C.P. No. 1127

C.P. No. 1127

MINISTRY OF TECHNOLOGY

AERONAUTICAL RESEARCH COUNCIL

CURRENT PAPERS

Liquid and Vapour Cooling Systems for Gas Turbines

By J. P. Edwards,
NGTE, Pyestock

LONDON: HER MAJESTY'S STATIONERY OFFICE

1970

PRICE 5s 0d [25p] NET

C.P. No.1127*

November 1969

Liquid and vapour cooling systems for gas turbines

- by -

J P Edwards
NGTE Pyestock

SUMMARY

This Note discusses the application of several types of liquid and vapour cooling systems to gas turbine blade cooling, with emphasis on systems suitable for continuous operation. The application of the 'heat pipe' concept to stator blade cooling is discussed in some detail and a tentative design study presented.

*Replaces A.R.C.30 986

CONTENTS

APPENDICES

ILLUSTRATIONS

1.0 Introduction

At the present time, all gas turbine cooling requirements are met by forced convection air cooling. The limitations of this method are well known, namely low heat transfer coefficients and high 'pumping work' losses. The former implies that the coolant temperature must be much lower than the metal temperature and that good cooling in high heat flux regions is difficult to achieve, and the latter that cooling entails a loss in cycle efficiency which partially nullifies the benefit of higher turbine entry temperature. Also, large cooling air requirements reduce the amount of combustion dilution air available, thus making it more difficult to obtain a good temperature traverse at combustion chamber outlet.

The schemes discussed in this Note may be classified as follows.

Open systems

1. Direct internal cooling by fuel or water.

2. External spray cooling of rotor blades.

3. Air cooling of rotors and stators using compressor delivery air cooled by (a) water injection, (b) fuel/air heat exchanger.

Closed primary coolant circuit with secondary coolant

4. Rotor cooling using a closed thermosyphon or vapour chamber in the blade to transmit heat to the root with root cooling using (a) water or (b) fuel.

5. Stator cooling using (a) a pumped liquid circuit or (b) a heat pipe to reject heat into compressor delivery air which subsequently passes through the combustion chamber.

2.0 Thermodynamic considerations

Schemes using fuel or compressor delivery air as a heat sink prior to its entry into the combustion chamber (i.e., 1,5) incur no thermodynamic penalty. In fact the feedback of heat from partly expanded gas in the turbine to compressor delivery slightly increases the cycle efficiency for a given combustion temperature rise. The augmentation of air cooling using aftercooling by fuel (3(b)) is also unlikely to reduce cycle efficiency.

Schemes using an expendable inert coolant such as water (2, 3(a)) involve loss of the latent heat from the cycle and so reduce thermal efficiency somewhat; the coolant must also be carried. Such schemes are therefore only suitable for short term thrust boosting. The most efficient method of thrust boosting for small thrust increases is simply raising the turbine entry temperature. To achieve this, extra cooling can be obtained by water injection into the blade cooling air[1]. Consider an engine with a compressor delivery temperature of 700°K, turbine entry temperature 1450°K and blade cooling flow 4 per cent of main flow. The injection of 10 per cent water into this cooling flow reduces its temperature by 270°C thus permitting a rise of 100° in turbine entry temperature and thus a 10 per cent thrust increase. The water used is about 30 per cent more than the increment in fuel flow required to raise the temperature. The boost effectiveness of this system, defined in Reference 1 as the percentage thrust increase over the percentage increase in liquid consumption is 0.26 - still better than for most other systems.

With the exception of scheme 4(a), in aircraft engines heat is not lost from the cycle and indeed the heat extracted by the cooling system must be rejected at relatively high temperatures. Whilst this is good thermodynamically, it puts a premium on our ability to transport large amounts of heat for the smallest possible temperature drop. Means of achieving this using liquids and vapour systems will be discussed.

3.0 Use of fuel as a heat sink

In high speed aircraft, the fuel is very much in demand as a heat sink for the cooling of electrical and other systems as well as for the engine oil. For kerosine we might assume that fuel becomes available for blade cooling at about 150°C. The amount of heat which can be absorbed is thus largely controlled by the temperature at which solid residues are formed by reactions in the fuel[2]. With proper precautions[3] this could be raised to the critical temperature (373°C) or above. The heat absorbed at the critical pressure 350 p.s.i. would then be 158.4 C.h.u. lb^{-1} (6.62×10^5 J kg^{-1}) (Reference 4). Somewhat more heat would be absorbed at lower pressures, but the tendency to form solid deposits would increase.

For a fuel/air ratio of 0.025, a cooling airflow of 6 per cent could be cooled by 260°C (scheme 3(b)). Some development of the burner would be necessary to cope with fuel at such a high temperature but this does seem feasible. If the use of liquid methane as a fuel is envisaged, the situation is even better. However much more heat will be absorbed by the fuel tanks so let us suppose that the latent heat, 121 C.h.u. lb^{-1} (5.06×10^5 J kg^{-1}) has been supplied and that after compression and passage through the oil cooler, the gas is at 150°C. It should be possible to heat the gas by another 400°C, thus absorbing 311 C.h.u. lb^{-1} (13×10^5 J kg^{-1}). Allowing for the lower fuel flow (87 per cent of that for kerosine), the heat sink capacity is 70 per cent greater[5]. These temperature limits are more conservative than those quoted in Reference 6 i.e., 810 and 920°K for kerosine and methane. Decomposition of methane into hydrogen and solid carbon is reduced by increasing pressure but is strongly influenced by the nature of the heating surfaces[7]. A heat exchanger for this system could be mounted between the HP turbine shaft and the combustion chamber inner annulus.

The supply of fuel to a rotor system as secondary coolant poses severe sealing problems since any fuel vapour leaking out will tend to burn and perhaps form carbon near the seals creating a hot spot. Some modern engines have no bearings between the turbine and HP compressor. It may thus be possible to discharge heated fuel from the rotor system after the last compressor stage thus obviating the need for a 'hot' fuel seal (Figure 1). There is also the prospect of using the fuel to cool the last stage of the compressor, which is quite unamenable to air cooling unless the air itself is precooled. Solid impurities, decomposition products and polymers which would normally remain in suspension will be centrifuged out, forming a barrier to heat transfer.

4.0 Closed circuit heat transfer devices

4.1 Thermosyphons for rotating blades

These consist merely of a hollow blade shell filled with a liquid metal. In order to obtain good effectiveness, the thermosyphons should be of the semi-closed type[8] for which the blade cavity communicates with a large chamber at the base. A thermosyphon of constant cross-section is not as effective[9]. The semi-closed thermosyphon eliminates some of the disadvantages of the open thermosyphon; namely a proneness of mechanically unbalanced operation and the deposition of solid material in the blind ends of the blade cavities. The device described in Reference 8 operated with a liquid metal to blade heat transfer coefficient of 32.5 kW m^{-2} $^{o}C^{-1}$ and a liquid to root value of 5.08 kW m^{-2} $^{o}C^{-1}$ giving temperature differences of 22.2 and 262.8^{o}C respectively.

4.2 Vapour chambers for rotating blades

An alternative to the semi-closed thermosyphon is the vapour chamber[10] which is merely an evacuated cavity containing a small mount of a fluid such as a liquid metal. Vapour condensed in the root region runs up the walls of the blade cavity in a film evaporating as it goes up, whilst the resulting vapour goes down towards the root (Figure 2). Heat transfer coefficients for evaporation and condensation are exceedingly high, especially for alkali metals. The processes are probably only limited by conductivity through the film; in any case the conductivity of the liquid metal is greater than that of the blade metal. Even for nucleate boiling, the heat transfer rate rises exponentially when the difference between surface and saturation temperature exceeds about 50^{o}C, for a heat flux of the order of 0.6 MW m^{-2} (Reference 11). Nucleate boiling is unlikely in this system since the wall film is so thin that the required degree of superheat is unlikely to be achieved at the container surface[12]. One possible limiting factor for vapour chambers is local drying out of the walls[10]. A grooved or porous surface on the walls as in a 'heat pipe' may help to overcome this or alternatively the whole of the aerofoil cavity could be filled with liquid metal circulating by natural convection and evaporating from its surface only with condensation taking place in the root cavity. Another limiting factor is pressure drop on the vapour side as choking conditions are approached. Choking in the vapour passage (the liquid return occupies negligible area) is considered to be limiting. The choking mass flow however increases rapidly with increasing vapour pressure and doubles as the temperature is raised from 1100^{o}K to 1200^{o}K. For potassium, the latter represents a heat flux of 1240 MW m^{-2} of passage cross-sectional area[13]. In practice, choking would occur at a considerably lower mass flow due to the fact that mass leaving the evaporating surface has momentum in an opposite direction to that of the vapour stream. This will increase resistance to flow and produce a less full velocity profile. However, the limiting heat flux will still be extremely high, and the temperature difference between evaporating and condensing surfaces will be mainly due to the difference in vapour pressure caused by momentum plus centripetal pressure drop.

The temperature drop in a vapour chamber would most likely be less than that in a thermosyphon and the former device would only require a very small fraction of the quantity of working fluid. However, with both devices the very difficult problem of removing heat from the rotating system remains, and because the temperature is almost uniform along the blade span the worst combination of stress and temperature will occur at the root and thus, to achieve the same creep life a greater overall degree of cooling may be required than in an air cooled blade where the temperature increases towards the tip.

4.3 'Heat pipes' for stator cooling

The heat pipe [14,15,16] is a type of vapour chamber which can be used in the absence of a gravitational field since capillary action is used to circulate the working fluid. The walls of the chamber are covered by a porous membrane wetted by the liquid. When one wall is heated, heat is conducted through the liquid filled membrane and liquid evaporates from the menisci in the pores. Its place is taken by liquid drawn in by capillary action from other parts of the 'wick'. Liquid is thus pumped from the condensing to the evaporating region. The 'head' available is proportional to the curvature of the meniscus in a pore of the wick and is of course proportional to the surface tension (Figure 2).

The fluid must have a high thermal conductivity so that heat can be conducted through the liquid under the membrane without the surface superheat becoming high enough to initiate nucleate boiling. The total pressure drop (liquid side + vapour side + gravitational head) must not exceed that obtainable by capillary action. The only fluids worth consideration for the turbine cooling application are lithium, sodium and potassium. Although lithium has the largest latent heat and surface tension, its vapour pressure is too low at the temperatures envisaged at present (i.e., around 1200°K). As in the case of the rotating vapour chamber the heat carrying capacity increases rapidly with temperature (Figure 3) owing to the increase in vapour mass flow for a given pressure drop, though in a heat pipe, there is some decrease in the available 'driving' pressure due to decrease in surface tension and increase in viscous loss in the liquid stream with increasing mass flow. The condenser section must on no account be overcooled whilst appreciable heat fluxes are being supplied to the boiler section, otherwise the boiler may be 'starved' and heat pipe action will cease abruptly. Under these conditions the vapour may attain sonic velocity, in which case further reduction in the vapour pressure in the condenser will not increase the flow but will prevent liquid from returning to the boiler. Hence heat pipe systems should reject heat to a gas stream which is already fairly hot i.e., compressor delivery air as in scheme 5(b). Some care will be needed during the engine starting cycle. Also when the dynamic head of the vapour stream is high, droplets of liquid may be detached from the wick surface and entrained in the vapour stream. This increases the fluid circulation without benefit to heat transfer, and pressure drop increases until, again, the heat pipe fails.

Appendix I contains a tentative design study by the author for a first stage stator blade cooling system for use on a large engine with a turbine entry temperature of 1620°K. The heat pipe operates within the 'entrainment limit' as given by the formula in Reference 17.

4.4 Pumped closed circuit liquid cooling

Amongst the fluids which have been proposed for this application are hydrocarbons such as diphenyl[18], fluorinated hydrocarbons such as perfluorodecalin (which can be used as the working fluid in an auxiliary cycle) and of course liquid metals. Chemical stability requirements are more stringent for closed systems and the hydrocarbons have operating temperature limits around 400°C, not much in excess of those for fuels. However, the prospects for liquid metal cooling of static parts are very good and the use of electromagnetic pumps, which are being developed for high temperature use, will permit a hermetically sealed system to be designed. Appendix II gives some design details of a liquid sodium system in a simialr application to the heat pipe mentioned above. However, the system may be limited to non-aeronautical use by the difficulty of servicing a system where all the pipe joints are welded up. In the heat pipe system, each blade with its heat pipe and cooler is treated as a separate self-contained unit and so poses no dismantling problems.

5.0 Liquid metal technology

Some past attempts to use liquid metals for gas turbine cooling have failed due to chemical reaction between impurities, the liquid metal and the container. However a great deal of work has been done in recent years in the USA and in Britain in connection with nuclear power systems, in particular those for use in space and for fast reactors. Using the correct techniques it should be possible to obtain lives of thousands of hours from liquid metal devices. A molybdenum alloy-lithium heat pipe has operated for 3820 hr at 1767°K (Reference 14). The corrosion problem can largely be overcome by eliminating all traces of oxygen from the system and using 'getters' such as barium and zirconium. These also improve the wetting properties of the fluid. Another problem, however is that of mass transport. Although nickel based alloys are not readily attacked by alkali metals they are slightly soluble at high temperatures. In evaporating systems, this will cause a transport of material from the condenser section to the evaporator section. In non-evaporating systems metal will dissolve preferentially in the hotter parts of the system and be deposited in the cooler part. Refractory metals such as tungsten and molybdenum have very low solubilities in alkali metals but have very little oxidation resistance and are thus unsuitable for parts in contact with hot combustion gases. The problem could most likely be solved by using a refractory metal lining in a nickel alloy blade assembly. This could either be a fabricated liner with the nickel alloy cast around it, or a vapour deposited internal coating. This liner would also serve as a barrier against diffusion of alloying elements and adsorbed gases into the liquid metal. In heat pipes, it may also be necessary to remove all gases, since they may give rise to nucleate boiling[12].

6.0 Conclusions

Several ways in which liquids can be applied to turbine cooling have been described. In the rotor cooling system using primary coolant, the problem of removing heat from the rotor is still very severe. If liquid cooling is used, there is a sealing problem. With air cooling the problem is that of providing a large area heat transfer surface near the root. The use of fuel as heat sink to augment air cooling would seem more promising. the application of the 'heat pipe' to the cooling of stator blades and for equalising the temperatures of casings and stationary shrouds in order to minimise distortion seems well worth investigating. Both applications promise improved thermal efficiency by reducing the proportion of working fluid diverted for cooling purposes and reducing leakage losses.

REFERENCES

No.	Author's	Title, etc.
1	J. F. Barnes F. W. Armstrong	A brief review of thrust boosting techniques. (A paper produced for the 75th meeting of the G.T.C.C.) April 1965
2	J. D. Smith	Aviation turbine fuels. Factors affecting thermal stability. 'Shell' Research Limited, Report No. K.160, November 1964
3	J. D. Smith	Aviation turbine fuel thermal stability. The effect of nitrogen purging. 'Shell' Research Limited, Report No. K.159, May 1964
4	K. Ross	The specific heat of aviation turbine fuels in the liquid phase. 'Shell' Research Limited, Report No. K.161, November 1964
5	-	Selected values of properties of hydrocarbons. National Bureau of Standards November 1947
6	F. S. Stepka	Considerations of turbine cooling systems for Mach 3 flight. NASA TN D4491, April 1968
7	G. Egloff	Reactions of pure hydrocarbons Reinhold 1937
8	V. A. Ogale	On the application of the semi-closed thermosyphon system to gas turbine blade cooling. Thesis. Technological University of Delft, September 1968
9	F. J. Bayley G. S. H. Lock	Heat transfer characteristics of the closed thermosyphon. Trans. ASME Series C, Vol. 87, February 1965
10	H. Cohen F. J. Bayley	Heat transfer problems of liquid cooled gas turbine blades. Proc.I.Mech.E., Vol. 169, No. 20, 1955

REFERENCES (cont'd)

No.	Author(s)	Title, etc.
11	C. B. Jackson et al	Liquid metals handbook. Sodium (NaK) supplement. Atomic Energy Commission, Department of the Navy, Washington, D.C., July 1955
12	J. G. Collier P. G. Kosky	Natural convective boiling of the alkali metals - A critical review. AERE R.5436, September 1967
13	A. H. O. Brown	Sodium and NaK. The natural convection cycle with reference to turbine blade cooling. British Siddeley Engines Limited, Ref. GN 4257, February 1961
14	K. T. Feldman G. H. Whiting	The heat pipe. Mechanical Engineering, Vol. 89, No. 2, pp.30-35, February 1967
15	T. P. Cotter	Theory of heat pipes. Los Alamos Scientific Laboratory LA-3246-MS, March 1965
16	-	Proceedings of Joint Atomic Energy Commission/Sandia Laboratories Heat Pipe Conference. Sandia Corp., Alberquerque, N. Mex., October 1966
17	J. E. Kemme	High performance heat pipes. Los Alamos Scientific Laboratory LA-DC-9027. Presented at the 1967 Thermionic Conversion Specialist Conference Palo Alto Calif., 30th October, 1967
18	H. May	Investigations of forced convection liquid cooled gas turbine blades. ASME Journal of Engineering for Power, January 1965

APPENDIX I

Heat pipe n.g.v. cooling system

Turbine entry temperature	1620°K
Compressor delivery temperature	910°K
Total pressure	1.1×10^{6} N m^{-2}
Blade Reynolds number	10^{6}
Nusselt number	700
Chord	5.08 cm
Span	10.16 cm
Gas thermal conductivity	9.65×10^{-2} W m^{-1} °K^{-1}
Blade metal conductivity	27.4 W m^{-1} °K^{-1}
Gas mass flow per blade	1.455 kg s^{-1}
Blade outer surface temperature	1220°K
Heat flux per blade	6.35 kW
Evaporator surface temperature	1169°K
Degree of superheat at inner wall	20°K
Sodium latent heat	4.21×10^{6} J kg^{-1}
Sodium coolant mass flow	1.5 g s^{-1}
Coolant vapour pressure	1.172×10^{5} N m^{-2}
Area of vapour passage out of blade	1.26 cm^{2}
Surface tension	0.112 N m^{-1}
Velocity heat of vapour in pipe ($\frac{1}{2}\rho V2$)	2.85×10^{3} N m^{-2}
Let overall pressure drop be 2 x velocity head in vapour pipe, then liquid pressure drop is:-	2.85×10^{3} N m^{-2}
Making a 10 per cent allowance for additional pressure drops:- pore diameter is:	0.7×10^{-2} cm
Liquid viscosity	0.015 kg m^{-1} s^{-1}
Pipe length	15 cm

Required cross-sectional area of liquid return wick	1.03 cm^2
Entrainment limited heat flux	11.1 KW

The cooler fitted to each blade is a laminar flow finned tube matrix with the following characteristics:

Equivalent diameter of passages	1 mm
Airflow (10 per cent main flow)	0.145 kg s^{-1}
Flow area	3.7×10^{-3} m^2
Nusselt number	4.35
Heat exchange area (air side)	0.127 m^2
Fin efficiency	74 per cent
Air temperature rise	39°K
Mean matrix temperature	1100°K
Fin thickness	0.2 mm
Solidity of fin matrix	40 per cent
Weight of fin matrix	0.2 kg

APPENDIX II

Pumped liquid sodium system
(blade operating conditions similar to previous example)

Coolant flow 2 per cent of engine mass flow	29.1 gs^{-1}
Temperature rise of coolant	173.5°C
Flow passage in blade (flat section)	50 x 1.5 mm
Blade outside surface temperature (mean)	1220°K
Blade inside surface temperature (mean)	1180°K
Coolant inside surface temperature difference	4°K
Flow velocity in cooling passage	0.517 ms^{-1}
Velocity head ($\frac{1}{2}pV^2$) in cooling passage	100 N m^{-2}
Let total pressure drop in coolant circuit be equal to 10 x velocity head in blade passage and pump efficiency be 15 per cent then electrical pumping power is	0.259 W

FIG. 1

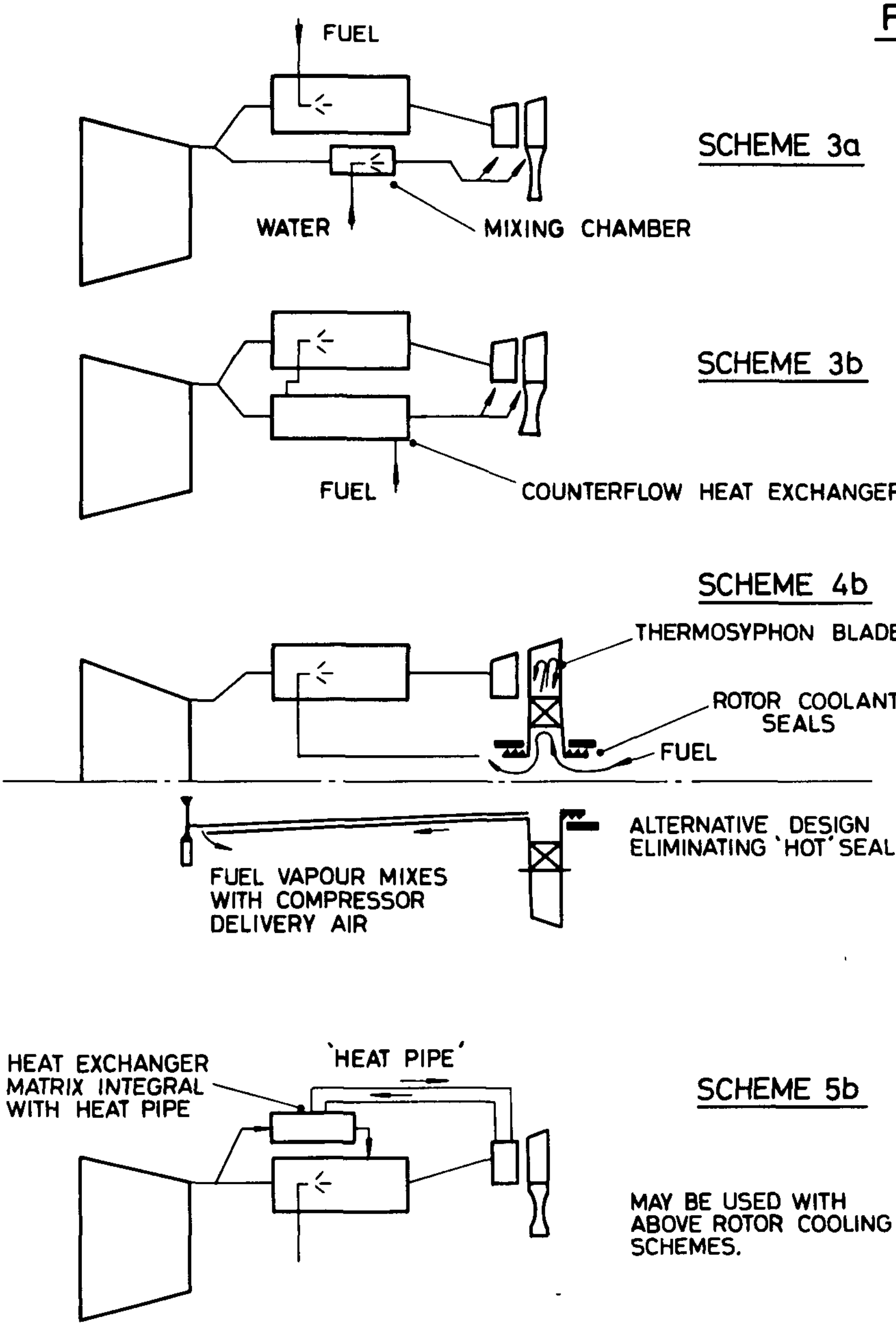


FIG.1 SOME COOLING SCHEMES

FIG. 2

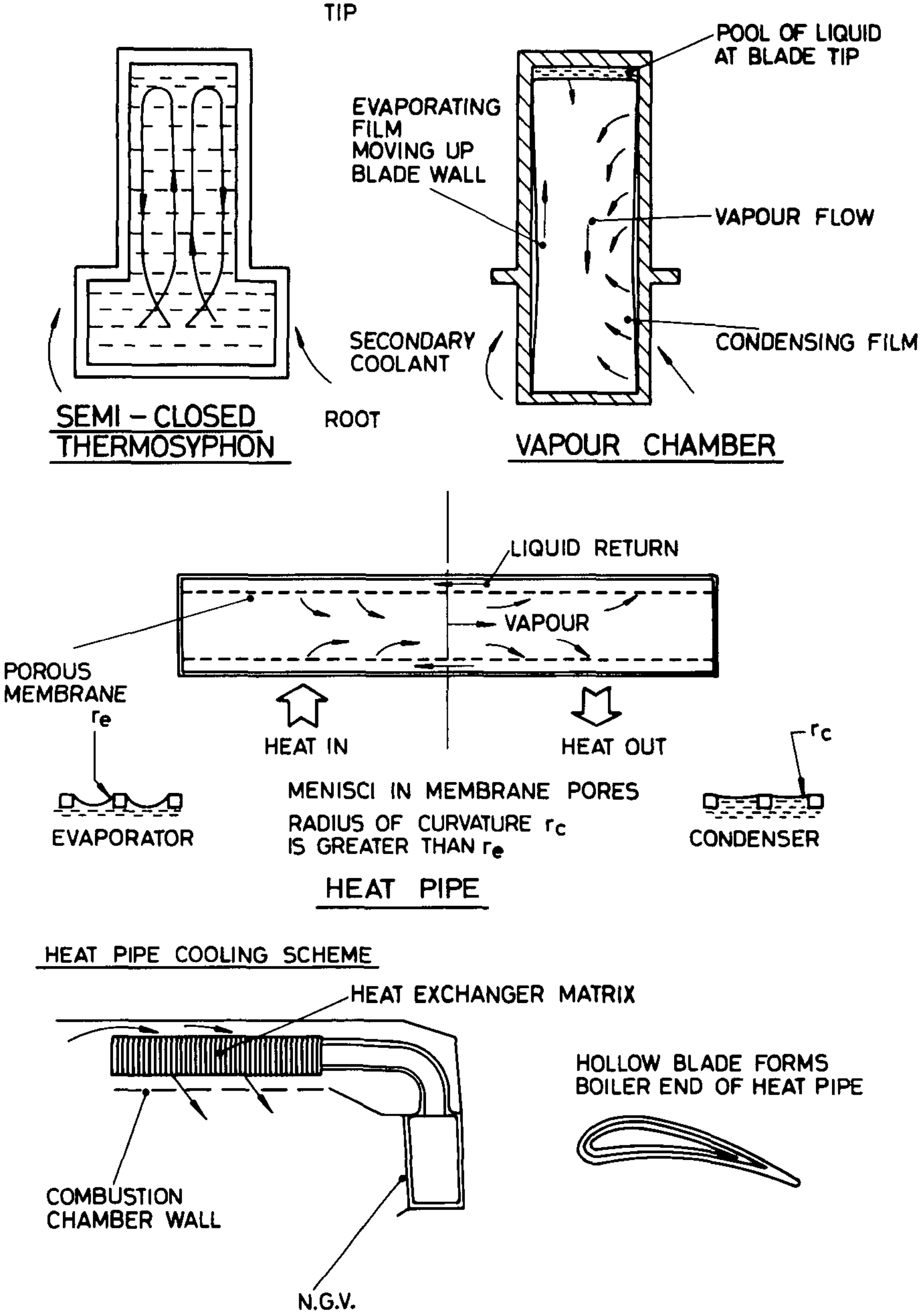
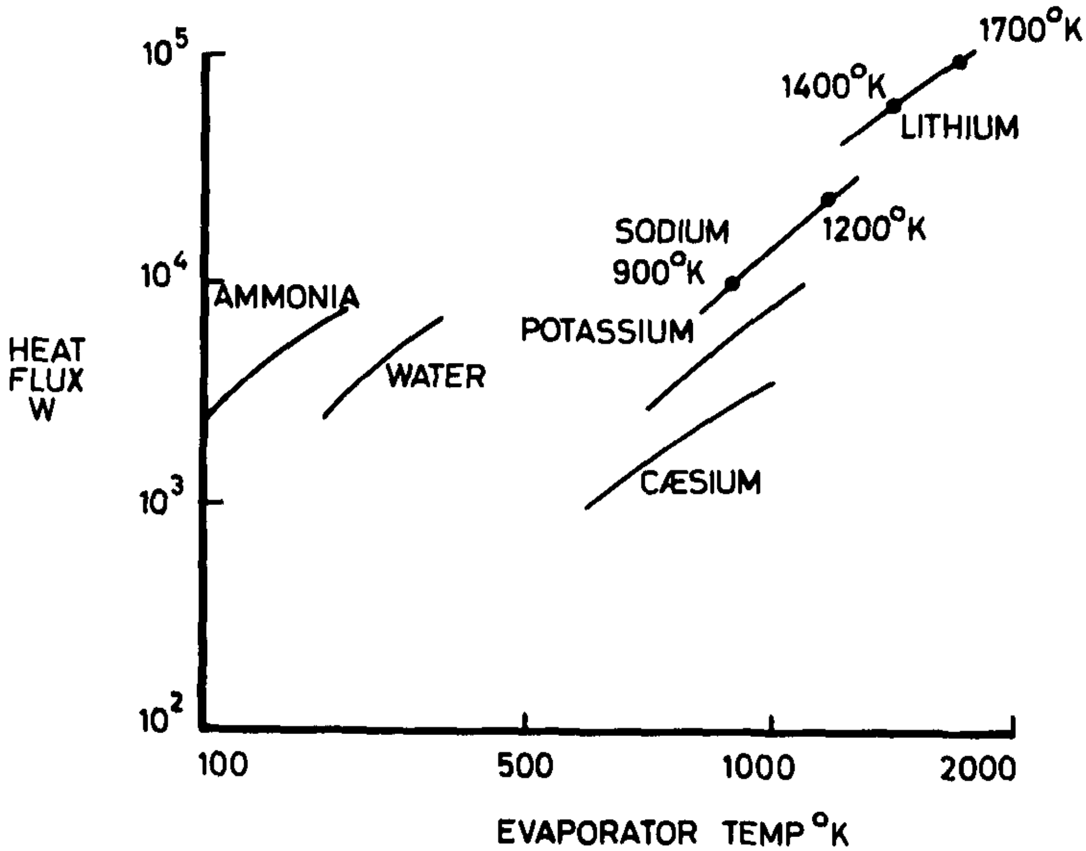


FIG.2 HEAT TRANSFER DEVICES

FIG. 3



THEORETICAL HEAT TRANSFER RATES FOR A CYLINDRICAL HEAT PIPE 1 FT. LONG X 1 IN. DIA. WITH A WOVEN MESH WICK USING VARIOUS WORKING FLUIDS.

REFERENCE 10.

FIG.3 HEAT PIPE PERFORMANCE

A.R.C. C.P. No.1127
November 1969
J.P. Edwards - NGTE Pyestock

LIQUID AND VAPOUR COOLING SYSTEMS FOR GAS TURBINES

This Note discusses the application of several types of liquid and vapour cooling systems to gas turbine blade cooling, with emphasis on systems suitable for continuous operation. The application of the 'heat pipe' concept to stator blade cooling is discussed in some detail and a tentative design study presented.

A.R.C. C.P. No.1127
November 1969
J. P. Edwards - NGTE Pyestock

LIQUID AND VAPOUR COOLING SYSTEMS FOR GAS TURBINES

This Note discusses the application of several types of liquid and vapour cooling systems to gas turbine blade cooling, with emphasis on systems suitable for continuous operation. The application of the 'heat pipe' concept to stator blade cooling is discussed in some detail and a tentative design study presented.

A.R.C. C.P. No.1127
November 1969
J. P. Edwards - NGTE Pyestock

LIQUID AND VAPOUR COOLING SYSTEMS FOR GAS TURBINES

This Note discusses the application of several types of liquid and vapour cooling systems to gas turbine blade cooling, with emphasis on systems suitable for continuous operation. The application of the 'heat pipe' concept to stator blade cooling is discussed in some detail and a tentative design study presented.

DETACHABLE ABSTRACT CARDS

C.P. No. 1127

Printed and published by
HER MAJESTY'S STATIONERY OFFICE

To be purchased from
49 High Holborn, London WC1V 6HB
13a Castle Street, Edinburgh EH2 3AR
109 St Mary Street, Cardiff CF1 1JW
Brazennose Street, Manchester M60 8AS
50 Fairfax Street, Bristol BS1 3DE
258 Broad Street, Birmingham 1
7 Linenhall Street, Belfast BT2 8AY
or through any bookseller

Printed in England

C.P. No. 1127

SBN 11 470315 9