HIGH TEMPERATURE GAS TURBINES FOR NAVAL USE

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Introduction

The position has been reached in the development of gas turbines for aircraft where, if further advances in performance are required, it will be necessary to increase the maximum operating temperature to an extent which will reduce the life of the engines by an unacceptable amount, unless measures are taken to limit the metal temperature of the turbine blades.

Manufacturers of high temperature materials have expressed the opinion that no major advances in the properties of turbine blade materials can be foreseen at present, particularly if long life is required.

From very early days the disks of gas turbines have been air cooled ; the technique of doing this has been improved to such an extent that it is now possible to use ferritic instead of austenitic materials for disks, and still leave a good factor of safety. This means that higher gas temperatures could be used if only the blading could withstand it. The outer turbine casings are air cooled, and as they are subject only to static stresses, no difficulty is foreseen in designing these for a higher gas temperature.

There remains the problem of designing combustion chambers for high gas temperatures. This is not a major one, and, in any case, these parts are of relatively simple construction, are inexpensive, and can readily be changed if need be.

What is true of aircraft gas turbines is even more so of naval gas turbines. Not only is there a need to improve the performance by using higher temperatures but also a need for a longer life between overhauls than is acceptable for aircraft engines. Turbine blade cooling, by reducing the metal temperature, can be used to achieve a better performance and a longer life, but these are only two of several advantages offered by this expedient.

Advantages Offered by Using Higher Gas Temperatures with Blade Cooling

These can be summarized as follows :—

(*a*) Improved fuel consumption.

(*b*) Reduced engine dimensions.

(*c*) Reduced sectional areas of intakes and uptakes.

(*d*) Less strategic material required.

(*e*) Increased life.

These are not all necessarily achievable in the same engine.

To give an indication of the improvements to be expected in full power fuel consumption, engine dimensions, and areas of intakes and uptakes by the use of higher gas temperatures, FIGS. 1 and 2 show the results obtained from calculations for an aircraft propeller gas turbine. These are not absolute values, as insufficient allowance has been made for losses. They do, however, show trends for a type of engine, which when fitted with a separate turbine to drive the propeller, is similar to the type at present in use in naval craft.

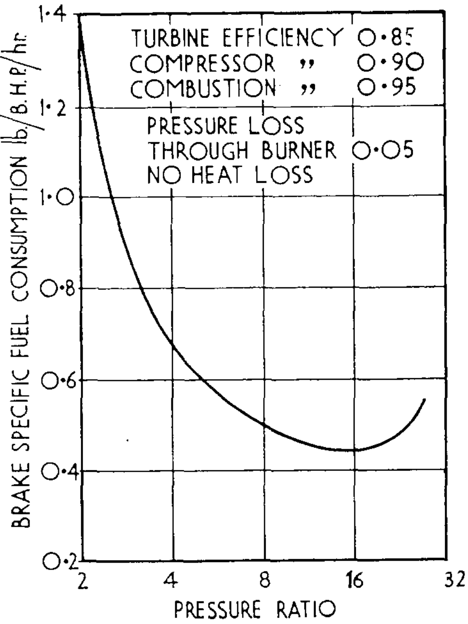

FIG. 1—BRAKE SPECIFIC FUEL CONSUMPTION/PRESSURE RATIO AT TEMPERATURE OF 837°C

FIG. 1 shows how the fuel consumption at a constant gas temperature varies with pressure ratio. FIG. 2 shows the improvement in fuel consumption with increased gas temperature for constant pressure ratios of 7 and 8, and also when using the optimum pressure ratio for the particular gas temperature.

It will be clear from these figures that a higher gas temperature implies a higher pressure ratio and the latter should be as near to the optimum as possible, or somewhat higher (to give better performance at part load).

It will be seen from FIG. 1 that increasing the pressure ratio beyond 32 in these calculations pays decreasing dividends, and this may be about the upper limit of pressure for simple gas turbines, even assuming that temperatures of the order of 1,500–1,700°C can eventually be used.

FIG. 3 shows how the specific output improves with increasing gas temperature. The mass flow of air required to develop a certain power from a gas turbine is inversely proportional to the specific output, that is the s.h.p./lb of air/sec. It follows, therefore, that the higher the specific output, the smaller the dimensions of the engine and the sizes of openings required in decks for air intakes and exhausts.

On the question of strategic materials ; if a temperature drop of 200°C (360°F) in blade metal temperatures could have been achieved in *R.M.60* it would have been possible to use a ferritic material containing only very small percentages of strategic elements instead of the highly strategic alloy Nimonic 90, which consists almost entirely of nickel, chromium and cobalt.

The life of the turbine blades in usually assessed on the basis of the time required to produce a certain amount of permanent strain (creep) at the designed working stress and temperature. FIG. 4 shows the time required to produce 0·1 per cent creep strain in Nimonic 90 at various stresses at temperatures of 750°C (1,382°F) and 815°C (1,499°F). It will be seen that if the tensile stress in the blade is 10 tons/sq inch, reducing the metal temperature from 815°C to 750°C prolongs the life from 100 to 3,000 hours.

There are, of course, disadvantages associated with turbine blade cooling and these are discussed in the following paragraphs, which describe various methods of cooling.

METHODS OF COOLING TURBINE BLADES

There are five principal methods of cooling the blades of marine gas turbines :—

(*a*) Air passing through the blades.

(*b*) By generating steam within the blades.

(*c*) Direct water spray.

(*d*) Internal water cooling by forced convection.

(*e*) A liquid coolant inside the blade with water as a secondary coolant.

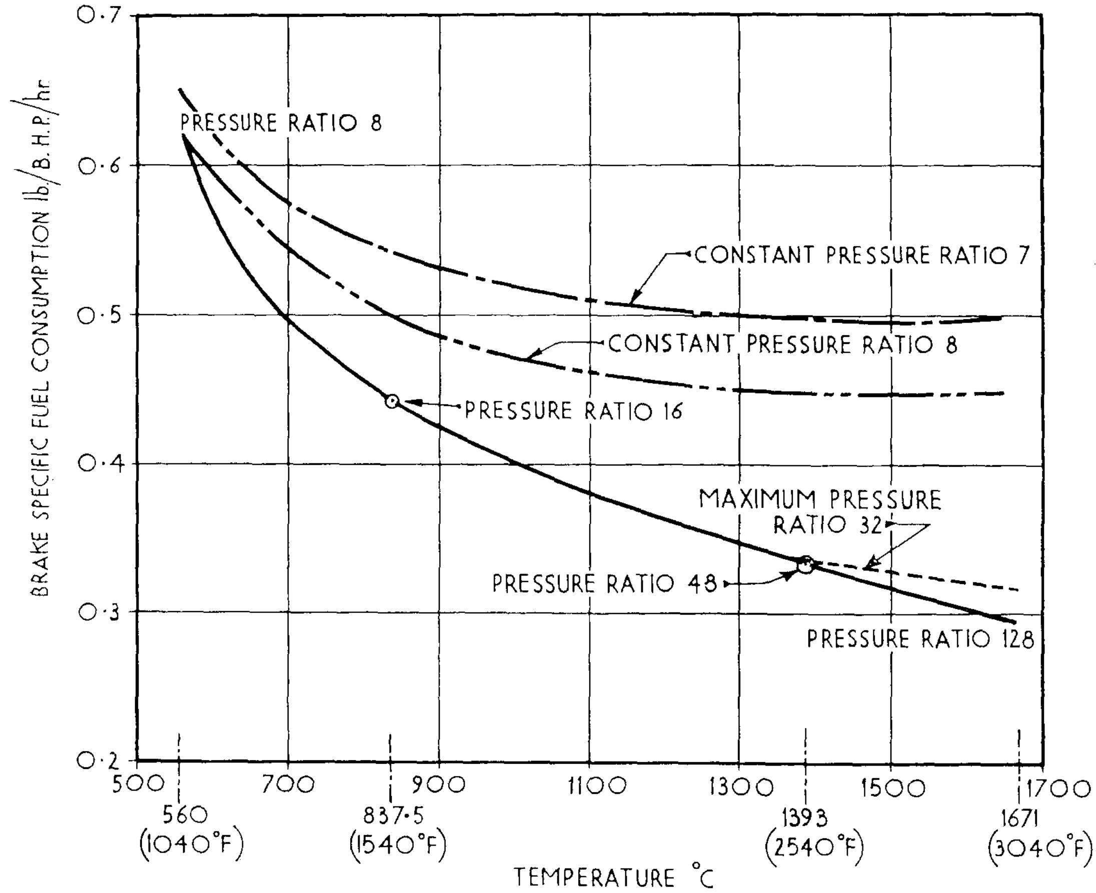

FIG. 2—BRAKE SPECIFIC FUEL CONSUMPTION/TEMPERATURE WITH OPTIMUM PRESSURE RATIO

Air Cooling

FIG. 5 shows one method of air cooling the blades in an experimental high temperature turbine at the National Gas Turbine Establishment. Many mechanical variations of this method are possible but all are the same in principle.

It consists of bleeding air from a stage in the compressor where there is sufficient pressure to allow it to enter the gas stream after passing through the blades. The air is usually passed from the compressor to the bore of the turbine rotor and through radial passages to the blades. A separate lead supplies air to the nozzles.

To ensure high heat transfer rates and even cooling over the blade section it is essential that the axial holes be numerous, of small diameter or width and sectional area, and distributed around the profile in a manner which will ensure effective cooling of the hottest part of the blade, i.e., the leading and trailing edges. Disadvantages are the high cost of manufacture and the possibility of the clogging of the holes with dirt. Air filters and coolers may be necessary.

The compressed air that is used for cooling involves a loss of work in the cycle with a consequent reduction of performance from the optimum without cooling, if the same maximum gas temperature is used. FIG. 6 shows the estimated performance of a simple gas turbine with separate power turbine employing a maximum gas temperature of 800°C (1,472°F) and 1,000°C (1,832°F) and a pressure ratio of 7 : 1 (not the optimum for maximum efficiency at either temperature) with varying percentages of the total mass flow for air cooling.

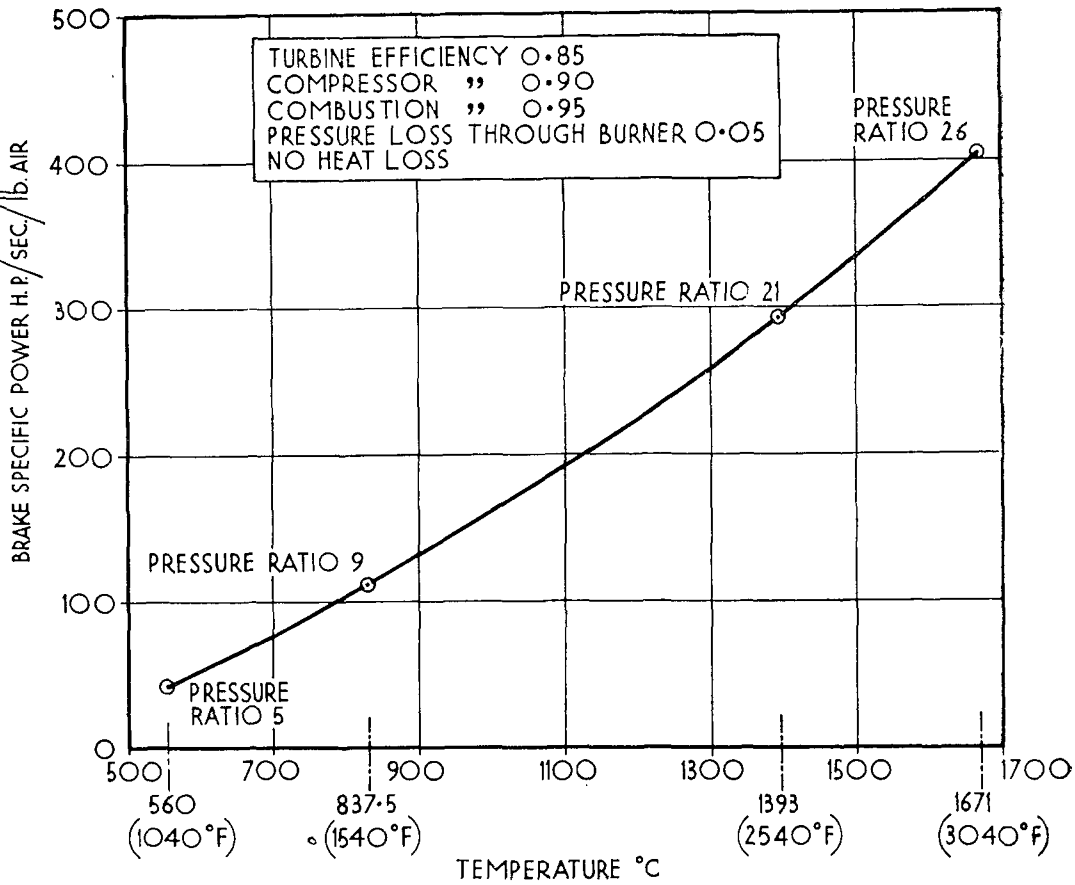

FIG. 3—OPTIMUM BRAKE SPECIFIC POWER/TEMPERATURE

The difference in performance for the two operating temperatures is most marked but would have been more so had the optimum pressure ratio for each gas temperature been used in the calculations. Also shown in FIG. 6 are the specific outputs of the two engines operating at 800°C and 1,000°C. It will be seen that the mass flow required for the 800°C engine is greater than that required for the 1,000°C engine for the same power output, and the linear dimensions will vary as

$$\sqrt{\frac{1}{\text{specific horsepower}}}$$

With regard to the actual quantities of air required, the N.G.T.E. tests have shown that a cooling air flow of 2 per cent of the mass flow permits of an increase of gas temperature of 270°C (486°F) in the blades of a single stage turbine.

The work of other investigators, however, indicates that a cooling air flow of 2·5 per cent will permit of an increase in gas temperature of only 110°C (198°F) in a single stage turbine. For an increase of 160°C (288°F) in gas temperature 3·0 per cent cooling air is required. The difference in the results can be attributed to the designs of the cooled blades. In the latter work the air passages are of comparatively large sectional area whereas in the N.G.T.E. design the air passages are very small and a higher heat transfer coefficient is obtained.

Assuming, therefore, that 2 per cent of cooling air is required to permit of an increase of gas temperature from 800°C to 1,000°C in a single stage and that it

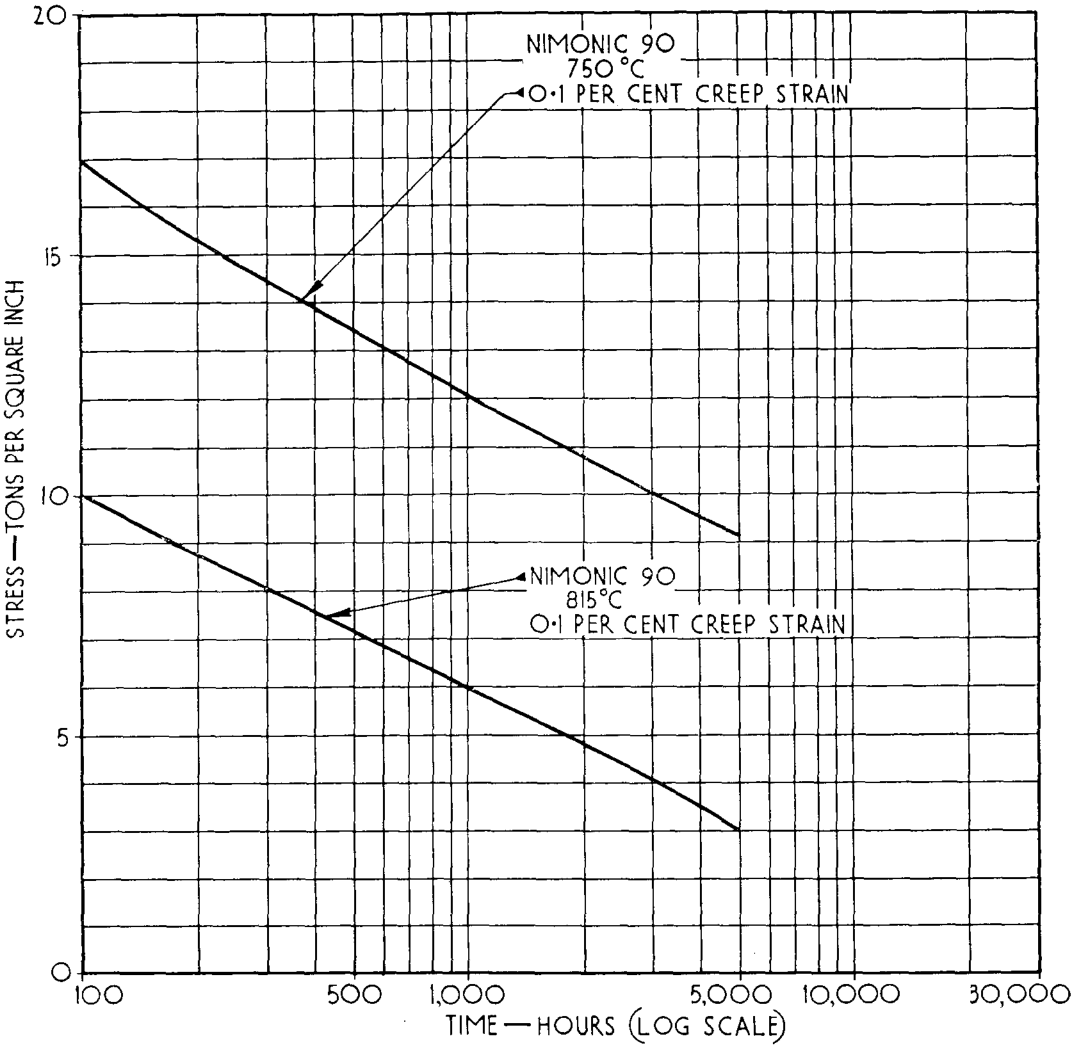

FIG. 4—CREEP STRAIN/TIME CURVES FOR NIMONIC 90

will be necessary to cool two stages, then a total quantity of 4 per cent will be required for turbine blades. On this basis, and assuming that another 2 per cent will be required for cooling ducts, the performance and specific output of the uncooled engine (no cooling air) operating at 800°C can be compared with the figures for the 1,000°C engine operating with 6 per cent cooling air.

From FIG. 6 :—

	Thermal efficiency per cent	*Specific horsepower s.h.p./lb air/sec.*
800°C engine uncooled	20·0	84·0
1,000°C engine cooled	23·1	117

The above figures are for a pressure ratio of 7 : 1. Had the optimum pressure ratio been used for each operating temperature it is estimated that the thermal efficiency would have been approximately 22·0 per cent for the 800°C uncooled engine and 28·0 per cent for the 1,000°C cooled engine.

One of the drawbacks of the simple cooled engine is that the air supply is always on, and at low load when the gas temperature is well down, cooling air is not required. This means that the low load performance will be relatively inferior to that of the uncooled engine although the actual fuel consumption

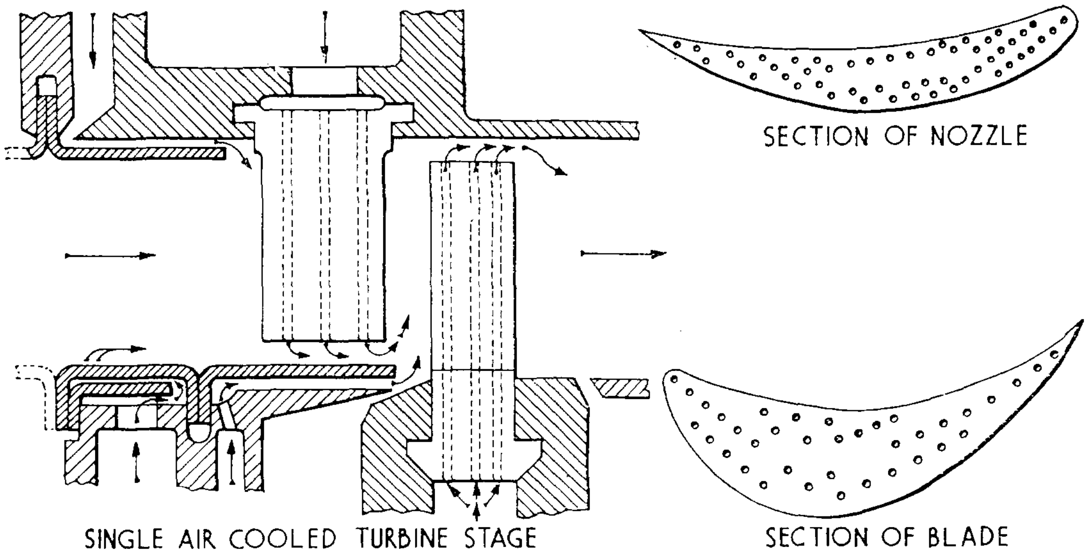

FIG. 5—AIR COOLED TURBINE STAGE

may be about the same or only slightly better, due to the increase in efficiency of the cooled engine at full power.

As an example of this, calculations have shown that increasing the gas temperature of a complex engine employing the same cycle as the *R.M.60* by 100°C (180°F) and using 3 per cent cooling air, the efficiency at full power is increased by 6 per cent and at $7\frac{1}{2}$ per cent power by only 2 per cent.

An installation employing a number of simple gas turbines would show to better advantage because one engine used for cruising would be operating at a higher percentage of power than a single complex engine. It may be possible to control the cooling air at part load but this will involve mechanical complication.

The use of porous material for air cooled turbine blades has been suggested, but the mechanical properties of this material are poor. New developments may, however, bring this idea to the fore again, particularly for turbine nozzles.

Steam Cooling

In 1945 a technical team investigating developments in Germany during the war discovered that Dr. Schmidt had built and tested a gas turbine with a water cooled rotor and steam cooled blading, the steam being generated in small diameter holes drilled through the blades and to which the water in the rotor had access. The advantages claimed for this design were :—

(*a*) It could operate at much higher gas temperatures, 1,200°C (2,192°F) was suggested, resulting in a much higher efficiency of plant.

(*b*) Ferritic materials could be used for rotor and blading, due to the low metal temperatures obtaining with water and steam cooling.

(*c*) As a result of using higher gas temperatures the air requirements are less, thus reducing the sizes of air and exhaust openings.

(*d*) The steam generated in the blading could be used for driving auxiliaries.

The Department expressed interest in this development and as a result the Schmidt turbine was brought over to England for testing by N.G.T.E.

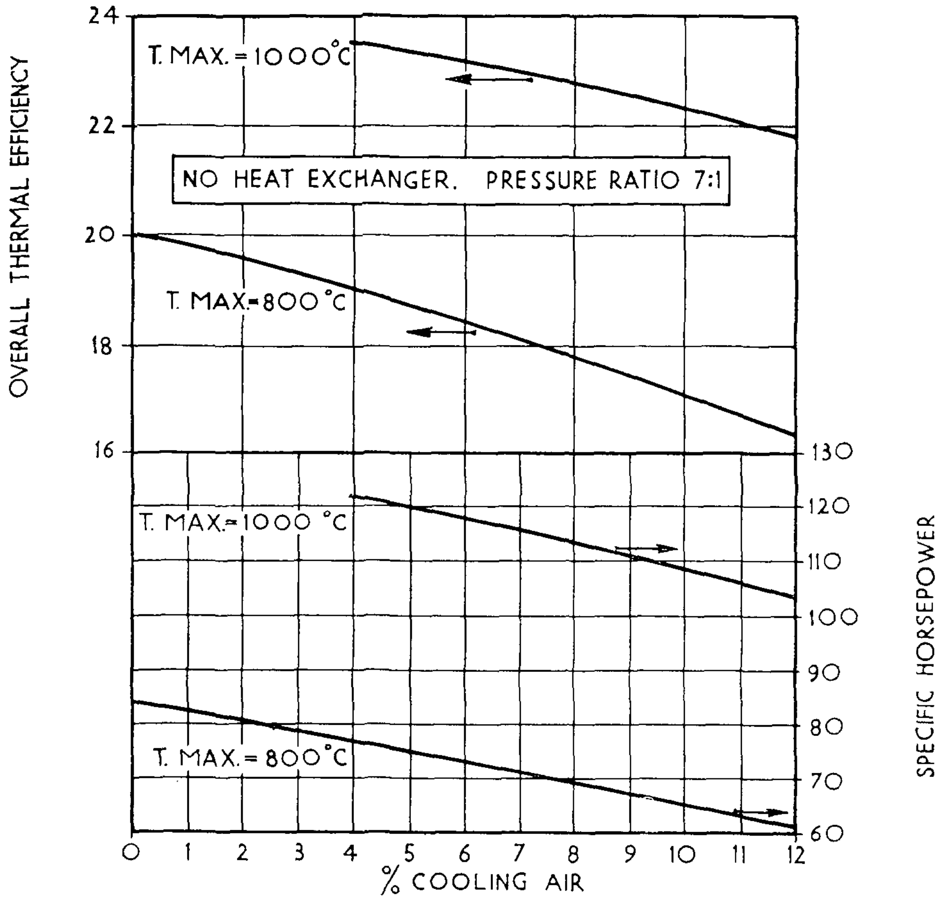

FIG. 6—ESTIMATED PERFORMANCE OF SIMPLE COOLED GAS TURBINE/COOLING AIR FLOW

The results of the tests were disappointing, the principal trouble being excessive vibration, the cause of which was not established, although it was thought to be associated with the water in the rotor. So far as known the turbine has never operated above 700°C (1,292°F). Control of the water level in the rotor also proved difficult.

A thermodynamic investigation has been carried out and it has been shown that only about 10 per cent of the heat given up by the gas to the water can be utilized in a steam turbine, due to the loss associated with the latent heat when the steam is condensed. There is also the problem of using a quantity of steam which varies with the power of the turbine.

Another disadvantage of this system of cooling is that distilled water must be used in order to prevent clogging of the small diameter holes in the blading. After a full investigation it was concluded that the Schmidt system extracts too much heat from the gas and introduces large negative reheat in the turbine which results in a loss of much of the efficiency otherwise gained from the high gas temperature.

Despite these findings it should be pointed out that the higher gas temperature reduces the dimensions of the turbine and compressors, as with air cooling. The additional complication of condensing equipment and distilled water requirements would, however, largely offset the advantages of reduced compressor and turbine dimensions with this system of cooling.

FIG. 7 shows a diagrammatic arrangement of the Schmidt cooling system.

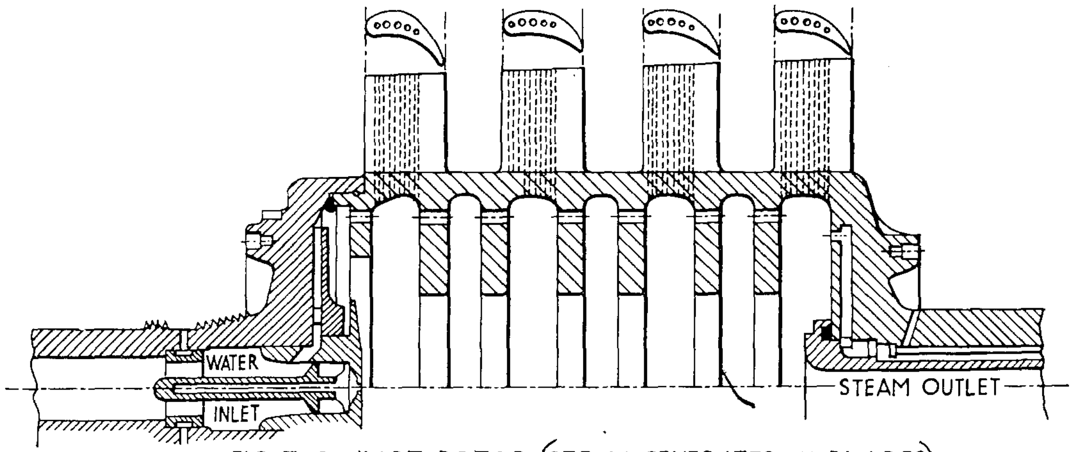

FIG. 7 SCHMIDT ROTOR (STEAM GENERATED IN BLADES)

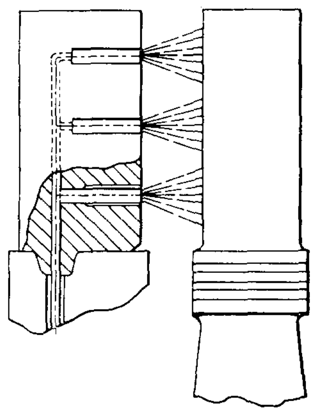

FIG. 8 SPRAY COOLING

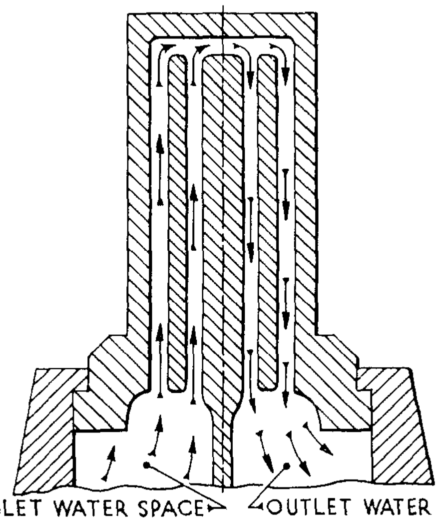

FIG. 9 FORCED CONVECTION

Spray Cooling

A series of tests were carried out at the N.G.T.E. using a Power Jet W2/700 aircraft gas turbine to determine what degree of cooling of turbine blades can be effected by direct water spray from the nozzles on to the moving blades. The arrangement is shown in FIG. 8.

The system was effective in reducing the mean blade temperature by some 300°C (540°F) but the quantities of water required were large. It was estimated that for a metal temperature drop of 300°C in a marine gas turbine of 30,000 s.h.p., the water quantities required may be as much as two to three times the fuel flow. As clean fresh water is required and all the water is lost no further interest has been taken in this method of cooling for naval gas turbines.

Water Cooling

This arrangement employs a system of forced convection in which water is forced through a series of passages in the nozzles and moving blades, as shown in FIG. 9.

Unpublished work by N.G.T.E. has shown that if the blades are to be cooled by water (and not by superheated steam) a very much reduced average blade temperature, as compared with air cooling, would have to be accepted. This would be of the order of 400°C (752°F) for the moving blades in which high water pressures can be created due to centrifugal force, and 150°C(302°F) for the stator blades in which the water pressure will be little above atmospheric.

To attain these low temperatures without creating steam, large quantities of water would be required. In addition, the heat flow into the blades would produce a large negative reheat in the turbine, with a lowering of turbine efficiency, and extract a great deal of heat from the gas, as in the case of the steam cooled design.

A theoretical comparison of the thermal efficiency of a water cooled and an air cooled gas turbine plant operating at 1,200°C and employing heat exchange and intercooling gave the following results :—

Type of cycle	Output per unit air mass flow s.h.p./lb/sec.	Thermal efficiency per cent
Uncooled	194·7	38·9
Aircooled	159·8	33·8
Watercooled Turbine stator blades at 150° C	170·3	31·5
Watercooled Turbine stator blades at 400° C	173·1	32·6

It will be seen that water cooling offers an improvement in specific output compared with air cooling but at some sacrifice of fuel consumption.

Liquid Cooling with Water as a Secondary Coolant

As a result of the investigations into the Schmidt system of steam cooling it was agreed that some form of liquid cooling that did not involve the formation of steam should be investigated.

At that time Pametrada were interested in some such form of cooling and put forward a proposal for cooling the moving blades employing two coolants. One, the primary coolant, to be inserted in closed passages in the turbine blades and the other, water, to pass through the rotor and around the blade roots. A liquid metal such as sodium was envisaged for the primary coolant. A contract was subsequently placed with Pametrada for the construction and test of a liquid cooled turbine employing these principles.

In order to obtain heat transfer data, Pametrada have constructed a rotating test rig incorporating two tubes representing dummy blades which are heated electrically. At the same time the manufacture of a turbine designed to operate at a temperature of 1,200°C is proceeding. This has brought to light some of the manufacturing problems associated with a design of this sort. The actual potentialities of this scheme cannot be evaluated until information is available from the test rig. This may not supply all the answers as the mechanical construction may place a limit on the speed at which it can operate.

This scheme may offer advantages over a straight water-cooled design in so far as higher blade temperatures may be possible and reduced water quantities may be required. The system cannot, however, be used for the stator blades, for which ceramic materials are proposed in the first design. The alternative is air-cooled stator blades.

In a paper recently presented to the Institution of Mechanical Engineers, Dr. Brown has shown that a liquid cooled turbine with a secondary coolant offers a slightly more efficient design than one with air cooled rotor blades. In an engine with a cycle similar to *R.M.60* but operating with a maximum temperature of 1,200°C, pressure ratio 15, the results were as follows :—

	Thermal efficiency per cent.		Air rate lb per s.h.p./hr.	
	With 0·75 heat exchange	Without heat exchange	With 0·75 heat exchange	Without heat exchange
Air cooled	37·8	31·1	16·0	15·3
Liquid cooled	40·9	32·3	15·3	14·7

Conclusion

It will be seen that the greatest emphasis is placed on air-cooling and the only alternative which appears practicable for naval gas turbines is the Pametrada scheme.

The one outcome of the use of higher gas temperatures which is assured, is a reduction in engine dimensions because of the increase in specific output. The theoretical gains in efficiency at low loads have yet to be established, but there is no doubt that gains will result as temperatures are increased, particularly if pressure ratios are increased to suit higher temperatures.

Air cooling is at present being employed in the turbine blades of aircraft engines on bench tests and flight engines are being designed to use it. It is recommended for future designs of naval gas turbines.

Liquid cooling has yet to be proved and the effectiveness of this system will not be established until the work at Pametrada is further advanced. It is unlikely that results, sufficient to form the basis of design of a gas turbine employing this system, will be available for some time. It appears to have the advantage over air cooling in that better control at part load will be possible.

The use of higher gas temperatures will create problems other than in the turbines. Reference has been made to difficulties with the combustion chambers. The temperature of the exhaust will also go up and this may present problems in ships, particularly in the case of auxiliary gas turbines fitted between decks. These are not major problems, however. In engines fitted with heat exchangers, a higher exhaust temperature may necessitate the use of more strategic materials in these components and increase the difficulties of dealing with expansion and distortion. This points to the need to dispense with heat exchangers.

No reference has been made to work in other countries on high temperature gas turbines, but all work in the United States runs parallel to that in this country, with emphasis on air cooling.

FIGS. 1, 2 and 3 are derived from a paper by Oscar W. Schey to the American Society of Mechanical Engineers in 1948.