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NEW TECHNOLOGY IN AN ALL-NEW HELICOPTER ENGINE. THE ENGINE DESIGNER'S ANSWER TO THE REQUIREMENTS OF TODAY

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Abstract

Although low specific fuel consumption is the main requirement in a modern helicopter engine, other requirements, such as low cost, weight and volume, prohibit elaborate and complex solutions. Substantiated by component development results, it is shown that the application of advanced technologies in the aerodynamics, materials and manufacturing fields to a newly designed engine with a basically simple configuration best satisfies the requirements of today.

1. Requirements

With the aircraft industry and aircraft owners still suffering from the unprecedented fuel price increase of more than 600% between 1970 and 1980 (Fig. 1) one could argue that there can hardly be any substantial requirement for a new helicopter engine other than lowest possible fuel consumption. Since 1982, however, fuel prices have fallen again and this reminds us that besides fuel consumption, there are still some other requirements which have to be met, and which influence the direct operating cost or life cycle cost of an aircraft.

Most modern helicopters are equipped with two engines for safety reasons. The main requirements for the engines - arising from this configuration - can be summarized as follows:

- Low specific fuel consumption, especially at part load, the prevailing operationg condition of the engines

- Ability to provide extra power well above the take-off rating in case of an engine failure in the "dead man's zone" (emergency rating)

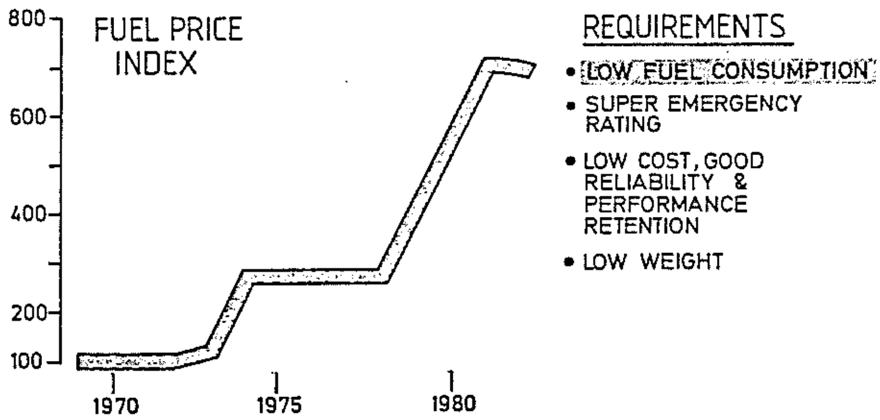


Fig. 1 Requirements for modern helicopter engines

- Low manufacturing and maintenance costs, good reliability and performance retention.

- Low weight and small engine envelope.

2. Engine Layout

Although low fuel consumption has the highest priority, the last two requirements cannot be ignored. They prohibit elaborate solutions with special systems and additional equipment which might be considered necessary for fulfilling the first two requirements without compromise, because of the increasing complexity, weight and volume of the engine.

The only reasonable solution therefore is to

- Strive for an essentially simple engine layout which represents the best compromise in fulfilling the different requirements;

- Apply latest research and development results in aereodynamics and thermodynamics, metallurgy and manufacturing technology to engine component design.

Thus the engine layout which forms the basis for the design of advanced turboshaft engine components at MTU features the following design characteristics:

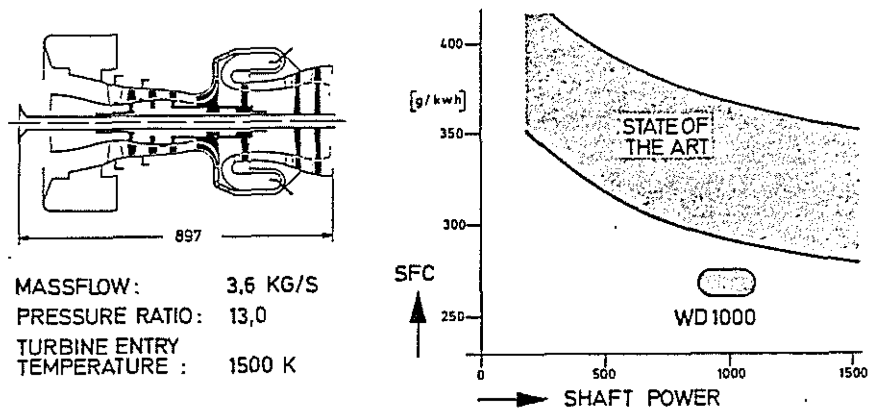


Fig. 2 Turboshaft Engine Demonstrator WD 1000

- An axial/centrifugal compressor with three axial stages and a compression ratio of 13:1;

- A reverse-flow annular combustion chamber

- A single-stage, air-cooled, transonic compressor turbine, operating at a turbine entry temperature of 1500 K at the aerodynamic design point; and

- A two-stage power turbine, directly connected to the power take-off at the front end of the engine by a long shaft running right through the hollow gas generator rotor.

The compressor, the combustion chamber and the compressor turbine are presently being developed at MTU as part of an advanced components development programme. A gasgenerator - designated the GNT1 - will be built from these components and tested, beginning next year (Ref. 1). By adding a power turbine the gasgenerator can be converted to a complete engine demonstrator - designated the WD 1000 - with a power output and fuel consumption as shown in Fig. 2.

The choice of a single-stage, gas generator turbine at a compression ration of 13:1 seems to be one of those compromises, mentioned above, between cost and weight, on the one hand, and performance, on the other. I will comment on this in more detail later on. One advantage of a single-stage gas generator turbine for engine performance becomes evident already from the schematic view in Fig. 2; namely, the short and rigid gas generator rotor, which enables blade tip clearances and gaps in rotating seals to be kept within tight tolerances, minimizing internal parasitic losses and supporting performance retention.

This is one reason why - in spite of the simple engine layout - the WD 1000 engine will have a distinctly better SFC than state-of-the art engines with power outputs of up to 1,500 kW.

Other reasons for the good fuel consumption will become clear in the following description of some component design features and development results.

Fig. 3 shows a cross-section of the WD 1000 engine and summarizes its design characteristics.

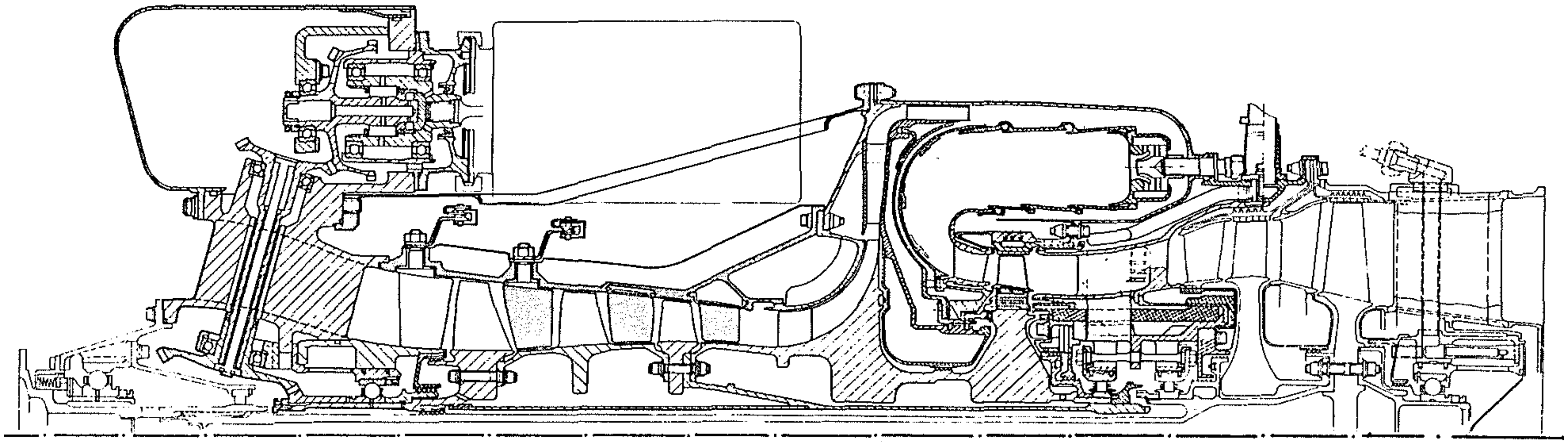
3. Compressor

3.1 Clearance Control

Clearance control and performance retention were main requirements in designing the WD 1000. As mentioned above, the short rigid gas generator rotor is one design feature which helps fulfil these requirements. The casing structure in the compressor area is another important feature.

As shown in Fig. 3, we have chosen a double-shell structure. The inner shell forms the flow path and accomodates the compressor vanes, whereas the outer shell transmits structural loads between the inlet casing and the combustion chamber casing, and controls the axial distance between the gas generator thrust bearing in the inlet casing and the impeller shroud ring and thus controls the blade tip clearance at the impeller exit, which has a big influence on compressor efficiency.

Because it lies on a large diameter, the outer shell can absorb manoeuvring loads without serious distortion, and it thus meets the requirement for performance retention. Furthermore, because it remains relatively cool, the outer shell is more suited for assisting to maintain narrow blade-tip clearances throughout the operating range of the centrifugal compressor than the inner shell. Maintenance of the required clearance becomes considerably easier when materials with a low coefficient of thermal expansion - preferably less than that of the titanium material used for the rotor (approx. $9 \cdot 10^{-6}/^{\circ}C$) are used (and this applies to both shells). For this reason,

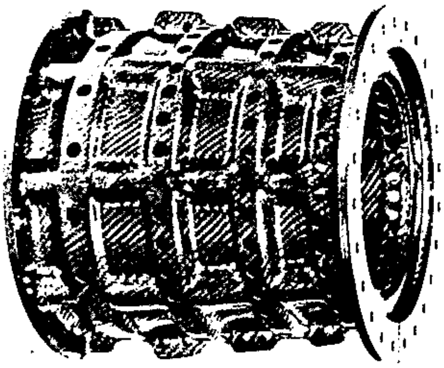


DESIGN CHARACTERISTICS

- ADVANCED AERODYNAMICS
- NEW MATERIALS & MANUFACTURING TECHNOLOGY
- COOLING AIR ECONOMY & CLEARANCE CONTROL
- SIMPLE BASIC CONFIGURATION

Fig. 3 Turboshaft Engine Demonstrator WD 1000

casings in CFRP material were included in the development work from an early stage. An experimental version of the inner shell is shown in Fig. 4.
In addition to exhibiting a lesser thermal expansion than a steel casing, a CFRP casing has a considerable weight advantage.



ADVANTAGES

	CFR	STEEL
• WEIGHT:	1,2 KG	5,9 KG
• THERMAL EXPANSION:	$4 \cdot 10^{-6}/K$	$11 \cdot 10^{-6}/K$
• MACHINING:	LOW	HIGH

MATERIAL: CARBON FIBRES IN HIGH TEMPERATURE RESIN

Fig. 4 CFRP-Compressor Casing

3.2 Aerodynamics

Aerodynamic development work on the compressor was started early on (Ref. 2). Fig. 5 shows the complete compressor rotor.

Fig. 5 Compressor Rotor

The airfoil profiles of the three axial rotors are optimized for each stage according to the relative flow velocity, that is supersonic in the first and second stage and supersonic (tip) - transonic (hub) in the third stage.

Initially the axial stators were equipped with NACA profiles, but in a second test series these were replaced by "controlled diffusion" profiles.

The expected reduction in the losses of cascades in which the incoming flow is transonic with the latter profile, which is capable of decelerating local supersonic regions at the design point to below the speed of sound surge-free, was confirmed, as shown in Fig. 6. As a consequence, the efficiency of the axial-flow compressor has been considerably improved.

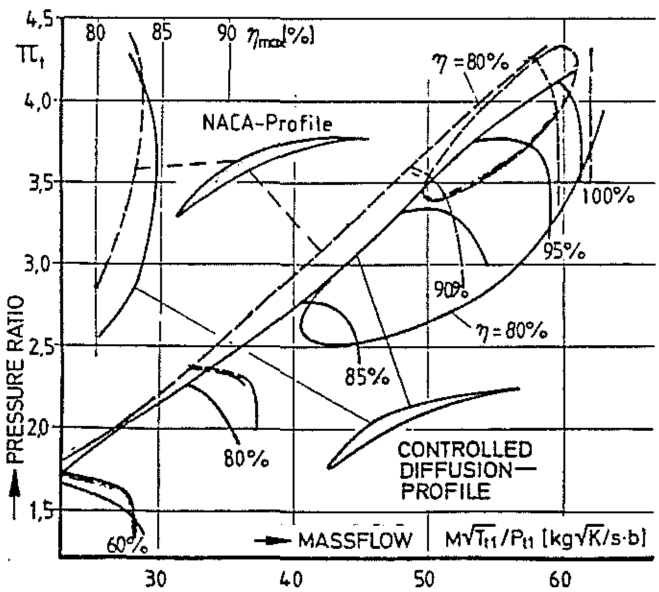


Fig. 6 Axial Compressor with Different Stator Profiles

An improvement in the efficiency of the centrifugal compressor is to be gained by diverting the casing-side boundary layer at the impeller exit. The highest gain in efficiency is obtained when about 2% of the mass flow is blown off.

This improvement has a significant effect especially at partload, thereby fulfilling one of the requirements mentioned above which calls for low part load fuel consumption.

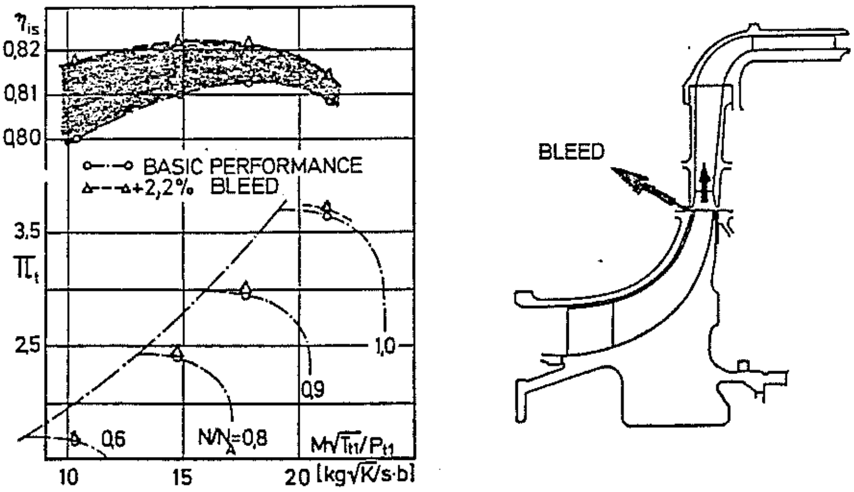


Fig. 7 Radial Compressor Improvements

Development of the axial/centrifugal compressor is not yet complete. Nevertheless, a high technological standard has already been attained and the feasibility of the desired low fuel consumption of the WD 1000 has been confirmed.

4. Combustion Chamber

A reverse-flow annular combustion chamber has been chosen for the WD 1000 (Fig. 8). This type of combustion chamber offers several advantages for a small engine:

- It allows close coupling between the compressor and the compressor turbine, resulting in a short and rigid gas generator rotor with the advantages already mentioned.
- It minimizes the overall length of the engine without penalties to the engine diameter, because it fits into the annulus defined by the outer diameter of the centrifugal compressor diffuser and the outer diameter of the compressor turbine.
- It can be removed from the engine for servicing without disassembling the gas generator rotor.

The reverse-flow annular combustion chamber, however, has one big disadvantage, namely its large flame-tube wall area, which needs cooling, compared to the relatively small mass flow that can provide cooling.

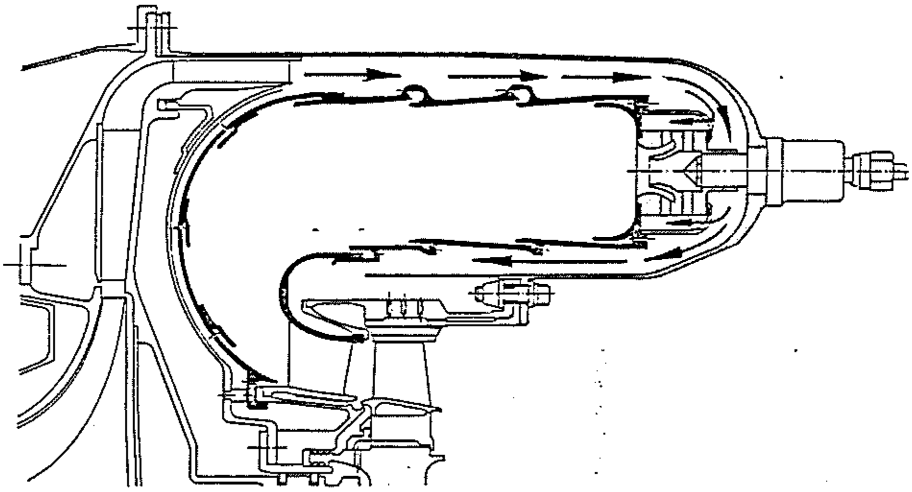


Fig. 8 Combustion Chamber

We have chosen this type of flame-tube despite this known disadvantage. The problem of keeping the flame-tube wall temperatures within acceptable limits is approached in two ways (Ref. 3).

1. By the introduction of efficient wall-cooling schemes in conjunction with thermal-barrier coatings.

2. By an improved fuel injection and flame stabilization system, which concentrates the burning process in the centre of the primary zone, thus avoiding contact of the hot burning products with the flame-tube walls.

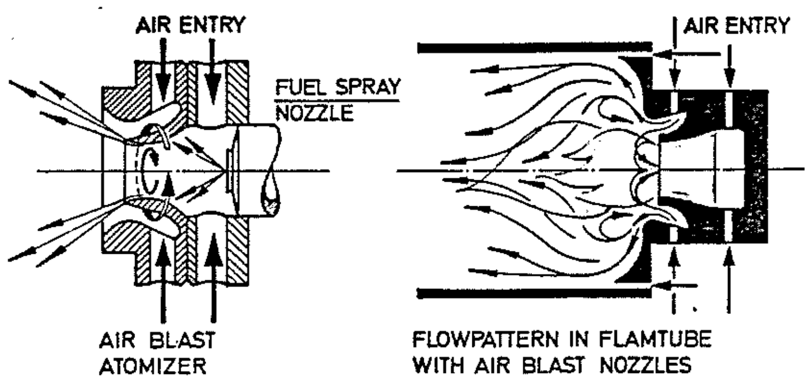


Fig. 9 WD 1000 Fuel Injection and Flametube Flowpattern

The fuel injection and flame stabilization system is illustrated in Fig. 9. Fuel is sprayed onto the inner surface of a venturi-nozzle, where it builds up a thin film. A rotating airstream drives this film to the nozzle exit, where it is atomized with the aid of the inner airstream and an outer airstream, which surrounds the outer surface of the venturi-nozzle and which rotates counter to the inner airstream. The desired flow pattern in the primary zone is shown by the arrows in Fig. 9. Immediately after leaving the nozzle, the outer airstream turns in the vertical direction, forming a layer on the flame-tube back-plate. Following the contour of the back-plate, the outer airstream turns back in the axial direction and approaches the first cooling film tangentially. Recirculation is directed towards the centre of the flame tube, where burning takes place. Air for wall-cooling and the air needed for the burning process are separated and do not interfere with each other, so that a high wall-cooling efficiency is attained. The basic nozzle geometry that would provide the desired flow pattern in the primary zone was developed in a water analogy test rig using a 5:1 scale perspex model of the nozzle.

The burning quality of this nozzle is now being investigated, as part of the complete combustion chamber development programme in a full scale combustion chamber rig. Fig. 10 shows the flame-tube and gives an impression of the burning quality. Looking upstream into the primary zone, it can be seen that fuel distribution is fairly uniform and that burning takes place, as desired, in the centre, away from the flame tube walls. The blue colour of the flames indicates good fuel atomization and air-fuel mixing.

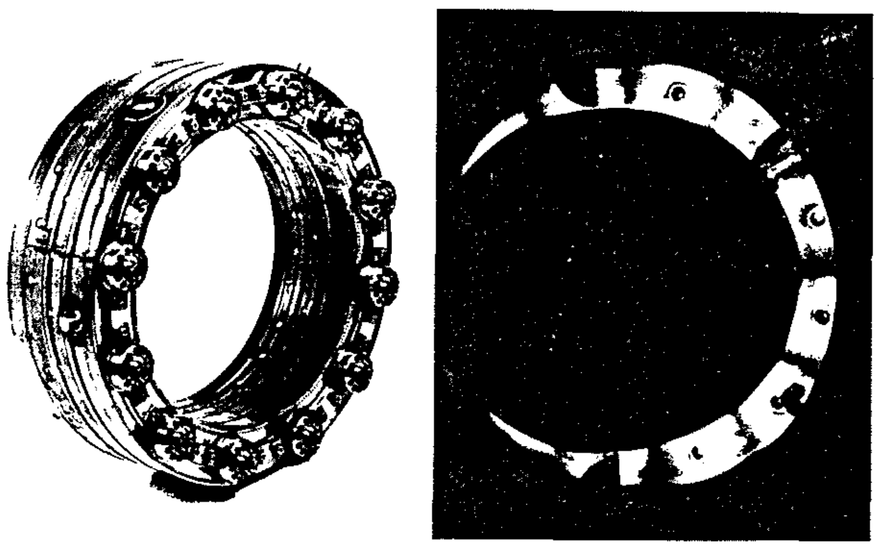


Fig. 10 Combustion Chamber Rig Test

5. Compressor Turbine

5.1 Aerodynamics

Marked progress in turbine aerodynamics, based on refined analytical design methods and test results, as well as the availability of improved turbine disk materials (powder metal) which tolerate higher rotational speeds, have encouraged us to provide the engine with a single-stage compressor turbine (Fig. 11) in spite of the high engine pressure ratio of 13:1. Besides the obvious advantages of lower cost, lower weight and shorter gas generator rotor, the single-stage solution, compared with a two/stage design, offers the possibility of increasing the turbine entry temperature and still reduce the cooling flow demand, because of the high temperature drop in the stator and the lack of a second stage, which would need cooling at least for its stator. Thus aerodynamic efficiency differences between the single-stage and the two-stage design, which certainly exist, are offset by the thermodynamic advantages of the single-stage design.

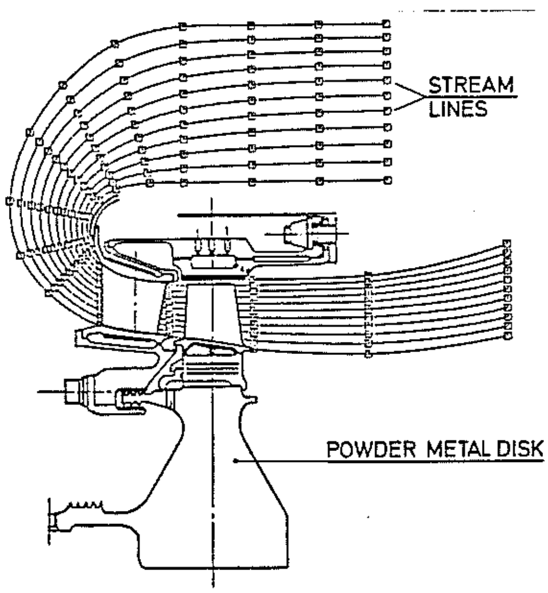


Fig. 11 WD 1000 Compressor Turbine

In the aerodynamic design of the WD 1000 compressor turbine (Ref. 4), attention has been paid especially to the reduction of end-wall losses. The flow path design in the stator area - favoured by the application of a reverse-flow annular combustion chamber - results in a radial pressure gradient near zero, minimizing boundary layer flows in the radial direction.

To further reduce the generation of secondary flows and the resulting losses, the areodynamic loading in the hub and tip regions of both vanes and blades is diminished by a lower deflection in these areas. This can clearly be seen when looking upstream onto the trailing edges of the nozzle guide vanes (Fig. 12).

The areodynamic performance of the turbine has been tested on a rig. The measured efficiency at the aerodynamic design point of approx. 85% (Fig. 12) includes the effects of vane and blade cooling, which was simulated on the rig. In the course of the ongoing development work we expect further improvements by reprofiling the turbine airfoils using latest fully three-dimensional calculation methods that also take into account the effects of the cooling air flowing out into the gas stream.

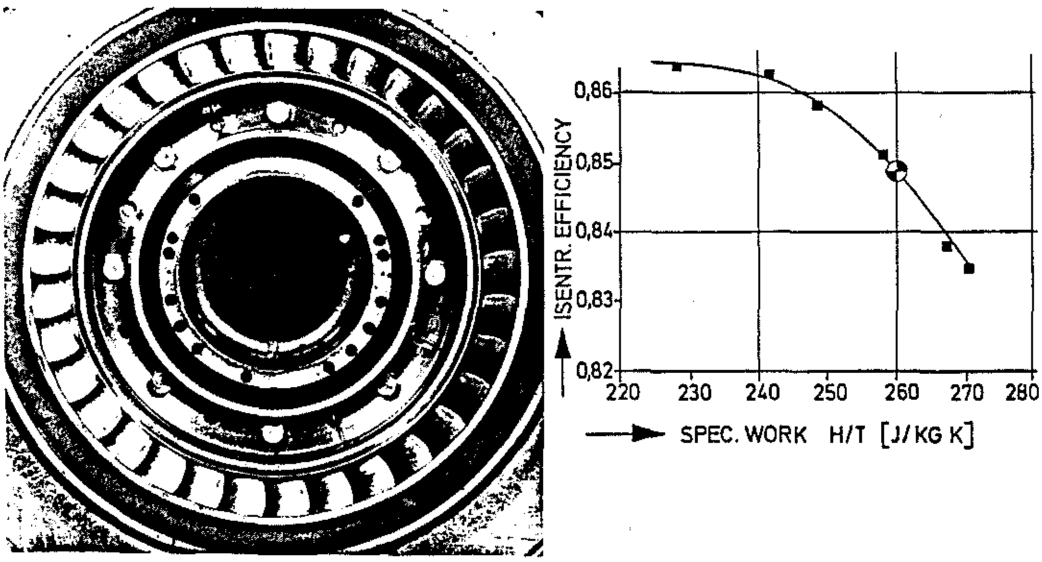


Fig. 12 Compressor Turbine Rig Test

5.2 Cooling-Air Economy

The air required for cooling the components in the turbine section exerts a negative effect on the fuel consumption of the engine in two ways:

- The air must indeed be compressed, but it is then taken out of the process and does no useful work;

- On accomplishing its task of cooling, the air flows into the gas stream, upsetting the aerodynamics and reducing the efficiency of the turbine.

To diminish the disturbance caused by the cooling air as it enters the gas stream, or to eliminate it altogether, is one of the problems which will be dealt with in the course of the further aerodynamic development of the turbine.

One objective in designing the whole engine is to bring the harmful effects of withdrawing the cooling air from the process to a minimum. As far as the WD 1000 engine is concerned, this is sought after by a variety of measures - especially by the careful design of seals for limiting leaks that are unavoidable in the vicinity of cooling-air streams - which we are unable to go into in detail here. Fig. 13, therfore, is limited to illustrating just one vital feature of the cooling-air system.

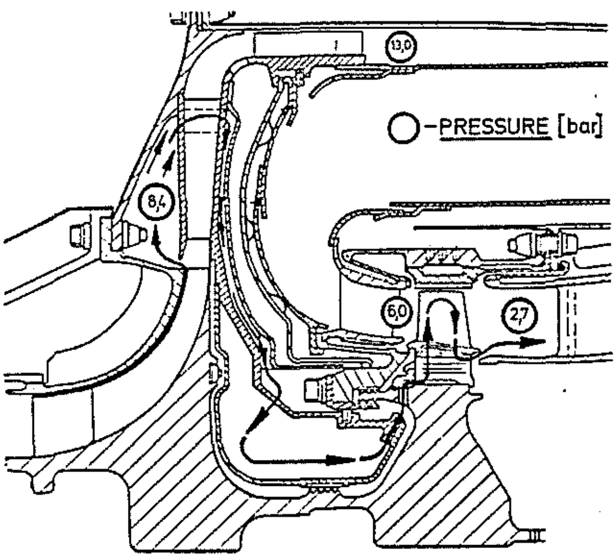


Fig. 13 WD 1000 Cooling-Air Economy

Mention was made in chapter 3.2 of the positive effect on the compressor efficiency of diverting the boundary layer at the exit from the centrifugal compressor impeller. The optimum quantity to be diverted roughly corresponds to the amount of air required for compressor-turbine blade cooling, and in the WD 1000 engine the air diverted is indeed used for this purpose. And here we have another advantage of the single-stage compressor turbine; only with the marked pressure drop in the stator and rotor of the single-stage turbine is it possible to supply this "waste" air with its relatively low pressure of 8.4 bar to the blades and to pass it through the cooling passages in the blades without risking the intrusion of hot gas upstream of the rotor.

5.3 Clearance Control

Because of the much greater thermal expansion, effective radial clearance control is much more important in the turbine section than with the axial-flow compressor. As shown in Fig. 14, an increase in the relative blade tip clearance in the compressor turbine from 1% to 2%, for example, results in a loss in efficiency of about 1.5% and, finally, in an increase in fuel consumption of around 1.8%. With a blade length of only 18 mm, a 1% relative tip clearance means just 0.18 mm. This fine relationship between the fuel consumption and the slightest changes in tip clearance make it obvious that great attention must be paid to clearance control.

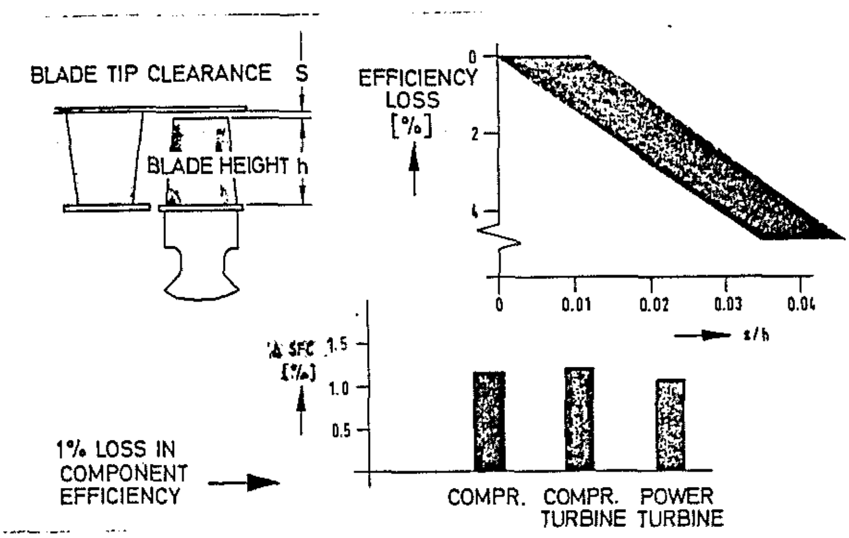


Fig. 14 Blade Tip Clearance Control

The growth characteristics of the rotor depend on the rotational speed and gas and cooling-air temperatures. Consequently, they are easy to calculate, but can hardly be influenced. Thus, to attain a narrow blade-tip clearance, the more easily influenced growth characteristics of the turbine shroud ring must be matched to those of the rotor. To achieve this in the WD 1000, a design was selected that had already been tried out at MTU and which is described in greater detail in Ref. 5.

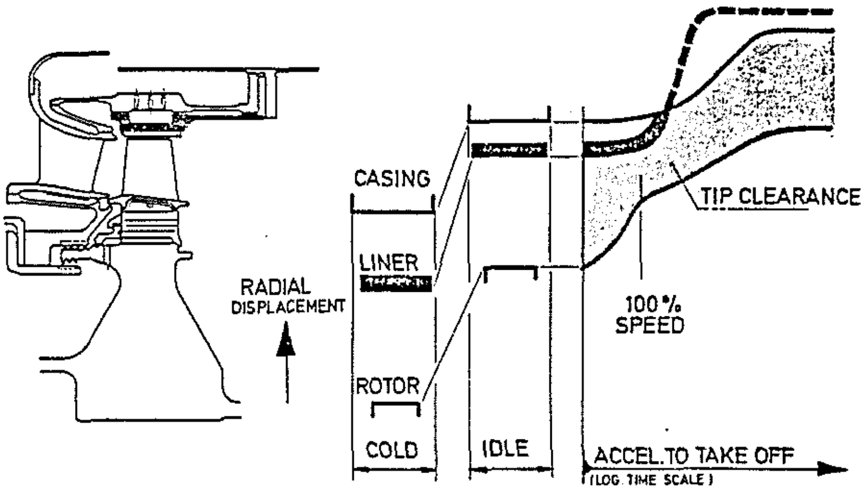


Fig. 15 Blade Tip Clearance Control

The shroud ring itself is relatively thin-walled with a ZrO_2 thermal insulation spray coating supported by a honeycomb at the gas-side and effective impingement cooling at the casing-side (Fig. 15). This thin-walled shroud ring is surrounded concentrically by a thick-walled casing ring. Whilst the tip clearance is determined to a great extent by the temperature of the shroud ring, which itself depends on a properly matched relationship between the thermal insulation and the impingement cooling, during steady-state operation, during transient conditions the expansion of the shroud ring is controlled by the thick-walled casing ring.

As shown in Fig. 15, this prevents, for example, temporary large increases in the tip clearance as a result of thermal expansion of the turbine disk during acceleration from Idle to full load. As discovered in various engines, such increases and the associated loss in efficiency can cause loss of power just when full take-off power is urgently required, i.e. about 20 - 40 seconds after acceleration to full load, in other words on lift-off or shortly thereafter.

6. Conclusion

It has been possible to present just a few design features of a new 1,000 kW helicopter engine here. Nevertheless it is hoped that the presentation gives an idea of how the complex and challenging requirements of today can best be met; not by greater complexity of the engine, but by applying advanced aerodynamics, materials and manufacturing technologies to an all-new engine with a basically simple configuration, by the economical use of cooling air and by the careful design of clearance control systems.

7. References

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