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VAPORIZATION COOLING FOR GAS TURBINES, THE RETURN-FLOW CASCADE

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ABSTRACT

A new paradigm for gas turbine design is treated, in which major elements of the hot section flow path are cooled by vaporization of a suitable two-phase coolant. This enables the blades to be maintained at nearly uniform temperature without detailed knowledge of the heat flux to the blades, and makes operation feasible at higher combustion temperatures using a wider range of materials than is possible in conventional gas turbines with air cooling. The new enabling technology for such cooling is the Return-Flow Cascade, which extends to the rotating blades the heat flux capability and self-regulation usually associated with heat-pipe technology. In this paper the potential characteristics of gas turbines that use vaporization cooling are outlined briefly, but the principal emphasis is on the concept of the Return-Flow Cascade. The concept is described and its characteristics are outlined. Experimental results are presented that confirm its conceptual validity and demonstrate its capability for blade cooling at heat fluxes representative of those required for high pressure ratio high temperature gas turbines.

1.0 INTRODUCTION

In the conventional conception of the gas turbine, the materials of the working parts of the engine including the compressor, turbine and combustion chamber operate hot. In the earliest engines they operated at the local gas' stagnation temperature; this set a rather low limit on the turbine inlet temperature and resulted in correspondingly low power per unit of airflow, and low efficiencies. Most modern engines use air cooling to maintain the metal temperatures in the combustor and turbine substantially below the gas temperatures. The compression ratios are limited by the temperature of the compressor outlet air which in most cases is the coolant for the turbine. Under the constraints of currently available cooling technology the high coolant temperature also limits the firing temperatures to values well below those corresponding to stoichiometric combustion of hydrocarbon fuels. Since the compression ratio and turbine inlet temperature are linked for maximum power, increasing the compression ratio to improve thermal efficiency results in higher cooling air temperatures as well as a reduction in the amount of air available for cooling, exacerbating the cooling problem. Thus, the efficiencies and power densities attained, though very impressive, are significantly below those that are potentially achievable with turbine inlet temperatures corresponding to stoichiometric combustion, and correspondingly high compression ratios. Furthermore, there are substantial thermodynamic penalties associated with the bleed of compressor discharge air for cooling, penalties considerably in excess of those dictated by the heat exchange to the cooling system. The technologies required for solution of these problems and the cost and fabrication difficulty of the high temperature superalloy turbine materials have caused high performance gas turbines to be expensive.

A new approach to gas turbine design is proposed here, in which the gas path is cooled by a flowing liquid or by evaporation of a fluid, providing such uniform and effective cooling that the gas temperature can be that which is dictated by thermodynamic considerations of maximum power and efficiency, more or less independently of materials considerations. With the cooling capacity available with liquids or by evaporation, the temperature of the material defining the gas flow path can be kept low enough that a much wider choice of engineering materials is available compared to the very expensive superalloys and other high temperature materials which are currently employed in gas turbines. For example it seems conceivable that aluminum alloys might be utilized for the stationary parts, and high strength low

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alloy steels for the rotating components, although the choice of materials and the mechanical details are outside the scope of this paper.

It is relatively easy to conceive of designs for liquid or evaporative cooling of the stationary components of the gas turbine. Depending on the location either liquid cooling or some form of heat pipe would be appropriate. Neither the cooling of these stationary parts nor the system aspects of the engine will be considered further here, although their importance to ultimate applications is appreciated. The principal subject of this paper is a technique for *evaporatively cooling* the rotating turbine blades, by means of what will be termed a *Return Flow Cascade*. It is proposed as the enabling element for a new class of heat engines.

This concept (for which US Patent Number 5,299,418, entitled "Evaporatively Cooled Internal Combustion Engine" was issued on April 5, 1994) has been investigated experimentally, leading to proof of the conceptual viability of the Return Flow Cascade. Preliminary modeling has established some factors that control the heat flux limits. In addition some system studies have been carried out to assess the thermodynamic aspects of its application to aircraft engines. We wish to emphasize however, that the configurations built and tested are in no sense optimized, nor have the systems investigations as yet been carried to the point that the ulimate promise of this new technology can be assessed.

2.0 GENERAL DESCRIPTION OF VAPORIZATION-COOLED GAS TURBINE

The general features of a gas turbine incorporating vaporization cooling are indicated schematically in Fig. 1. Liquid coolant might be circulated through cooling jackets which surround the combustion chamber and also through the stationary turbine blades. Alternatively, heat pipes could be used to cool these parts of the flow path. Since the technologies for cooling these parts are well established if the heat-pipe option is used (see Cotter (1972) for a basic description of heat pipes), or need not be very different from those used in reciprocating engine practice if liquid cooling is used, they will not be discussed further here. Thus, the focus of the following discussion is on the cooling of the rotor

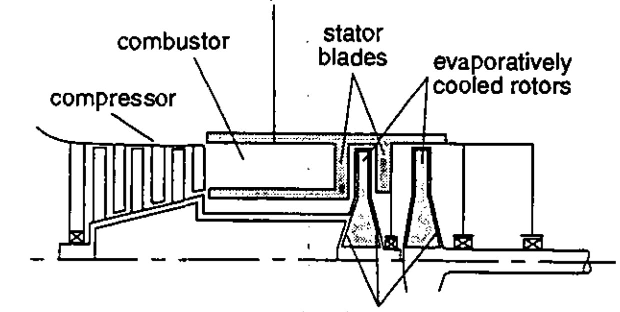


Fig. 1: Schematic of vaporization-cooled gas turbine.

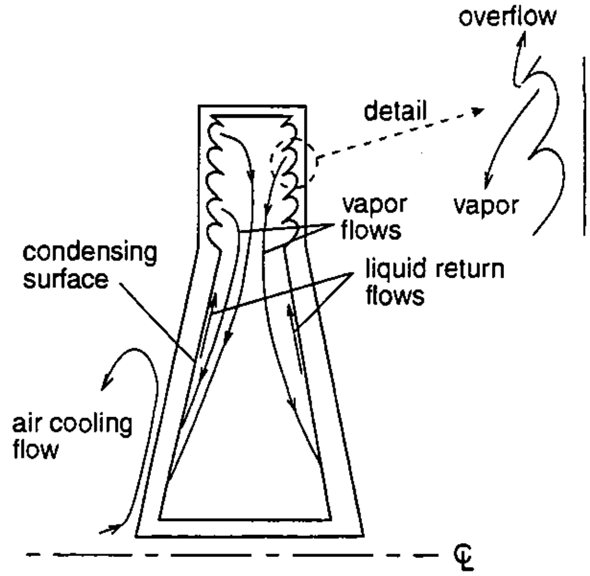


Fig. 2: Schematic diagram of evaporative-cooling system for rotor.

blades. Although liquid thermosiphon or through-flow cooling have been considered in the past for the rotor, they have been found lacking for various reasons. For example, in the first case because of stresses induced by hydrostatic heads in the rotor passages and in the latter because of the high liquid consumption required. Nor did either of these concepts offer the self-regulating characteristic of vaporization cooling, as will be explained.

The conceptual basis for evaporatively cooling the turbine rotors is shown in Fig. 2. It comprises a closed heat transfer system, probably separate for each individual blade, in which the cooling fluid cascades outward in the centrifugal field of the rotating blade, falling from one capture shelf to another, with enough fluid being evaporated at each shelf to absorb the heat flux in its neighborhood. All fluid which is evaporated passes inward as vapor to the condensing section (conceptually) in the rotor disc, where it is reliquified, and then flows back outward in the centrifugal force field, to the cooling cascade in the blades. All the liquid passes to the first (most inward) shelf where some is evaporated and the remainder passes to the second and subsequent shelves, just enough being evaporated at each shelf to absorb the heat load at its location.

This concept has several important attributes. First, it is capable of accepting very high heat fluxes, comparable to those realized with heat pipes, the heat flux being limited only by the liquid flow rate or by the return flow of the vapor. Second, it enforces a nearly constant temperature of the cooled surfaces so long as there is liquid available over the whole surface covered by the shelves. That is, the shelves function as does the wick of a heat pipe. Therefore it is not necessary to predict the local heat flux accurately in order to maintain a nearly constant wall temperature. Thirdly, as conceived it is self controlling, both locally and globally. Any fluid which is vaporized returns to the cascade so that the flow rate of coolant is automatically that required by the total heat load to the blades. So long as there is fluid in the capture shelves, variations in the local heat load result only in variations in the local evaporation rate. The net result is to hold the blade approximately at a constant temperature set by the temperature of the condenser. The working fluid would be chosen so that its vapor pressure at the desired temperature is in a range appropriate from the viewpoints of the vapor pressure loads on the blade, and the vapor flow velocities required to carry the heat load to the condenser.

It has been noted repeatedly that this concept is similar in function to that of the now quite generally used heat pipe; however it is important to note that the heat pipe as such is not suitable for application in the rotating system of a turbomachine because the capillary pumping that it depends upon to saturate the wick with fluid cannot distribute the fluid in the strong centrifugal field of the rotor. Typically, capillary pumping produces a few inches of rise with water against normal gravity (Cotter, 1972). In the several thousand G field of a turbine rotor this would become a few mils. As noted above, it may be that the heat pipe concept will be appropriate for cooling the stationary components of the gas turbine, in place of liquid cooling. This choice, while important to the design of a vaporization cooled gas turbine, is not critical to the concept addressed here.

The heat rejected by the vapor in the condenser can be removed in any number of ways. Convective cooling of the condenser by air which flows through the rotor cavity is one possibility, as shown schematically in Fig. 2. In this case, since the cooling air need not be at the pressure of the turbine working fluid, it could be extracted from the compressor at a lower pressure and temperature than is usual, reducing the amount of air required and the cycle penalty associated with its extraction. Alternatively, it could be desirable under some circumstances to introduce a liquid cooled condenser near the axis in the rotating system, the liquid coolant being fed to the rotor by some system of seals allowing its transfer to and from the rotating system. Again, though important in the application of the evaporative cooling concept, these considerations are not central to the concept, and a number of arrangements are possible.

2.1 Operating Characteristics

Assuming that the cooling system has come to some steady condition as described above, it is important to determine whether this operation is stable, and in particular what the response of the system is to a perturbation. Although the final determination of such stability must be experimental, and strong evidence for it will be presented below, some discussion of the mechanisms that may control the behavior of the cascade is in order at this point.

For example suppose the gas temperature is raised. This will occasion an increase in the evaporation rate in the blade, and hence an increase in the vapor flow. The increase in vapor flow to the condenser will cause an increase in its temperature, and therefore an increase in the vapor pressure in the blade, which will in turn cause an increase in the blade temperature, reducing the heat flow to it. But because of the very rapid increase of vapor pressure with temperature, the increase in blade temperature required to accommodate the increased heat load will be relatively small, and in most cases dominated by the cooling in the condenser. This argument indicates that the system should be stable to such perturbations.

A second question is how the system will behave during startup from an initially cold condition, during shutdown from hot operation, and during transients from one operating condition to another. In the cold startup the fluid in the rotor will be initially in liquid (or solid) form and when rotation begins it will tend to accumulate in the tip regions of the blades. The temperature being low, its vapor pressure will be low as well, and the vapor flow relatively small. As the blades heat up the fluid will vaporize, liquefy in the condenser, and flow into the cascade, filling the capture shelves successively from the innermost radius. Eventually a steady state will be reached in which the capture shelves are full and the normal operation described above will continue. It is possible that the portions of the blades in which the capture shelves are unfilled at cold startup, could be overheated if the heating rate is initially too high, however it appears that this situation can be avoided by gradually increasing the operating temperature at startup. Thus a general requirement is that the time required for adjustment of the cooling system to a new operating condition, be short compared to the time scale for changes in operating condition. The adjustment time is in fact determined by the flow times of the vapor to the condenser and the liquid from it. These are in the order of milliseconds, so it appears that the response of the cooling system should be sufficiently fast, in most situations. In the opposite case of cooling of the system, the liquid will be retained in the capture shelves until the rotation stops.

These plausibility arguments have been confirmed by the experimental results presented in Section 3.

2.2 Cooling Limits

Some requirements for evaporative cooling systems can be understood from rough estimates of the heat flux which must be accepted by the cooled blades. In first approximation the heat flux to the blade is given by

$$q_w = \rho u c_p (T_t - T_w)\, St \qquad (1)$$

where ρu is the mass flux density in the flow passage to be cooled, c_p is the specific heat of the gas, $T_t - T_w$ is the difference between the gas' stagnation temperature and the temperature of the cooled wall, and St is the Stanton number. For a typical gas turbine, with $\rho u = 2{,}000$ kg/m^{2}s, $c_p = 1{,}000$ J/kg K, $T_t - T_w = 500$ K, and $St = .001$, this gives an estimate of 100 watt/cm^2. In the vaporization cooled turbine this heat flux must be conducted through the wall of the turbine blade, to the coolant. This conduction process is governed by Fourier's law of heat conduction,

$$q_w = k\frac{\Delta T}{\Delta x} \tag{2}$$

where k is the thermal conductivity of the blade material, ΔT is the temperature difference between the inside and the outside of the blade, and Δx is the thickness of the blade's skin. The objective of the cooling system being to maintain the blade material at as nearly as possible a constant temperature, this relation sets a limit on the permissible thickness of the blade's skin, Δx. For copper, $k = 480$ watt/m K, and for the above estimated heat flux, we find an allowable thickness of approximately 2.5 cm if the allowable temperature difference is 50 K. On the other hand, for an alloy with a conductivity of about one tenth that of copper, the allowable thickness is reduced to 0.25 cm. This seems still to be in a practical range.

2.3 Cascade Requirements

Since the heat must be conducted from all points on the blade to the immediate neighborhood of the fluid in the capture shelves in order that the evaporation of the fluid may absorb the heat, the spacing of the capture shelves, as well as the thickness of the blade's skin, must be as small as the dimension which has been estimated in the preceding paragraph. Thus an essential feature of the concept is a close spacing of the capture shelves, with spacing inversely proportional to the heat flux. This requirement for a close spacing in turn leads to further requirements on the structure of the cooling cascade.

In order that the cascade function as intended in the acceleration field of the rotor, it is necessary that the capture shelves be "level" in the "effective gravity" of the rotor, so that each shelf will fill before the liquid spills over its lip into the next shelf. This implies two requirements, namely:

a) that the lip of each capture shelf be at a nearly constant radius from the axis of rotation, over the entire internal circumference of the blade, the deviation being small compared to the spacing between shelves, and

b) that disturbances of the rotational force field, by either gravity or rotational or lateral acceleration of the entire engine, be small compared to the rotational force field. That this second requirement is readily met may be seen from an estimate of the rotational force field. The centripetal acceleration is v^2/r, where v is the tangential velocity of the blade and r is the radius. A typical value of v is 400 m/s or more, so for $r = 0.5$ m, the acceleration is 3×10^5 m/s^2 or about 3×10^4 times the acceleration of gravity. Thus gravity itself should introduce only very minor perturbations, and the system should be very insensitive to lateral accelerations as high as 100 times gravity. If the time for angular acceleration of the rotor is in the order of 1 sec, the equivalent peripheral acceleration of the blades is on the order of 400 m/s^2, which is still a factor of 1,000 below the acceleration due to the steady rotation.

The experiments to be described in the next section largely confirm the cascade behavior postulated and rationalized above.

3.0 EXPERIMENTAL MODELING

Since the Return Flow Cascade is a wholly new concept for cooling rotating blades, the first need is to define the factors controlling its feasibility and limiting the performance. This includes the interaction of fluid characteristics such as surface tension, wetting and viscosity with acceleration forces in controlling fluid flow rate and distribution. Nor is the best geometry for such a cascade known. The model test sections used for the tests to be reported here were designed essentially to implement the idealized concept shown in Fig. 2. They consist of cylindrical evaporator sections with axisymmetric (about the radial direction) shelves on their inner surfaces, coupled to conical condensers. The evaporator sections which model the turbine blade are electrically heated on their outer surfaces while the condenser is air-cooled. The evaporators were approximately 2 cm in inner diameter and 5 cm in radial extent, the condensers of similar dimensions but tapered as shown. In light of the noted uncertainties about the liquid return behavior, the evaporators were provided with flow-control inserts designed to ensure that the liquid flowing from the condenser enters the first capture shelf and then passes sequentially through the remaining shelves, while at the same time allowing the vapor to enter the central passage through which it returns to the condenser. The arrangement for this purpose is shown in Fig. 3. It consists of a tubular insert, with hemispherical depressions on its outer surface that provide passages from one shelf to the next, and holes aligned with the inner portion of each space between the capture shelves to allow the venting of the vapor

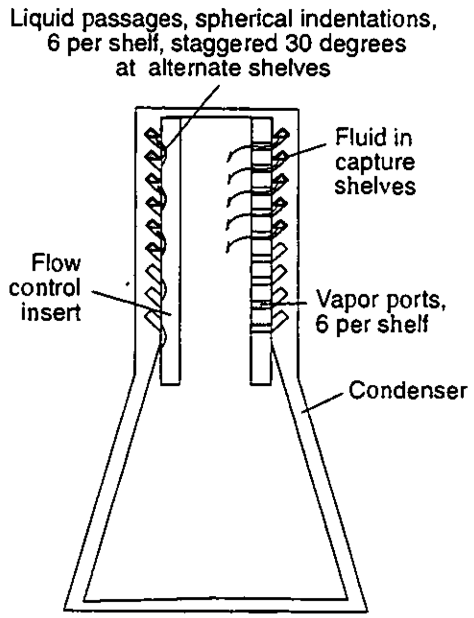
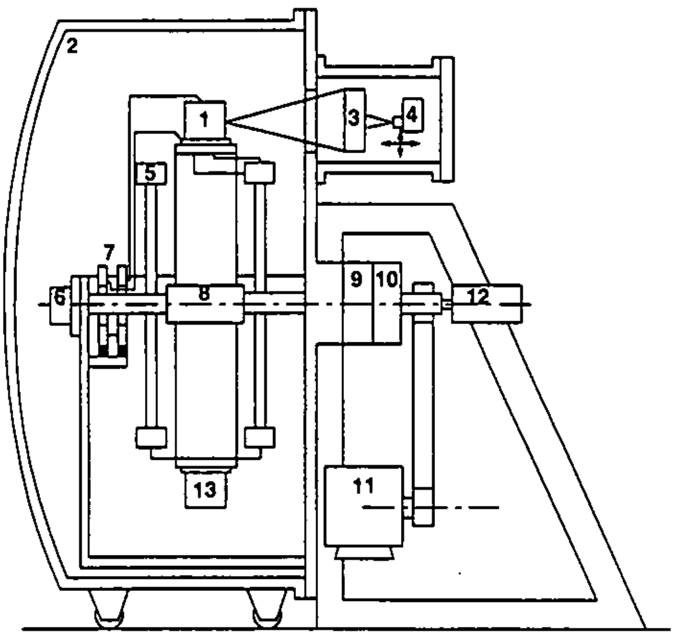


Fig. 3: Schematic of a typical evaporator and coupled condenser, showing flow control insert and capture shelves.



Legend:

1) Test model
2) Vacuum enclosure
3) Detection mirrora
4) Infrared detector
5) Counterflow heat exchanger
6) Shaft encoder
7) Power slip rings
8) Pressure transducer
9, 10) Shaft seals
11) Drive motor
12) Slip rings
13) Counterweight

Fig. 4: Overall schematic of rotating heat transfer facility.

generated in the shelves. Two test models using this geometry have been tested, one with a shelf spacing of approximately 0.050 inches (TS1) and one with a spacing of 0.080 inches (TS2). In each case there were six of the liquid-passage depressions and six of the vapor holes, per shelf. It will be argued later that the liquid flow through the first (inner) set of depressions limited the heat flux capacity of these models under some conditions. Clearly, the capacity could be increased by providing more depressions for the inner shelves.

The instrumentation consisted primarily of several thermocouples distributed radially in grooves on the outer surface of the evaporator and three similarly distributed on the condenser. In addition the rotational speed, cooling air flow and its temperature at inlet to and exit from the condenser, were recorded.

These test models were mounted on the rotating arm of the MIT Rotating Heat Transfer Facility, which is shown schematically in Fig. 4, at location 1, labeled "test model." Briefly, the facility enables a test assembly to rotate in a vacuum, to eliminate external heat transfer and windage. Cooling fluid is provided through rotating seals and electrical power and electrical signals are carried through slip rings. The radius of the rotating arm is approximately 0.5 m and the rotation rate can be as high as 30 rps, although in the tests to be described it did not exceed 10 rps. The flow circuit includes counter-flow heat exchangers that operate between the incoming cooling air and that leaving the test section. This allows the test section to operate with elevated cooling air temperature without the need for high temperature airflow through the shaft and seals of the facility.

3.1 Test Procedures

The experiments were designed to explore the characteristics of the Return Flow Cascade, and in particular to determine if it is self regulating and capable of maintaining a nearly uniform evaporator temperature as described above. A typical experiment consisted in initiating rotation of the test model, and gradually increasing the electrical power to the evaporator while observing the variation of the evaporator and condenser temperatures. Typically these were closely coupled, showing that the thermal conductance of the cascade is very high (to be quantified below), as hypothesized. Due to the function of the counterflow heat exchangers the overall temperature level of the condenser and therefore of the test section would continue to rise even at constant heating power. After the cascade stabilized at some heat flux, (as high as 40 watt/cm^2 in the experiments reported) with the temperatures gradually rising as the heat exchangers heated up, the rotation rate was then gradually reduced (at about 0.025 rps per second) until the rapid (in a few seconds) rise of one or more of the temperatures on the evaporator signaled that the heat flux limit of the cascade had been reached. At this time the rotation rate was increased and generally the cascade recovered to its former temperature distribution. In this way the limits of the cascade were explored as a function of rotation rate and heat flux

Most of the experiments have used water as the working fluid, but methanol was substituted for it in some experiments, to explore the effects of changes in the fluid properties. In high temperature gas turbine applications the most suitable working fluids are likely to be alkali metals, but for practical reasons they have not yet been used in these experiments.

3.2 Experimental Results

Time histories of two experiments conducted with the test section having 0.080 inch shelf spacing, along the lines described above, are shown in Figs. 5 and 6. In the case of Fig. 5 the working fluid was water, while for Fig. 6 it was methanol. For this test section, even at rotation rates as low as 1 rps it was not possible to reach the heat flux limit of the cascade within the available heating power with water as the working fluid. Hence Fig. 5 does not exhibit a heat flux limit. With methanol, the limit of the cascade was reached, as will be described in discussion of Fig. 6.

In Fig. 5, the variables of principal interest are the four temperatures measured on the evaporator, the condenser temperature, the heater power which was stepped up in increments of 50 to 100 watts, and the condenser cooling mass flow, which was also modulated as the power was increased. The spanwise acceleration was held at about 100 G's over most of the experiment. The evaporator temperatures are numbered increasing outward, 6 being near the tip and 2 about one third of the distance from the

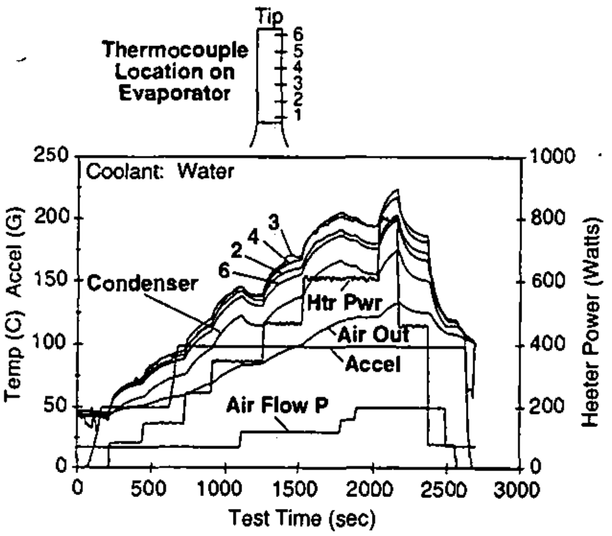


Fig. 5: Time history of an experiment with TS2 using water as working fluid.

inner end of the evaporator to the tip. The primary result of the experiment is that the four temperatures measured in the evaporator are all nearly equal, the maximum difference being about 20 C at the highest power, and they all closely follow the condenser temperature, the difference between the lowest of them and the condenser being about 20 C. This low temperature difference indicates a very high thermal conductance, a characteristic typical of heat pipes. It is also significant that a step increase in the cooling flow, resulting in a drop of the condenser temperature, caused a corresponding drop in the evaporator temperatures. The time scale for response to step changes in power or cooling flow was of the order of a minute for these conditions, corresponding approximately to the thermal inertia of the test section.

The thermocouples in the evaporator are situated in grooves in its outer wall, adjacent the heater, and those in the condenser are several millimeters from the condensing surface, so a portion of the measured temperature difference between evaporator and condenser is due to conductive drop in the two walls. At a heat flux of 40 watt/cm^2, the highest value attained in Fig. 5, the conductive drop is estimated to be about 10 K, or half the temperature difference between the condenser and the coolest point of the evaporator.

Figure 6 shows the behavior of this same test model with methanol as working fluid, and in particular its response to changes in rotative speed (indicated by the radial acceleration in G's). After an initial heating period, the power and cooling flow were held constant. At about 1500 seconds the cascade had reached a stable operating point at about 350 watts and an acceleration of 100 G's, where the temperature differences within the evaporator and from evaporator to condenser were comparable to those found in Fig. 5. The rotative speed was then lowered linearly in time commencing at about 1800 seconds. At about 60 G's, two of the evaporator temperatures began to rise, indicating that the cascade had reached its heat flux limit at this speed. Increasing the speed caused it to recover to the former operating condition. The variation in speed was repeated at about 2400 and 2900 seconds with similar results.

Similar results are shown in more detail for the test section with 0.050 inch shelf spacing in Fig. 7. For this test water was the working fluid. Four evaporator temperatures are shown, the number indicating the distance in inches from the tip of the evaporator, so that the one labeled 1.456 is closest to the condenser. In this data sequence the heater power and condenser temperature were nearly constant, while the rotative speed was decreased until the heat flux limit was reached, then immediately ramped back up. This was done twice, at about 2500 and 2900 seconds. At the beginning of the first sequence (2500 seconds) all evaporator temperatures were nearly equal at about 140 C. As the speed was reduced the location at 0.865 inches showed signs of divergence at an acceleration of about 50 G's. The temperature at 1.456 inches similarly increased at about 45 G's, whereupon the speed was

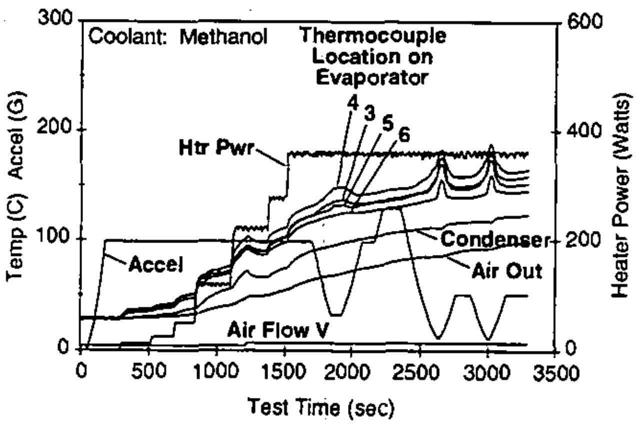


Fig. 6: Time history of an experiment with TS2 using Methanol as a working fluid and exhibiting heat flux limit.

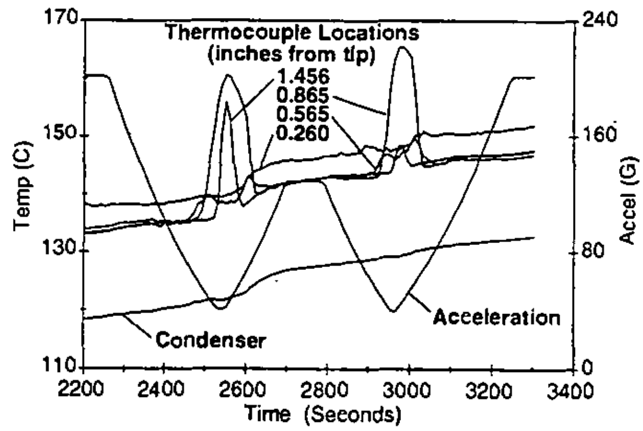


Fig. 7: Time history of an experiment with TS1 using water as a working fluid and exhibiting heat flux limit.

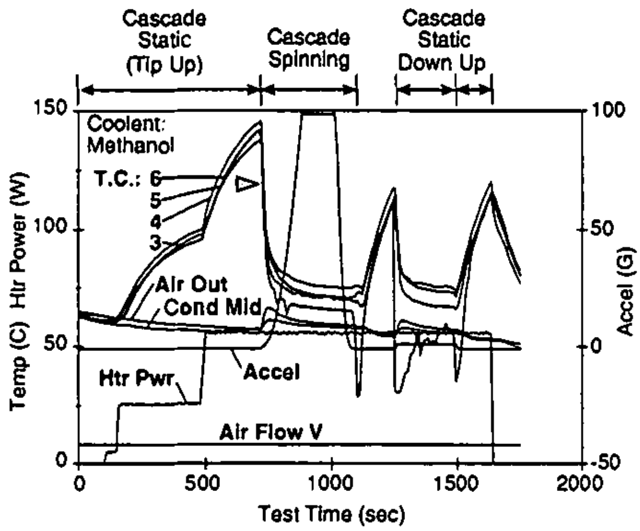


Fig. 8: Comparison of conductances of cascade when dry and in normal operation, with water as working fluid.

increased and both returned to their initial values. The same behavior occurred beginning at about 2930 seconds. That the cascades recovered after each of these divergences is taken as proof of the self-regulation capability and stability of the cooling mechanism of the cascade.

To further illuminate the general characteristics of the Return Flow Cascade an experiment was conducted with the test section with 0.080 inch shelf spacing, to compare its conductance in normal operation with the conductance of the metallic structure of the evaporator and condenser. As recorded in Fig. 8 this was done by applying power to the evaporator with it stationary and upright, so that the working fluid (methanol) was isolated in the condenser. As indicated by the time interval from about 100 to 700 seconds the evaporator temperatures rose rapidly, while the condenser temperature remained nearly constant at its initial value, until the temperature difference between evaporator and condenser was about 100 C and still rising rapidly. At 700 seconds, rotation was initiated, building to an acceleration of about 100 G's at 1000 seconds. The evaporator temperatures quickly dropped and the condenser temperature rose, till the temperature differences were on the order of 10 to 20 degrees and stable. The rotation was then stopped with the test section pointed up as initially and the temperatures diverged again. It was then pointed down, giving the effect of a 1 G acceleration and the temperatures returned to the levels attained with rotation. The conclusion is that at this low heat flux, a 1 G acceleration is sufficient for operation of the cascade. More than that, the cascade quickly refills and resumes normal operation after having been dried out. This again confirms the self-regulating characteristic hypothesized earlier.

From a series of experiments such as those discussed above, the global characteristics of the two test models have been adduced. It is important that these are the characteristics of these test models only; no claim to generality can be made at this point in our investigation. On the other hand the observed characteristics are thus far as initially supposed.

3.3 Heat Flux Limits

In this section a simple flow and heat transfer model will be proposed that seems to provide a rationale for the observed heat flux limit.

The model is based on the assumption that in the configurations tested, the limiting heat flux is determined by the rate of liquid flow through the small passages that direct liquid from one shelf to the next. In this case the heat flux limit, Q, is given by the product of the liquid mass flow and the latent heat

$$Q \approx \dot{m} h_{fg} \tag{3}$$

The head available to drive flow through the passages is approximately

$$\Delta p \approx \rho \omega^2 r \Delta r \tag{4}$$

where an upper bound on Δr is defined by the spacing of the shelves. The flow velocity through the passages is then

$$u^2 \approx 2\Delta p / \rho = 2\omega^2 r \Delta r \tag{5}$$

and the resulting Q is

$$Q \approx h_{fg} \rho \omega A [2r\Delta r]^{1/2} \tag{6}$$

For water and $r = 0.5$ m this becomes

$$Q \approx 2.3 \times 10^4 \, \omega A [\Delta r]^{1/2} \tag{7}$$

in watts where Δr is in cm and A is in cm^2.

Thus, according to this model the limiting heat flux should be proportional to the rotative speed, or the square root of the acceleration. It is a bit difficult to quantify A and Δr, but taking reasonable values of $A = 1$ mm^2 and $\Delta r = 1$ mm, we have an estimate for the total heat load allowable on the blade of

$$Q \approx 400\, n \approx 280\, g^{1/2}, \text{ watts} \tag{8}$$

where n is the rotative speed in rps and g is the acceleration. The numerical factor however must be viewed strictly as an estimate, certainly subject to error by a factor of 3 or more.

The limiting heat flux given by this model is compared in Fig. 9 to a large number of limiting points identified for the test section with 0.050 inch shelf spacing, operating with water. The experimental points for each thermocouple are located on the abscissae at the rotational speed for which its temperature began to increase as the speed was decreased, at the heating power indicated on the ordinate. A series of lines indicate the limiting heat fluxes predicted by the above model with flow dimension defined by the actual cascade geometry. The assumption is that since all the liquid feeding the cascade must pass through the innermost row of depressions in the flow control insert, and some evaporates on each shelf, the outermost shelves should evidence the heat flux limit first as the rotative speed and therefore the liquid supply is reduced.

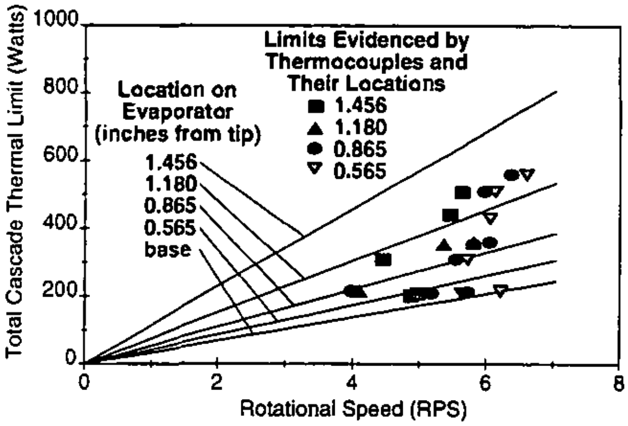


Fig. 9: Heat flux limit for TS1 with water as working fluid, as a function of rotative speed. Model results assume liquid flow rate through the inner end of the flow control insert as the limiting factor in heat flux.

The first point to be made from comparison of the model to the data is that the magnitude of the limiting heat flux is about right, suggesting that the liquid supply is in fact the limit on heat flux. Closer inspection shows that in general the thermocouples nearest the tip did usually exhibit the limit at the lower heat fluxes. On the other hand the trend of limiting heat flux with rotative speed does not seem to follow the simple proportionality suggested by the model.

A second test of the model is provided by the two sets of data from the test section with shelf spacing of 0.080 inches, described above in Figs. 5 and 6, one with water and the other with methanol. With water this test section showed no heat flux limit within the range of heating available, up to about 800 watts. But with methanol the heat flux limit was identified at a power of about 350 watts. The heat flux limit for acceleration-driven methanol flow is about 0.4 that of water, so that other things being equal one would expect the cascade to be stable up to 800x0.4=320 watts with methanol. This is somewhat below the observed limit of 350 watts.

Clearly, more must be done to define the heat flux limits of the Return Flow Cascade, but it is important to note that the accelerations required to achieve stability in these experiments are far below those found in turbines. As indicated above a typical level is 3×10^4 G's. Thus it seems doubtful that the limits found here will be controlling in applications to gas turbines, particularly when use of alkali metals as coolants is considered.

4.0 ENGINE SYSTEM CONSIDERATIONS

To judge the viability of the Return Flow Cascade in an engine application several factors must be considered. These include the mechanical details of incorporating it in the turbine rotor and the provision of cooling for the condenser and other matters that are quite specific to the particular engine. In addition there are the more general effects on the engine's thermodynamic cycle of a higher permissible firing temperature, increased heat loss from the turbine due to lower blade temperatures, and the possibility of rejecting this heat elsewhere than to the turbine flow path. This last class of effects has been treated by Martinez-Tamayo (1995) for aircraft engine applications. Amongst his findings are the conclusion that in a turbofan engine there is considerable benefit in rejecting the heat from the turbine cooling system to the fan air stream, rather than to the turbine exhaust. He also found that the deleterious effects of the increased heat loss in the turbine can be more than offset by the beneficial effects of increased turbine inlet temperature, so that both the specific power and the thermal efficiency of the engine are improved by incorporation of vaporization cooling. In this analysis the Return Flow Cascade was treated as a replacement for air cooling, so that the basic architecture of the engine was assumed to be conventional.

Much remains to be done to achieve a full evaluation of the potential of the vaporization cooled gas turbine. As indicated in the Introduction, if its characteristics are exploited fully the overall characteristics of the engine may be very different from those of present gas turbines. This is a subject for future discussion.

5.0 CONCLUSIONS

The primary conclusion of this work is that the Return Flow Cascade does function as conceived, to provide self-regulated heat flow from the evaporator to the condenser. While the experiments reported have not fully covered the range of heat flux found in modern engines, when combined with simple models they indicate that these characteristics will be preserved at the very high heat fluxes that will exist in such modern engines if vaporization cooling is incorporated in them. The Return Flow Cascade therefore provides a means for cooling the rotating blades of a turbine to a very nearly constant temperature at temperature and pressure levels characteristic of modern gas turbines. In this sense it is the enabling element for the concept of the vaporization cooled gas turbine, which presents a new paradigm for gas turbine design.

Much remains to be done however, in examining both the detailed design issues and the system issues that must be resolved to incorporate the Return Flow Cascade in practical engines. These include the management of the waste heat from the condenser and the design of practical blades incorporating the evaporator and condenser, as well as the opportunities presented by new material choices.

6.0 ACKNOWELDGEMENTS

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