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HEAT PIPE COOLING OF TURBOSHAFT ENGINES

William G. Anderson
Thermacore, Inc.
Lancaster, PA

Sandra Hoff
Power Systems Division
U.S. Army
Fort Eustis, Virginia

Dave Winstanley
John Phillips
Scott DelPorte
Garrett Engine Division
Allied Signal Aerospace Co.
Phoenix, Arizona

ABSTRACT

Reduction in turbine engine cooling flows is required to meet the IHPTET Phase II engine performance levels. Heat pipes, which are devices with very high thermal conductance, can help reduce the required cooling air. A survey was conducted to identify potential applications for heat pipes in turboshaft engines. The applications for heat pipe cooling of turbine engine components included the power turbine first stage vanes, shroud, and case, the HP turbine vanes and shroud, and the T5 temperature probe. Other potential applications for heat pipe cooling include regenerative cycle and intercooling, bearing cooling, IR signature reduction, and active clearance control. Calculated performance benefits included an increase in specific shaft horsepower, and a decrease in specific fuel consumption, as determined with an IHPTET Phase II turboshaft engine performance model. For example, using heat pipes to cool the power turbine vanes, shroud, and case would increase the specific shaft horsepower by 6 percent, while decreasing the specific fuel consumption by 2.2 percent. While this study examined turboshaft engines, most of the applications are also applicable to turbofan engines.

INTRODUCTION

When advancing gas turbine state-of-the-art, the two main goals are to achieve higher engine power-to-weight ratios, and lower specific fuel consumption. Typically, these advanced engines run hotter, so that conventional cooling methods require an increased flow of cooling air. Heat pipes, which have a very high effective thermal conductance, offer an alternative cooling method. A typical heat pipe is shown in Figure 1. Heat is supplied to the evaporator section, vaporizing liquid. The vapor travels down the heat pipe through an adiabatic section to the condenser section, where it condenses and releases its latent heat. The liquid is returned to the evaporator by capillary forces in a wick lining the tube wall. Since the evaporating/boiling and condensing heat transfer coefficients are large, the heat pipe has an effective thermal conductivity that is hundreds of times larger than copper. Because of their large effective thermal conductivity, most heat pipes are essentially isothermal, with temperature differences between the evaporator and condenser ends on the order of a degree celsius. The only significant temperature drop is caused by heat conduction through the heat pipe envelope and wick.

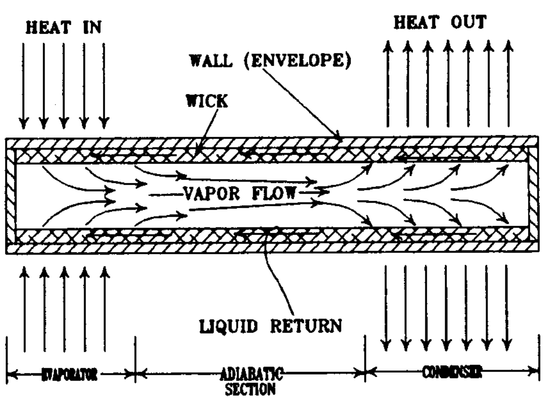

Figure 1. Typical Heat Pipe.

Heat Pipes

In addition to a very high effective thermal conductivity, heat pipes have several additional advantages over other cooling methods: (1) Heat pipes are passive devices without moving parts, which increases reliability, (2) Heat pipes are hollow tubes or other enclosed structures which can be integrated into the engine structure, providing cooling capacity without significantly increasing engine weight, (3) By increasing the surface area in the condenser, heat pipes can transform a high heat flux at the

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evaporator to a lower heat flux at the condenser, making it easier to remove heat by gas flow over the condenser, and (4) The evaporator and condenser surfaces are generally nearly isothermal. For example, the evaporation rate increases in the high heat flux areas of the evaporator, which minimizes the temperature variation.

Heat pipes rely on capillary forces to supply liquid to the heat pipe evaporator, overcoming the pressure drops in the heat pipe. The vapor pressure in the heat pipe evaporator is set by the evaporator temperature, since the fluid is at the saturated conditions. As the vapor travels through the adiabatic section to the condenser, the pressure drops due to frictional forces. The vapor in the condenser is at a slightly lower pressure (and temperature) than the vapor in the evaporator. The condensed liquid travels back to the evaporator, with a frictional drop due to laminar flow in the heat pipe wick. For the heat pipe to operate, the capillary pumping capability must be greater than the sum of the pressure drops in the loop:

$$\Delta P_{capillary} \geq P_{Total} = \Delta P_{vapor} + \Delta P_{liquid} + \Delta P_{gravity} \qquad (1)$$

where:

ΔP_{vapor} is the pressure drop in the vapor between the evaporator and condenser, Pa (N/m^2)

ΔP_{liquid} is the pressure drop in the liquid return line and wick, Pa

ΔP_g is the pressure drop due to gravitational and accelerational forces, Pa

$\Delta P_{gravity}$ is the pressure drop due to hydrostatic forces that occurs if the evaporator is elevated above than the condenser, or if the heat pipe is being accelerated:

$$\Delta P_{gravity} = \rho_l\,(g+a)\,h \qquad (2)$$

where

$\Delta P_{gravity}$ pressure drop due to gravity and maneuvering accelerations, (Pa)

ρ_l working fluid liquid density, (kg/m^3)

$(g+a)$ gravitational and maneuvering accelerations (m/s^2)

h heat pipe height, m

Due to the relatively large maneuvering accelerations (9-10 g), the hydrostatic head for heat pipes in gas turbines is relatively large, limiting heat pipe lengths to under about 15 cm.

The fluid circulation in the heat pipe is driven by the liquid/vapor interfacial forces in the evaporator wick. The capillary pumping capability of the wick, which is the maximum capillary pressure that the wick can sustain, is given by:

$$\Delta P_{capillary} = P_{vapor} - P_{Liquid} = 2\sigma/r_c \qquad (3)$$

where

P_{vapor} Pressure in the vapor, Pa

P_{Liquid} Pressure in the liquid in the wick, Pa

$\Delta P_{capillary}$ pumping capability of the wick (capillary pressure), Pa

r_c radius of the pores in the wick, m

σ surface tension, N/m

Alkali metal working fluid heat pipes with sintered metal wicks will typically have a pumping capability of 5,000 to 15,000 Pa (0.05-0.15 atmosphere).

The design of a heat pipe wick involves a trade-off between capillary pumping capability and permeability (ease of flow). As the pore radius is decreased, the capillary pumping capability increases. However, the permeability of the wick will decrease, increasing the liquid pressure drop. Typical heat pipe wicks include grooved wicks and spaced annuli for space applications where gravity can be neglected, and screen wicks for applications with a low gravity head. Sintered powder metal wicks are often chosen for heat pipes where the evaporator must operate above the condenser, since these wick designs have the highest pumping capability. They are also used in high heat flux applications, since they are less susceptible to blockage of liquid flow in the wick due to vapor generation.

TURBINE ENGINE APPLICATIONS

This study examined the application of heat pipes to existing or planned IHPTET Phase II turboshaft engines, rather than examining an engine redesigned around the use of heat pipes. A review of earlier work on heat pipe applications in turbine engines can be found in Beltran *et al.* (1984)

During the first stage of the study, a number of potential applications for heat pipe cooling were identified in the following categories:

- Cooling Turbine Engine Components
- Engine Cowl Anti-Icing
- Regenerative Cycle and Intercooler
- Bearing Cooling
- Miscellaneous

Table 1 lists the heat pipe applications, the potential benefits, the heat source, and the heat sink. Each potential application was then examined for feasibility. The primary selection criteria for ranking the feasible applications was specific fuel consumption, since it is more difficult to decrease specific fuel consumption than increase specific horsepower in IHPTET turboshaft engines.

The design requirements for heat pipe cooling of turboshaft engines include the following parameters: (1) Heat flux, (2) Source temperature, (3) Heat rejection temperature, (4) Heat pipe geometry constraints, and (5) Acceleration profiles. Table 2 lists approximate thermal boundary conditions for the heat pipe cooling applications based on the operating environment for the Integrated High Performance Turbine Engine Technology (IHPTET) Phase II turboshaft engine. Heat transfer coefficients are reduced due to the presence of a thermal barrier coating (0.25 mm thick for vanes/case; 0.51 mm thick for HP shroud). Anti-icing conditions are based on the T800/LHTEC Design Conditions. The maximum accelerations are typically 9 to 10 g in any direction.

Table 3 shows estimates of the effect of heat pipe cooling on turbine engine performance. Cooling flows for conventionally cooled components in an Integrated High Performance Turbine Engine Technology (IHPTET) Phase II engine were estimated. Improvements in engine specific fuel consumption (SFC) and specific horsepower (SHP/Wa) were determined with the IHPTET Phase II engine performance model. Note that some of the cooling flow reductions are already assumed in the IHPTET Phase II engine performance cycle, and therefore are not additive to Phase II performance levels. In these cases, the heat pipes provide a method of achieving the desired cooling flow reduction. Note that while the benefits were calculated for a turboshaft engine, similar results would be achieved for a turbofan engine.

TABLE 1. POTENTIAL HEAT PIPE APPLICATIONS

Application	Benefits	Heat Source	Heat Sink
1. Anti-Ice Engine Cowl	Reduced Compressor Bleed	Engine exhaust Engine oil	Engine inlet
2. PWT Case Cooling (passive)	Reduced cost over active control Alternative to ceramics, composites	Case	Compressor discharge
3. HPT Vane Stator Cooling	Reduced cooling flow Reduced endwall/airfoil gradient Reduced combustor pressure drop Reduced mixing losses	Cooled stator	Compressor Discharge
4. Turbine Shroud & Duct Cooling	Reduced Cooling flow Improved Clearance Control Eliminate brittle materials	Cooled component	Compressor discharge
5. Reduce IR Signature	Reduced nozzle losses Simpler geometry	Exhaust duct	Eductor Air
6. Pre-Cool HPT Cooling Air	Reduced Cooling Flow Increased Fuel Temperature	Compressor Discharge	Eductor (nacelle) or fuel
7. T5 probe for average temperature	Improved measurement accuracy	Hot spot in profile	Cooler portions of profile
8. Regenerative Cycle	Increased cycle efficiency	Exhaust	Compressor Discharge
9. Intercooler	Increased cycle efficiency	Compressor air	Eductor or Fuel
10. Active Clearance Control	Same as above	Cooled component	Compressor discharge
11. Bearing Cooling	Easier geometries Reduced Transient Temperature Eliminate emergency oil reserve	Bearing	Eductor (nacelle) Oil
12. Cool Rear Bearing Sump	Reduced oil temperatures Reduced transient temperatures	Oil Sump	Fuel
13. Cool Turbine Blades	Reduce Blade Cooling Flow	Blades	Disk rim or cavity air

These calculated benefits assume that the heat pipe can be built. As discussed below, several of the applications were eliminated when heat pipe design calculations demonstrated that the heat pipe was not feasible.

Note that the heat pipe benefits appear to be strongly dependent on the specific turbine engine examined, with some engines showing larger benefits than the benefits in Table 3. For example, the calculated benefits from heat pipe cooling of vanes in an IHPTET Phase II engine was a reduction in specific fuel consumption of 1.6%, and an increase in specific shaft horsepower of 1.8%. Gottschlich and Meininger (1992) examined heat pipe cooling of military fighter turbine engine vanes and found a reduction in SFC of 0.2%, and an increase in thrust of 7.2%.

TABLE 2. HEAT PIPE THERMAL BOUNDARY CONDITIONS

	Source				Sink				
Application	Temperature, °C	h, W/m² °C	Area, cm²	Metal Temperature, °C	Temperature, °C	h, W/m² °C	Area, cm²	Max. Power, W	Heat Flux, W/cm²
1. Anti-Icing	340	160	1610	>4	-28	200	3230	2930	1.8
2a. Power Turbine Vane (40)	1040-1180	870-1440*	820	<870	400-570	1740-2890	480-1130	35,700	44
2a. Power Turbine Endwall	980-1090	720-1300*	480	<930	400-570	1740-2890	480-1130	8,500	18
2b. Power Turbine Shroud	980-1090	580-640*	290	<870	400-570	1740-2890	480-1130	4980	17
2c. Power Turbine Case	870-980	430-600*	1780	<650	400-570	1740-2890	480-1130	34,300	19
3a. HPT Vane (28)	1320-1870	1160-1970*	500	<840	400-570	1740-2890	480-1130	94,300	190
3b. HPT Endwall	1150-1650	1160-1930*	270	<820	400-570	1740-2890	480-1130	42,500	157
4. Turbine Shroud	1040-1510	580-970*	245	<870	400-570	1740-2890	480-1130	14,600	60
5. Reduce IR Signature	–	–	–	–	–	–	–	–	
7. T5 Probe	1040-1180	2030-3770	4.1	1040-1180	1040-1180	2030-3770	4.1	0 (net)	0
10. Active Clearance Control	–	–	–	–	–	–	–	–	
11. Ball Bearing	180	—	100	<200	25	–	–	4,100	42
11. Face Seals (2)	180	—	100	<200	25	–	–	2,500	24
12. Rear Bearing Sump	180	—	40	<200	25	–	–	1,100	29
12. Face Seals (2)	180	—	80	<200	25	–	–	2,800	36

	Source				Sink				
Application	Inlet Temp., °C	Exit Temp., °C	Area, cm²	Flow Rate, kg/s	Inlet Temp., °C	Exit Temp., °C	Flow Rate, kg/s	Max. Power, W	Heat Flux, W/cm²
6. Precool Air	400-570	–	310	0.16-0.32	25	<177	0.09-0.32	108,000	350
8. Regenerator	538-704	524-691	–	4.5-7.7	400-567	412-580	4.5-7.7		
8. Regenerator	538-704	496-663	–	4.5-7.7	400-567	442-608	4.5-7.7		

* Heat transfer coefficients are reduced due to the presence of a thermal barrier coating (0.010" thick for vanes/case; 0.020" thick for HP shroud)

Table 3. Preliminary Heat Pipe Performance Benefits

Application	ΔSFC	ΔSSHP	ΔW_c
1. Anti-Icing			
Max. Continuous Power	-2.95	3.27	
50% MCP	-4.31	3.56	
20% MCP	-5.50	4.77	
2. Power Turbine 1st Stage			
Vane and Endwall	-1.42	4.02	-2.6
Shroud	-0.57	1.15	-0.5
Case	-0.25	0.82	-0.5
Total	-2.24	6.02	-3.6
3. HPT Vane/Endwall	-1.58	1.77	-6.9
4. HPT Shroud	-1.14	2.42	-1.2
5. IR Suppression	-1.11	1.08	
6. Pre-cool Cooling Air	-0.66	0.99	-1.1
7. T5 Probe	-0.44	3.31	N.A.
8. Regenerative			
+25°F	0.19	-1.31	N.A.
+75°F	-0.44	-2.92	N.A.
9. Intercool			
+25°F	0.35	1.38	N.A.
+75°F	0.98	4.31	N.A.
10. Active Clearance		TBD	
11. Forward Bearing Cooling		Reliability	N.A.
12. Rear Bearing Sump		Reliability	N.A.

ΔSFC: Change in Specific Fuel Consumption, %
ΔSSHP: Change in Specific Shaft Horsepower, %
ΔW_c: Reduction in Engine Core Flow, %

COOLING TURBINE ENGINE COMPONENTS

The turbine engine components examined included:

- Power Turbine First Stage Vane,
- Power Turbine First Stage Shroud,
- Power Turbine Case,
- HP Turbine Vane Cooling,
- HP Turbine Shroud Cooling,
- T5 probe,
- Precooling the Cooling Air, and
- Turbine Blade Cooling.

The thermal boundary conditions are similar for most of these applications. All of these applications would have an alkali metal working fluid, either lithium or sodium. The choice of working fluid is determined by the operating temperature. Sodium works well from about 550°C (P_v = 0.012 atmosphere) to 900°C (P_v = 1.2 atmosphere). The vapor pressure and vapor density are very low at temperatures below 550°C, so a high vapor velocity is required to transport a significant amount of energy. As the temperature is decreased, the velocity continues to increase, until the flow is sonically limited. Potassium replaces sodium as the working fluid for lower temperatures, since its vapor pressure is higher at any given temperature.

At temperatures above 900°C, the vapor pressure of sodium increases rapidly. A thicker heat pipe wall would be required to contain the vapor pressure, increasing weight, and the temperature drop across the heat pipe. In addition, the surface tension of sodium decreases with increasing temperature, decreasing the capillary pumping capability of the wick. Lithium is chosen as the working fluid at higher temperatures, since it has a lower vapor pressure and a higher surface tension.

Power Turbine and HP Turbine Vanes

In gas turbine engines, a series of stationary vanes are located just upstream from the turbine power blades. Fuel and air are burned in the combustor, pass through the stationary vanes, then drive the rotating turbine blades. The gas temperature near the vane is too high for the vane materials to withstand without cooling. Gas temperatures are in the range of 1320-1870°C for the HPT vanes, and in the range of 1040-1180°C for the power turbine vanes. Currently, the vanes are cooled with high pressure air blown through the interior of the vane. The air passes through an intricate series of internal passages in the vane, then is discharged through slots in the vane walls. The vane trailing edge is also cooled by air blown through slots at the trailing edge.

While current turbine vanes can be cooled by high pressure bleed air, the use of cooling air decreases net thrust and increases fuel consumption. Substantial cooling flow reductions in the vanes are necessary to meet IHPTET Phase II goals. One possibility is the use of a nonmetallic ceramic or ceramic matrix composite (CMC). The ceramic or CMC vane would operate at or near the hot gas path temperature, and thus require little or no cooling. However, replacing the current metallic vanes with ceramic or CMC vanes has several problems, including: (1) metallic to nonmetallic attachment—thermal growth mismatch can cause destructive stresses in the brittle nonmetallic structure, (2) oxidation coatings currently can not survive in hot streak conditions, and (3) the manufacturability of the complex vane segment geometry requires significant development effort.

Heat pipes can offer an alternate cooling method. The technical challenges for heat-pipe vane cooling include the small size and high heat fluxes, up to 200 W/cm^2 for the HP vanes. Very thin, high performance heat pipes must be developed. To avoid aerodynamic penalties, the trailing edge of the vanes must be very thin. The wick and vapor space of the heat pipe near the trailing edge must also be very thin. The options for heat pipe cooling are (1) Heat Pipe cooling of the entire vane, and (2) Heat pipe cooling near the leading edge, with bleed air used at the trailing edge. Reducing the heat pipe thickness increases the internal heat pipe pressure drop, and it may not be possible to develop a heat pipe that cools the entire vane.

Due to the 600° to 900°C operating temperature range, the heat pipes will use sodium or potassium as the working fluid. The heat sink for vane cooling would be the compressor discharge (CD) air. The heat pipe condensers could be embedded in the compressor exit deswirl vane outer diameter, which has good convection heat transfer to the CD air. The 480 cm^2 area shown in Table 2 represents the amount of space that heat pipes could conveniently occupy without affecting other structures or causing performance losses. An additional 650 cm^2 of convective surface area could be utilized on the deswirl vane inner diameter, with some modifications or concessions to other components. Turbofan engine applications could use the fan bypass duct at the heat sink.

Due to the location of the heat sink, the HP stator vanes will be easier to cool than the power turbine vanes. The adiabatic section of the heat pipes from the power turbine vanes will have to cross several pressure boundaries, so routing and sealing must be considered. On the other hand, the HP stator vanes are already contained within the combustor (compressor discharge) plenum, which also contains the deswirl vane (heat sink). The "adiabatic" section of the heat pipe from the HP vanes to the deswirl vane will be cooled in compressor discharge air with a velocity in the range of 30 to 100 m/s, allowing some of the heat to be rejected in this region.

Turbine Shrouds and Power Turbine Case

Cooling the HP turbine shroud, power turbine first stage shroud, and power turbine case were identified as potential applications. The rotor shrouds are non-moving, cylindrical components that surround the rotating turbine blades. The power turbine case is a conical structure located after the power turbine shroud. The geometries are similar, and it may be possible to cool both the shroud and case with a single series of heat pipes. Turboshaft engines sometimes have a 1 to 2 inch-long duct before the power turbine. The duct geometry is similar to the geometry for the HP turbine rotor shroud, but at a lower temperature, and with a lower heat flux. The heat pipe used to cool this inter-turbine duct would be similar in design to the heat pipe for the HP shroud. The turbine shrouds and power turbine case are currently cooled with high pressure bleed air. However, bleed air to cool the shroud reduces engine performance, since the air could otherwise perform useful work. Heat pipe cooling could eliminate the use of high pressure bleed air for cooling these components, increasing power, and decreasing fuel consumption.

For cooling with heat pipes, the cylindrical shroud would consist of a series of heat pipes; each heat pipe would form an arc segment. The energy from the turbine shroud would be transferred to the compressor exit deswirl vane. A number of heat pipes are used, rather than one large heat pipe, to make it possible for capillary forces in the wick to return the liquid to the heated surface. The hydrostatic forces that the wick must pump against are 1 g of gravity, and the 9 g of accelerational force during maneuvers. Assuming that the heat sink is the compressor exit deswirl vane, the heat pipes would have a maximum diameter of about 30 cm, which is too large to return the liquid to the evaporator. By forming the shroud from a series of arcs, the height over which the wick must pump the liquid can be reduced to roughly 8 cm, which at 10 g's is equivalent to a height of 80 cm under 1 gravity. This is within the range of liquid metal heat pipe wicks.

The heat pipes are an alternative to using non-metallics to conserve cooling air, which is a required IHPTET Phase II technology. Case cooling also helps to reduce the thermal load into the mount for the composite shroud which surrounds the power turbine case. Scorching of this composite structure at the mount has been experienced on other programs, and could be eliminated with heat pipe case cooling.

T5 Temperature Probe

A heat pipe could be used as an average gas temperature indicator behind the HP turbine (station 5.0 or T5 air). The heat pipe was assumed to be a 1/4-inch-diameter tube positioned radially in the flow path behind the HP turbine. The heat pipe vapor space will operate at the averaged temperature of the gas, so a thermocouple inserted in the heat pipe will relay the average temperature measurement to the engine control unit. The improved temperature measurement accuracy (over discrete thermocouples) could allow engines to run hotter while maintaining acceptable safety margins, due to reduced conservatism in turbine component design. Table 3 shows the engine performance improvement for a 50°F increase in turbine rotor inlet temperature (TRIT).

However, it should be noted that the temperature measured inside the T5 heat pipe probe is a function of both the temperatures and heat transfer coefficients outside the probe. The averaged temperature would be weighted toward the temperatures in the fastest part of the stream, since the heat transfer coefficient is largest there. Further studies would be required to determine if this type of "average" temperature would be useful in controlling the turbine engine. For these reasons, the T5 temperature probe was not considered further.

Table 4. Precooling Heat Pipe Properties

	Cesium	Potassium
Acceleration, m/s^2	98	98
Heat Pipe Length, cm	15	15
Temperature, °C	425	450
Liquid Density, kg/m^3	1730	735
Vapor Pressure, Pa	4210	1596
Grav. Pressure Drop, Pa	25431	10810
Sonic Limit, kW	181	145

Precool the Cooling Air

Air flows through tubes to the turbine blades to be used as blade cooling air. Heat pipes might be used to lower the cooling air temperature by transferring heat to the fuel. From 0.09 to 0.32 kg/s of fuel pass through two 0.95 cm diameter lines prior to entering the fuel manifolds. Up to 100,000 W could be transferred from the air to the fuel, increasing the temperature of the fuel to 177 °C before it enters the combustor. The blade cooling air can be reduced by 1.13% W_c since its lower temperature provides more cooling capacity. In addition, the higher fuel inlet temperature brings slightly more energy into the combustor, so both source and sink conditions provide performance payoffs.

It can be shown that this application is not feasible for the IHPTET Phase II engine operating conditions. The two potential working fluids are cesium and potassium in the operating temperature range of 400-450°C. Since there are two heat sinks on opposite sides of the engine, the heat pipe length would extend from the fuel line, to the tubes which are located 90° away, giving a heat pipe length of roughly 15 cm. Gravitational pressure drops for the heat pipes at an acceleration of 10 g are shown in Table 4, based on eq. 2. The pressure drop due to gravity is over 10,000 Pa for potassium, and over 25,000 Pa for cesium. While it would be possible to design a wick capable of handling these larger than normal pressure drops, a second problem is that the gravitational pressure drop is larger than the cesium vapor pressure. For example, the vapor pressure in the condenser with cesium is about 4000 Pa. Assuming that the evaporator is located above the condenser, the minimum liquid pressure at the top of the condenser would be -21,000 Pa. It is unknown if the cesium could sustain such a large tension, or if the cesium would flash into vapor, blocking liquid return from the evaporator to the condenser. Negative liquid pressures have been demonstrated in liquid metal heat pipes, but the tension was only -2500 Pa.[Anderson, 1992]

A second problem with either cesium or potassium is the sonic limit. As the operating temperature is reduced for any working fluid the density decreases, since the heat pipe operates at saturated conditions. To maintain the same heat transfer rate, the velocity must increase. At low enough temperatures, the flow can become sonically limited, with a Mach number of 1. The equation for estimating the sonic limit with mass addition is:[Chi, 1976]

$$Q_{max} = P_0 \lambda A \sqrt{\frac{\gamma}{2 R T_0 (\gamma + 1)}} \tag{4}$$

where

Q_{max} = sonically limited heat transfer, W
P_0 = stagnation pressure, Pa
λ = Latent heat of vaporization, J/kg
T_0 = stagnation temperature, K
A = flow area, m^2
γ = ratio of specific heats, 1.67
R = gas constant, J/kg K

The sonic limits shown in Table 4, based on the cooling tube area of 310 cm^2, are less than twice the required heat transfer rate of 100,000 W for both working fluids. A generally accepted rule of thumb is that the Mach number should be less than 0.2 at the operating conditions for the heat pipe to operate isothermally. For both working fluids, the evaporator area would have to be increased substantially.

In addition to the problems with negative pressures and the sonic limit, the heat transfer rate can be up to 100,000 W, which could require large surface areas to transfer heat into the fuel without boiling. While precooling the cooling air with heat pipes does not appear to be feasible for this engine design, it may be feasible on a turbofan engine, if an alternate heat sink can be identified.

Turbine Blades

Cooling of the turbine rotor blade was examined as a potential application of heat pipes; however, the heat sink location and temperature resulted in no net benefit to the engine cycle. The locations considered for the heat sink were the blade attachment area, the disk itself, and the disk cavities surrounding the rotor. The rotor speeds required for IHPTET Phase II are such that superalloy disk materials must be used. These materials have an absolute upper temperature limit of 650°C (1200°F). If heat is transferred to the disk, either the heat pipe metal temperature must be below 650°C at the point of contact with the disk, or an intermediate insulator with additional active cooling is required. If the heat sink is located in the disk cavity, then the air flow must cool the heat flow from both the disk and the blades. In all three heat sink cases (disk, blade attachment, and disk cavity) the cooling flow to maintain the disk cavity air temperature limit was nearly the same (or in the case of the attachment, actually more) than the blade cooling flow required using the existing Phase II blade cooling technologies. Therefore, heat pipe cooling of rotor blades was not pursued further.

Table 5. Required Heat Sink Areas at the Compressor Exit Deswirl Vane for Cooling Turbine Engine Components

Application	Sink Area, cm^2	Power, W	Heat Flux W/cm^2
1. PT1 Vane	290	22300	27
2. PT1 Endwall	60	5400	11
3. PT1 Shroud	41	3200	11
4. PT1 Case	780	25500	14
5. HPT Vane	948	58200	117
6. HPT Endwall	415	24400	90
7. Turbine Shroud	100	7600	31

Heat Sink for Cooling Turbine Engine Components

The compressor exit deswirl vane was examined as the heat sink for the turbine shrouds, vanes and case, since this has good convection heat transfer to the CD air. The available area ranges from 480 cm^2 if the heat pipes are located on the O.D. of the vane, to about 1100 cm^2 when the I.D. is considered also, with some performance losses. Because of this limited area, not all of the proposed applications could be use the deswirl vane as the heat sink. However, other heat sinks with larger areas (but lower heat transfer coefficients) have been identified that are also suitable for heat pipe cooling.

To determine if the available heat transfer areas were sufficient, the required heat sink areas for the vane, shroud, and case cooling were calculated, assuming mid-range values for the heat transfer boundary conditions given in Table 2. The sink temperature was set at 482°C, with a heat transfer coefficient to the sink of 2315 W/m^2k. The heat transfer equation included temperature drops due to heat transfer from the source to the wall, conduction though the walls and wicks at the evaporator and condenser, and heat transfer from the condenser to the sink:

$$T_{source}-T_{sink} = Q\left[\frac{1}{A_{source}h_{source}}+\frac{t_{wall}}{K_{wall}A_{source}}+\frac{t_{wick}}{K_{wick}A_{source}}+\frac{t_{wick}}{K_{wick}A_{sink}}+\frac{t_{wall}}{K_{wall}A_{sink}}+\frac{1}{A_{sink}h_{sink}}\right] \quad (5)$$

The temperature drop inside the heat pipe can be neglected. Required condenser (sink) area, total power, and heat flux in the evaporator are shown in Table 5, for a wick thickness of 0.064 cm. The sink area shown in Table 5 is the minimum required to maintain the wall temperature below the specified maximum. The results shown in Table 5 indicate that sufficient area would be available for all of the proposed applications, although the required area for the HPT vanes and PT1 case are fairly high. Heat transfer to additional sinks would probably be required. For turbofan engines, the fan bypass duct would be used as the heat sink.

ENGINE COWL ANTI-ICING

The current procedure for anti-icing is to bleed hot air from the compressor, which is detrimental to performance. Engine inlet anti-icing showed the largest reduction in specific fuel consumption, with reductions of up to 5.5 percent at low power. However, note that this fuel savings occurs only when anti-icing is required. The most dangerous icing conditions typically occur while an aircraft is in a holding pattern prior to landing. Under these conditions, the high percentage bleed required for anti-icing increases the danger to the aircraft and its mission. Heat pipe anti-icing has previously been shown to be effective in marine vessels operating in the Arctic, using ammonia or ethanol as the working fluid.[Masuda *et al.*, 1982] However, the heat flux required was only 0.12 W/cm^2, which is an order of magnitude lower than the heat flux required for the anti-icing of engine cowls.

The anti-icing application differs in several ways from the other heat pipe applications. First, the working fluid would be either ethanol (-117°C freezing point) or methanol (-94°C freezing point), to prevent freezing of the working fluid at the minimum air temperature of -28°C. In addition, a pumped loop design would be required, rather than conventional heat pipe. The anti-icing distance is longer than the distance for the other heat pipes. In addition, the surface tension of the alcohol is an order of magnitude lower than the surface tension of alkali metals, which reduces the wicking capability of the system. Since the system can be subject to 10 g's of accelerations/gravity, the required pore size is too small for the system to be practical. As shown in Table 6, the required pore size is roughly 1 micron just to overcome the acceleration/gravity head, assuming a 0.61 m (2 ft long) heat pipe. Because of the small pore sizes, a pumped system would be required. This does have the advantage that the anti-icing can be shut off when it is not needed.

With the anti-icing application, fuel consumption is reduced only when anti-icing is required. The application would require a pumped loop, not a heat pipe. Finally, anti-icing capability will not be demonstrated during IHPTET Phase II demonstrator engine testing. For these reasons, an anti-icing pumped loop design was not considered further.

REGENERATIVE CYCLE AND INTERCOOLER

In a regenerative cycle, air exiting the compressor is pre-heated with turbine exhaust air, which in turn is cooled. Heat pipes could be used to transfer the heat between the two air streams. The use of heat pipes in a regenerative cycle was briefly examined to: (1) determine if incorporating heat pipes into existing structures could provide benefits to the cycle, and (2) perform a sensitivity study to determine if further examination was warranted. As expected, the regenerative cycle shows almost no benefits for the planned IHPTET Phase II concept, with a very high pressure ratio compressor. The pay-off from the regenerative cycle would be more substantial at lower overall pressure ratios, where pressure losses are less costly.

Table 6. Anti-Icing: Pore Size Required to Overcome Gravity/Acceleration

	Ethanol	Methanol
Surface Tension (10°C)	0.0236	0.0257
Density, kg/m^3 (10°C)	798	800.5
Hydrostatic Head, Pa	47673	47823
Heat Pipe Length, m	0.610	0.610
Acceleration, m/s^2	98	98
Pore Size, micron	0.99	1.07

In the intercooled cycle, the compressor interstage air is cooled by heat exchange with turbine exhaust air. A sensitivity study was performed to illustrate how the engine performance is affected if the gas temperature changes by 25 and 75°F. Intercooling provides denser compressor exit air, which in turn demands more fuel, so specific fuel consumption increases (note that SHP/Wa increases significantly). For a 75°F change, the specific fuel consumption increases by 1%, while SHP/Wa increases by 4.3%. Since the specific fuel consumption is the driving performance factor, this option was not pursued further.

For other turbine engines which could benefit from regenerators or intercooling, the major technical challenge is the development of lightweight, high performance heat pipe exchangers with low pressure drop. Silverstein (1968) and Kraft (1975) briefly examined heat pipe gas turbine regenerators. Kraft examined the design of a heat pipe regenerator on a Pratt and Whitney STF 429 engine. The use of the regenerator could improve specific fuel consumption by 3.3 percent. However, the heat exchanger calculations showed that the heat-pipe heat exchanger was relatively heavy, and was large enough to require a bulge in the core nacelle. The increased weight and aerodynamic drag more than offset the regenerator benefits, so the specific fuel consumption would actually increase. Kraft stated that heat pipe regenerators may be applicable to small, slow flying turboprops, when the core flow is very small relative to the entire engine flow.

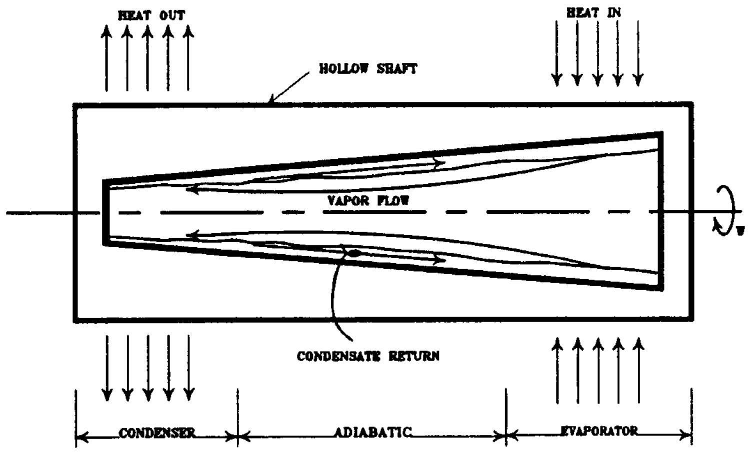

Figure 2. Hollow shaft with an integral rotating heat pipe, showing liquid returned to the evaporator by centrifugal forces.

Silverstein examined heat-pipe regenerators with cesium working fluid and Hastelloy X or titanium envelopes, and examined a generic regenerator design. The advantages of a heat-pipe regenerator included basic design simplicity, ease of integration with the gas turbine, and reduced exhaust gas duct losses. However, the weight was comparable to or higher than other heat exchanger designs. Silverstein assumed that the heat pipes in the regenerator would operate in the 400° to 500°C range, so that cesium was required as the working fluid. At these temperatures, the heat pipe diameter must be large enough to avoid compressible flow effects, which can cause significant temperature gradients in the heat pipe. At higher regenerator temperatures, the vapor density is high enough that sonic effects can be neglected, and the heat pipe diameter can be reduced. This in turn would significantly reduce the heat pipe size and weight.

While both Kraft and Silverstein were pessimistic about heat-pipe regenerators, it should be noted that their studies were completed over twenty years ago, when heat pipes were still a new technology. While not applicable to the IHPTET Phase II engine, materials and engine conditions have changed enough that heat-pipe regenerators may be feasible with present technologies. Kraft assumed that the heat pipes had stainless steel walls that were 0.076 cm (0.030 in) thick. Recently, heat pipes using alkali metal working fluids have been fabricated with titanium composite walls that were only 0.03 cm (0.012 in.) thick, which would reduce the heat pipe weight by a factor of three to five (Rosenfeld *et al.*, 1991). These composites would also greatly reduce the weight of the regenerator designed by Silverstein.

BEARING COOLING

Heat pipes could be used to cool the front and rear bearing sumps and seals. Heat generation occurs in the bearings and seals, and additional heat is conducted from the surrounding structure. Although no significant performance benefit is expected, heat pipes could be used to improve reliability. Helicopter engines are required to operate for 5 minutes without oil to simulate damage to the lubrication system during battle. This is a difficult requirement and could lead to the creation of backup systems. Heat pipes could keep bearings and seals cooler while lubrication is removed, improving the ability of the engine to meet this goal. The heat sink would be compartment (ambient) air. Air entering the inlet particle separator (IPS) could also be considered as a potential heat sink. Higher heat transfer coefficients would be available at the IPS at the expense of operating this component full time.

In some cases, it may be possible to use a integral heat pipe to transfer heat through the interior of the shaft. An example of an integral heat pipe in a shaft is shown in Figure 2. The rotating heat pipe consists of a sealed hollow shaft, with a slight taper along the shaft, and containing a fixed amount of working fluid. Centrifugal forces are used to return fluid from the condenser to the evaporator, along the taper of the shaft. In general, a wick is not required, for cases where entrainment of liquid into the vapor is not a problem.

MISCELLANEOUS

IR Signature Reduction

Two other applications are (1) reduce infra-red (IR) signature, and (2) active clearance control. Heat pipes could provide reliable lightweight passive thermal control for IR signature reduction by incorporating the heat pipes into the hot engine structures. The generated heat would be transferred via heat pipes to an alternate section of the aircraft and rejected over a larger surface area, reducing the peak temperatures of the heat affected area. The result would be an aircraft with less sensitive thermal focal points making ground and air based IR detectability more difficult.

While a potential exists for weight and complexity reductions by using heat pipes for IR signature reduction, the details of an IR signature reduction structure would be quite involved, and were felt to be beyond the scope of the present study. The benefits shown in Table 3 are for the reduction in pressure losses associated with the elimination of the current elaborate eductor exhaust systems.

Active Clearance Control

Ideally, the clearance between the gas turbine blades and the surrounding shroud would be as low as possible for optimum performance. However, additional clearance is required to avoid tip rubs during engine transients, due to centrifugal and thermal displacements. The shroud and blades have different transient thermal expansions, due to differences in heating rates (Evans and Glezer, 1984). Heat pipes may be able to improve engine performance at part power engine conditions by using actively controlled heat pipes in the static structures of the HP turbine. With active clearance control, the heat flow would be regulated with a variable conductance heat pipe. This would allow the

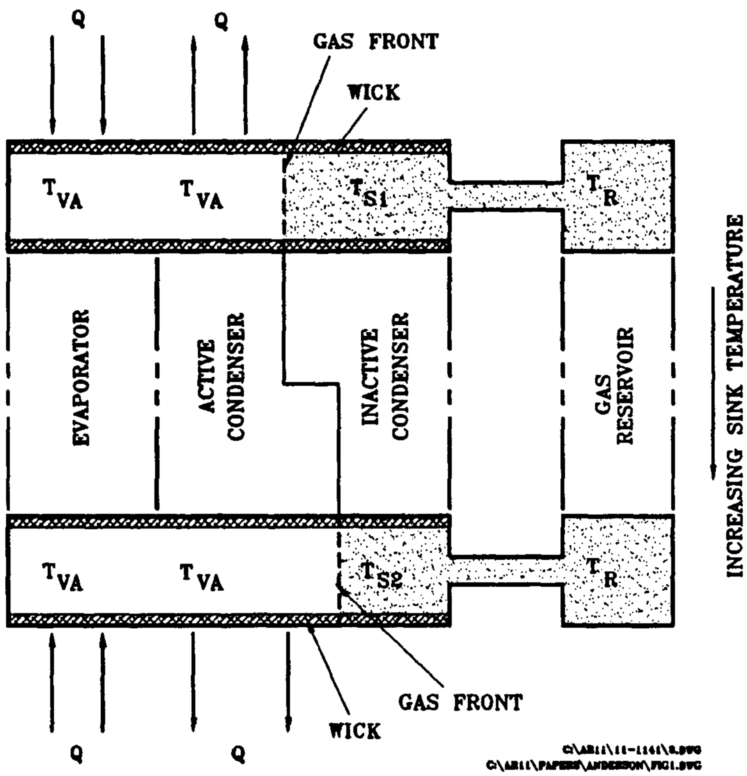

Figure 3. Variable Conductance Heat Pipe (VCHP).

turbine shroud to grow with the turbine blades, minimizing turbine clearances and optimizing performance.

A variable conductance heat pipe (VCHP) uses the contraction and expansion of a non-condensible gas to control the condenser heat transfer area. The VCHP shown in Figure 3 is a passive system that maintains a nearly constant temperature in the evaporator as the heat flux changes. As shown in the lower half of Figure 3, when the heat removal rate is high, the vapor pressure in the heat pipe increases slightly. This compresses the noncondensible gas, increasing the active condenser area for the heat pipe. When the heat removal rate decreases, the vapor pressure drops slightly, the gas expands, and the condenser area is decreased, decreasing the heat removal rate. With this VCHP, the evaporator temperature is almost independent of the heat transfer rate. It might be possible to control the heat removal rate from the VCHP with the addition of heaters and a controller, matching the thermal growth of the heat pipe to the turbine blades.

Note that the active clearance control application is based on the shroud heat pipe, which is discussed in more detail above. The benefits from active clearance control were not calculated, since it would be more realistic to consider this application after a shroud heat pipe was fabricated and tested to verify the basic heat pipe cooling concept.

HEAT PIPE DESIGN CONSIDERATIONS

In addition to the heat transfer aspects of the heat pipe design, three other factors have been identified which must be addressed during the design, fabrication, and testing: (1) Materials Compatibility, (2) Hydrogen Permeation, and (3) Cold Start-Up. These factors are discussed briefly below.

Materials Compatibility

For any heat pipe, life tests must be conducted to verify the compatibility of the heat pipe working fluid, wick, and the heat pipe container. There are a number of problems that can occur when the fluid and container are incompatible, including (1) corrosion of the container or wick, (2) plugging of the wick, and (3) non-condensible gas generation. Leaks due to corrosion of the container wall will allow the atmosphere to enter the heat pipe, stopping the heat pipe from working. Solids particles from corrosion or erosion of the wick or wall can be transported by the flowing liquid to the evaporator wick, which can cause partial blockage of the wick.

The third potential problem is non-condensible gas generation. For example, water is incompatible with aluminum, since small amounts of non-condensible gas are continuously generated. The non-condensible gas is swept by the condensing vapor to the condenser, where it forms a gas pocket that blankets a portion of the condenser. The area available for heat transfer is reduced, and the temperature difference across the heat pipe must increase to transfer the same power.

The heat pipe container and working fluid will depend on the operating temperature of the chosen application. For a high temperature alkali metal heat pipe, alloys currently used in turbine engine fabrication would be considered first. Many superalloys are known to be compatible with alkali metals. For example, alkali-metal-working-fluid (sodium, potassium, or cesium) heat pipes have been fabricated with the following envelopes: Hastelloy X, Haynes 188, Haynes 230, Haynes 556, Incoloy 800, Incoloy 800H, Inconel 600, Inconel 601, Inconel 625, Inconel 718, and RA 330.

Heat pipe fabrication, cleaning, and fluid handling procedures are also important in manufacturing heat pipes with long operating lives. For example, oxygen contamination must be avoided in alkali metal working fluid heat pipes. These oxides are much more corrosive than the sodium, and can greatly accelerate non-condensible gas generation or corrosion. With the proper fabrication procedures, heat pipes can be manufactured with very long operating lives. For example, Thermacore tested an Incoloy 800 heat pipe with sodium at 800°C for over 50,000 hours without degradation.

In a heat pipe life test, a series of heat pipes are fabricated with the chosen container material, wick, and working fluid. A schematic of a high temperature heat pipe life test set-up is shown in Figure 4. Heat is supplied to the heat pipe with a silicon carbide heater, and rejected by thermal radiation from the

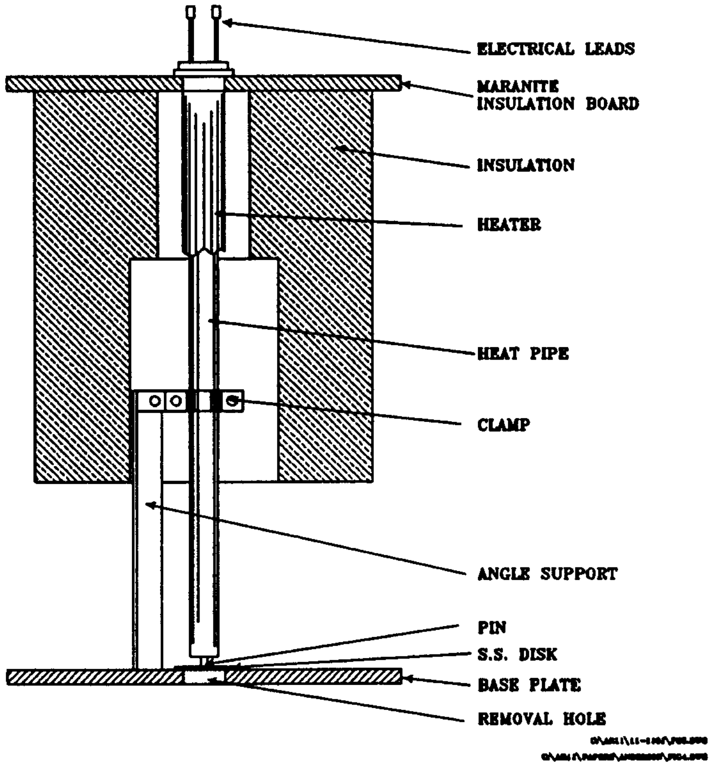

Figure 4. Typical Heat Pipe Life Test Setup.

uninsulated condenser. The evaporator is elevated above the condenser, to demonstrate that the heat pipe can operate in any orientation. Heat pipes are clamped in the test rig, then operated at the chosen temperature while the temperature difference between the evaporator and condenser is monitored, to determine if non-condensible gas is blocking a portion of the condenser. At selected times, i.e., 100 and 1,000 hours, heat pipes are removed from the life test, cut open, and examined for problems with corrosion or wick plugging.

Hydrogen Permeation

For high temperature heat pipes, hydrogen permeation through the heat pipe wall must be minimized. Small amounts of atomic hydrogen can be generated by the dissociation of molecules such as water vapor at the exterior surface of the heat pipe. At the temperatures found in turbine engines, atomic hydrogen passes through most metals. Once hydrogen is inside the heat pipe, it recombines to molecular hydrogen and forms a pocket of non-condensible gas in the heat pipe condenser section. As discussed above, this gas pocket can significantly increase the temperature difference required to transport the desired power.

Hydrogen permeation through a metal is the overall process of moving hydrogen from the bulk gas phase on one side of the wall to the bulk gas phase on the other side. This process consists of interactions at both the entry and the exit surfaces of the metal, as well as the actual transport through the metal. Hydrogen enters and diffuses through metals as atoms dissociated from the molecule, or as positively charged ions (protons). The hydrogen molecule (H_2) does not diffuse in metals.

In many cases the entry of hydrogen at the surface, rather than diffusion through the bulk metal, is the controlling factor in permeation of hydrogen through steel, so variations in the steel surface can greatly change the permeation rate.[Thermacore, 1981] A coating can be applied to the heat pipe evaporator to reduce hydrogen permeation to acceptable levels, allowing the heat pipe to operate for long periods of time. Thermacore has demonstrated in-situ and applied coatings which have proven effective in reducing hydrogen permeation to acceptable levels. Two to three orders of magnitude reduction in permeation have been achieved, which reduces the inflow of hydrogen to below the outflow rate, allowing equilibrium to be achieved with low levels of hydrogen in the pipe.

One method of reducing hydrogen permeation (U.S. patent 4,082,575) uses a thin layer of aluminum oxide to provide a permeation barrier.[Thermacore, 1980] This process is used on ferritic alloys that contain aluminum. The alloy is heated to diffuse aluminum to the surface, where the aluminum oxidizes to aluminum oxide. This procedure was used at Thermacore to fabricate a permeation-resistant stainless-steel/sodium heat pipe for operation at 700°C from a flame heat source. Heat pipe performance would normally degrade rapidly in the hydrogen rich products of combustion. Hydrogen permeation was reduced by a factor of roughly 200. These hydrogen permeation reduction factors were confirmed by measurements at Argonne National Laboratory on treated steel samples supplied by Thermacore.[MacLaren, 1977; Van Devanter, 1980]

Thermacore has also developed a FeCrAlY (24% CR, 4% Al 0.5% Y) coating that can reduce hydrogen permeation to acceptable levels.(U.S. patent 4,478,275) The heat pipes were developed for fluidized bed reactors. Hydrogen permeation in fluidized beds is a serious problem that is compounded by the extremely corrosive and abrasive environment which erodes most coatings. Several Incoloy 800 heat pipes were fabricated with FeCrAlY coatings. An erosion resistant coating of Magnesium Zirconate (MgO-ZrO_2) was used on top of the FeCrAlY permeation barrier, to provide abrasion resistance for the fluidized bed particles. The heat pipes operated near 800°C, with a gas flame heat source. The heat pipes successfully completed over 1000 hours operation without experiencing problems due to hydrogen permeation. Heat pipes without protective coatings develop large hydrogen gas pockets in the condenser after several hours of operation.

Cold Start-Up

During start-up of an alkali metal heat pipe, the heat pipe working fluid is initially frozen. When heat is applied to the heat pipe, the liquid first thaws, then starts to transmit heat from the evaporator to the condenser. In general, a thermal front travels down the heat pipe, with frozen working fluid before the front.

Once the front reaches the condenser, the entire heat pipe warms up in a uniform fashion. It is possible to design heat pipes with sintered powder metal wicks for very rapid start-up from the frozen state. For example, a molybdenum heat pipe with lithium working fluid was heated from room temperature to 2,000 K in under 90 seconds. The heating rate was limited by the power supply, not the heat pipe. As a second example, a rocket thrust chamber was cooled by a surrounding lithium heat pipe. In this system, incident heat fluxes up to 800 W/cm^2 were demonstrated. Very rapid heating rates were applied to the heat pipe from the combustion gases in the engine, with the incident heat flux ranging from zero to the 800 W/cm^2 level in well under one second.

CONCLUSIONS

A number of potential applications for heat pipe cooling of turbine engine components were identified, including:

Power Turbine First Stage Vane
Power Turbine First Stage Shroud
Power Turbine Case
HP Turbine Vane Cooling
HP Turbine Shroud Cooling
T5 probe

Other potential applications for heat pipe cooling include regenerative cycle and intercooling, bearing cooling, IR signature reduction, and active clearance control. While this study examined turboshaft engines, most of the applications are also applicable to turbofan engines. In some cases, the heat pipe design is simplified in turbofan engines, since the bypass air can be used as the heat sink.

Calculated performance benefits included an increase in specific shaft horsepower, and a decrease in specific fuel consumption, as determined with a IHPTET Phase II turboshaft engine performance model. Some of the cooling flow reductions are already assumed in the IHPTET Phase II engine performance cycle and therefore are not additive to Phase II performance levels. In these cases, the heat pipes provide a method of achieving the desired cooling flow reduction.

The design and fabrication of a heat pipe cooled component must consider materials compatibility, hydrogen permeation, and cold start-up.

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