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COMPOSITE WALL CONCEPT FOR HIGH TEMPERATURE TURBINE SHROUDS—HEAT TRANSFER ANALYSIS

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ABSTRACT

A heat transfer analysis was made of a composite wall shroud consisting of a ceramic thermal barrier layer bonded to a porous metal layer which, in turn, is bonded to a metal base. The porous metal layer serves to mitigate the strain differences between the ceramic and the metal base. Various combinations of ceramic and porous metal layer thicknesses and of porous metal densities and thermal conductivities were investigated to determine the layer thicknesses required to maintain a limiting temperature in the porous metal layer. Analysis showed that the composite wall offered significant air cooling flow reductions compared to an all-impingement air-cooled all-metal shroud.

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THE WALL SHROUDS above high pressure turbine blades of current aircraft gas turbine engines are metallic structures and need to be cooled to reduce thermal distortions, cracking, and oxidation. This cooling is obtained by air bled from the engine compressor. Since the bleeding of the air from the compressor reduces engine cycle efficiency, reduction of the cooling air requirement is desired.

The use of a layer of ceramic on the hot gas-side of the shroud structure, as studied in references (1 to 3)* can significantly reduce the coolant flow requirement and/or reduce the shroud metal temperatures. Wear measurement of conventional shrouds of large, high bypass turbofan engines indicate that local metal removal caused by turbine blade rub is generally in the range of 0.76 mm (0.030 in.) deep; therefore, in a composite shroud, the ceramic layer thickness must be at least 0.76 mm (0.030 in.) to preclude exposure of the metal support structure by blade rubs.

Several approaches have been investigated in regard to adherence of thermally sprayed ceramics to the metal support structure. The simplest approach is to thermally spray ceramics directly on the solid metal support structure. However, data in reference (3) indicate that the adherence of ceramics is poor when thick ceramic layers in the range of 22 mm (0.08 in.) are sprayed directly on the metal substrate. Two possible reasons for poor adherence are (1) large thermal stresses through the thick ceramic layers under transient operation and (2) high stress due to abrupt change in material thermal expansion at the ceramic/metal interface. These effects are mitigated and the adherence of a thick ceramic layer is enhanced by employing graded ceramic/metal layers (4) in which the thermal expansion coefficient is tailored by changing the percentage of metal in the intermediate layers. However, reference (4) reports that excessive stresses exist in the ceramic top layer and a means to build in beneficial residual stresses is needed. Another approach, the one of concern in this paper, is to use a compliant, generally low density and low modulus, interlayer between the ceramic and metal base. In this composite wall concept, the compliant layer acts to mitigate the strain difference between the

*Numbers in parentheses designate References at end of paper.

ceramic layer and the metal base. Possible interlayer materials are various types of porous metals such as felt, woven and foam metals. Thermal shock studies (5) revealed that this compliant concept is more effective in reducing thermal stresses than the graded layer concept.

The objective of the study reported herein was to analytically examine the variables that affect the design of a composite wall shroud consisting of a metal base, an interlayer of porous metal and an outer layer of yttria stabilized zirconia.

Based on considerations of low oxidation and long life, the maximum allowable temperatures of the porous metal of the composite shroud, and the all-metal shroud were set at 1144 K (1600° F) for current materials and 1200 K (1700° F) for advanced materials. Both the composite and all-metal shrouds were assumed to only be impingement air cooled. The gas and coolant conditions assumed were those of an advanced gas turbine.

The overall thickness of the composite wall shroud was kept the same as on typical all-metal shroud (5.59 mm (0.220 in.)). The solid metal wall thickness of the composite shroud was held constant at 2.03 mm (0.080 in.) to maintain structural integrity. The variables investigated for the composite shroud were (1) ceramic thicknesses from 0.5 to 3.06 mm (0.02 to 0.12 in.), (2) corresponding porous-metal thicknesses from 3.06 to 0.5 mm (0.12 to 0.02 in.), (3) porous-metal density from 10 to 50 percent of a fully dense material, and (4) two porous metals with thermal conductivities that differed by a factor as much as 6, and (5) ratios of cooling airflow to turbine gas flow from near zero to 0.03.

The data are presented as curves of temperatures through the composite shroud for various coolant- to gas-flow ratios for selected thicknesses of layers and for various porous-metal densities. Comparisons are made between the various composite shroud combinations and the all- metal shroud.

SYMBOLS

A	parameter, see eq. (4)
A_c	impingement cooling airflow area
B	blade chord
c_1, c_2	constants
D_c	diameter of impingement cooling holes

d	porous metal relative density
h	heat transfer coefficient
K	thermal conductivity
m	exponent, see eqs. (3) and (6)
Nu	Nusselt number
Pr	Prandtl number
R	thermal conductance, K/T
R_c	coolant- to gas-flow ratio
Re	Reynolds number
T	temperature
V	velocity
W	flow rate
X_c	distance between impingement holes
Z	impingement jet-to-wall distance
δ	tip-clearance to blade-span ratio
μ	viscosity
τ	thickness

Subscripts:

b	ceramic thermal barrier
c	cooling air
g	gas
i	inside (toward coolant side), see eq. (1)
o	outside (toward gas side)
p	porous metal
p1	porous-metal type 1
p2	porous-metal type 2
w	metal support structure
x	flow rate, cross flow air

ANALYSIS AND CONDITIONS

HEAT BALANCE - An element of the composite turbine shroud, shown in Fig. 1, was analyzed for the assumed engine conditions and geometry shown in Table 1. Radiation was neglected and heat flow was assumed one-dimensional through the ceramic thermal barrier, the porous-metal interlayer, and the metal wall. The effective gas and cooling air temperature T_g and T_c were assumed equal to their respective total temperatures. For these assumptions the heat flow equations are

$$h_g(T_g - T_{bo}) = R_b(T_{bo} - T_{bi}) = R_p(T_{po} - T_{pi})$$
$$= R_w(T_{wo} - T_{wi}) = h_c(T_{wi} - T_c) \qquad (1)$$

This set of equations was used in a computer program to calculate the desired surface and interface temperatures. Equations (1) required gas- to-surface and surface-to-coolant heat transfer coefficients. The gas-to-surface heat transfer coefficient was obtained from the equation of heat transfer reported in reference (6). This equation, developed from experimental heat transfer studies on a turbine shroud, is

$$Nu_g = h_g C/K_g = 0.052\ Re_g^{0.8}(1 - 2\delta^{0.8}) \quad (2)$$

where δ is the tip-clearance to blade-span ratio. The gas-side Reynolds number Re_g was evaluated at an assumed average gas Mach number of 0.8, the characteristic dimension of the blade chord, and the gas properties near the shroud surface. The transport gas properties were obtained from the data of reference (7).

Impingement cooling of the metal wall was assumed. For this cooling method, the coolant-side heat transfer coefficient was obtained from the correlation from reference (8), which in the notation of this report is

$$Nu_c = h_c D_c/K_c = AB\ Re_c^m\ Pr_c^{0.33}(Z/D_c)^{0.091} \quad (3)$$

where, for an assumed $3000 < Re_c < 30\ 000$,

$$A = \exp\left[0.026(X_c/D_c)^2 - 0.8259(X_c/D_c) - 0.3985\right] \quad (4)$$

$$B = 1/\left[1 + 0.4696(W_x Z/W_c D_c)^{0.965}\right] \quad (5)$$

$$m = -0.00252(X_c/D_c)^2 + 0.06849(X_c/D_c) + 0.50699 \quad (6)$$

The following fixed geometry values were assumed for the previous equations: ratio of jet-to-wall distance to hole diameter $Z/D_c = 15$, a ratio of hole spacing-to-diameter $(X_c/D_c) = 10$, a ratio of crossflow to jet-flow $(W_x/W_c) = 1.0$, and a hole diameter $D_c = 0.51$ mm (0.02 in.). Substituting these values into equations (4) to (6) and then into equation (3) along with appropriate coolant properties values from reference (7) results in

$$h_c = c_1 Re_c^{0.94} \quad (7)$$

where

$$c_1 = 3.73\text{x}10^{-2}\ W/m^2/K$$

The coolant Reynolds number was determined from the following equation:

$$Re_c = W_c D_c / A_c \mu_c \qquad (8)$$

where for the given values of the coolant temperature, assumed geometry of the impingement holes, diameter of the turbine shroud, and shroud width gives

$$Re_c = c_2 R_c W_g \qquad (9)$$

where

$$c_2 = 2.0\times10^4 \text{ sec/kg}$$

This equation is then substituted into equation (7), which for assumed turbine gas flow rate and values of coolant- to gas-flow ratio R_c, and provides the needed values of coolant-side heat transfer coefficients.

COMPOSITE SHROUD CONFIGURATIONS AND MATERIALS - Fig. 1 depicts the composite shroud configuration consisting of a metal support, an interlayer of porous metal, and a sprayed ceramic layer which is exposed to the turbine gas flow. The overall radial thickness of the shroud was 5.59 mm (0.220 in.) and was based on the consideration of replacing a specified all-metal shroud with ceramic composite shrouds. Analysis of the relative structural strengths of the shrouds was not made since it was considered outside the scope of the present paper. The radial thickness of the metal support base was selected to be 2.03 mm (0.080 in.) for all configurations. Therefore the combined radial thickness of the ceramic and porous-metal layers was 3.56 mm (0.140 in.) for all configurations. The thicknesses of the ceramic and porous-metal layers were varied and the temperatures and cooling flow requirements were determined. The porous-metal thickness was varied from 0.5 to 3.06 mm (0.02 to 0.12 in.) which corresponds to a ceramic thickness variation of 3.06 to 0.5 mm (0.12 to 0.02 in.) (see Table 2). The density of the porous-metal interlayer was also an independent variable in the study and was varied from 0.1 to 0.5 of solid metal.

The values of thermal conductivity needed for the conductance terms in equation (1) for the metal wall (MAR-M-509) and the ceramics (yttria stabilized zirconia) were obtained from references (9) and (10), respectively. Two different porous metals composed of the same materials (FeNiCrAlY) were considered; they differed in structure and thermal conductivity. Porous-metal 1 was a felt-type material and porous-metal 2 was an open-cell foam-type material. The thermal conductivity for porous-metal type 1 was obtained from the following equation which was fitted to the data of reference (11) for various relative densities d_p and average temperatures T_p:

$$K_{p1} = 4.5\ d_p^{1.38}\ e^{(5.4 \times 10^{-4} T_p)}\ W/m/K \qquad (10)$$

since there was no empirical data for the thermal conductivity for porous metal 2, it was assumed that the thermal conductivity was a linear function of material density. The published thermal conductivity data for 100 percent dense FeNiCrAlY alloy (12) was adjusted for density by the following equation:

$$K_{p2} = d_p(6.38 + 0.018\ T_p)\ W/m/K \qquad (11)$$

RESULTS AND DISCUSSION

The results of the heat transfer analysis of composite shroud designs and comparisons with an all-metal shroud (where both shrouds were impingement air-cooled) are presented in Figs. 2 to 4. The composite shroud is illustrated in Fig. 1, and the assumed engine conditions and geometry are given in Table 1.

COMPARISON OF COMPOSITE AND ALL-METAL SHROUDS - The calculated results for the all-metal shroud for coolant- to gas flow-ratios as high as 0.06 are shown in Fig. 2(a). The figure shows little reduction in metal temperature with increasing coolant flow ratio. The calculations showed that the specified maximum metal temperature of 1144 K (1600° F) could not be obtained even with a coolant- to gas-flow ratio as high as 0.10. Therefore, for the conditions of the analysis, impingement cooling alone is not sufficient or practical for the all-metal shroud. The impingement cooling would need to be supplemented by film-cooling, or the shroud would require low thermal conductivity rub material on the gas side to reduce the heat flux, metal temperatures, and cooling flow ratio.

By way of comparison, Fig. 2(b) shows the temperatures in the composite shroud. The composite shroud in Fig. 2(b) consisted of 1.78 mm thickness of ceramic and a 1.78 mm thickness of porous-metal 2 with a 0.2 density. The metal temperatures for the composite shroud are significantly lower than for the all-metal shroud at the same or lower coolant flow ratios. Data in Fig. 2(b) for the composite shroud indicate that the maximum allowable metal temperature is reached at a coolant- to gas-flow ratio of 0.02. The data in Fig. 2(b) show the very large insulative effect of the ceramic. For example, at a coolant- to gas-flow ratio of 0.02 the temperature drop through the ceramic layer is 426 K (766° F). Another observation is that the combined insulative effect of the ceramic layer and porous-metal layer causes the gas-side ceramic surface temperature to be very near the gas temperature.

EFFECTS OF POROUS-METAL DENSITY AND THERMAL CONDUCTIVITY - Fig. 3 shows the temperatures of composite shrouds (ceramic and porous-metal layers each 1.78 mm thick) as a function of cooling flow ratio for porous-metal types 1 and 2 (with different densities and with inherent differences in thermal conductivities). For the two densities and for the range of porous-metal temperatures in Fig. 3, it can be determined from equations (10) and (11) that porous-metal 2 has a thermal conductivity four to six times higher than porous-metal 1. For a density of 0.2 and a temperature of 1144 K (1600° F), porous-metal 2 has a thermal conductivity 5.9 higher than porous-metal 1.

Fig. 3 shows the effect of increasing the densities of the porous materials from 0.2 to 0.5. As density is increased, the porous-metal gas-side temperature is reduced; this occurs because thermal conductivity increases with density, thus resulting in increased heat transfer from the porous metal. The beneficial effect of the greater thermal conductivity of porous-metal 2 compared to porous-metal 1 is apparent when comparing Fig. 3. Fig. 3 also shows that the coolant- to gas-flow ratio required to obtain the 1144 K (1600° F) maximum porous-metal temperature is 0.02 for porous-metal 2 with a 0.2 density. However, porous-metal 1 with a 0.2 density could not be cooled to this temperature even with very high coolant flows. If the density of porous-metal 1 was increased to 0.5, the maximum allowable porous-metal temperature could be obtained at a coolant flow ratio of 0.024.

In the design of a composite shroud, the configuration parameters include the layer thicknesses as well as the porous metal density and thermal conductivity. The porous-metal density, layer thickness, and structure (felt, foam, or woven), in addition to their effect on heat flow and shroud temperatures, are factors which will affect the stress, adherence, and durability of the composite wall. Although stress analysis is not addressed in this paper, a general observation from thermal stress and cyclic life considerations is that lower porous-metal densities with their associated lower modulus of elasticity are desired. From a consideration of handling and structural integrity, a lower limit on the density of the porous wall is assumed to be about 0.2.

EFFECTS OF LAYER THICKNESS - Fig. 4 presents the results of the analysis in a form which shows how porous-metal temperature varies with changes in thicknesses of the ceramic and porous metal layers for a given coolant flow ratio.

The effect of the different thermal conductivities of the two different porous materials is apparent from Fig. 4. A general observation is that, as the ceramic layer thickness decreases, the porous-metal density must increase to maintain a given temperature. This is due to the fact that a higher thermal conductivity in the porous-metal is required to accommodate the increased heat flux and to maintain the allowable temperature limits on the porous metal at 1144 K (1600° F) for current materials or 1200 K (1700° F) for advanced materials.

Fig. 4 shows that the higher the allowable porous-metal temperature (1200 K (1700° F)) is, the lower the porous-metal densities at a given ceramic/porous-metal thickness ratio can be. Inspection of Fig. 4 also reveals that the porous layer temperature decreases with decreasing porous layer thickness and increasing ceramic layer thickness for a constant porous layer density. As the porous-metal density is increased, the porous-metal temperature is decreased for given ceramic and porous-metal thicknesses. As an example, for a 0.2 density and a 1144 K (1600° F) temperature limit, porous-metal 1 would need to be about 0.91 mm (0.036 in.) thick with a layer of ceramic 2.64 mm (0.104 in.) thick to satisfy the heat load and cooling conditions selected. On the other hand, porous-metal 2 with the same density would be 1.78 mm (0.07 in.) thick with a 1.78 mm (0.07 in.) thick layer of ceramic. The choice, as stated in the previous section, would be influenced by thermal stress considerations which dictate a selection of the lower modulus porous layer, that is, porous- metal 2 with the thicker porous metal layers and the correspondingly thinner ceramic layer.

SUMMARY OF RESULTS

The analysis provided a basis for evaluating the effects of variables on the design of composite turbine shrouds which consisted of the metal case wall, an interlayer of a porous metal, and an outer layer of ceramic (yttria stabilized zirconia). The results were as follows:

1. Significant reductions in the cooling-air to gas-flow ratio are indicated for the composite shrouds compared to an all-metal shroud that was only impingement air cooled. This is based on the same maximum allowable temperature for the all-metal shroud and for the porous metal interlayer of the composite shroud.
2. The good insulating properties of the ceramic significantly reduced the temperatures of the porous metal and support wall, but also caused the gas-side surface temperature of the ceramic to be essentially at the gas temperature.
3. For a given porous metal density and coolant- to gas-flow ratio, decreasing the thickness of the porous-metal and increasing ceramic thickness resulted in lower support wall temperatures.
4. To maintain given allowable interlayer temperatures and coolant- to gas-flow ratios, porous-metal density or thermal conductivity must increase as the ratio of the thickness of the ceramic-to-porous metal decreases.

CONCLUDING REMARKS

In general, thermal cycle life considerations (refs. 5 and 13) would indicate more compliant, lower densities of porous metal for the composite wall designs. Also the porous layer thickness must be large enough to provide the needed strain isolation between the ceramic layer and the metal base. Therefore a high thermal conductivity at a low density in the porous layer is needed. Based on the foregoing and an assumed lower limit on porous-metal density (considering the composite wall handling and structure integrity) a 1.78 mm (0.07 in.) thickness of porous material 2 with a density of 0.2 and a 1.78 mm (0.07 in.) thickness of ceramic appears to be a good composite wall configurations for the assumed conditions.

The insulating property of the ceramic layer causes the ceramic surface temperature to be almost equal to the engine gas temperature. A detrimental result of the high surface temperature would be an increase in radiation to turbine parts. This may require supplemental cooling of these parts. A beneficial effect of the high ceramic temperature may be improved abradability of the ceramic during a blade rub.

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Table 1. - Assumed Engine Conditions and Geometry

Gas total temperature at shroud, K (°F)	1589 (2400)
Gas total pressure, atm	25
Gas flow rate, kg/sec (lb/sec)	113.6 (250)
Turbine tip diameter, cm (in.)	96.5 (38)
Blade chord, cm (in.)	3.05 (1.2)
Blade tip clearance-to-span ratio	0.02
Gas average absolute Mach number	0.8
Cooling air temperature to shroud, K (°F)	756 (900)
Allowable porous-metal temperatures, K (°F):	
Current material	1144 (1600)
Advanced material	1200 (1700)

Table 2. - Composite Wall Layer Thicknesses

Configuration	Ceramic layer thickness, τ_b		Corresponding porous-metal layer thickness, τ_p	
	mm	in.	mm	in.
1	0.50	0.02	3.06	0.12
2	0.76	0.03	2.79	0.11
3	1.27	0.05	2.29	0.09
4	1.78	0.07	1.78	0.07
5	2.29	0.09	1.27	0.05
6	2.79	0.11	0.76	0.03
7	3.06	0.12	0.50	0.02

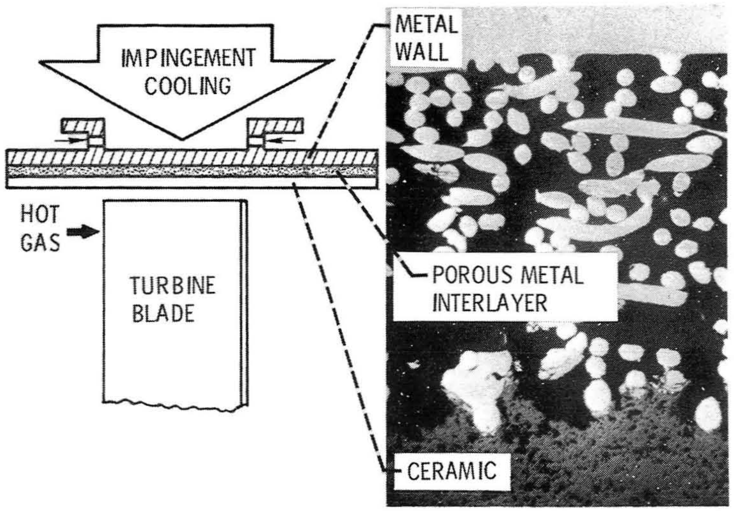


Figure 1. - Composite turbine shroud.

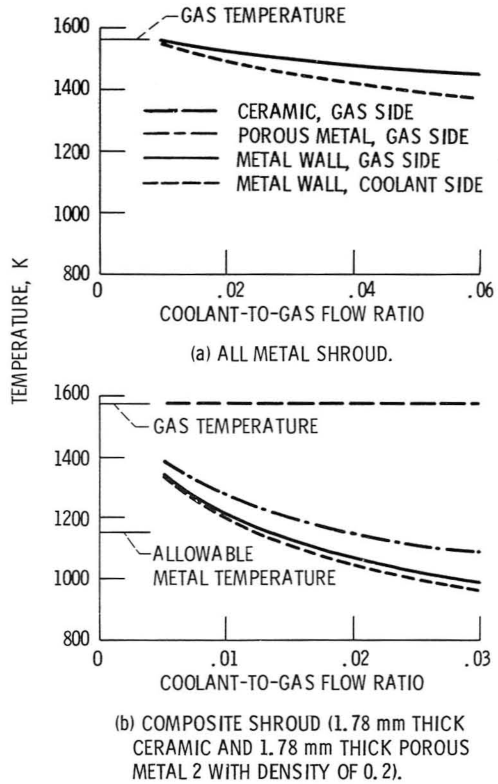
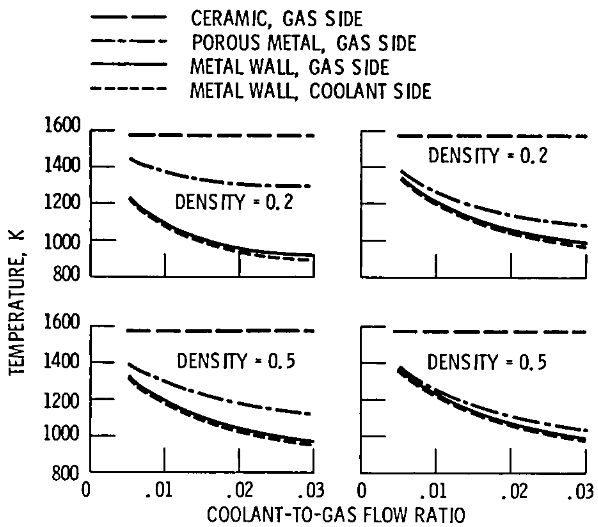


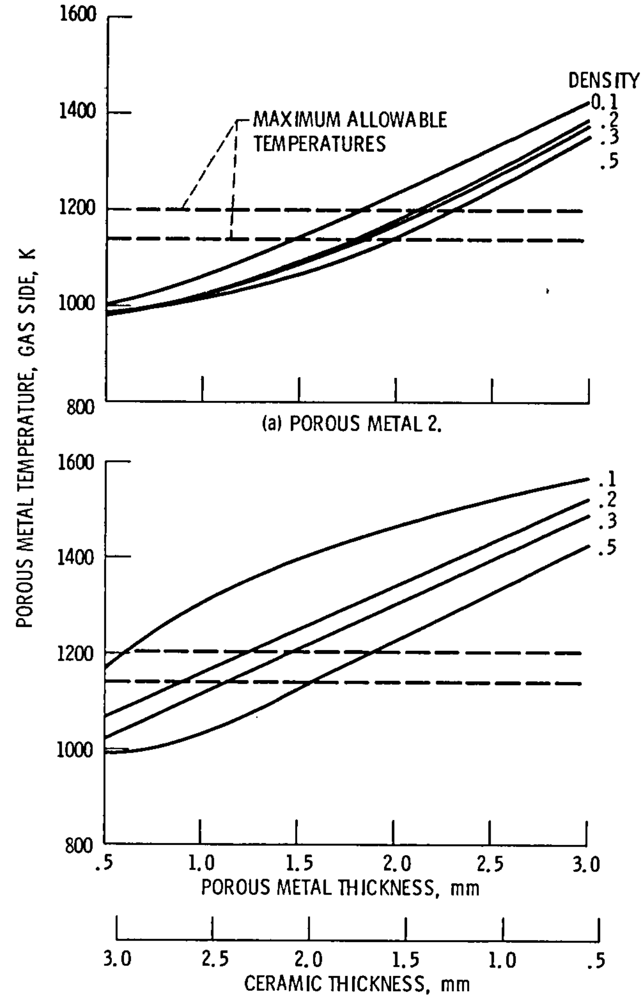
Figure 2. - Comparison of an all metal shroud with a composite shroud.



(a) POROUS METAL 1.

(b) POROUS METAL 2 (THERMAL CONDUCTIVITY 4 TO 6 TIMES GREATER THAN POROUS METAL 1).

Figure 3. - Effects of porous metal density and thermal conductivity.



(a) POROUS METAL 2.

(b) POROUS METAL 1.

Figure 4. - Porous metal gas side temperature as a function of porous metal and ceramic thickness; coolant-to-gas flow ratio, 0.02.

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