also often a significant component of nozzle exhaust

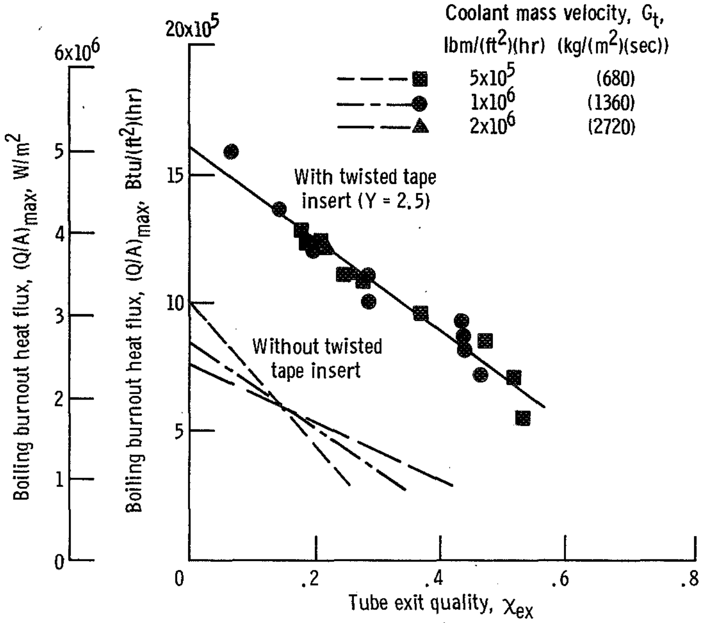

Figure 1. - Burnout heat flux for boiling in hollow tubes with and without twisted tape inserts. The tubes are uniformly heated, straight, and of constant cross section. (Figure taken from ref. 5.) Coolant tube pressure, 2000 psia (13.8 MN/m²); coolant tube diameter, 5/16 inch (0.008 m); boiler length, 21 inches 0.54 m).

products. The melting temperature of this coating material (about 3720° F (2320 K)) is such that the solid coating will melt in some areas of the nozzle, depending on the local coating thickness and heat flux; while in other areas the exhaust product Al_2O_3, which is mostly in liquid form, may impact on the nozzle walls and freeze. Because of the high gas velocities, the melt layer flowing over the solid coating will be wavy and tend to be ripped off wherever it is relatively thick. Figure 2 is a schematic representation of a region of a nozzle that is coated with a meltable insulating coating and cooled by a boiling coolant.

In the analysis that follows, the thickness of this solid coating required to prevent boiling burnout and coolant tube wall overheating is determined for a hypothetical large, solid rocket nozzle cooled by a liquid. In addition, steady-state values and the transient change in the coating thickness, heat flux, and wall temperature are determined. This thermal protection method is successful if the coating thickness does not melt so much during the burn time that the burnout heat flux is exceeded, or the wall overheated, anywhere in the nozzle.

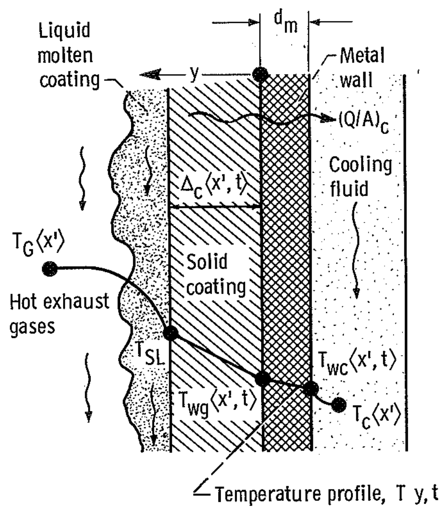

Figure 2. - Schematic representation of segment of nozzle wall at position x'. Meltable insulating coating is shown applied to one side of nozzle wall. A coolant flows behind other side of wall.

Analysis

Numerous simplifying assumptions are necessary in this analysis, because of the many physical complications of this problem. The liquid (melt) layer is wavy and tends to be blown off it it gets too thick (see fig. 2). Therefore, the liquid layer is assumed to be very thin such that its thermal resistance is appreciably less than the thermal resistance of the solid coating and/or hot-gas thermal resistance. Numerous components are present in the exhaust gases of the nozzle considered, including about 6 percent liquid particles of Al_2O_3, which may impinge on the nozzle wall. This is a major constituent of the solid coating so that the solid-liquid interface temperature could be affected. With the expected thin liquid layer, in the case of the nozzle, and low partial pressure of Al_2O_3 vapor in the exhaust gas, the liquid-vapor interface temperature should approach the solid-liquid temperature T_{SL}. Because of the many uncertainties, it is assumed that there is little error in incorporating the relatively small liquid layer thermal resistance in the hot-gas-side coefficient resistance. The analysis is further simplified by including the effect of thermal radiation in h_G. Based on these assumptions, an effective gas-side heat-transfer coefficient h_G is defined by equation (1), which incorporates the many possible effects of the thin liquid layer:

$$\left(\frac{Q}{A}\right)_G \equiv h_G(T_G - T_{SL}) \tag{1}$$

(All symbols are defined in the appendix.) It is further assumed, in the absence of data for coated nozzles, that h_G is approximately equal to known bare wall coefficients h'_G. This is a conservative estimate since the melt layer adds resistance such that h_G would be less than h'_G. Equation (1) also incorporates the assumptions that (1) the melt layer covers the solid, so that local equilibrium can exist at the solid-liquid interface, and (2) the melt temperature is single valued or at least covers a small range in temperature. In other words, it is assumed that T_{SL} is a constant. For the limited accuracy possible in this analysis, it is further assumed that all the coating and wall properties are constant. The startup times of the rocket engine (gas side) and the coolant side are assumed to be very much less than the characteristic time of the melting process so that the gas-side and coolant-side variables may be taken to be only dependent on the location along the nozzle. It is also assumed that little energy, compared with the heat fluxes considered, is stored in the metal wall and in the thin solid coating layer (i.e., the thermal capacity of the wall and coating layer are neglected). This assumption leads to linear temperature profiles in these layers.

There are considerable variations and uncertainties in the boiling and melting processes and wide variations in the conditions within a given nozzle. Because of this problem, calculations are based on a range of representative hot-gas and boiling conditions.

<u>Melting of an insulating coating</u>. - With the preceding assumptions the analysis in reference 6 for condensing (or evaporation) and freezing (or melting) on a plate is simply modified to describe this problem. The solid layer thickness Δ_c at a position x' is given by equation (2):

$$\underbrace{\rho_s L_{SL} \frac{d\,\Delta_c}{dt}}_{A} = \underbrace{\frac{T_{SL} - T_c}{\dfrac{d_m}{k_m} + \dfrac{1}{h_c} + \dfrac{\Delta_c}{k_s}}}_{C} - \underbrace{h'_G(T_G - T_{SL})}_{G} = \left(\frac{Q}{A}\right)_c - \left(\frac{Q}{A}\right)_G \tag{2}$$

The properties are constant and T_c, h_c, h'_G, and T_G are given functions of position along the nozzle x', while $\Delta_c = \Delta_c\langle x', t\rangle$. For now it is assumed that only the heat of fusion of the coating absorbs heat before it reaches the wall. Later, this restriction is relaxed such that heat is absorbed by melting and partial vaporization of the coating. In either case, the liquid-vapor interface temperature will remain at nearly the melting temperature because of the low partial pressure of gaseous Al_2O_3 in the exhaust gases. Equation (2) relates the net heat absorbed or liberated by melting or freezing (term A) to the heat transferred to the coolant (term C) minus the fixed heat input from the hot gas (term G). If term G exceeds term C, there is melting (i.e., $d\,\Delta_c/dt$ is negative). Equation (2) is an ordinary differential equation in Δ_c because the small effect of the flowing melt layer, which would require partial differential equations if considered, was incorporated in the gas-side coefficient h'_G. Because of the many simplifications made previously, equation (2) is essentially the equation Stefan used to represent the freezing of ice on a pond, which is described in reference 7. The heat flux to the coolant is the same in each layer, because of the no-thermal-capacity assumption, so that the following relations result

$$\left(\frac{Q}{A}\right)_c = \left(\frac{Q}{A}\right)_c \langle x', t\rangle = \frac{T_{SL} - T_c}{b + \frac{\Delta_c}{k_s}} \tag{3a}$$

$$= \frac{T_{wc} - T_c}{\frac{1}{h_c}} \tag{3b}$$

$$= \frac{T_{SL} - T_{wg}}{\frac{\Delta_c}{k_s}} \tag{3c}$$

where

$$b = \frac{d_m}{k_m} + \frac{1}{h_c}$$

For a given burnout heat flux $(Q/A)_{max}$, the minimum coating thickness required to prevent burnout Δ_Q can be derived from equation (3a):

$$\Delta_Q = k_s \left[\frac{T_{SL} - T_c}{\left(\frac{Q}{A}\right)_{max}} - b\right] \tag{4}$$

A minimum coating thickness must also be based on the maximum allowable wall temperature $T_{w,max}$. The coating thickness to prevent overheating Δ_T is obtained from equations (3a) and (3c), where $T_{wg} \equiv T_{w,max}$:

$$\Delta_T = \frac{bk_s(T_{SL} - T_{w,max})}{T_{w,max} - T_c} \tag{5}$$

The steady-state coating thickness Δ_{ss} is derived from equation (2) by setting $d\,\Delta_c/dt \equiv 0$:

$$\Delta_{ss} = k_s\left[\frac{T_{SL} - T_c}{h'_G(T_G - T_{SL})} - b\right] \tag{6}$$

Bare-wall nozzle. - If there is no coating on the nozzle walls (bare wall), the heat flux to the coolant would be

$$\left(\frac{Q}{A}\right)_c = \left(\frac{Q}{A}\right)_c \langle x' \rangle = \frac{T_G - T_c}{b + \frac{1}{h'_G}} \tag{7a}$$

$$= \frac{T_{wg} - T_c}{b} \tag{7b}$$

Solving for T_{wg} results in

$$T_{wg} = T_{wg}\langle x' \rangle = T_c + \frac{T_G - T_c}{1 + \frac{1}{bh'_G}} \tag{8}$$

Representative Thermal Conditions in Nozzle

Representative bare-wall gas-side heat-transfer coefficients $h'_G\langle x' \rangle$ and gas temperatures for a hypothetical 6-foot- (1.8-m-) diameter throat solid rocket nozzle in the 3×10^6-pound (13×10^6-N) thrust range have been estimated by W. L. Jones of the Lewis Research Center on the basis of analysis and experiments at Lewis. These conditions are listed in table I for five stations along the nozzle, which are designated in figure 3. Two extremes of h'_G are listed because of significant uncertainties in its value. Pessimistic values of h'_G are analytical values, while the lower optimistic values are ex-

TABLE I. - THERMAL CONDITIONS IN LARGE HYPOTHETICAL SOLID ROCKET NOZZLE WITH 6-FOOT- (1.8-m-) DIAMETER THROAT, 3×10^6 POUND (13×10^6 N) THRUST, AND REPRESENTATIVE COOLANT CONDITIONS FOR AXIAL BOILER TUBES SHOWN IN FIGURE 3

[Properties of Al_2O_3 coating: melting point, 3720° F (2320 K); thermal conductivity, 1.57 Btu/(ft)(hr)(°F) (2.7 W/(m)(K)); density, 200 lbm/ft^3 (3200 kg/m^3); heat of fusion, 460 Btu/lbm (1100 kJ/kg); heat of vaporization (estimated at 1 atm (100 kN/m^2)), 1630 Btu/lbm (3800 kJ/kg); boiling point (estimated at 1 atm (100 kN/m^2)), 5440° F (3100 K). Stainless steel wall: thermal conductivity, 11 Btu/(ft)(°F) (19 W/(m)(K)).]

Nozzle station	Case	Bare-wall gas-side heat-transfer coefficient,[a] h'_G		Hot-gas temperature,[a] T_G		Coolant-side heat-transfer coefficient, h_c		Coolant temperature,[b] T_c	
		Btu/(ft^2)(hr)(°F)	W/(m^2)(K)	°F	K	Btu/(ft^2)(hr)(°F)	W/(m^2)(K)	°F	K
1	Pessimistic	280	1600	5500	3300	500	2 850	485	525
	Optimistic	280	1600	5500	3300	5000	28 500	445	500
2	Pessimistic	665	3800	5490	3300	500	2 850	470	515
	Optimistic	665	3800	5490	3300	5000	28 500	415	485
3	Pessimistic	1500	8500	5400	3250	5000	28 500	455	510
	Optimistic (throat)	905	5150	5400	3250	5000	28 500	390	470
4	Pessimistic	800	4550	4750	2900	1000	5 700	430	495
	Optimistic	600	3400	4750	2900	5000	28 500	340	445
5	Pessimistic	325	1850	4390	2700	1000	5 700	400	480
	Optimistic	325	1850	4390	2700	5000	28 500	330	440

[a]Based on bare-wall nozzle data and/or calculations.

[b]Determined by assuming saturation conditions and following linear pressure variations: Pessimistic: 600 to 250 psia (4100 to 1700 kN/m^2 abs); Optimistic: 400 to 100 psia (2800 to 690 kN/m^2 abs).

trapolated from experimental data with small rocket engines. The pressure within the nozzle P_N varies from about 600 psia (4100 kN/m^2) in the chamber to 300 psia (2080 kN/m^2) in the throat and finally near the exit it has fallen to about 6 psia (41 kN/m^2). Also listed in table I are representative estimates of the coolant-side heat-transfer coefficients h_c and coolant temperatures T_c for a situation where the coolant flows and boils in the axial nozzle length tubes that are shown in figure 3. Because of uncertainties in the boiling region locations and other uncertainties in the boiling process, two representative extremes of these cooling parameters are listed in the table. The optimistic values are based on the assumption that effective boiling occurs at that station and that the pressure varies in the tube from 400 psia (2800 kN/m^2) at the entrance

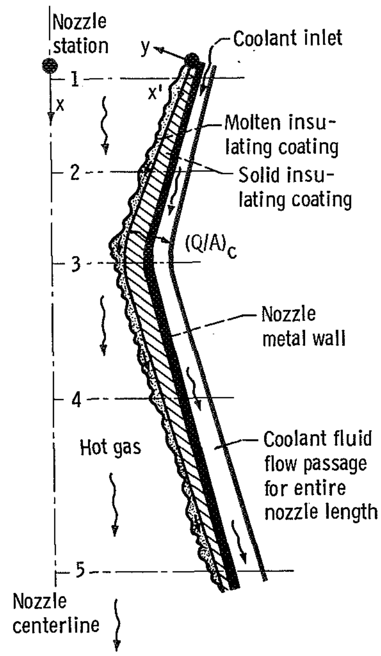

Figure 3. - Schematic representation of nozzle cooled by boiling in axial nozzle length tubes covered with melting insulating coating.

to 100 psia (690 kN/m^2) at the exit. Pessimistic values are based on the much lower values of h_c associated with significantly less effective heat transfer and a pressure variation of from 600 to 250 psia (4100 to 1700 kN/m^2). The throat will be assumed to have the same boiling coefficient in either case. The representative coolant temperatures, which have a far lesser effect than expected h_c variations, are specified by assuming a linear pressure drop and a saturated coolant. The simplicity of this analysis is justified by the results which indicate that no further refinement is necessary in order to determine whether the thermal protection method investigated could be practical.

Results and Discussion for Boiling in Tubes

Bare-wall nozzle. - Table II indicates the heat flux and hot-surface wall temperature at each station for the pessimistic and optimistic conditions listed in table I. It should be expected that the tube will fail if the wall temperature exceeds some maximum allowable wall temperature or the heat flux exceeds an allowable maximum heat flux (i. e., burnout flux). It is difficult to estimate a specific value of burnout flux in this case because the heat flux varies greatly along the nozzle, the coolant tube is curved and heated on one side, and the tube cross section varies such that it is smallest

TABLE II. - HEAT FLUX AND WALL TEMPERATURE FOR BARE-WALL HYPOTHETICAL NOZZLE SUBJECTED TO THERMAL CONDITIONS OF TABLE I

[Wall thickness of stainless steel, d_m, 0.04 in. (10^{-3} m).]

Nozzle station	Case	Wall temperature on gas side, T_{wg}		Heat flux to coolant, Q/A	
		^oF	K	Btu/(ft^2)(hr)	W/m^2
1	Pessimistic	2450	1600	8.54×10^5	2.7×10^6
	Optimistic	1070	850	1.24×10^6	4×10^6
2	Pessimistic	3505	2200	1.32×10^6	4.2×10^6
	Optimistic	1690	1200	2.53×10^6	8.1×10^6
3	Pessimistic	2580	1700	4.23×10^6	1.35×10^6
	Optimistic	1960	1350	3.1×10^6	1×10^7
4	Pessimistic	2635	1700	1.69×10^6	5.4×10^6
	Optimistic	1360	1000	2.03×10^6	6.5×10^6
5	Pessimistic	1585	1150	9.1×10^5	2.9×10^6
	Optimistic	895	750	1.13×10^6	3.6×10^6

in the throat region. There are not sufficient data to estimate the local variation of burnout flux along the coolant tube in this case. In the absence of such data, it is assumed that the burnout data shown in figure 1, which is for uniformly heated tubes of constant cross section, are adequate for this comparison. From figure 1 it can be seen that, even with a swirl insert in the tube and a low exit quality, the maximum allowable heat flux (i.e., the burnout flux) will be about 1×10^6 to 1.5×10^6 Btu per square foot per hour (3.16×10^6 to 4.75×10^6 W/m^2). The maximum allowable wall temperature for an inexpensive material such as stainless steel is about 1000^o to 1500^o F (810 to 1100 K). Even for the optimistic conditions, both the maximum allowable wall temperature and burnout heat flux are surpassed in many regions of the nozzle so that the boiling-water-cooled bare-wall thermal-protection method should be expected to fail.

Wall with insulating coating. - The bare-wall results clearly indicate the need for an insulating coating on the tubes in order to reduce the tube wall temperatures and heat flux to the coolant. The insulating coating considered, which is largely composed of Al_2O_3, is applied, as shown in figure 2, over the coolant tubes. The steady-state coating thicknesses at the five stations along the nozzle, which are subject to the thermal conditions listed in table I, are calculated from equation (6). These steady-state results are plotted as points in figure 4. This will determine whether the coating will melt or tend to freeze when compared with some given initial thickness. If, for ex-

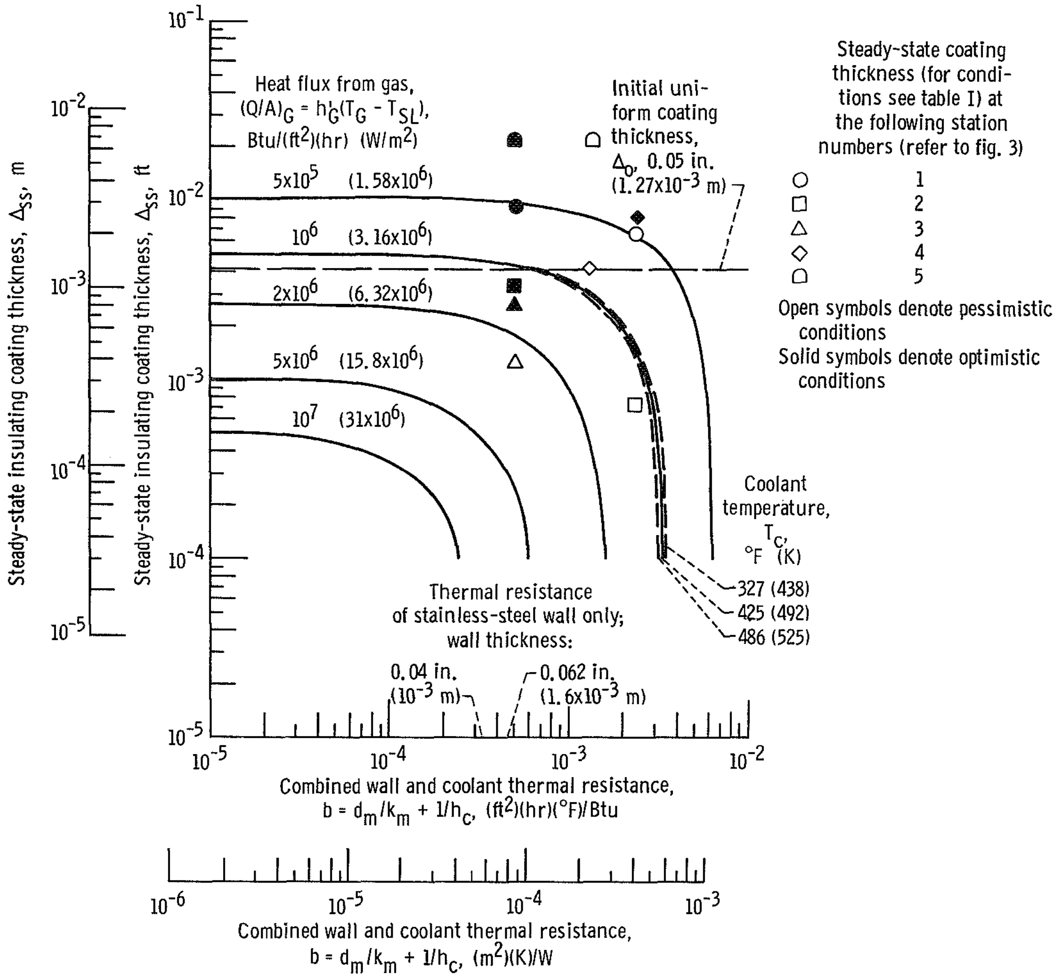

Figure 4. - Steady-state insulating coating thickness (Al_2O_3) as function of combined wall and coolant thermal resistance with heat flux from gas as parameter.

ample, the steady-state thickness is less than the initial coating thickness, the coating will melt to the steady-state thickness. Since there is significant Al_2O_3 in liquid form, although little Al_2O_3 gas, in the exhaust, freezing may occur where the initial coating thickness is less than the steady-state thickness. However, it is not certain that the steady-state thickness could be achieved by freezing under these conditions. Consider a given uniform initial coating thickness of $\Delta_0 = 0.05$ inch (1.27×10^{-3} m) and stainless tubes of 0.04-inch (10^{-3}-m) wall thickness. Figure 4 indicates that there would be melting at stations 2 and 3. The most severe conditions occur in the nozzle throat region and that region will be considered in the most detail.

Equation (2) is solved numerically for the coating thickness during the transient melting process. Figure 5 is a plot of the change with time of the coating thickness Δ_c at a few of the critical stations for the thermal conditions listed in table I. The tube

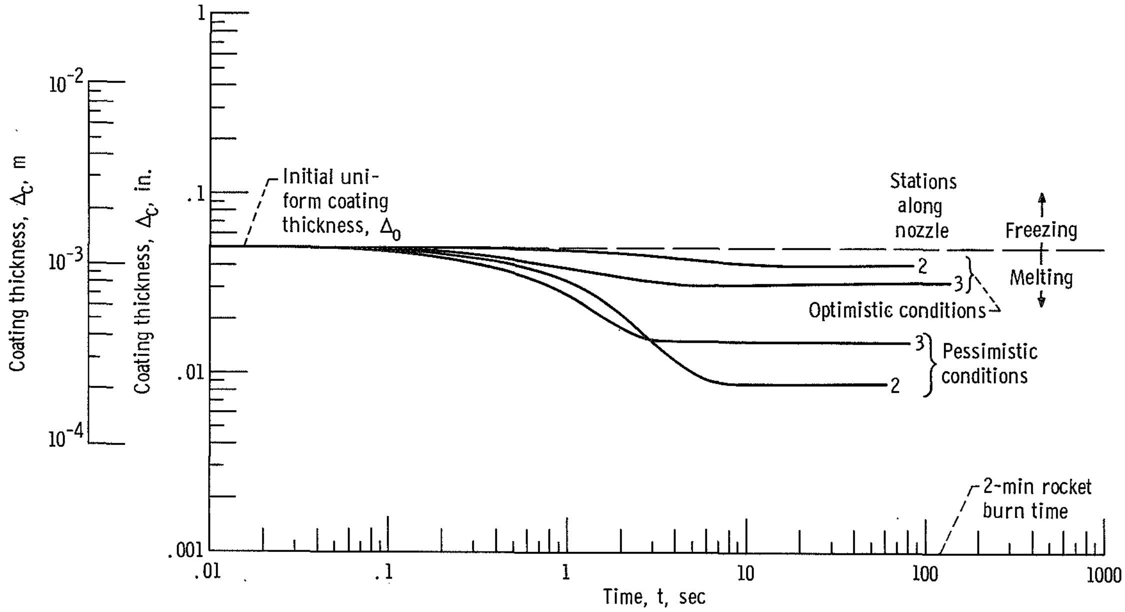

Figure 5. - Melting of coating with time at some critical stations along nozzle. Calculation is based on conditions in table I: thickness of metal wall, 0.04 inch (10^{-3} m); initial uniform coating thickness, 0.05 inch (1.27×10^{-3} m).

wall thickness d_m is taken to be 0.04 inch (10^{-3} m), and for this calculation a uniform initial coating thickness Δ_0 of 0.05 inch (1.27×10^{-3} m) is used. Unfortunately, the coating at the critical throat region melts to a steady-state thickness very rapidly (in about 3 to 20 sec). Increasing the initial coating thickness Δ_0 by a factor of 10 does not alter this result there appreciably; nor does a tenfold increase in h_c. Increasing the metal wall thickness from 0.04 to 0.062 inch (10^{-3} to 1.6×10^{-3} m) increases the wall temperature thereby making matters worse. Therefore, it is clear that a simple melting insulating coating does not significantly retard the rapid attainment of a steady-state coating thickness because the heat absorbed by the melting process (heat of fusion), compared with the heat transferred, is typically small for this process. Thus, the analysis of this transient problem simplifies to a simple steady-state determination. If it is assumed that additional heat is absorbed by evaporating all the melt layer, such that the heat of fusion is increased by the addition of the heat of vaporization, there will be a further delay in the attainment of steady state, but not nearly enough. Therefore, if the steady-state coating thickness Δ_{ss}, as determined from equation (6) and plotted in figure 4, falls below either of the minimum coating thicknesses required to prevent burnout Δ_Q, or simple overheating Δ_T, that design will fail at that point. The minimum coating thickness to prevent simple overheating is determined from equation (5) and is plotted for various assigned values of $T_{w,max}$ in figure 6. The minimum coating thickness to prevent burnout is determined from equation (4) and is plotted in figure 6 for various assigned values of the burnout heat flux $(Q/A)_{max}$. Should burnout

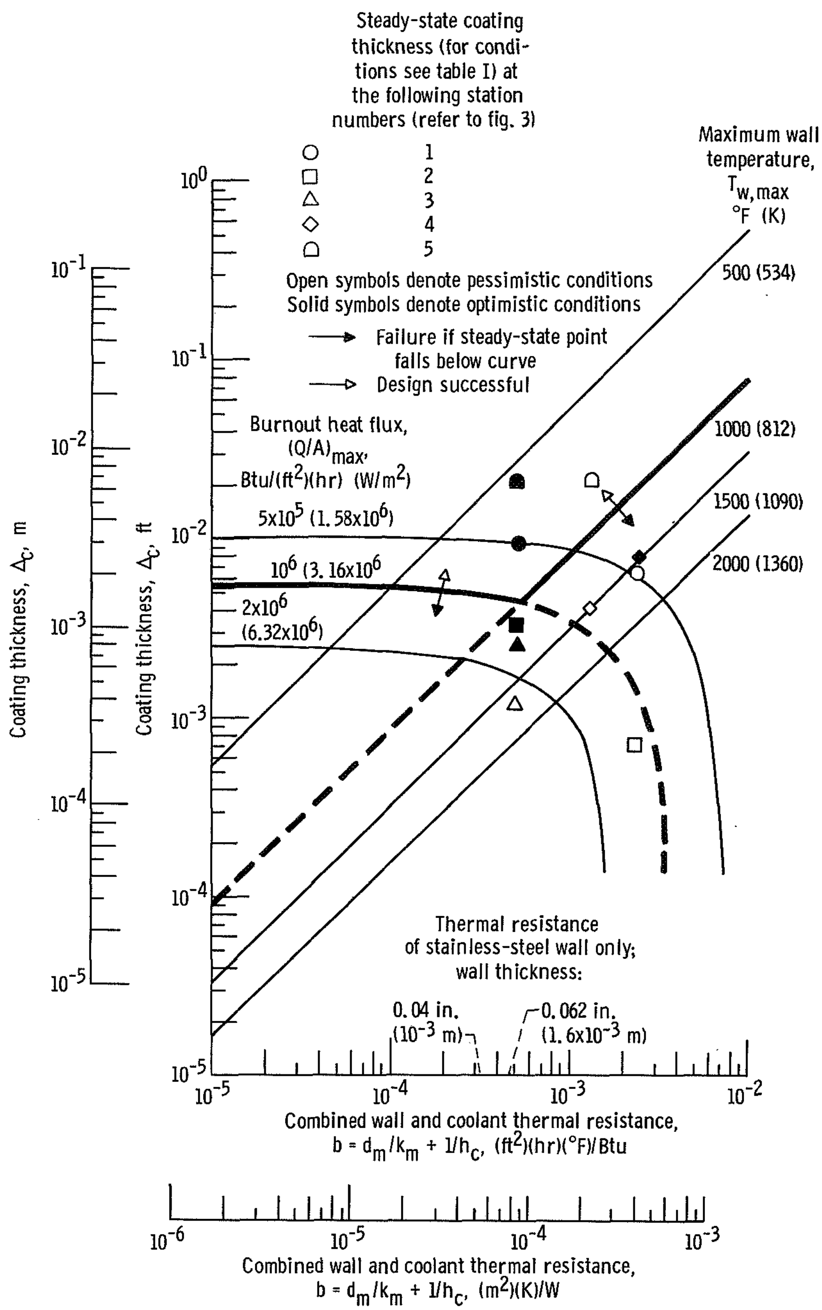

Figure 6. - Comparison of steady-state coating (Al_2O_3) thicknesses, for conditions of table I, with the minimum coating thicknesses to prevent burnout Δ_Q and tube overheating Δ_T comparison shows where such failures occur.

occur, the coolant heat transfer coefficient h_c drops appreciably, and tube overheating results.

From the burnout data of figure 1 it is unlikely that the maximum allowable heat flux $(Q/A)_{max}$ can exceed 10^6 Btu per square foot per hour (3.2×10^6 W/m^2) with a swirl insert and reasonable exit quality. The reader is again cautioned that figure 1 is based on burnout data for uniformly heated tubes of constant cross section. In the absence of more applicable data, figure 1 is assumed to be adequate here. The requirement of inexpensive tube materials and construction methods essentially limits the maximum wall temperature $T_{w,max}$ to between 1000° to 1500° F (810 to 1100 K) for stainless steel. A conservative value of 1000° F (810 K) is used for $T_{w,max}$. These limitations form the envelope which is described by the two intersecting heavy curves in figure 6. The steady-state coating thicknesses[1] at the five nozzle stations, which are subject to the thermal conditions listed in table I, are plotted as points in figure 6. There will be tube failure for those points that fall below the envelope as described. It is clear from figure 6 that, in most areas of the nozzle, the coolant tubes can be expected to fail either by burnout or simple overheating. The nozzle throat region is the most severe area because both burnout and simple overheating can be expected there. Failure occurs at station 4 because of simple overheating. The region at station 2 is nearly safe for the optimistic case, where h_c is high because the coolant is boiled there, but there is considerable overheating at station 2 for the pessimistic case where there is no boiling. A high value of h_c, which results when the coolant is boiled there, is also required at station 1 in order to avoid overheating.

Clearly, some changes are necessary in order to improve this thermal-protection scheme, which is at best marginal for the Al_2O_3 coating. If the coating were composed of zirconium oxide, with its appreciably high melting point, the thermal protection would be adequate. However, this coating may react chemically with the aluminum in the exhaust product. Even considering only the optimistic points with Al_2O_3, an increase in the maximum allowable burnout flux to about 1.5×10^6 Btu per square foot per hour (4.75×10^6 W/m^2) is required at stations 2 and 3. Figure 1 indicates that, in order to achieve this higher burnout flux, the exit quality must be reduced to about 10 percent and a swirl insert must be used. In order to absorb this heat flux, with such a low exit quality, the coolant flow must be greatly increased. In addition, simple overheating at stations 2 and 3 is excessive even with a boiling coolant. To reduce simple overheating there, a reduction of the wall thickness would be necessary since the wall is the controlling thermal resistance. It is desirable to reduce the overall coolant mass flow required to cool the entire nozzle. In the throat region, high flow rates and coolant velocities are required, whereas elsewhere much higher qualities and correspondingly

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[1]For this comparison, the coating applied initially is assumed to be at least as thick as the steady-state thickness.

lower coolant flows are possible. It would therefore appear to be desirable to use circumferential or short axial coolant tubes with swirl inserts rather than the nozzle length tubes discussed previously. In this way, the coolant flow, coolant tube cross section, and coefficient could be more readily varied axially along the nozzle in order to avoid failure and also to reduce the overall coolant requirements.

An estimate of the coolant flow requirement at any axial location in the nozzle is now made for circumferential or short axial tubes. The coolant flow rate per unit nozzle surface area G_l is given, approximately, by equation (9):

$$G_l \cong \frac{h'_G(T_G - T_{SL})}{c_l(T_{LV} - T_c) + \chi_{ex}L_{LV}} \tag{9}$$

Figure 7 contains the results of such a calculation for a number of exit qualities χ_{ex} where the nozzle thermal conditions correspond to the worst location, the throat. These are given by P_N = 300 psia (2080 kN/m^2), T_G = 5400^o F (3250 K), and $250 \le h'_G \le 2000$

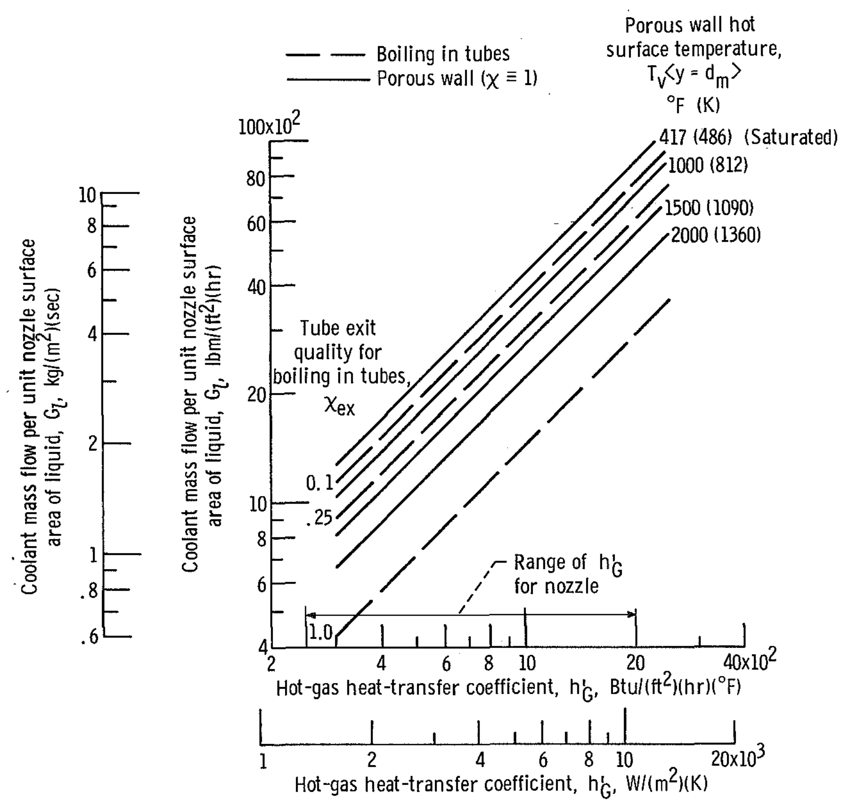

Figure 7. - Comparison of coolant flow rates for boiling water in tubes or within porous wall that lines rocket nozzle. Nozzle conditions: hot-gas temperature, 5400° F (3250 K); nozzle pressure, 300 psia (2080 kN/m^2); coolant temperature, 60° F (290 K).

Btu per square foot per hour per ^{o}F ($1400 \le h'_G \le 11\ 000$ W/(m^2)(K)); the coolant is optimistically taken at $T_c = 60^o$ F (290 K). From before, the throat region of the nozzle should be subject to a heat flux of about 1.5×10^6 Btu per square foot per hour (4.75×10^6 W/m^2) and an h'_G = 1000 Btu per square foot per hour per ^{o}F (5700 W/(m^2)(K)) so that the exit quality required to prevent burnout in this region, with an effective swirler, would be about 10 percent. From figure 7, the necessary coolant flow flux required to attain this maximum exit quality in the throat region is about 3800 pounds mass per hour per square foot (5.2 kg/(m^2)(sec).

Cooling by boiling other liquids. - A boiling-water coolant appears to be better than either a subcritical boiling cryogen (e.g., liquid nitrogen) or a boiling liquid metal. This occurs because the heat of vaporization of water is much higher than the cryogen, allowing for a lower mass flow rate for cooling water. Also, the boiling point for water is far lower than for the liquid metal, so that the overheating problem of an inexpensive tube material, such as stainless steel, is far less severe with water.

COOLING BY BOILING IN OR ON POROUS NOZZLE WALLS

In this section, thermal protection is accomplished by lining the nozzle walls with a porous wall (see fig. 8). The coolant (e.g., water) enters the porous wall from a

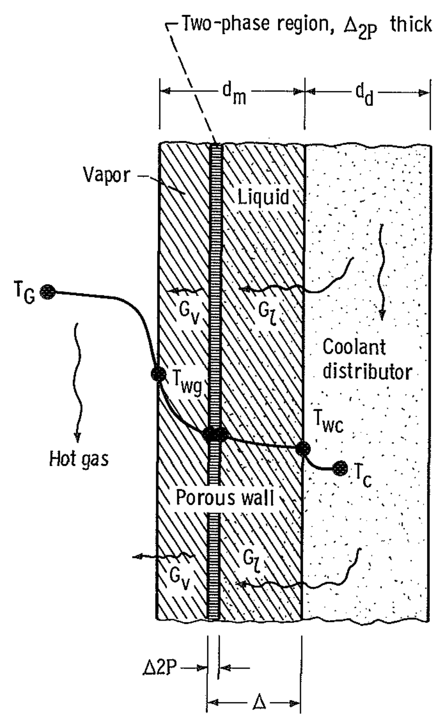

Figure 8. - Hot gas flowing past axial segment of porous wall lining of nozzle cooled by boiling within porous wall.

coolant distributor system behind the porous wall. The coolant vaporizes within the pores or at the hot-gas surface of the porous wall. Heat transferred to the wall is thereby largely absorbed by the coolant vaporization process and partly by heating up the incoming coolant liquid to the saturation temperature. Conceptually, the overheating and burnout limitations of the previous method could be avoided.

In the case where there is boiling within the pores, the following quantities are determined: the porous wall temperatures, heat flux to the wall and the coolant distributor, and the location of the phase change boundary. Where there is boiling at the hot surface, the flow required to maintain a vanishingly thin layer of liquid at the hot surface is determined; porous-wall temperatures and heat flux to the wall and coolant distributor are also found. No attempt is made to analyze the serious startup problem of the porous wall and coolant distribution problem caused by the large axial variations in nozzle pressure and heat flux. This study is limited to thermal performance.

Analysis

In porous-wall gas-transpiration studies, where a gas flows into the boundary layer, it has been shown that there is an increase in the boundary layer thickness so that the heat-transfer coefficient h_G is reduced compared with the coefficient for a solid nonporous wall h'_G. An experimental study of vaporization of a liquid coolant within and at the surface of a porous wall was performed in reference 8, where h_G was measured and found not to be lower, as supposed from gas-transpiration studies, but somewhat higher than the h'_G for a solid wall. There is still considerable discussion about this result, and the temperature and heat flux range of this experiment are not nearly within the range of the nozzle. Therefore, for now it is best to assume that h_G is equal to values used for a bare-wall nozzle (i.e., $h_G \equiv h'_G$). This problem is assumed to be adequately described by a boiling region, Δ_{2P} thick, between wholly liquid and vapor regions. So long as the porous channels are of small cross section, it can probably be reasonably assumed that this boiling (two phase) region will be negligibly thin (i.e., $\Delta_{2P} \equiv 0$) unless the porous wall is thin. A qualitative experiment conducted by this writer indicated that a fine liquid spray (i.e., fine mist) passed into the hot-gas stream regardless of whether the porous wall was heated or cold. When more coolant is used than necessary for vaporization at the hot surface, a layer of liquid will flow over the hot surface, which may not be vaporized entirely. Therefore, to cover both of these possibilities it is assumed that some fraction of the coolant liquid flow $1 - X$ is not vaporized and passes through the vapor region of the porous wall without further change. A somewhat common assumption is that the porous void surface area is large such that the liquid or vapor flowing in the voids is locally at the temperature of the adjacent solid. With these assumptions, the analysis greatly simplifies. Energy transfer in the vapor and liquid regions is described, respectively, by

$$k_v' \frac{d^2T_v}{dy^2} - G_v c_v \frac{dT_v}{dy} = 0 \tag{10}$$

and

$$k_l' \frac{d^2T_l}{dy^2} - G_l c_l \frac{dT_l}{dy} = 0 \tag{11}$$

The effective thermal conductivities are given in terms of the thermal conductivities of the vapor, liquid, and the solid part of the porous wall, and the wall porosity:

$$k_v' = k_v p + k_m(1 - p) \tag{12a}$$

$$k_l' = k_l p + k_m(1 - p) \tag{12b}$$

Mass flow continuity in the vapor region is given by

$$G_v = \chi G_l \tag{13}$$

while G_l is constant within the liquid region. When $\chi = 1$, there is complete vaporization of the coolant liquid within the pores.

The integration of the describing differential equations (i.e., eqs. (10) and (11)) is performed with the properties and porosity assumed constant. In a practical case, the porous wall may not be of constant porosity across the wall. For example, the porous wall may be made of layers, with each layer made of a different mesh screening. In such cases, the differential equations will have to be solved by a direct numerical method.

The boundary conditions for the problem are given in equations (14) to (17). An energy balance at the hot surface $y = d_m$ is given by equation (14):

$$k_v' \frac{dT_v}{dy}\langle y = d_m\rangle = h_G'(T_G - T_v\langle y = d_m\rangle) \tag{14}$$

The two-phase region has been assumed to be of vanishing thickness (i.e., $\Delta_{2P} \equiv 0$), so that the boundary conditions at the liquid-vapor transition $y = \Delta$ are

$$T_v\langle y = \Delta\rangle = T_l\langle y = \Delta\rangle = T_{LV} \tag{15}$$

and

$$k_v' \frac{dT_v}{dy}\langle y = \Delta\rangle = k_l' \frac{dT_l}{dy}\langle y = \Delta\rangle + \chi G_l L_{LV} \tag{16}$$

If the coolant flow to the hot surface is greater than that necessary to have boiling there, a liquid layer will be present (i.e., film cooling). However, because of the high gas velocity, this layer will be wavy and tend to be ripped off so that the thermal resistance of the layer can be assumed to be relatively small compared with that of the gas. By this assumption, this analysis will approximately describe the film cooling also, where $\chi < 1$.

A heat balance at $y = 0$, the cold side of the porous wall, where the coolant in the distributor is flowing past and into the porous wall, is given by the following relation:

$$k_l' \frac{dT_l}{dy}\langle y = 0\rangle = h_c(T_l\langle y = 0\rangle - T_c) \tag{17}$$

The coolant side coefficient h_c is a function of G_l in this case because the coolant flows through a coolant distributor channel (e.g., a hoop), past the cold side of the porous wall, and then through the porous wall. From boundary layer suction theory (ref. 9), h_c can be approximately determined, for a reasonably deep channel, by the following relation:

$$h_c = \frac{G_l c_l}{1 - \exp\left(-\frac{G_l c_l}{h_{co}}\right)} \tag{18}$$

The coefficient for zero flow through the porous wall h_{co} is determined by using the following empirical equation of reference 7 for flow through a channel of depth d_d and around a nozzle of diameter D_N:

$$h_{co} = 0.023\, c_l\left(\frac{G_l \pi D_N}{d_d}\right)\left(\frac{G_l \pi D_N}{\mu_l}\right)^{-0.2}(Pr_l)^{-0.6} \tag{19}$$

The solution to the differential equations (17) and (18), where ξ_i is constant, is of the form

$$T_i = C_{1i}\, e^{\xi_i y} + C_{2i} \tag{20}$$

where i stands for liquid or gas and ξ_i is defined as

$$\xi_l \equiv \frac{G_l c_l}{k_l'} \tag{21a}$$

$$\xi_v \equiv \frac{G_v c_v}{k_v'} \tag{21b}$$

This solution and the boundary conditions given in equations (14) to (17) are used to determine the constants C_{1v}, C_{1l}, C_{2v}, and C_{2l}:

$$C_{1v} = \frac{h_G'(T_G - T_{LV})}{k_v'\xi_v\, e^{\xi_v d_m} + h_G'\left(e^{\xi_v d_m} - e^{\xi_v \Delta}\right)} \tag{22}$$

$$C_{1l} = \frac{h_c(T_{LV} - T_c)}{k_l'\xi_l - h_c\left(1 - e^{\xi_l \Delta}\right)} \tag{23}$$

$$C_{2v} = T_{LV} - C_{1v}\, e^{\xi_v \Delta} \tag{24}$$

$$C_{2l} = T_{LV} - C_{1l}\, e^{\xi_l \Delta} \tag{25}$$

The hot-side surface temperature of the porous wall is given by

$$T_v\langle y = d_m\rangle = C_{1v}\, e^{\xi_v d_m} + C_{lv} \tag{26}$$

The heat flux from the hot gas is

$$\left(\frac{Q}{A}\right)_G = h_G'(T_G - T_v\langle y = d_m\rangle) \tag{27}$$

The temperature at the cold-side porous wall surface is given by

$$T_l\langle y = 0\rangle = C_{1l} + C_{2l} \tag{28}$$

and the heat flux into the coolant passage by

$$\left(\frac{Q}{A}\right)_c = h_c(T_l\langle y = 0\rangle - T_c) \tag{29}$$

Equation (30) is derived from equations (16) and (20) to (25) and is solved for the location of the two-phase region Δ:

$$f\langle\Delta\rangle = 0 = \frac{k_l' h_c(T_{LV} - T_c)\xi_l\, e^{\xi_l \Delta}}{k_l' \xi_l - h_c\left(1 - e^{\xi_l \Delta}\right)} - \frac{k_v' h_G'(T_G - T_{LV})\xi_v\, e^{\xi_v \Delta}}{k_v'\xi_v\, e^{\xi_v d_m} + h_G'\left(e^{\xi_v d_m} - e^{\xi_v \Delta}\right)} + \chi G_l L_{LV} \tag{30}$$

In solution of equation (30), it is assumed that the pressure drop across the vapor region is small so that T_{LV} and L_{LV} can be simply evaluated at the adjacent nozzle pressure P_N. If the solution of equation (30) indicates that $0 < \Delta < d_m$, boiling occurs within the porous wall. If $\Delta \leq 0$, boiling occurs in the coolant passage, and this analysis is not applicable. If $\Delta \geq d_m$, boiling occurs essentially at the hot surface of the porous wall, since the thermal resistance of any layer of liquid there is neglected compared with the hot-gas resistance. For boiling at the hot surface, where $\Delta = d_m$, equation (30) simplifies to equation (31). Equation (31) is therefore solved for the minimum coolant flow $G_{l\,s}$ necessary to have vaporization at the hot surface:

$$f\langle G_{l\,s}\rangle = 0 = 1 - \left[\frac{h_G'(T_G - T_{LV}) - \chi G_{l\,s} L_{LV}}{G_{l\,s} c_l (T_{LV} - T_c)}\right]\left[1 + \left(\frac{G_{l\,s} c_l}{h_c} - 1\right) e^{-\xi_l d_m}\right] \tag{31}$$

Equation (30) can be manipulated into the following form when boiling occurs within the porous wall (i.e., $0 < \Delta < d_m$):

$$\frac{G_l c_l (T_{LV} - T_c)}{1 + \left(\frac{G_l c_l}{h_c} - 1\right) e^{-\xi_l \Delta}} + \chi G_l \left[L_{LV} + c_v(T_v\langle y = d_m\rangle - T_{LV})\right] = h_G'(T_G - T_v\langle y = d_m\rangle) \tag{32}$$

In practice, it is often true that $\left[(G_l c_l/h_c) - 1\right]e^{-\xi_l \Delta} << 1$ and positive, so that this equation simplifies to equation (33). Equation (33) is recognized as the result that would be obtained from a simple overall heat balance, and would result regardless of the effect of Δ_{2P}. Therefore, in practice, the effect of the porous-wall variables (i.e., porosity, conductivity, and thickness), which do not appear in (33), may not be important. Equation (33) gives the approximate coolant flow rate required to maintain the hot surface at a given temperature $T_v\langle y = d_m\rangle$:

$$G_l \cong \frac{h_G'(T_G - T_v\langle y = d_m\rangle)}{c_l(T_{LV} - T_c) + \chi\left[L_{LV} + c_v(T_v\langle y = d_m\rangle - T_{LV})\right]} \tag{33}$$

When boiling occurs at the hot surface, equation (33) or (31) simplifies to equation (34). Film cooling would be described here by assigning values of $\chi < 1$:

$$G_l = G_{l\,s} \cong \frac{h_G'(T_G - T_{LV})}{c_l(T_{LV} - T_c) + \chi L_{LV}} \tag{34}$$

Comparison of equations (33) and (34) shows that far less coolant flow is required when the hot surface is above the saturation temperature because of the reduction in the heat flux from the gas (the numerator of eq. (33)).

TABLE III. - COMPARISON OF EXPERIMENTAL RESULTS OF REFERENCE 10 WITH ANALYTICAL RESULTS FROM EQUATION (33) FOR BOILING OF WATER IN A POROUS WALL

[Nozzle pressure, 14.7 psia (100 kN/m^2 abs); porous-wall - fluid temperature in vapor region, $T_v\langle y = d_m\rangle = 215^\circ$ F (375 K); porosity of porous wall, 0.25; fraction of liquid vaporized, $\chi = 1$.]

Hot-gas-side temperature, T_G		Heat-transfer coefficient				Coolant temperature, T_c		Coolant mass flow per unit nozzle surface area liquid, G			
		Gas side, h_G		Coolant side, h_c				Experimental		Calculated (eq. (33))	
$^\circ$F	K	Btu/(ft^2)(hr)($^\circ$F)	W/(m^2)(K)	Btu/(ft^2)(hr)($^\circ$F)	W/(m^2)(K)	$^\circ$F	K	lbm/(ft^2)(hr)	kg/(m^2)(sec)	lbm/(ft^2)(hr)	kg/(m^2)(sec)
692	640	16.6	94	55	314	185	358	10.4	14×10^{-3}	7.9	10.5×10^{-3}
		15.1	86	72	410	191	361	9.7	13.2	7.3	9.9
		9.7	55	74	420	196	364	8.2	11	4.7	6.4
410	485	26.9	155	21	120	196	364	5.75	7.8×10^{-3}	5.3	7.2×10^{-3}
		24.9	143	75	427	197	364	4.8	6.5	4.9	6.65
		18.2	104	73	415	199	365	3.7	5	3.6	4.9

At this point, it would be helpful to compare the results of equation (33) with the comparable experimental results of reference 10. Table III indicates an adequate overall agreement.

Results and Discussion for Porous Wall

The analytical results for the porous wall are discussed in two parts. The first part considers boiling at the hot surface, while the second part considers boiling within the pores.

In order that the porous wall may be compared with the previous thermal protection method, where the coolant boiled in tubes, the thermal conditions listed in figure 7 are used in all porous-wall calculations; that is, the gas-side coefficient h'_G will be varied from 250 to 2000 Btu per square foot per hour per $^{\circ}$F (1400 to 11 000 W/(m^2)(K)) and a single representative gas and coolant temperature of 5400° and 60° F (3250 and 290 K), respectively, will be used.

Boiling at hot surface. - The coolant flow required to have vaporization at the surface G_{ls} is determined from equation (31). The cooling parameter $G_{ls}c_l/h'_G$ is plotted in figure 9 as a function of porosity (e.g., $0.05 \leq p \leq 0.95$) for a practical range of wall thickness ($1 \leq d_m \leq 0.1$ in. ($0.025 \leq d_m \leq 0.0025$ m)) and h'_G ($250 \leq h'_G \leq 2000$ Btu/(ft^2)(hr)($^{\circ}$F) ($1400 \leq h'_G \leq 11\ 000$ W/(m^2)(K)). This figure indicates that $G_{ls}c_l/h'_c$ is essentially independent of p, h'_G, G_{ls}, and d_m for these values. This result and equations (12) and (21) imply that k_m is also not important in the determination of G_{ls}. Therefore, the simplifications leading to equation (34) are justified in a practical situation. The largest reduction from a constant value of $G_{ls}c_l/h'_G$ (about 3 percent) that is shown in figure 9 occurs for the lowest values of p, h'_G, and d_m.

The coolant mass flow per unit nozzle surface area required for boiling at the hot surface in the severest area, the throat, where the nozzle pressure is 300 psia (2080 kN/m^2 abs), is computed by equation (34) and plotted in figure 7 as $T_v\langle y = d_m\rangle = 417^{\circ}$ F (486 K) (saturated). Because of the higher heat flux that results from the low, hot surface temperature (i.e., the coolant saturation temperature), the mass flow requirements are equivalent to the coolant tube method where the maximum exit quality is about 10 percent in order to prevent tube burnout. The nozzle pressure changes from 600 psia (4100 kN/m^2 abs) near the chamber to about 6 psia (41.0 kN/m^2 abs) near the exit. Because of this pressure variation, the required coolant flow would be about 13 percent higher near the exit than at the throat; near the chamber, the required flow would be about the same as at the throat. Based on figure 7 the initial weight of water coolant, for a 2-minute burn, is estimated to be about 1 percent of the thrust of the rocket nozzle described in table I (p. 9).

Figure 10 is derived from equations (22) to (26), (28), and (31) by setting $\Delta \equiv d_m$.

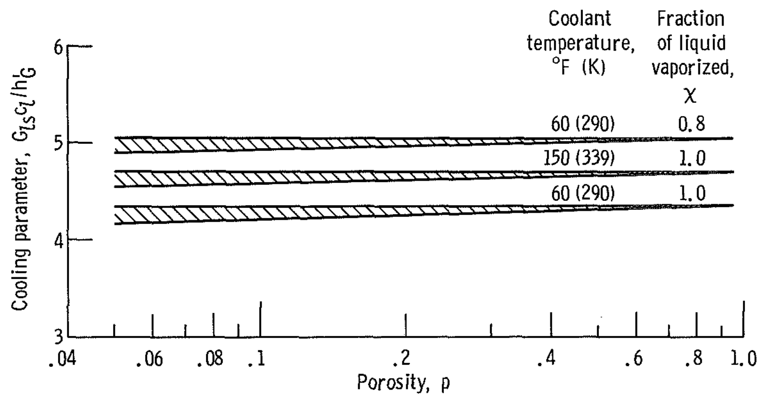

Figure 9. - Cooling parameter and its maximum variation as function of porosity for range of gas-side heat-transfer coefficient, $250 \leq h'_G \leq 2000$ Btu per square foot per hour per °F ($1400 \leq h'_G \leq 11\ 000$ W/(m^2)(K)); wall thickness, $0.1 \leq d_m \leq 1.0$ inch ($0.0025 \leq d_m \leq 0.025$ m); and coolant flow; where nozzle pressure, 300 psia (2080 kN/m^2); depth of coolant passage, 1.0 inch (0.025 m); hot-gas temperature, 5500° F (3300 K); and wall thermal conductivity, 10 Btu per foot per hour per °F (17.3 W/(m)(K)).

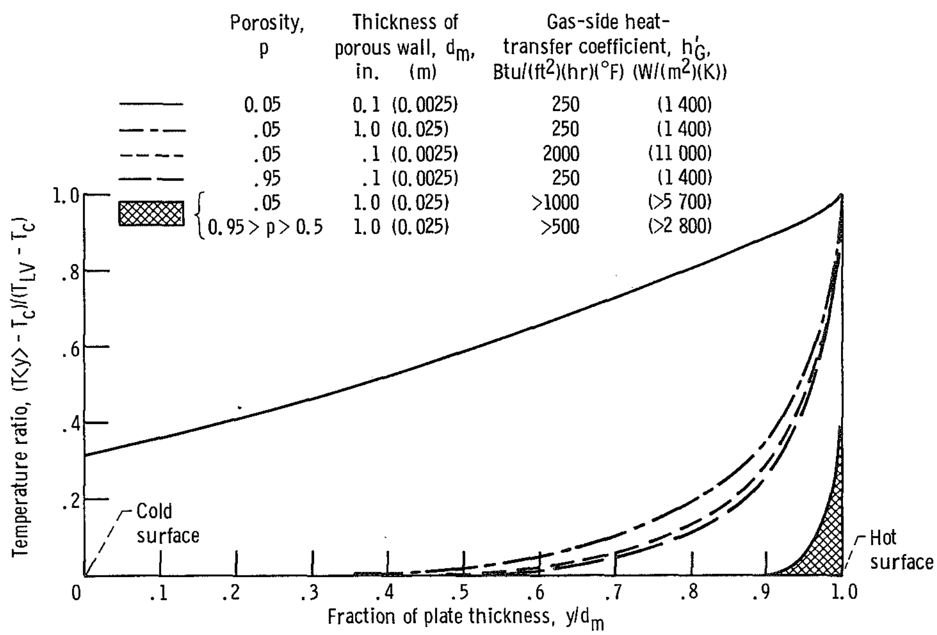

Figure 10. - Temperature profiles in porous wall where coolant water boils at hot surface. Hot-gas temperature, 5500° F (3300 K); coolant temperature, 60° F (290 K); fraction of liquid vaporized, 1.0; nozzle pressure, 300 psia (2080 kN/m^2); depth of coolant passage, 1.0 inch (0.025 m).

It gives a few representative temperature profiles across the porous wall. The porous-wall parameters must be considered in any calculation of local internal temperatures. Figure 10 indicates that, for the porosity range considered, the 1-inch- (0.025-m-) thick porous wall remains essentially at the incoming coolant temperature ($T_c = 60^\circ$ F (290 K)) throughout, except near the hot surface. The temperature within a porous wall of 0.1 inch (0.0025 m) thickness and porosity $p = 0.05$ is a good deal closer to the saturation temperature for $h'_G = 250$ Btu per square foot per hour per $^\circ$F (1400 W/(m^2)(K)). (This value of h'_G corresponds to a low coolant flow.) The temperature profiles become more nearly the uniform inlet coolant temperature T_c, as G_{ls}, (G_{ls} is proportional to h'_G), p, or d_m increase.

Figure 11 indicates the heat transfer from the hot gas and into the coolant distributor. This figure clearly shows that practically no heat passes through the wall to the coolant distributor even though the heat flux transferred from the hot gas to the wall can approach 10^7 Btu per square foot per hour (3.2×10^7 W/m^2). This is especially

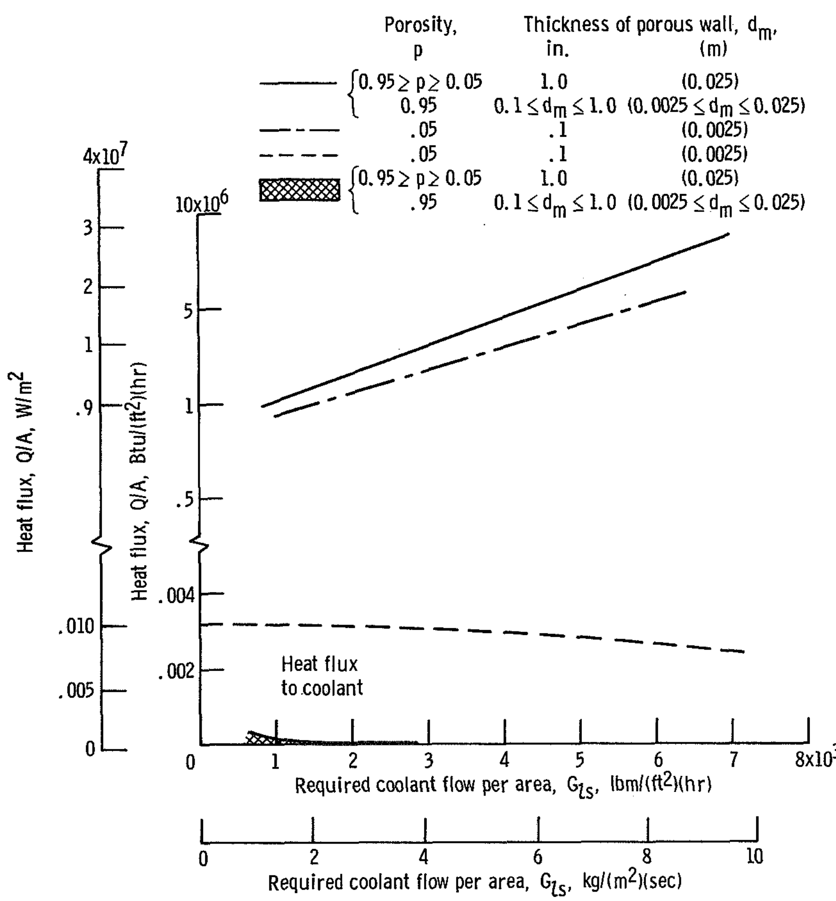

Figure 11. - Heat flux to coolant and from gas for boiling at hot surface. Hot-gas temperature, 5500° F (3300 K); coolant temperature, 60° F (290 K); fraction of liquid vaporized, 1.0; nozzle pressure, 300 psia (2080 kN/m^2); depth of coolant passage, 1.0 inch (0.025 m).

true at high coolant flow rates.

Lower vaporization fractions ($\chi < 1$) may be caused by liquid spitting out of the pores and/or liquid being ripped off from an excessively thick liquid film on the hot surface by the fast moving gas. Figure 9 indicates that somewhat more coolant is necessary in the event that $\chi < 1$, or the coolant temperature is higher. There is little effect on the temperature profiles if less than 100 percent of the liquid is vaporized at the hot surface (i.e., $\chi < 1$).

The results show that the porous wall can be maintained below the saturation temperature of the coolant, provided a sufficient coolant flow $G_{l\,s}$ is provided. Because of the resulting cold wall and low heat flux to the coolant distributor, the choice of porous materials is large and the structural and fabrication problems of the nozzle and porous wall are greatly lessened. The porous wall material could be an ablative material so that any occasional local overheating would not be serious.

Because of axial variations in pressure and heat flux, a circumferential (rings) coolant distribution system would be desirable; otherwise an axially varying porosity is necessary. Reducing startup problems with this coolant system may require an umbilical connection on the ground until lift-off.

<u>Boiling other liquids</u>. - Lithium would allow an order of magnitude greater reduction in coolant flow requirements compared with water; however, its exceptionally high

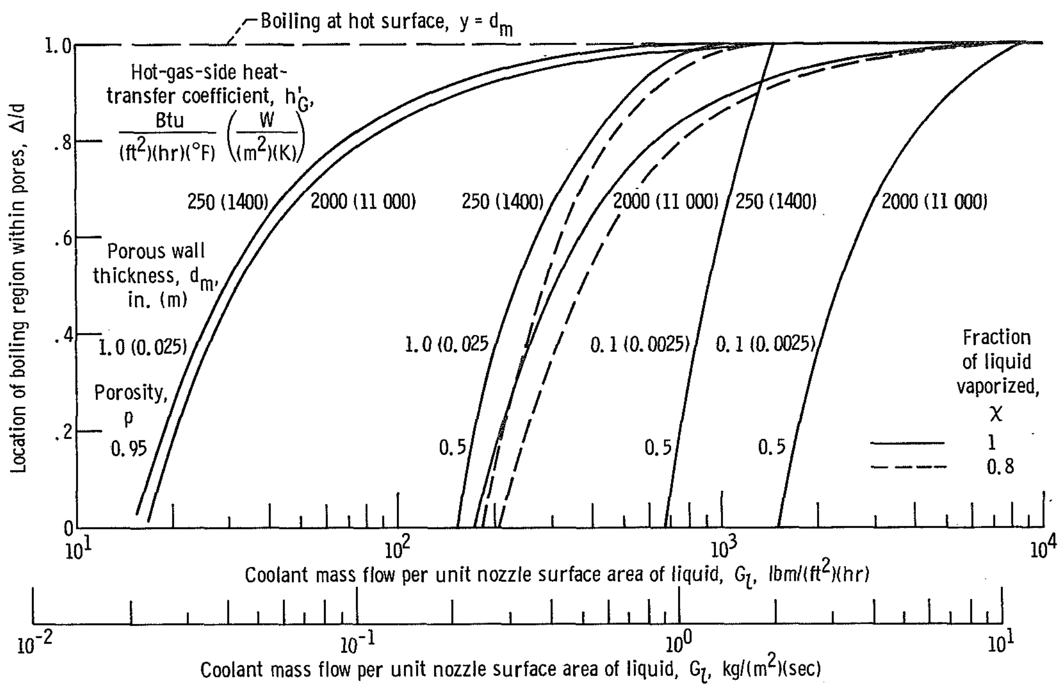

Figure 12. - Sensitivity of boiling location in the pores to gas-side heat-transfer coefficient and coolant mass flow per unit nozzle area of liquid. Hot-gas temperature, 5500° F (3300 K); nozzle pressure, 300 psia (2080 kN/m²); coolant temperature, 60° F (290 K); depth of coolant passage, 1.0 inch (0.025 m); wall thermal conductivity, 10 Btu per foot per hour per °F (17.3 W/(m)(K)).

boiling point (in excess of 3500° F at 300 psia (2200 K at 2080 kN/m^2)) severely limits the choice of porous-wall materials. A subcritical cryogen offers no advantage over water in this application because of its typically low heat of vaporization.

Boiling within porous wall. - Allowing vaporization to occur within the porous wall requires less coolant flow. Based on equation (33), figure 7 clearly shows this reduction in coolant flow for arbitrary hot surface temperatures in excess of the saturation temperature. Figure 12, determined from equation (30), indicates the location of the two-phase region as a function of the water coolant flow area for d_m = 0. 1- and 1-inch- (0.0025- and 0.025-m-) thick walls of porosity p = 0.95 and 0.5. Figure 13 indicates the hot surface temperature of the porous wall for the same porosities and a 1-inch- (0.025-m-) thick wall. From figure 13 it is clear that the coolant flows must approach the coolant flows needed to have surface boiling in order to keep the temperature of the porous wall within conventional porous-wall-material limits. Porosity has no appre-

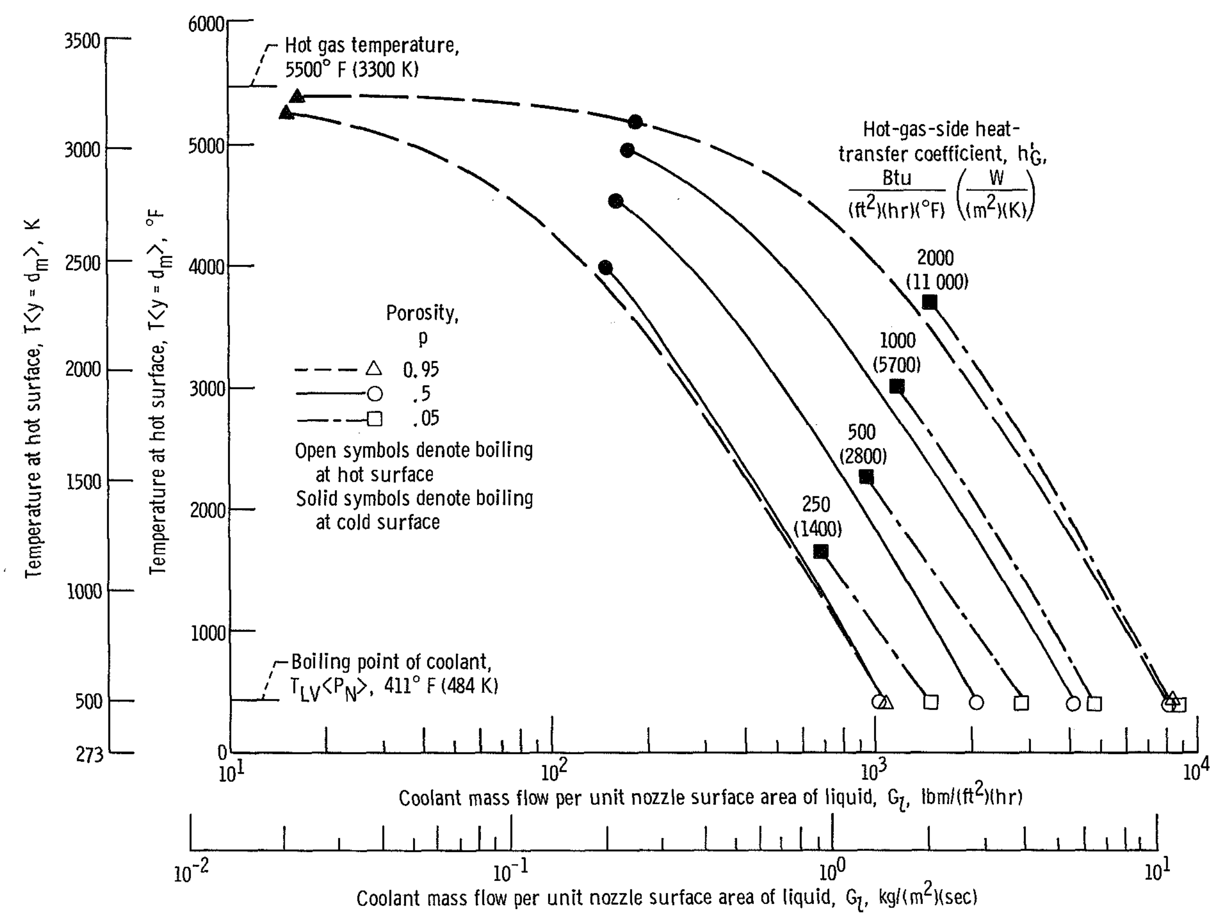

Figure 13. - Sensitivity of hot-gas-side porous wall surface temperature to gas-side heat-transfer coefficient and coolant mass flow per unit nozzle surface area of liquid. Hot-gas temperature, 5500° F (3300 K); nozzle pressure, 300 psia (2080 kN/m^2); coolant temperature, 60° F (290 K); thickness of porous wall, 1.0 inch (0.025 m); wall thermal conductivity, 10 Btu per foot per hour per °F (17.3 W/(m)(K)); fraction of liquid vaporized, 1.0.

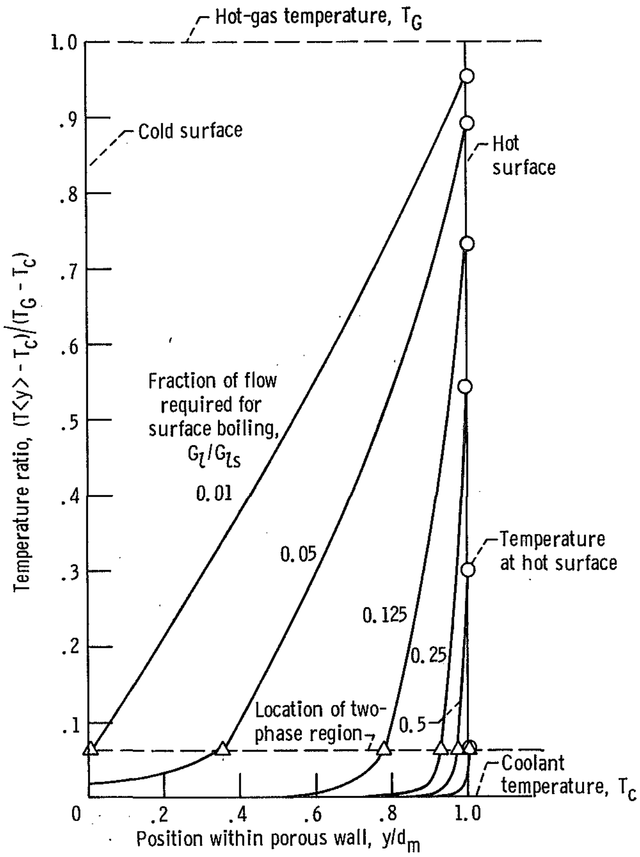

Figure 14. - Temperature profiles in porous wall as coolant flow approaches flow for surface boiling. Hot-gas temperature, 5500° F (3300 K); nozzle pressure, 300 psia (2080 kN/m^2); coolant temperature, 60° F (290 K); thickness of porous wall, 1.0 inch (0.025 m); depth of coolant passage, 1.0 inch (0.025 m); wall thermal conductivity, 10 Btu per foot per hour per °F (17.3 W/(m)(K)); fraction of liquid vaporized, 1.0; porosity, 0.5; hot gas-side heat-transfer coefficient, 1000 Btu per square foot per hour per °F (5700 W/(m^2)(K)).

ciable effect on that result. Figure 12 shows that these high coolant flows require that the boiling location be close to the hot surface. Plotting these two figures linearly indicates that small changes in coolant flow will not have a large effect on the hot surface temperature or the two-phase location, at least not for a wall thickness of 1 inch (0.025 m) or a high porosity. Figure 14 shows a few representative temperature profiles within the porous wall. Clearly, the temperature gradient, even at flow rates approaching that required to have surface boiling, is sharp near the hot surface such that a temperature measurement at the hot surface could be considerably in error.

SUMMARY OF RESULTS

The goal of this preliminary study was to investigate thermal-protection methods for a large solid rocket nozzle that use inexpensive materials and construction methods, and which can be fired many times with only minor repairs. Two thermal-protection methods that could conceivably satisfy this goal were investigated herein.

The first method involved an inexpensive insulating coating, which is cooled by boiling a small flow of water coolant in tubes placed behind the coating. For the high heat fluxes that occur in the nozzle, an initially thick coating will melt rapidly to its steady-state thickness. In the throat region, the heat flux is high, surpassing reasonable boiling burnout heat flux limits such that the tube wall temperature exceeds the limits of inexpensive tube materials. In order to avoid burnout in this region, coolant exit qualities must be low; therefore, the coolant flow rate must be correspondingly high. Circumferential or short axial coolant tubes would allow different coolant flow rates at each region in the nozzle so that the burnout and overheating problems could be reduced. In any event, this thermal-protection method is at best marginal.

The second method involved lining the nozzle walls with a porous wall through which the coolant flows. The coolant vaporizes within the pores or at the hot-gas porous-wall surface. The burnout and overheating problems of the first method can be readily avoided by this method provided that there is sufficient coolant flow to maintain boiling at the hot surface of the porous wall. With a water coolant, the maximum porous wall temperature would be about 400° F (480 K) so that a great variety of porous-wall materials is possible. A porous ablative material would give additional insurance.

The melting-coating boiling-in-tubes method would require about the same amount of water coolant as the porous wall under conditions where there is no burnout or overheating. This result occurs because of the high water flow resulting from the low exit qualities that are necessary to prevent burnout in the tubes. This consequence tends to balance the fact that the heat flux is lower in the tube case compared with the porous-wall case (because the melting temperature of the coating is much higher than the saturation temperature of the water that vaporizes at the porous-wall surface). Neither a boiling liquid metal nor a boiling subcritical cryogen offer any real improvement over the water coolant for either thermal-protection method considered.

Lewis Research Center,
National Aeronautics and Space Administration,
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120-27.

APPENDIX - SYMBOLS

A	surface area of nozzle, ft^2; m^2
b	thermal resistance of wall and coolant, $(ft^2)(hr)(^oF)/Btu$; $(m^2)(K)/W$
$C_{1v}, C_{2i}, C_{2v}, C_{1l}$	constants
c_l, c_v	specific heat of liquid or vapor, $Btu/(lbm)(^oF)$; $W/(kg)(K)$
D_N	nozzle diameter, ft; m
D_T	coolant tube diameter, ft; m
d_d	depth of coolant passage, ft; m
d_m	thickness of metal or porous wall, ft; m
G_l, G_v	coolant mass flow per unit nozzle area of liquid or vapor, $lbm/(ft^2)(hr)$; $kg/(sec)(m^2)$
G_{ls}	minimum coolant flow G_l to have boiling at hot surface, $lbm/(ft^2)(hr)$; $kg/(sec)(m^2)$
G_t	coolant mass velocity (flow per unit tube cross-sectional area), $lbm/(ft^2)(hr)$; $kg/(sec)(m^2)$
h_c	coolant-side heat-transfer coefficient, $Btu/(ft^2)(hr)(^oF)$; $W/(m^2)(K)$
h_{co}	coolant distributor heat-transfer coefficient for no suction
h_G	gas-side heat-transfer coefficient (defined by eq. (1)), $Btu/(ft^2)(hr)(^oF)$; $W/(m^2)(K)$
h'_G	gas-side heat-transfer coefficient, bare wall, $Btu/(ft^2)(hr)(^oF)$; $W/(m^2)(K)$
k_l, k_v	thermal conductivity of liquid or vapor, $Btu/(ft)(hr)(^oF)$; $W/(m)(K)$
k_m	thermal conductivity of metal wall, $Btu/(ft)(hr)(^oF)$; $W/(m)(K)$
k_s	thermal conductivity of coating, $Btu/(ft)(hr)(^oF)$; $W/(m)(K)$
k'_l, k'_v	effective thermal conductivities defined by eq. (12), $Btu/(ft)(hr)(^oF)$; $W/(m)(K)$
L_{LV}	heat of vaporization of coating or coolant, Btu/lbm; kJ/kg
L_{SL}	heat of fusion for coating, Btu/lbm; kJ/kg
P_c	pressure in coolant tube, psia; kN/m^2 abs

p_N	pressure in nozzle, psia; kN/m^2 abs
p	porosity of porous wall (i.e., fraction of total cross-sectional area that is porous)
Q/A	heat flux, $Btu/(ft^2)(hr)$; W/m^2
$(Q/A)_c$	heat flux to coolant, $Btu/(ft^2)(hr)$; W/m^2
$(Q/A)_G$	heat flux from hot gas, $Btu/(ft^2)(hr)$; W/m^2
$(Q/A)_{max}$	boiling burnout heat flux, $Btu/(ft^2)(hr)$; W/m^2
T_c	coolant temperature, ^oF; K
T_G	hot-gas temperature, ^oF; K
T_{LV}	boiling point of coolant, ^oF; K
T_l, T_v	porous-wall - fluid temperature in liquid region or in vapor region, ^oF; K
T_{SL}	melting point of coating, ^oF; K
T_{wc}	coolant-side wall temperature, ^oF; K
T_{wg}	gas-side wall temperature, ^oF; K
$T_{w,max}$	maximum allowable wall temperature, ^oF; K
t	time, sec
w_T	coolant mass flow, lbm/hr; kg/sec
x	distance along centerline of nozzle, ft; m
x'	distance along wall of nozzle, ft; m
Y	ratio of tube diameter to distance required for each half twist of twisted tape
y	distance normal to nozzle wall measured inward either from metal wall or cool side of porous wall, ft; m
Δ	location $y = \Delta$ of two-phase transition, ft; m
Δ_c	thickness of coating, ft; m
Δ_o	initial uniform coating thickness, ft; m
Δ_Q	minimum coating thickness to prevent burnout, ft; m
Δ_{ss}	steady-state coating thickness, ft; m
Δ_T	minimum coating thickness to prevent metal wall overheating, ft; m
Δ_{2P}	two-phase region "thickness," ft; m

χ_{ex}	exit quality of coolant in tubes
χ	fraction of liquid vaporized within or on porous wall
ξ_i, ξ_l, ξ_v	$\xi_i = G_i c_i / k_i'$, where i is liquid l, or vapor v
ρ_s	density of coating material, lbm/ft^3; kg/m^3
$f\langle x' \rangle$	f as function of x'
$f\langle x'; t)$	f as function of x' and t

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