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THE FEASIBILITY OF HEAT PIPE TURBINE VANE COOLING

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ABSTRACT

This paper evaluates the basic feasibility and anticipated benefits to using heat pipe technology to cool the turbine vanes of gas turbine engines.

This concept involves fitting out the vane interior as a heat pipe, extending the vane into an adjacent heat sink and then transferring the vane incident heat through the vane to the heat sink. The baseline is an advanced military fighter engine and the bypass air is the chosen heat sink. The results of this study show a 7.2% increase in engine thrust, a 0.2% decrease in specific fuel consumption with engine weight increased by less than 1% by using this technology.

HEAT PIPE TURBINE VANE COOLING CONCEPT

Turbine inlet temperature has a strong effect on the performance of gas turbine engines and is limited by material temperature capabilities in the engine. Current approaches to increase turbine inlet temperature include the development of high temperature materials and cooling the turbine vanes with compressor discharge air. At turbine inlet temperatures needed to meet future Air Force goals, engine performance losses resulting from the diversion of compressor discharge air for cooling will significantly reduce the performance gains that would otherwise result from the elevation in turbine inlet temperature.

Heat Pipe Turbine Vane Cooling (HPTVC) is a technique for cooling turbine vanes that does not require the diversion of compressor discharge air around the combustor. A wick structure is placed on the inside surface of the vane and an appropriate heat transport fluid is introduced that fills the wick's pores. The vapor of the fluid occupies the remaining internal volume of the vane. The vane thus acts as a heat pipe. Then, instead of cooling the vanes by the injection of diverted compressor discharge air, cooling is accomplished by transporting incident heat internally through the vanes to an appropriate heat sink.

A heat pipe transfers thermal energy by circulating a working fluid in its vapor and liquid phases over a desired distance (Ref. 1 and 2) as shown in Figure 1. Heat evaporates the working fluid in the evaporator. This vapor then travels down the vapor core until it condenses in the condenser. The condensed liquid then makes its way back to the evaporator in the wick structure. This results in a nearly isothermal heat transfer process. Surface tension forces in a capillary wick enable the pumping head. The meniscus between the vapor and liquid phases naturally adjusts to allow the exact pumping head required to match the flow losses associated with the applied heat load. This wick performs the function of a "vapor diode," allowing vapor to flow to the condenser but preventing it from flowing back into the wick structure and thus depriming the heat pipe. Heat pipes require no external power, have no moving parts and are thus inherently reliable. Heat pipes have been used extensively for spacecraft thermal management.

Conceptually, the vane looks and behaves exactly like the heat pipe shown in cross section in Figure 1. The heat flow between the heat input and rejection zones of the vane is then as shown schematically in Figure 2.

In response to the need for innovative turbine vane cooling techniques, Wright Laboratory sponsored contract F33615-91-C-2101 to CCS Associates to determine the basic feasibility and anticipated benefits to using heat pipe technology to cool the low pressure turbine vane of an advanced military fighter engine. This paper summarizes those results (Ref. 3).

PROBLEM DEFINITION

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We chose an advanced military fighter turbine engine as the baseline for this analysis. The Allison Gas Turbine Division of General Motors Corporation supplied the design information. Table 1 shows the dimensions of the vane.

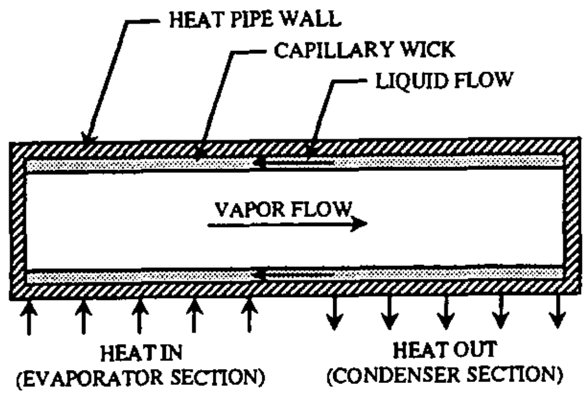

Figure 1. Heat Pipe Cross-Section

We chose two engine design points to perform the analysis. Design Point 1 corresponds to Mach 1.6 flight at an altitude of 12,200 m (40,000 ft). Design Point 2 corresponds to Mach 1.2 flight at an altitude of 150 m (500 ft). The specific design parameters of the heat pipe cooled turbine vane determine the critical design point for the most severe heat transfer, heat transport and structural conditions. Design Point 1 is the critical condition in some cases, while Design Point 2 is the critical condition in others. Table 2 summarizes the design criteria used in the concept analysis. Besides the combustion gas duct, it is necessary to define the heat sink air duct in which the heat rejection portion of the vane is located, as well as any intermediate path through which the vane must pass. As shown in Figure 3, the vane passes through an intermediate cooling duct before entering the bypass air duct that serves as the heat sink. The area of the extended vane in the bypass air duct is not sufficient to provide the needed heat rejection area. Two ways to augment this area are to either bend the vane at an angle β to the radial direction or add chordwise fins to the vane as shown in Figure 3.

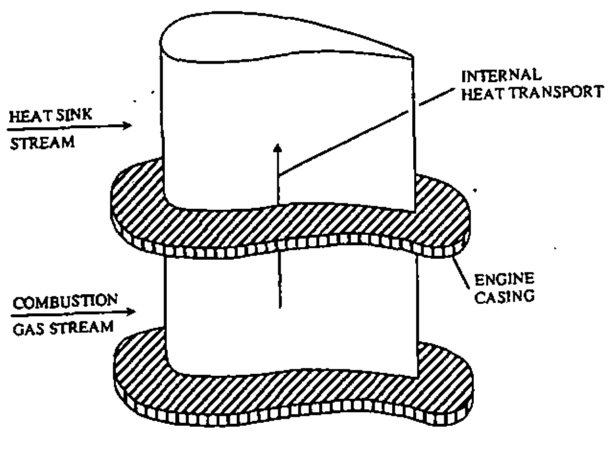

Figure 2. Heat flow Path in Heat Pipe Cooled Turbine Vane

Table 1. Representative Vane Dimensions	
Span, cm. (in.)	8.25 (3.25)
Chord, cm. (in.)	7.87 (3.10)
Leading edge radius, cm. (in.)	0.41 (0.16)
Area of cross section, cm^2. (in^2.)	6.20 (0.961)
Maximum thickness, cm. (in.)	1.21 (0.475)
Surface length around cross section, cm. (in.)	16.5 (6.498)
Number of vanes	28

HEAT PIPE WICK TYPES

The fundamental heat pipe design relation is expressed by

$$p_c = p_b + \Delta p_{lf} + \Delta p_{vf}$$

where p_c available "capillary pressure," N/m^2

p_b body forces on the fluid, N/m^2

Δp_{lf} liquid pressure drop, N/m^2

Δp_{vf} vapor pressure drop, N/m^2

The available "capillary pressure" is assumed to be equal to the pressure as determined by the Laplace relation

$$p_c = (2\sigma/r_c) * \cos\phi$$

where σ surface tension, N/m

ϕ attachment angle, degree

r_c effective pore radius, m

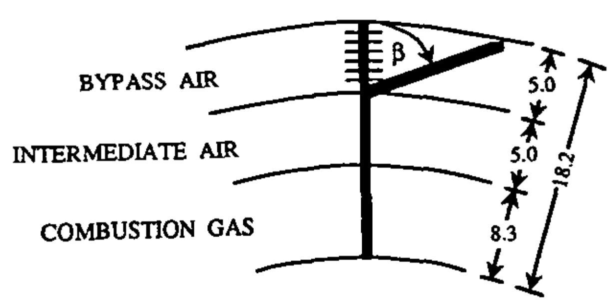

Figure 3. Duct Geometry (cm.)

Table 2. Design Criteria for Heat-Pipe-Cooled Turbine Vane Study		
Parameter	Design Point 1 Mach 1.6; 12.200 m	Design Point 2 Mach 1.2; 150 m
Combustion Gas		
Mean heat transfer coefficient, w/m^2-K (Btu/hr-ft^2 - °F)	520 (300)	1040 (600)
Peak heat transfer coefficient, w/m^2-K (Btu/hr-ft^2 - °F)	1040 (600)	1690 (975)
Mean heat transfer coefficient at trailing edge, w/m^2-K (Btu/hr-ft^2 - °F)	430 (250)	970 (560)
Peak gas temperature, K (°F)	1980 (3100)	2120 (3356)
Mean gas pressure, kPa (psia)	19,300 (2800)	
Gas pressure, kPa (psia)	585 (85)	1380 (200)
Bypass Air		
Mass velocity, kg/m^2-s (lb/ft^2-s)	156 (32)	440 (90)
Inlet air temperature, K (°F)	530 (500)	560 (550)
Air pressure, kPa (psia)	275 (40)	690 (100)
Intermediate Air*		
Inlet air temperature, K (°F)		700 (800)
Mass velocity, kg/m^2-s (lb/ft^2-s)		25 (5)
Gas pressure, kPa (psia)		1720 (250)
Acceleration on Vanes		
Radial acceleration, g	10	10

* Flows through duct between combustion gas and bypass air ducts.

The homogeneous wick structure shown in Figure 1 cannot efficiently perform the dual functions of liquid flow channel and vapor diode that are required of any wick. Smaller pore radii lead to a better vapor diode function but also higher liquid pressure drop. If these two functions are made separate functions of the wick structure, heat pipe heat transport capability is greatly enhanced.

In this study, we evaluated the three types of wicks illustrated in Figure 4. In each case, the liquid flow channel and the capillary differential pressure generation are completely separate and independent of one another.

With the two-layer screen wick, liquid flow proceeds via a relatively coarse-pored screen layer next to the wall, and the pressure differential arises in a fine-pored screen layer next to the vapor space. With the screened groove wick, the liquid flow channel is a series of grooves in the heat pipe wall, extending in the direction of heat flow. With the artery wick, the liquid flow channel is in the form of small diameter tubes formed from the same fine-pored capillary wick layer which now lines the heat pipe wall. The artery tubes also extend along the direction of heat flow.

In each succeeding case, the pressure drop of the returning liquid flow is reduced and heat pipe performance increases, significantly. The performance enhancement of artery designs versus the homogenous wick design shown in Figure 1 can be as much as one order of magnitude when measured in kW-m (i.e., heat transported times distance transported). However, as the complexity of the wick increases, so does technical risk.

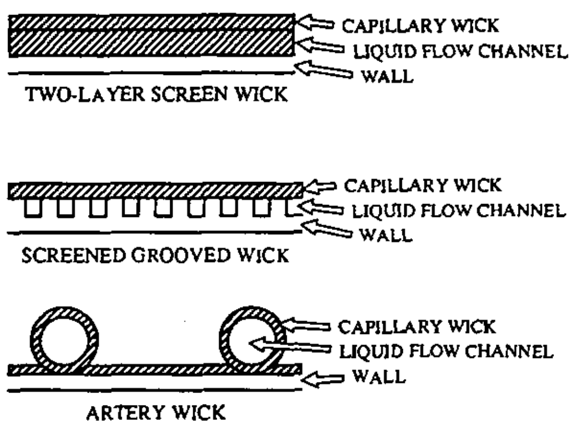

Figure 4. Heat Pipe Wick Types

VANE CONCEPT

We chose a CMSX-4 vane for this study. CMSX-4 is a single crystal alloy with excellent high temperature strength and resistance to oxidation. However, its upper temperature limit of 1310 K (1900° F) and its somewhat low thermal conductivity represent significant disadvantages for the HPTVC application. Higher HPTVC payoff would have been obtained using a vane material of TZM molybdenum or carbon-carbon. There are, however, significant development programs required before these materials can be used for turbine vanes. CMSX-4 is a relatively state-of-the-art material.

CONCEPTUAL DESIGN

For this CMSX-4 vane, we evaluated the heat rejectioa area, internal heat transport through the vane, trailing edge temperature and leading edge stress and stability for Design Points 1 and 2. Generally, the design point with the most stringent design requirements was the critical design condition.

Heat Rejection - The two chief concerns during the heat rejection analysis were to limit the CMSX-4 material temperature to 1250 K and to minimize the weight of the heat rejection scheme. These two parameters are also interrelated since an increase in vane side wall thickness results in hotter vane wall temperatures as well as a larger heat rejection requirement. The desire for thin vane wall thickness affected the design in such areas as vane structural integrity, heat pipe wick selection and thermal barrier coating (TBC) layer thickness.

The critical design condition for the determination of heat rejection characteristics is Design Point 1, because of the poorer heat transfer coefficients in the bypass duct for this Mach 1.6, 12,200 meter condition.

We used fins on the vane surface in the bypass duct since the required heat rejection area ranged from 3 to 6 times the vane surface area in the combustion gas duct. Bent vanes could only be considered if the heat rejection area was less than 1.8 times the hot side vane surface area, i.e., the vane angle would be less than 90°.

We chose nickel as the fin material for most effective heat rejection. Although nickel is not usually considered for operation in the temperature range of interest for the fins (1200-1220 K), it should have adequate oxidation resistance and its strength should be adequate to withstand the modest external pressure (100 psi maximum) in the bypass duct. It also should be possible to braze nickel fins to the CMSX-4 vane.

Heat Pipe Design Considerations - Our desire for a minimum temperature drop through the vane wall and wick resulted in our selection of an artery type wick heat pipe. This design, with its thin distribution wick lining, best satisfies this requirement while also meeting the need for low liquid frictional pressure drop.

Primarily because of the 10 g radial acceleration requirement, the effective wick pore radius of the heat pipe must be less than 14 microns. The largest readily available mesh results in an effective pore of 60 microns. Kemme (Ref. 4) has shown that by drawing down multiple layers of the 400 mesh screen the effective pore radius can be reduced to 10 microns. For nominal design purposes, we selected an artery radius of 0.5 mm (.020 in).

Trailing Edge - We calculated peak vane temperatures at the tip of the trailing edge in the combustion gas duct on the assumption that the trailing edge can be represented as a rectangular reverse fin whose thickness is equal to the mean vane thickness at the trailing edge. With a reverse fin, heat flows from the hot surrounding gas into the cooler fin instead of from the hotter fin into the cooler surrounding gas. We assumed one-dimensional heat flow along the trailing edge. For short fins where two-dimensional heat flow is significant, this assumption underestimates the peak trailing edge temperature.

In Figure 5, we plotted the peak trailing edge temperature as a function of trailing edge length at Design Point 1 for CMSX-4 and nickel trailing edges that were covered by a 0.15 mm (0.006 in) layer of TBC.

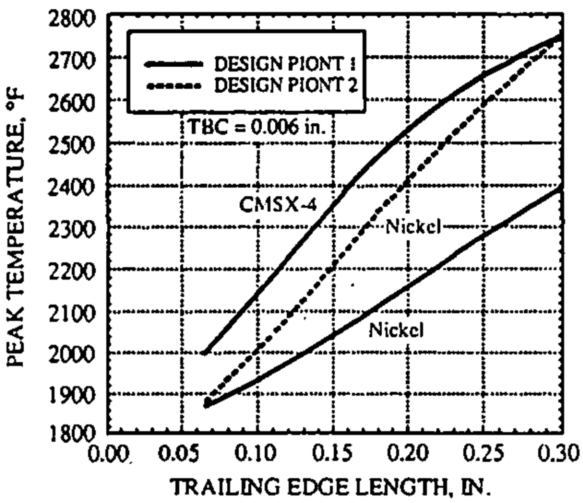

Figure 5. Trailing Edge Temperatures for CMSX-4 Vane

The temperature increases rapidly with trailing edge length, and is substantially higher for CMSX-4. Even at a trailing edge length of 1.65 mm (0.065 in)(equal to the mean thickness), the temperature or the CMSX-4 trailing edge is 1362 K (1993° F), 70 K (126° F) greater than that of the nickel trailing edge.

It appears that a short trailing edge (i.e., one for which the active heat pipe region extends close to the end of the trailing edge) is necessary to avoid excessive temperatures. Although trailing edge temperatures are lower with nickel, CMSX-4 is preferable from a fabrication standpoint if the higher temperatures can be tolerated.

Structural Aspects - We determined the stresses and structural stability at the vane leading edge by assuming the leading edge to be equivalent to a cylinder subjected to uniform pressure and temperature differentials across the cylinder walls. We calculated the thermal and hoop stresses in the leading edge using standard formulas. The maximum wall stress is compressive, and occurs at the outer wall surface. It is equal to the sum of the thermal and hoop stresses.

Thermal stress dominates, with maximum thermal and hoop stresses occurring at the Design Point 2 operating condition. For both operating conditions, the maximum stress is considerably below the stress for 1% creep in 10,000 hours. Thus the vane leading edge is structurally adequate.

The remainder of the vane is a pressure vessel of nonoptimum shape, also subjected to pressure and temperature differentials across the vane walls. The analysis shows that internal support in the form of spanwise ribs will be required to prevent buckling of the main vane body. For a vane wall thickness of 0.914 mm (0.036 in), at least two ribs with a thickness of 0.762 mm (0.030 in) are required.

DETAIL DESIGN

We examined the CMSX-4 vane in greater detail, introduced modifications needed to refine and enhance the design and calculated the principal characteristics of the modified design.

We introduced the following design modifications:

Addition of TBC - The vane wall thickness at leading edge and in the bypass air duct is 0.041 mm (0.016 in). This is too small for a practical vane design. By reducing the vane heat load, thicker walls can be specified without significantly affecting the size and weight of the heat rejection fins. We accomplished this by increasing the thickness of the TBC layer on the vane surface in the combustion gas duct from 0.015 to 0.030 cm (0.006 to 0.012 in).

Addition of Ribs - The external gas pressure on the vane walls is substantially greater than the internal vapor pressure. We added three internal spanwise ribs to stabilize the vane walls against buckling from the net external pressure on the vane. Gaps are cut in the ribs to permit chordwise flow of the sodium liquid and vapor within the vane interior.

Addition of Wick Layer - The distribution wick on the inner surface of the vane is fabricated from the capillary pumping layer. Drawing down multiple layers of 400 mesh screen forms the capillary layer reducing its effective pore radius to 10 microns. The small pore size will greatly increase the frictional resistance of the liquid sodium that flows through the distribution wick, to an extent that is not readily determinable analytically. To insure that the liquid sodium pressure drop will not be excessive, one layer of unaltered 400 mesh screen is inserted between the capillary pumping layer and the vane wall.

Vane Dimensions - Geometric considerations preclude extending the active heat pipe region inside the vane to the tip of the trailing edge. Because of the need to avoid excessively high temperatures at the vane trailing edge in the combustion gas duct, we restricted the uncooled length of the trailing edge to 0.015 cm (0.060 in). This results in a shortening of the original vane chord length by 0.25 cm (0.1 in). Also, in the conceptual design, we added the TBC layer and its associated bonding coat to the original vane profile, extending the outer vane surface by 0.043 cm (0.017 in) beyond the original vane profile. The detail design adopts these changes, hut relatively minor design changes could retain the original vane profile and dimensions.

Trailing Edge - Even with the shortened trailing edge, temperatures at the top of the trailing edge are higher than desired but still tolerable with CMSX-4 as the trailing edge material. A nickel trailing edge will run much cooler, but additional fabrication development will be required to successfully braze a nickel trailing edge to the CMSX-4 vane body. We have retained the nickel trailing edge as an option in the event that CMSX-4 trailing edge temperatures are excessively high.

VANE HEAT TRANSPORT CAPABILITY

At Design Point - Vane design heat transfer calculations established that vane temperatures are higher and heat transport limits closer to the actual heat transport rate at Design Point 1 (Mach 1.6 at an altitude of 12,200 m) than at Design Point 2 (Mach 1.2 at an altitude of 150 m). The following are significant results. Heat incident on the vanes in the combustion gas duct can be dissipated by the fins in the bypass air duct without exceeding vane temperature limits or heat pipe heat transport limits. Temperatures at the tip of the trailing edge are higher than the leading edge, but still tolerable. The heat pipe heat transport mechanism remains functional at radial accelerations (i.e., during maneuvers) as high as 10 g's.

During Startup - Transient studies performed on the conceptual vane design show that, during startup, vane temperature limits are not exceeded, and heat transport limits will remain equal tn or greater than the actual heat transport rate. The studies also show that the liquid sodium in the heat pipe wick may be under tensile stress (negative pressure) of 0.70 to 0.90 kPa which increases to 8.9 kPa for a few seconds as the engine power level rises from idle to the full power condition. Operation of a sodium heat pipe under moderate tension has been demonstrated (Ref 4). However, the effect of liquid tension on heat pipe operation in an aircraft environment has not been determined. We believe that the actual vapor temperature, and therefore the vapor pressure, is higher than predicted by the transient analysis. Therefore, the predicted liquid tension may in fact not really be present.

The somewhat large increase in liquid tension observed results from the assumption that the maximum radial acceleration of 10 g's instantly develops when the full power operating condition is reached. At this point the vapor temperature and pressure are still well below full power values. If high-g maneuvering could be avoided for only 10 sec after full power has been reached, the liquid tension condition will be avoided. If tension actually develops in the liquid sodium, there is a question whether the wick arteries would remain filled with liquid. If depriming of the arteries should be a problem, repriming capability can be improved through the use of smaller diameter arteries. Alternatively, a screened groove wick with small groove dimensions can be considered.

SYSTEMS BENEFITS ASSESSMENT

Allison examined the impact on engine performance of substituting heat-pipe-cooled vanes for conventional air-cooled vanes in the LP turbine of the engine under study. The use of heat-pipe-cooled vanes affects the engine in the following ways:

- The diversion of 2.5% of the compressor inlet flow from the high pressure turbine for cooling the low pressure vane is no longer required. Fully heating this undiverted air in the combustor before entering the turbine results in a performance gain.

- Cooling air no longer issues from the vane interior and flows along the vane outer surface. Elimination of cooling air effluent results in a 0.5 percentage point increase in turbine efficiency, and a net performance gain.

- The vanes protrude into and extend across the width of the bypass air duct, producing a 1.14% pressure drop in the bypass air flow. This pressure drop causes a performance loss.

- TBC is added to the original vane profile in the combustion gas duct and the trailing edge is shortened. The thicker trailing edge reduces turbine efficiency by 0.2 percentage points, with an adverse effect on engine performance.

- The vane length is more than twice that of conventional air-cooled vanes, and additional TBC material (in the combustion gas duct) and fins (in the bypass air duct) are present. The total mass of 12.2 kg (27 lb) for the heat-pipe-cooled vanes exceeds the 5.2 kg (11.5) mass of the replaced air-cooled vanes by 7.0 kg (15.5 lb). Thus, there is a net increase in engine mass.

Other effects that also should be studied include the cost impact and potential failure mechanisms of the vanes, and the safety issues of including sodium in the engine. These will result in additional advantages and disadvantages.

Allison evaluated changes in engine thrust, specific fuel consumption, and weight. We have summarized the results below:

- At a constant turbine rotor inlet temperature (RIT), the engine thrust increases by 7.2%.

- Alternatively, holding the thrust constant, the RIT would be reduced by 55 K (100°F), thus increasing the durability of the turbine flow path components.

- Specific fuel consumption (SFC) decreases by 0.2%.

- The engine weight increases by less than 1%.

We obtained these significant results by assuming that the only change to the engine was in the method of cooling the low pressure turbine vanes. The performance benefits will be even higher by redesigning and optimizing the engine to fully take advantage of heat pipe cooled turbine vanes.

CONCLUSIONS AND RECOMMENDATIONS

Significant improvements in gas turbine performance can be achieved by using heat-pipe cooled turbine vanes in place of conventional air-cooled vanes.

A significant R&D program will be required to realize these performance improvements and bring heat pipe cooling to a state of technological readiness. Precision vane fabrication and assembly techniques must be developed. The capability of a heat-pipe-cooled vane to function in a representative engine environment during startup as well as for extended operating periods must be shown.

For the particular case examined, the heat pipe is a challenging design, chiefly because of the high g maneuvering requirement. For an application without this requirement, such as an advanced transport engine, or commercial applications, the heat pipe design would have much lower technical risk. In these cases the performance benefits of the heat-pipe-cooled vane could be used to optimize SFC or lower hot end temperatures and thus lengthen engine life.

Other possible applications of this technology are for small turbine engines, such as jet fuel starters, auxiliary power units, or missile engines. The hot end of these small engines is difficult or impossible to cool using bleed air because of size and geometry constraints. Heat pipes have been successfully used in applications requiring vapor diameters of less than 0.50 cm. Thus, small size is not a heat pipe constraint.

ACKNOWLEDGEMENT

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