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LECTURE SERIES No. 46

on

Small Gas Turbines for Helicopters and Surface Transport

NORTH ATLANTIC TREATY ORGANIZATION

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ADVISORY GROUP FOR AEROSPACE RESEARCH AND DEVELOPMENT

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SMALL GAS TURBINES FOR HELICOPTERS
AND SURFACE TRANSPORT

J. Weidhuner

Langthorn.

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PREFACE

This Lecture Series is sponsored by the Propulsion and Energetics Panel and the Consultant and Exchange Program.

In view of the increasing interest in Small Gas Turbines for the propulsion of surface vehicles, it was felt appropriate by AGARD Propulsion and Energetics Panel to set up a team of selected experts from various NATO nations to present a series of lectures on this subject.

After a survey of the field of application for Small Gas Turbines, to replace reciprocating engines for helicopters and surface vehicles propulsion, the present state-of-the-art will be reviewed together with problems related to reliability, life time, pollution regulation, weight and volume according to various applications.

Conventional and advanced cycles (cycle of Nernst cycle with heat exchange) will be compared. A description of components (compressors, combustion chamber, turbines, nozzles, shafts with various configurations) will be followed by a review of industrial and technological problems.

The use of Small Gas Turbines for power generation, auxiliary stand by or emergency power plant is then presented. The last paper will be a Survey of Future Possible Developments and performance improvements (mixed diesel and turbines – use of high temperature materials).

A round table discussion with the participation of all the speakers will conclude the Lecture Series which will be presented in four different NATO nations (France, UK, Canada and USA) from 21 June to 2 July 1971.

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Lecture Series Director

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SMALL GAS TURBINES FOR HELICOPTERS AND SURFACE TRANSPORT

INTRODUCTION

by

Jean FABRI

The performance increase of aircraft engines during the last 15 years is mainly due to the development of large or medium sized gas turbines. Small gas turbines can however find also a broad field of application and it is the aim of the present Lecture Series to review the state of the art of small gas turbines from the various view points of civilian and military applications, present and future technology, cycle developments for maximum specific power and minimum operating costs, analysis of component development, industrial applications or auxiliary power generation as well as the estimated future developments.

A set of specialists from various NATO countries has been invited to outline these different questions and this gives to this Lecture Series a new dimension, since it shows the interest in small gas turbine development all around NATO countries.

The survey of "Military and Civilian Needs for Small Gas Turbines" by D.D. Weidhuner (USA) reviews the expected fields of application of small gas turbine engines. Helicopter and small fixed wing aircraft propulsion are well known applications, unmanned aircraft, combat vehicles such as heavy tanks and truck tractor trailers are some important military applications from which civilian transport facilities such as smooth and dependable passenger carrying buses will be developed sooner or later.

The greatest field potential application is certainly that of automotive engines, but the cost barrier will have to be broken before the antipollution advantage of gas turbines may become significant on an industrial or economic basis.

Further applications derive from some uses in which the large gas turbines have already found their place : electric power generation as main or auxiliary power plant and marine application. In all these problems, the cost of small gas turbines against that of diesel engines remains certainly the main drawback, but the high reliability and the long life of small gas turbines will finally impose the use of these engines.

The "State of the Art of Gas Turbines for Helicopters and Surface Transport" is given by H.H. Langshur and B.J. Palfreeman (Canada). The industrial background of the authors gives them the ability of discussing thoroughly the present state of the art as well as the development expected for the coming 10 years. The main present application is helicopter propulsion and this will prevail for many years. Actual trend is towards small gas turbines with still higher performances, and higher compression ratios, higher turbine inlet temperatures are expected. Technological progress will bring operational multispoon small gas turbines as well as cooled blade turbines and this progress will be continuously transferred to the ground vehicle propulsion systems, for which the low pollution characteristics of gas turbines will be a source of development.

But gas turbine development is not only due to technological progress. A better understanding of "Gas Turbine Cycles" as shown by P. Alesi and R. Laurens (France) will also be a source of improvments.

A detailed study of the various parts of small gas turbines is given in "Analysis of Small Gas Turbine Engine Components" by E. Schnell (Germany). Successively compressor design, turbines, combustion chamber and shaft configuration are reviewed. High pressure ratio compressors are required, and the respective advantages of multi-radial and/or centrifugal compressors vs. single stage axial compressor are analyzed. For the turbine, the trend is toward high inlet temperature operation and this requires progress in blade cooling whenever possible or in materials : evidently, both are necessary but much is expected from new techniques in blade cooling. Combustion chambers are usually very efficient and a choice exists between the annular type of combustion chambers when compactness is required and can type combustion chambers for ground application. Similarly heat exchangers, though often desirable are difficult to be placed in engines for helicopters or aircraft, but their use in ground transportation engines are more and more recommended.

All this development became possible only as a consequence of the improvement of technology. R.M. Lucas (U.K) reviews the "Industrial and Technological Proglems of Small Gas Turbines for Helicopters and Ground Transport". Scaling down an efficient large engine is not an obvious operation since many components can simply not be derived from a simple scaling law. This is the case of tolerances for instance which require better tools and more skilled workers, that may increase manufacturing costs. The technological bottlenecks that the small gas turbine manufacturer encounters are described and clearly solutions for all of them are ready in all cases.

"Power Generation", which may be one of the main future applications of small gas turbines is reviewed by A. Jaumotte (Belgium). The potential of large gas turbines as a standby power source in power generation plants is well known. Their scaling down to auxiliary power plants is very appealing These emergency or standby plants will be used for starting the engines, but also for air conditioning on ground, in flight and also on space vehicles.

To conclude, J. Melchior (France) analyzes the "Future Developments" one can expect. Gas turbines for helicopters will still be improved towards higher specific power, higher efficiency. But the gas turbine for ground transport will also become more current. Not only improvements in component design, but the choice of new cycles, new concepts of gas turbines (the rotating combustion chamber of the Nernst Turbine for instance) will give the high efficiency necessary for small gas turbines to compete industrially and economically with the diesel engine for ground transport.

It is hoped that the present Lecture Series will help to promote the development of small gas turbines.

MILITARY AND CIVILIAN NEEDS FOR SMALL GAS TURBINES

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Summary

The most important consideration in the selection of a power plant for any application is to maximize return on investment, or to maximize cost effectiveness. Whether for military application or commercial use, it must be indicated that the small gas turbine can yield more return or profit to the purchaser while satisfying operational requirements, than other engines, or its choice cannot be justified. The point of being cost effective is generally a more difficult task in the small turbine engine than in larger sizes because of the large production base of reliable, efficient, and low cost piston engines in the small horsepower category, and the greater difficulty of producing efficient small turbine components. This paper presents the critical requirements of the propulsion system for helicopters, vehicles, marine craft, electrical power generation, and total energy systems for buildings, and indicates the engine characteristics necessary for the turbine to be the preferred choice. Installation requirements and ancillary components are discussed, exhaust emission levels and certain other technical goals are specified. Small gas turbines are arbitrarily considered to be less than 2,000 HP or less than 10 lb/sec airflow.

The helicopter will be the first application discussed. It is also probably the easiest area in which to justify the selection of gas turbine engines. Virtually all new helicopters in the last ten years have been turbine powered. Only in small sizes, under 300 HP, can the aircraft piston engine approach being competitive. Probably the most significant parameter of the helicopter propulsion system is the engine plus fuel weight per mission, which must be minimized. Figure 1 shows turbine vs. piston engines for mission propulsion weight. The modern gas turbine with its very light weight and low fuel consumption shows considerable advantage. Low propulsion weight helps to reduce aircraft acquisition cost, inasmuch as one less pound of propulsion weight usually results in at least 4 pounds less airframe weight, which may cost approximately $60 per pound. Since aircraft piston engines above 300-400 HP are not generally in production, engine prices cannot be compared above these powers. It is generally believed that when compared on an equivalent production basis, there is not a significant difference in price, except in the small sizes. Experience in production of the Lycoming T-53 helicopter engine shows a 90% learning curve, based on dollars per HP through several models, and indicates that the cost of the one thousandth engine was approximately $50 per horsepower. Small (under 300 HP) piston engines are generally less expensive than this, in the range of $20-30 per horsepower. The total system, mission and life must be carefully assessed to determine life cycle costs. Generally, experience indicates the turbine engine will average 2 to 3 times the life of the piston engine in a helicopter, partially because continuous high speed and high power level is detrimental to the piston engine. Since overhaul costs are usually 20-30% of engine acquisition costs, life is an important economic factor.

The other major consideration in turbine engine selection for helicopters is whether the engine should be a free-turbine or the fixed shaft (or coupled) configuration. It has been generally agreed that multi-engine helicopters should utilize free-turbine engines, since coupling and control of the engine is simplified. Large single engine helicopters favor the free-turbine engine since a heavy, expensive, and often troublesome clutch can be eliminated. In small, single engine helicopters, either engine type is acceptable, and both have been successfully demonstrated. The fixed shaft engine, operating at constant speed, has the important advantage of very rapid power response time, and is generally less costly to manufacture, whereas the free-turbine permits elimination of the clutch, and provides some inherent rotor speed stability if overloaded, due to the torque curve characteristics. It was the general conclusion of the majority of technical experts at an AGARD Helicopter Propulsion meeting in 1968, that the fixed shaft engine was preferred for small, single engine helicopters, although the practice in the United States has been the opposite.

Regenerative or non-regenerative engines may be used in helicopters, but it appears that the engine plus fuel weight per mission favors the high pressure ratio, non-regenerative engine due to usually short mission times. The regenerative engine is expected to cost perhaps 20% more initially, and may have somewhat higher maintenance requirements. Therefore, even though the specific fuel consumption is less, the regenerative gas turbine has not been considered cost effective in helicopters. There could be advantages to a regenerative cycle engine if infrared suppression is required, or if the mission requirements necessitated a large number of missions between refueling. However, since the fuel consumption of advanced high pressure ratio and high temperature engines is now quite good, on the order of .45 lb fuel/HP hr, and the specific fuel consumption curve is quite flat in the upper 50% of the power range, it does not seem likely that the regenerative engine will become attractive in helicopters.

While the gas turbine is the most cost effective selection for the helicopter today, further advancements in technology such as lower weight, lower fuel consumption, and longer life will increase the return on investment. A further return can be made by the use of diagnostic systems for inspection and maintenance. New helicopter engines being sponsored by the US Army will feature built-in sensors, vibration pick-ups, etc., to enable automated diagnostic systems to be used, thus reducing the skill level required of maintenance personnel. Borescope ports provided in an engine will enable inspection of almost any internal part without disassembly. Thus maintenance costs should be reduced. It may be desirable, with the use of proper diagnostic systems, to send the engine to overhaul based on observed condition, rather than at a scheduled operating time, which should increase the utilization of the engines owned.

Since helicopters operate much of the time in the ground environment, sand, dust, and foreign object ingestion must be protected against, to preclude engine damage. US Army helicopter experience in Vietnam shows that approximately 2/3 of engines removed from service prematurely were due to ingestion damage, before inlet particle separators were installed. Particle separation may be the barrier filter type or some type of centrifugal separator. Since turbine engine output is quite sensitive to inlet losses, the pressure drop across the particle separator must be minimized, a requirement which is normally incompatible with high filter efficiencies. Barrier filters have not proven to be practical, and current centrifugal separators are not as efficient as desired. Other requirements of the particle separator are anti-icing, means of visually inspecting the compressor face, and perhaps an integral water-wash manifold to clean the compressor after operation in a salt atmosphere. Figure 2 shows the T53 engine used in the HU-1 helicopter with particle separator installed.

Other installation needs may be an infrared suppressor, compressor bleed for cabin heating, and a starting system which may or may not incorporate an auxiliary power unit.

Small fixed wing aircraft can also show turbine engine advantages over the piston engine, although to a somewhat lesser degree. In the 200-300 horsepower category, reliable piston engines are available at a low cost, as mentioned before. Life of the piston engine in fixed wing aircraft is acceptable because high power is used only for short periods at take-off, and cruise power may be approximately 60% normal rated power. It is therefore somewhat difficult to convince a customer that he should pay perhaps twice the initial cost of the piston engine for the turbine in this power category, even though the turbine promises longer life, and due to its lighter weight, a payload advantage. The turbine engine may or may not be cost effective in this case at this time. In sizes above 300-400 horsepower, modern piston engines are no longer available, turbine performance is quite good, and the higher aircraft cost makes an additional increment of engine cost less significant in overall acquisition cost. Many of these aircraft are executive type where smoothness and quietness of operation of the turbine are desirable advantages. Nevertheless, significant emphasis must still be placed on reducing acquisition cost, and maintaining long life, without sacrificing engine performance, in order to make the small gas turbine the clearly superior choice. Small turbo-fan and jet engines are attractive for fixed wing aircraft, particularly the somewhat larger and higher performance executive type aircraft. They are only mentioned briefly since they are usually larger than the 10 lb/sec airflow limitation arbitrarily established for small engines in this paper, but the same criteria of cost, reliability, and life is applicable.

A further flying application for small gas turbines is in expendable, unmanned aircraft, such as target drones. Speed and altitude requirements are generally representative of high performance manned aircraft, therefore a jet engine or perhaps a turbo-fan engine is desired. It is necessary to meet performance requirements, but otherwise the most important characteristic must be very low acquisition cost since the engine is expendable. Normally, lower efficiencies can be tolerated, and the design can be compromised toward low pressure ratios. Since expected life is short, high operating stresses can be used and turbine materials can be utilized at higher temperature than in piloted aircraft since safety margins are not as significant. This type engine is quite specialized and normally derivatives of the engine are not as useful in other applications. Therefore, the development costs usually must be amortized over a single application, but development efforts can be minimized since flight safety requirements are minimal.

The next application category to be discussed is the combat vehicle, particularly the heavy tank. For many years the diesel engine has been the preferred engine choice. The diesel engine is usually a specialized engine for the application, may be air cooled or liquid cooled, and is usually much higher in specific output (i.e., smaller and lighter per horsepower) than conventional industrial diesel engines. The diesel engine has been attractive due to its low fuel consumption and good reliability. The parameter of engine plus fuel weight per mission which was paramount in the helicopter is of much lesser significance in a battle tank. A more important parameter is engine plus fuel volume per mission. It has been estimated that each additional cubic foot of propulsion volume adds approximately 1000 pounds to vehicle weight, due to the necessity of providing thick and heavy armor around the power plant compartment. Battle tanks have weight and width constraints due to the necessity of using existing roads, streets, and bridges.

The gas turbine, to be considered for battle tank application, must offer an installed power plant plus fuel volume no more than the diesel engine. Since mission durations may be as much as 48 hours, (although not necessarily 48 hours of operation) fuel consumption is very important. The typical average duty cycle of the tank engine may involve 40% idle time, 40% low to medium power, and 20% of time at high power levels. Therefore, good fuel consumption is required at low power levels, which is not a characteristic of the simple cycle gas turbine. The regenerative cycle gas turbine is clearly indicated, and it can be shown that the extra volume of the regenerative gas turbine, which may be three times that of the simple cycle, is offset by the difference in fuel volume over the long mission times. This is shown in Figure 3 for a typical modern battle tank. A regenerative gas turbine which will offer less mission propulsion package volume than modern diesel engines must be of rather advanced technology. If a modern diesel engine of .75-1 HP per cubic inch displacement is used as the standard of comparison, the regenerative gas turbine must have a maximum cycle temperature of at least 2000^{o}F, a pressure ratio of approximately 10, and regenerator effectiveness of 70%, with good component efficiencies, to compare favorably with the diesel. This cycle should yield a minimum specific fuel consumption of at least .45 lb/HP hr. Some means of maintaining high cycle temperatures at low power, such as variable turbine geometry, must be employed to achieve good fuel consumption at low power.

Considerable attention must be paid to several difficult development problems in the regenerative engine described above. Inasmuch as the combustor inlet temperature is high in a regenerative engine, combustor cooling is a considerably greater problem than in simple cycle engines. The high turbine inlet temperature and high secondary air or cooling air temperature necessitates very carefully designed convective cooling approaches, and very good quality control in manufacture. Due to the necessity of maintaining high permperature at low power levels, the percentage time at maximum temperature is much greater than the time at maximum power, and therefore the thermal loading of the engine hot section is much more severe than in typical simple cycle engines. Efficient turbine cooling systems are mandatory to achieve reasonable life. Variable turbine geometry must also operate reliably in the hot gas stream. These differences from the simple cycle engine require particular attention during engine development so that engine reliability is not less than the diesel engine.

While the above goals seem somewhat formidable, it is believed that technology has advanced to the point that successful development of such an engine is quite practical, and a measure of reliability and durability greater than the diesel is to be expected, although not yet proven.

Power levels of armored, tracked vehicles are now generally less than 20 HP per ton, but the trend is upwards to obtain greater maneuverability or agility, and perhaps twice this power level might be desired in the future. Gas turbines offer the promise of considerable growth potential, whereas growth of the diesel engine is more limited, and will probably result in increased package size.

Gas turbine installation in a battle tank must also fulfill some unique requirements. Inlet air filtration is quite important since a tank convoy results in very dusty conditions. It is generally required that the air filters be capable of operating 5 hours in zero visibility dust conditions without servicing. The preferred filter appears to be a combination of an inertial particle separator plus a barrier filter. There are some development programs in the US on self-cleaning filters in order to meet the operational requirement. These are usually based on a vacuum cleaner principle wherein a cleaner traverses the barrier filter to remove the imbedded dust particles. An example is shown in Figure 4. The air filter requirement is more stringent than helicopter requirements since dust concentrations are greater and exposure times are longer.

Another unique requirement is for vehicle fording of streams. This necessitates inlet and exhaust snorkels, and either a sealed engine compartment or the ability of the engine to be immersed in water. It appears at this time that the sealed compartment with hydraulically telescoping and elevating snorkels is the preferred solution.

The gas turbine engine must be installed with a suitable transmission providing steer, braking, and speed control. Controls compatibility for smooth operation is necessary. It has been demonstrated that a hydromechanical transmission (i.e., a split power path between a planetary gear train and a hydrostatic system) can accomplish the necessary control and operation functions, providing good acceleration, and also permitting the engine to operate on its optimum operating line for best fuel consumption.

Fringe benefits of the gas turbine engine in the battle tank include very low engine noise, smokeless exhaust, good cold starting, and low vibration.

Acquisition cost of this gas turbine engine is very important. Unless considerable attention is devoted to production methodology during development, the cost may be enough more than the diesel engine that it will not be selected. A typical diesel tank engine might be expected to cost approximately twice per horsepower than that of normal production diesel engines for industrial and commercial application, or in the range of \$30-35 per horsepower. It is believed that a regenerative gas turbine can be competitive with this price, with proper design. As mentioned before, history of a simple cycle helicopter engine in the HP category showed the production learning curve, through several models, to average 90% on a dollars per horsepower basis, with the average price over 5000 engines being \$45 per horsepower. The use of industrial quality castings and industrial gearing will reduce costs, the fuel control cost will be half that of an aircraft fuel control, but the regenerator will increase basic engine costs by 10-20%. It is considered that regenerative engine costs can be competitive with the diesel.

There may be large accessory loads, electrical or hydraulic, on a tank engine, and it is desirable that provision for this power be made on the vehicle transmission, or on the output of the engine. Rapid application of a 50 HP load on the gas producer section may cause overtemperature, compressor stall, or at very least preclude rapid engine acceleration if suddenly needed.

It appears, in summary of the tank engine considerations, that a modern, advanced technology regenerative gas turbine, properly developed and installed, can offer vehicle performance advantages over the piston engine vehicle, and by virtue of longer life expectation, offer the lowest life cycle costs.

The next application to be discussed is <u>the truck tractor-trailer</u> which is primarily used commercially in highway, inter-city service. Most factors are also applicable to military usage of similar equipment. Since the usual application is in a commercial, profit making operation, total costs are very important. It is necessary that the acquisition cost be similar to the standard commercial diesel engine, as installed, and operating and maintenance costs must be lower in order to increase profit potential.

Ford Motor Company, who will begin automotive or industrial type turbine engine production in August 1971 says that the production turbine will cost no more than comparably sized diesel engines. If a typical diesel engine installation rated at 450 horsepower were to cost \$12,000 to \$13,000 (for the basic engine and associated equipment such as radiator and cooling system) then Ford will offer a turbine engine of the same horsepower size for the diesel engine price. It is understood that Detroit Diesel Allison will price their turbine engines accordingly. Therefore, it is virtually established that acquisition cost for a turbine engine will be competitive. It remains to be established that the overhaul and maintenance costs will be lower, but extreme attention is being paid to this factor during development. Current diesel truck engines in highway service run 300,000 to 500,000 miles before removal for overhaul, although there may be mid point maintenance or partial overhaul performed in the truck. Maintenance comprises 10-11% of the revenue dollar, and fuel is 7-8% of the revenue dollar.

Thus, the general requirements for a successful gas turbine to compete with the diesel engine in highway truck-tractors are established. The type engine generally proposed to meet these requirements have characteristics as follows: single stage centrifugal compressor of 4-5 pressure ratio, 1700-1900^{o}F turbine inlet temperature, rotary regenerator, and free turbine with a means, such as variable geometry, to maintain high turbine inlet temperatures at low power. Ford's design is shown in Figure 5. The engines must be designed for minimum production cost, very long life, and minimum maintenance with easily replaced components as modules. The cycle chosen appears to be quite conservative, but is now judged to be the best compromise between cost, reliability, and low fuel consumption. Typically, this type engine will weigh one half the diesel engine and occupy 2/3 the volume. Thus, with the same axle loadings, somewhat more payload can be carried for additional revenue.

As state and federal laws change concerning minimum speed requirements on grades and the operation of double trailers on turnpikes, the power required for the truck-tractor will increase, further favoring the gas turbine, due to its excellent growth potential. If the above mentioned conservative gas turbine cycle can be shown to provide greater profits than the diesel, the future potential is indeed good, since all further technology gains should further increase profit, and considerable technology gains are fully expected.

As in other applications, the installation and ancillary equipment is quite important. Good air filtration and inlet silencing must be provided. A transmission well matched to the engine and load must be used. The use of diagnostic equipment will provide for proper maintenance. An interesting, but important installation requirement is the necessity to avoid salt or salt water ingestion from streets where salt is applied to melt snow and ice. Ford has installed a de-mister in the inlet air stream, similar to marine practice, to remove the salt water. Salt ingestion can be deleterious to certain materials in the air path of the engine, and also promotes sulfidation in the turbine section.

Engines similar to those in highway truck-tractors will serve equally well in passenger carrying buses, where the absence of noise and vibration will be an added benefit, contributing to passenger comfort. These engines could also be expected to serve well in military trucks, except that probably some militarization would be required, such as larger capacity air filter, and provisions for engine immersion in water or a water tight compartment to satisfy the requirement for fording streams or swimming.

While on the subject of vehicles, the automotive gas turbine must be considered. The automobile industry is of course the largest producer of engines, and while this industry in the past has not shown high interest in gas turbines, there is renewed interest at this time due to anti-emission requirements. Gas turbines have been evaluated in automobiles, and have demonstrated acceptable operational characteristics, acceleration and fuel consumption. Probably the greatest disadvantage today is production cost. While it appears that a gas turbine can be produced competitively with the diesel engine, the current spark ignition automobile engine is very much less costly to produce, being perhaps $3-4 per horsepower.

A major reason for the renewed interest in automotive gas turbines by the manufacturer's is that perhaps future exhaust emission requirements can be met with the gas turbine. It is currently indicated that unburned hydro-carbons and carbon monoxide emissions are very low, except at idle. Oxides of nitrogen are still too high to meet anticipated requirements, although there is promise of reducing them.

The gas turbine engine for automobiles is generally envisioned as being quite similar to the truck engine described before, but about one half the size. The smaller size further increases the difficulty of attaining desired performance. While the regenerative engine is usually considered necessary to attain the desired part power fuel consumption, there may be other considerations which affect cycle choice. The anti-emission requirements for automobile engines have resulted in a decrease of efficiency of the spark ignition engine, and the part load specific fuel consumption is not now very good. Therefore, the simple cycle gas turbine may become competitive, particularly if further advances in small compressors are achieved. The lower production cost and much smaller volume of the simple cycle engine would be very desirable. There is also some indication that the higher combuster inlet temperature of the regenerative engine tends to increase NO_x emissions.

The subject of automobile engines is vast and cannot be covered in depth in this paper. There is a very high level of activity in every automobile manufacturer's laboratory at this time to select the next generation engine, which must meet emission requirements. Competitors to the standard spark ignition engine are: regenerative and simple cycle gas turbines, closed Brayton cycle engines with various working fluids, the Wankel type engine, hybrid engine and battery systems, various versions of the piston engine such as stratified charge engines, and others. If the gas turbine is selected for volume production by the automobile industry, there will be a profound impact on the small turbine industry, and on all users of power in the 100-300 horsepower range.

Another use for the small gas turbine is electrical power generation. Several models of large jet engines have been modified and power turbines added, to provide for topping or in some cases addition to base load in large electrical power generating stations. These have been generally quite successful, but the small gas turbines (under 2000 HP) under discussion here are not usually suited to fixed installations since large amounts of power are required. The small gas turbine nicely fulfills the requirement for portable electric power. Portable military generator sets cover the spectrum from less than 1 KW to approximately 1000 KW, with the majority of electrical power being produced in the 60-100 KW sizes, although the majority of units procured are less than 10 KW.

Typically, engine generator sets under 20 KW are spark ignition piston engine driven. Over 20 KW, diesel engine driven sets are used, with the small gas turbine now being given more serious consideration. Like the other power plant areas, the desired engine is that one which will meet the requirement at the lowest cost, the cost in this case being ¢/KW hr, amortizing all costs of procurement, operation, maintenance, transportation, etc.. In the very small sizes, spark ignition engines are very inexpensive and perform generally satisfactorily, although reduced maintenance would be desired. Above 20 KW, acceptable diesel engines are available and their increased procurement cost is offset by longer life and lower maintenance.

From the performance standpoint, frequency regulation and ability to pick up load rapidly are characteristics of primary importance. Figure 6 shows the regulation desired with load changes. It should be noted that two standards are listed, one for typical utility power, and another for the precise regulation required for powering sophisticated electronics systems. In piston engines, this control is obtained by an isochynous governor. In turbine engines two approaches are possible. If the generator is gear driven, the engine power control and isochynous governor controls speed. A more advanced concept is to drive an alternator at turbine speed, generating very high and uncontrolled frequency, and to utilize a solid state cycloconvertor to achieve the frequency and frequency control desired. This latter approach still necessitates an engine power control which provides for the engine to pick up load rapidly, even though frequency is always controlled. Turbine engines for electrical power generation are usually fixed shaft engines, rather than free turbine, because of the excellent load response characteristics at constant speed. In the modern turbo-alternator approach, where the alternator is driven directly at turbine speed, design approaches integrating the alternator with the rotating turbine components can be used. Figure 7 shows an example where only two bearings are required for the total rotating assembly, providing a minimum of size, weight, and production cost.

Regenerative gas turbines are favored where fuel consumption is paramount, but the extra weight and volume reduces portability, and most portable gas turbine generator sets have been simple cycle. There have been studies which indicate that the simple cycle gas turbine is more cost effective than regenerative gas turbines for generator sets, due to the higher cost and presumed higher maintenance of the regenerative cycle unit, but these studies are sensitive to the type utilization of the unit. For application where portability is important, the simple cycle will be favored. If regenerative gas turbines developed and produced commercially for vehicles or other industrial uses can be utilized for generator drive, it is likely that they will be favored due to better fuel consumption, but more importantly, a significant production base will exist which will reduce costs. It could be expected that many military requirements will favor the simple cycle engine, and some military and most commercial applications will favor the regenerative engine.

As in other turbine engine installations, adequate inlet air filtration is required, and some special attention to silencing may be necessary since generator sets may operate in the proximity of personnel for long periods. Automated control for remote starting, operating, and shutdown may be desirable in a number of circumstances. Provision for paralleling sets may be necessary to assure that power is not interrupted in certain critical operations.

Another area to consider for small gas turbines is marine application. Throughout the world, Navies are successfully employing gas turbine power in ships. These engines are usually derivatives of aircraft engines, which have been developed into quite mature, long life engines, and have been marinized to be compatible with the sea-going environment. A number of the lessons learned in installation and marinization are applicable to the small engine case. It is of fundamental importance to provide protection against the corrosive salt atmosphere. This involves selection of materials which are corrosion resistant, and application of protective coatings. Magnesium in the air path of the engine is not desirable since even protective coatings can be eroded away. It is good practice to provide a water wash manifold in the engine inlet to facilitate washing down the engine internally with fresh water after exposure to salt environment. This, incidentally, is also good practice on helicopter engines which operate in salt water environments. Salt ingestion promotes another problem in turbine engines, which is that of sulfidation in the turbine section. The sulfur in fuels together with salt yields sodium sulfate which attacks the turbine buckets. A reasonable solution to the problem is to install a water separator or de-mister in the induction system, to prevent the salt from entering the engine.

The industrial gas turbines of 2-400 HP being developed for truck and other uses will likely be strong contenders for small boat or pleasure boat application. Since these engines are expected to be priced competitively with current diesel engines, and should satisfy the small boat requirements, offering low noise and vibration together with long life, it is to be expected that they will be quite attractive for this use. It is further likely that the small boat market is not sufficiently large to justify the development expense of a special marine engine, which would not have an appreciably different cycle (pressure ratio, temperature, etc.) than the industrial engines for other application. For the same reason, in the 500-2000 HP category, it seems likely that marinized versions of aircraft engines or a battle tank engine may be utilized rather than a specially developed marine engine.

The small gas turbine is also a very competitive contender as a power source to provide all the utility services of large buildings, schools, shopping centers, etc.. This system is usually termed the Total Energy Concept. A schematic of one version of this system is shown in Figure 8. Essentially, a small gas turbine provides shaft power to drive a generator for electrical power, and the exhaust heat energy is utilized in a boiler to provide steam for space heating, and hot water. Refrigeration is provided by absorption chilling, or a refrigeration unit can be driven by shaft or electrical power. Most of the heat energy of the fuel is utilized with final exhaust temperature being very low, and therefore overall thermal efficiencies of 60-80% are possible. Thus, the nomenclature of Total Energy Concept is derived.

This concept eliminates the dependency on large central utility generating plants. It is apparent, however, that initial capital expenditures are greater for this system than merely connecting to existing public utilities. Therefore, an economic analysis must show operating savings such that a cross over point is reached within some reasonable time period, probably five years, after which this system must show the lowest total cost. This economic analysis depends on many factors, mostly non-technical, such as Total Energy Concept effect in building cost, labor costs, depreciation, tax effects, local utility power costs, etc.. It also depends on the gas turbine life and reliability. It is therefore desirable to choose gas turbine design parameters which are conservative and conducive to long life. Compressor stage loadings may be low to reduce effects of compressor fouling, and cycle temperature must be modest to permit clean combustion and long life. A conservative cycle is still quite efficient in this case due to the high utilization of engine exhaust heat energy.

Like many other application areas, this market may not be sufficiently large at this time to warrant development of a specialized engine. Therefore, adaptations of engines developed for other purposes may be utilized. Engine power ratings of interest are generally in the 500-1000 HP category. Simple cycle industrial engines with conservative design or de-rated and modified aircraft engines can be considered.

There are many other application areas for the small gas turbine which have not been discussed, and probably many more applications which have not yet been developed. It is necessary that the gas turbine be analyzed for any application to determine whether it might be the most economic selection over the life cycle.

It is probable that the biggest market growth for small gas turbines in the next few years will be that where the diesel engine is now utilized. The gas turbine is already firmly entrenched in the aviation field, and the market rather well established. Since the regenerative gas turbine now seems able to compete with the diesel in terms of acquisition cost and fuel economy, there appears to be market opportunity in the industrial and truck area, and all the other widely varied applications of the diesel. There are approximately 700,000 diesel engines manufactured and sold commercially each year in the United States. The quantities are approximately as shown in Figure 9. This market is large enough to support several rather large producers, and therefore, there is sufficient market to sustain some turbine producers. It is to be expected that current diesel engine manufacturers will develop gas turbines, and perhaps current turbine engine manufacturers will develop industrial type engines to gain a share of this market. Capturing the markets now held by the diesel will necessitate well developed turbines which can prove economic through longer life, reliability, etc., and will also necessitate extensive market development and applications engineering to inform potential customers of operating advantages of gas turbines, and how all the fringe benefits can be utilized, and total costs analyzed.

There are fundamental reasons why the small gas turbine faces a more competitive situation than did large turbine engines, particularly jet aircraft engines. The jet engine permitted such large aircraft performance gains that the piston engine was quickly non-competitive. The performance gains possible with small shaft power gas turbines over their competition are much less spectacular, and in fact the development of a high efficiency small engine presents technical difficulties not encountered in large units. This is due to the effects of relatively larger compressor and turbine blade tip clearances, the relatively larger boundary layer in all flow passages, smaller Reynolds Numbers, greater possibility of leakage of air from the cycle through seals, flanges, etc., compromise of blade shapes and aspect ratios to accomodate cooling passages and maintain structural integrity, and many other scale effects. Scaled large engine axial compressors cannot be used since the result would be such small blade heights and sections as to be impractical to manufacture and quite vulnerable to foreign object ingestion. This in turn often causes design philosophies of small engines to be different, such as the use of radial rather than axial compressors and turbines. Besides the fundamental penalties of small components, there is the current disadvantage that by far the greatest expenditures in turbine technology have been in the large jet engine area for high performance military aircraft, and only relatively smaller expenditures made to advance small turbine component technology. Therefore, the technology bank is not as great to support small engines as large ones. It is for these reasons, plus the generally good experience and low cost of diesel and spark ignition engines that the small gas turbine has not progressed more rapidly. It is indicated that component technology programs for small turbine engines should be supported and more extensive efforts devoted to producibility of the engine.

The US Army has supported a number of component programs in the 2-5 lb/sec air flow category in the last 7-8 years. Initial goals were rather ambitious, being directed towards 12-1 pressure ratio single stage radial compressors, and approximately 2300^{o}F cooled turbines. Without reviewing these programs in detail, it can be reported that results were not outstanding, and that efficiencies were generally 5 or more percentage points below goals. While these results were disappointing, it did indicate the problems of attaining high efficiency small components, and where further efforts should be placed. Component technology is still being supported toward ambitious goals, some of which are outlined in Figure 10. This type program must continue, but also it is believed necessary to combine advanced components into a running engine or gas generator to evaluate them under engine conditions. Only after this is done can engine development goals be established which can be achieved on a predictable schedule and cost, without undue risk.

In covering the subject of this paper, "Military and Civilian Needs for Small Gas Turbines", quantity requirements of the military are omitted because of the security classification of inventory requirements. The civil quantity requirements depend only upon the technical progress made toward the turbine surpassing its competition on an overall cost-effectiveness basis. It would appear that at least one half the current diesel engine market is a reasonable potential. At the other end of the spectrum, if gas turbines are successful in automobiles, ten million or more gas turbines per year could be produced. Thus, the overall numerical requirements can be very great indeed.

Technical requirements differ considerably with applications, as has been mentioned throughout this paper, but common requirements of improved component efficiencies, higher cycle pressure and temperature limits, and improved producibility are important to every application. Engine acquisition costs and life cycle costs have been emphasized for every application. Particularly in the small gas turbine area, it is believed that the designer should consider costs with as much emphasis as the cycle parameters. By so doing, it is believed that the small gas turbine can capture a large share of the total market for power under 2000 horsepower, on the basis of providing the purchaser the greatest return on investment.

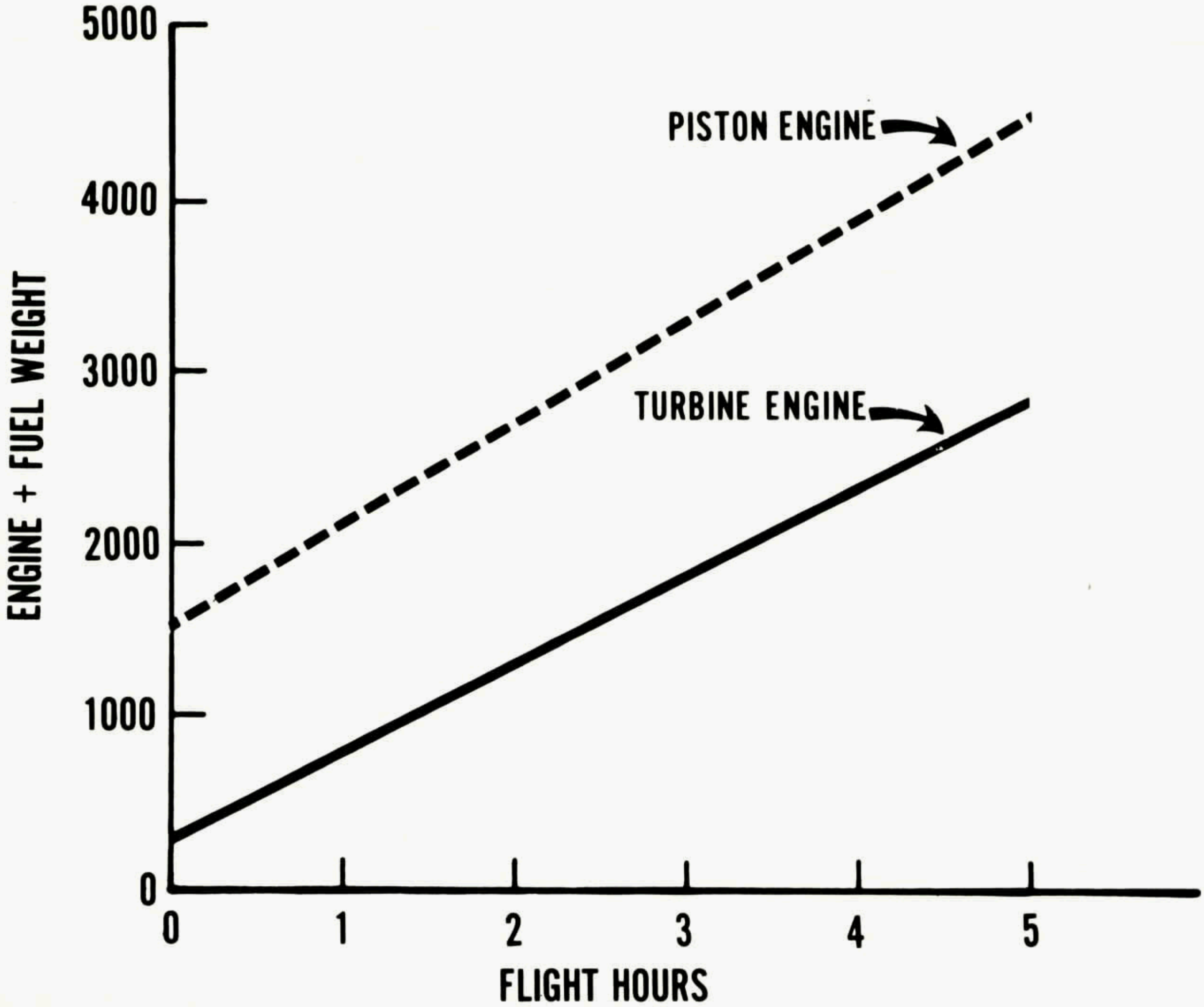

Fig.1 Helicopter engine + fuel weight per mission (typical 1500 horsepower engines)

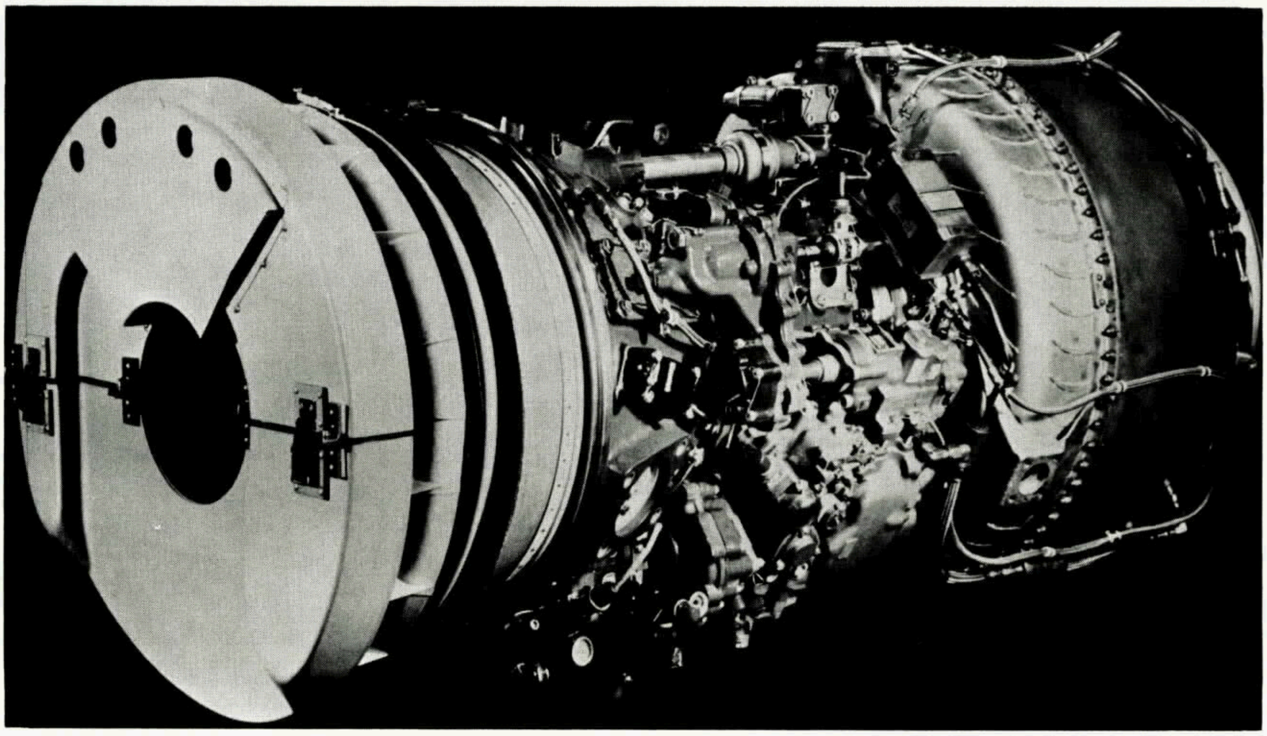

Figure 2

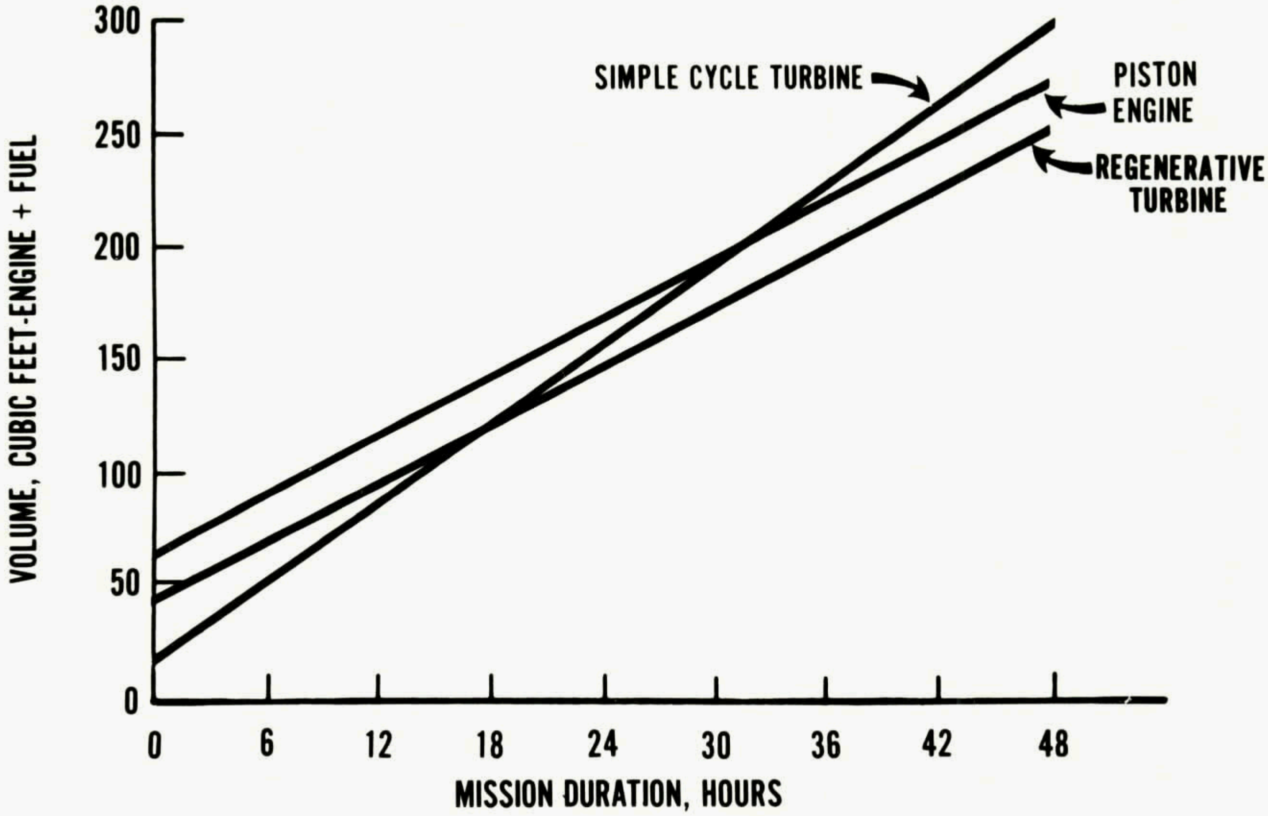

Fig.3 Engine plus fuel volume per mission for heavy tracked vehicle

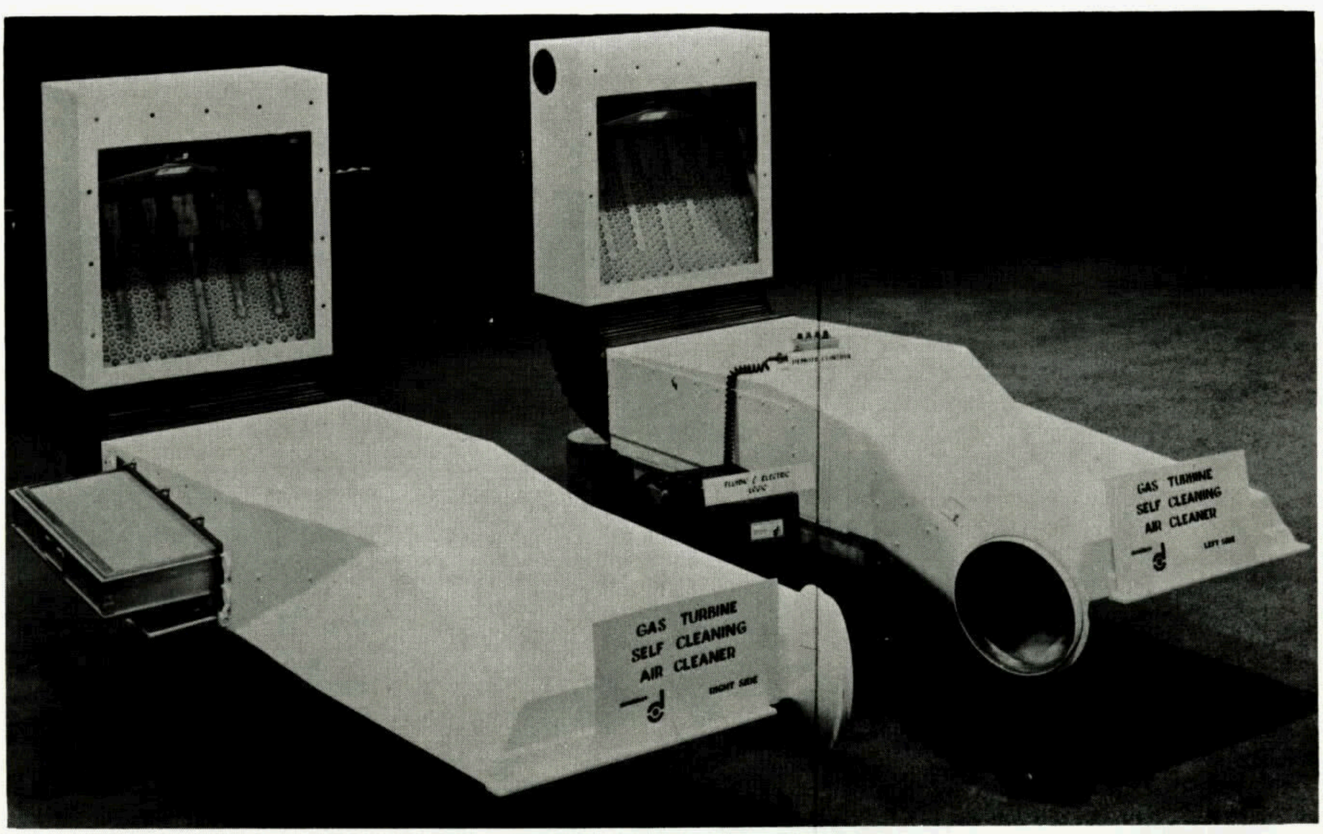

Figure 4

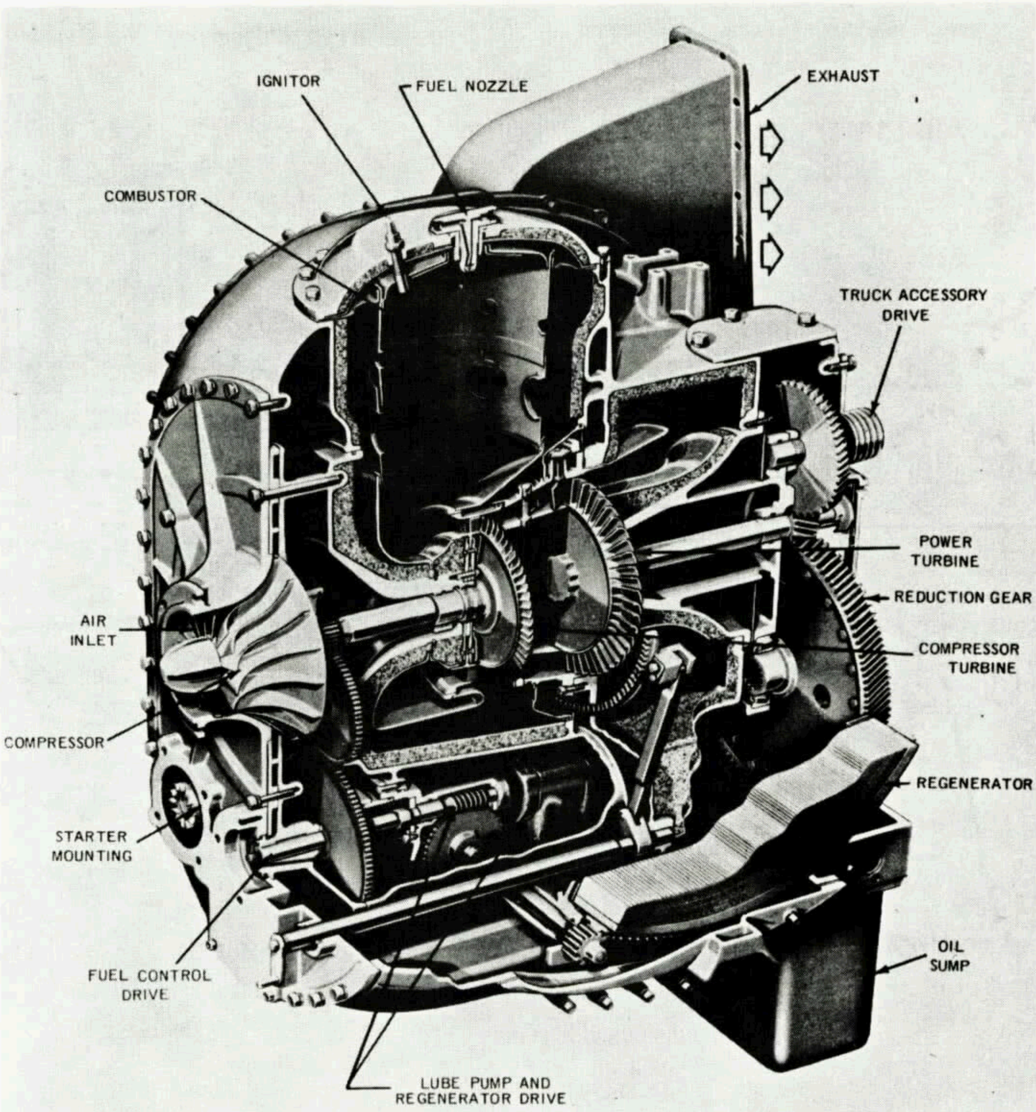

Figure 5

	UTILITY	PRECISE
● REGULATION, %	3	1
● STEADY STATE STABILITY (VARIATION)		
SHORT TERM (30 sec)	2	1
LONG TERM (4 hours)	4	2
● TRANSIENT PERFORMANCE		
● APPLICATION OF RATED LOAD		
DIP, %		
50 - 60 Hz	20	15
400 Hz	20	12
RECOVERY, (SECONDS)	3	.5
● REJECTION OF RATED LOAD		
RISE, %		
50 - 60 Hz	20	15
RECOVERY, (SECONDS)	3	.5
● APPLICATION OF SIMULATED MOTOR LOAD		
DIP, %		
50 - 60 Hz	40	30
400 Hz	40	25
RECOVERY TO 95% OF RATED VOLTAGE (sec)	5	.7

Fig.6 Electrical power generation performance characteristics

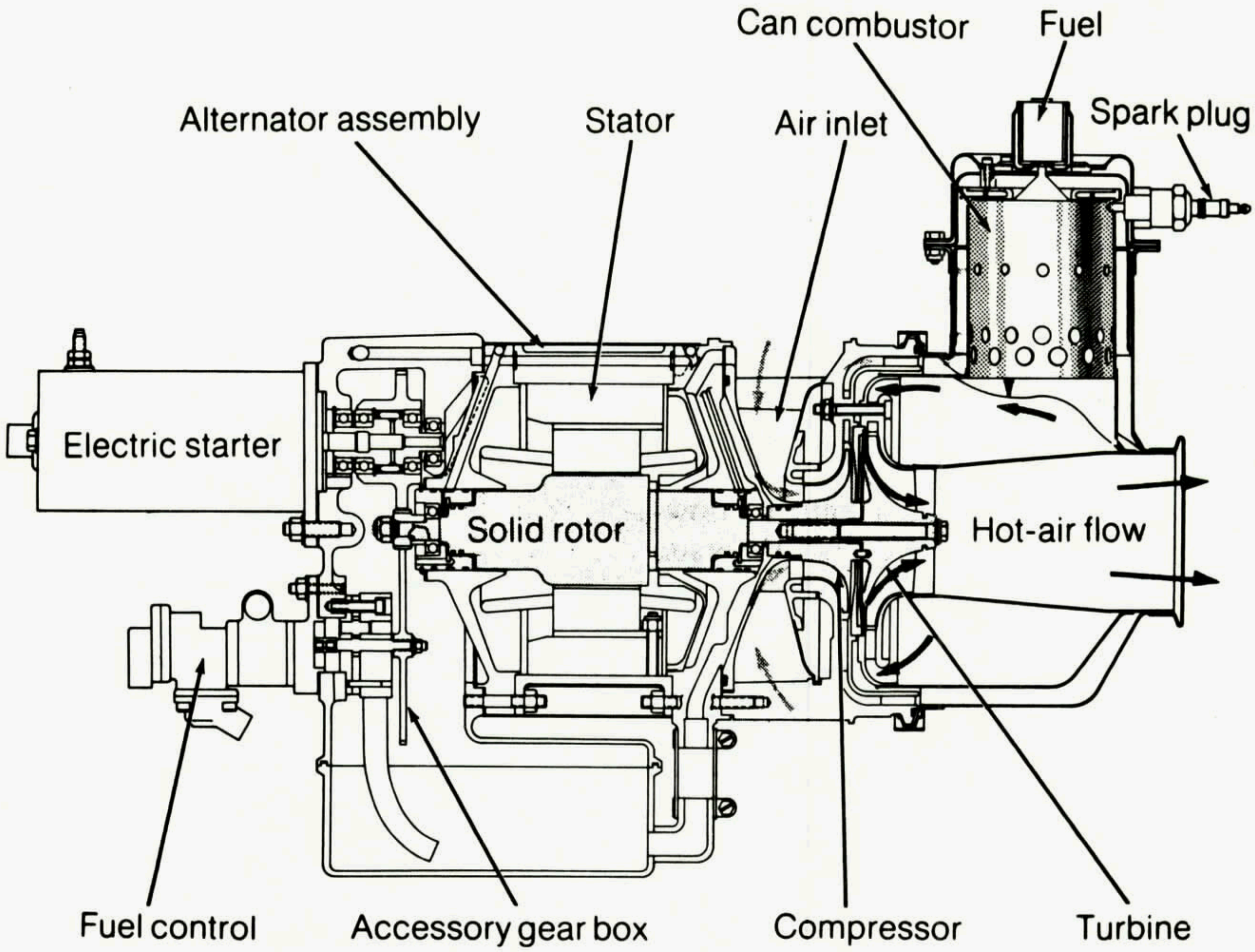

Figure 7

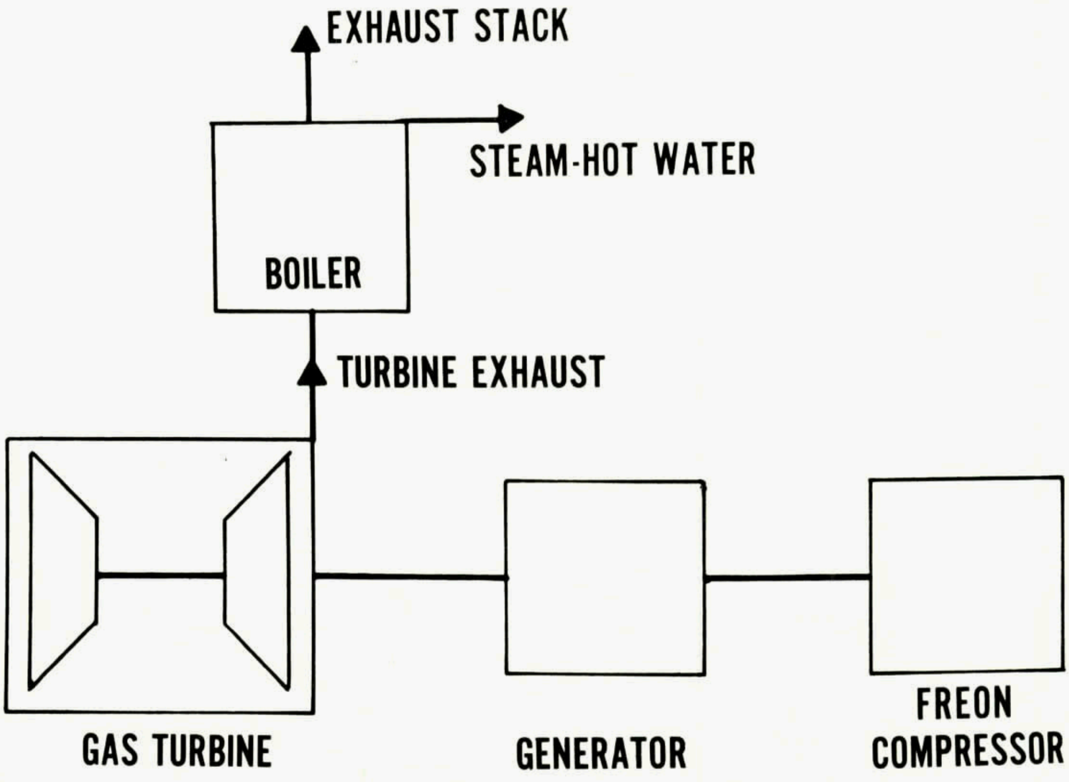

Fig.8 Total energy concept

UNDER 100 HP	330,000
100-200 HP	250,000
200-400 HP	150,000
400-1,000 HP	20,000
1,000 HP AND HIGHER	5,000
	755,000

Fig.9 Approximate yearly U.S. diesel engine market

- **RADIAL COOLED TURBINE GOALS**

MASS FLOW	5 lb/sec
ADIABATIC EFFICIENCY	87.5%
INLET TEMPERATURE	2300°F
PRESSURE RATIO	5
WORK EXTRACTION	220 BTU/lb

- **CENTRIFUGAL COMPRESSOR GOALS**

MASS FLOW	3 lb/sec
PRESSURE RATIO	8 AT 80% EFFICIENCY
PRESSURE RATIO	10 AT 75% EFFICIENCY

- **AXIAL-CENTRIFUGAL COMPRESSOR GOALS**

MASS FLOW	2 - 5 lb/sec
PRESSURE RATIO	12 - 18

- **COMBUSTOR GOALS**

MASS FLOW	3 lb/sec
PATTERN FACTOR	1-2
EXIT TEMPERATURE	2500°F
PRESSURE DROP	3%
EFFICIENCY	99%
TEMPERATURE RISE	1660°F

Fig.10 Second generation technology

THE STATE OF THE ART OF SMALL GAS TURBINE ENGINES FOR HELICOPTERS AND SURFACE TRANSPORT

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SUMMARY

The paper reviews the current technical and market status of below 1000 SHP turboshaft engines, as applied to helicopters and surface transport. Major data are given for the successful engines and comparisons of salient design features are made. Engines now in development are discussed. On the basis of an industry survey, advances to be expected in a 1980 helicopter engine are described and the expectations are critically reviewed. 1980 surface transportation engines are treated similarly, though in less technical detail. The main challenges for the engine designer and manufacturers in the surface transport field are brought out.

SOMMAIRE

Cet essai examine l'état actuel des techniques et des marchés pour les turbo-moteurs installés dans les hélicoptères ou véhicules pour le transport terrestre. On discute les données les plus importantes des moteurs qui ont réussis et on compare leurs principaux avantages techniques. On décrit aussi des moteurs en voie de mise au point. A partir d'une étude faite dans l'industrie pour ce genre de machines, on prédit ce que devra être le moteur pour hélicoptère en 1980 et, avec un oeil critique, ces prédictions sont passées en revue. Il en est de même pour ce qui concerne le moteur pour le transport terrestre en 1980, mais d'une manière moins détaillée. On a eu soin aussi de faire ressortir les principaux défis qui confrontent les bureaux d'étude et les ateliers de fabrication des moteurs pour véhicules de transport terrestre.

INTRODUCTION

The terms of reference given to us for this paper suggested a review of the present state of the art, as well as a forecast of future developments. The paper will accordingly be organised into two parts, dealing with the present and the future. By "present" state of the art we mean engines now in production. "The future" is defined as engines entering production in 1980.

The definition of a "small" engine is one of less than 1000 SHP, output power being the most useful criterion. This is arbitrary, but a convenient compromise for the purpose of the discussion. Rationally, one can say that engines below 1000 SHP must be configured technically with price prominent in the designers' minds, whereas above this size there is not quite the same pressure for maximum attention to cost, in that performance (low SFC) and other considerations tend to assume relatively more importance. Another justification is that surface prime mover applications may occur in considerable numbers up to this approximate size in future, warranting mass production; beyond 1000 SHP the market will be smaller.

The authors thank United Aircraft of Canada Limited for permission to produce this paper; Allison Division of General Motors and Pratt & Whitney Aircraft for permission to quote service statistics; and the following companies for participating in a survey related to future developments: Avco Lycoming, Fiat, Garrett, Leyland, Motoren und Turbinen Union of Munich, Rolls Royce, Solar, Teledyne Continental and United Aircraft Corporation.

We will now turn to the present state of the art, dealing first with helicopter powerplants and then with surface prime movers. In general the description emphasizes production engines rather than prototypes, though announced new developments are included. Auxiliary Power Units are not included in order to keep the paper to a manageable length. Engine data quoted are taken from reference 1, supplemented by brochures and material from the technical press.

PART I - CURRENT STATUS

Helicopter Engines

The Allison General Motors 250 or, in military designation, T63, is one of the most widely used helicopter powerplants today. In the HP range discussed here, this engine, the Turbomeca Artouste and the Bristol Siddeley Nimbus, which is a derivative of the Artouste and going out of production (reference 2), are the only ones flying in Western production helicopters. The MIL-2 helicopter is powered by a Russian 400 SHP engine, similar in layout to the Allison, but considerably heavier and poorer in fuel consumption. Fig. 1 shows major data of the Allison 250-C18 model which was type certificated in September 1965. About 5000 engines have been produced for both military and civilian usage.

Fig. 2 shows major data of the Turbomeca Artouste 3B which is powering the Alouette III helicopter. Reference 1 states that this model is derated from 870 SHP. (The 130 Kg weight in Fig. 2 is taken direct from a brochure; it is 24 Kg lower than given in reference 1). A smaller Artouste version powers the Alouette II and the Artouste 3N at 592 SHP powers the SA 341 Gazelle. Compared with the 3B, the 3N has lower mass flow, i.e. it does not appear to be a derated engine, it has 6:1 pressure ratio, and these combine to improve brochure SFC to .627 lb/SHP-hr. (284 gr/PS-hr.). These engines are all fixed shaft designs.

The issue of free versus fixed shaft was hotly debated in the U.S.A. some years ago and resolved there in favour of the free shaft. The free shaft engine's nearly constant power as speed is reduced promotes rotor speed stability. This is a feature that enhances safety in that it is more forgiving of pilot error, and this was the telling point in the U.S. debate. In the light of a recent analysis of fiscal 1970 U.S. military helicopter accidents (reference 3), which ascribed 50% of the causes to crew error, this seems to have been a good decision.

Fig. 3 shows major data of the UACL PT6B-16 engine which has powered the Lockheed 286 rigid rotor helicopter and several experimental helicopters. It has not to date been installed in any production helicopters.

The basic design of the North American engines mentioned dates back to the late fifties. Turbomeca's concepts originated earlier. The engines all feature compressors having a combination of one or more axial stages and a final centrifugal stage. At the time these engines were designed this was the preferred way of achieving pressure ratios in the range of about 6, with adequate efficiency at low mass flows. The 800 SHP Garrett TSE 331-3U-303, some of which have been used in Sikorsky S-55 conversions, and the prototype 375 SHP MTU 6022 use two centrifugal stages. There is a wide diversity in burner design, ranging from the centrifugal fuel injection Turbomeca style, to single injector designs of the Allison type and MTU 6022 scroll type, and the multi-injector, annular Garrett and UACL PT6 design. All engines use axial turbines except for the 240 SHP Garrett TSE 36 which is derived from an Auxiliary Power Unit. It uses a radial turbine (reference 4). The free shaft engines use both one and two stage gas generators as well as one and two stage power turbines. The construction of the axial turbines varies from all-cast wheels to one-piece forged and machined wheels to cast blades inserted into forged hubs. The layout of the two free shaft engines, the T63 and the PT6, is characterised by avoidance of concentric shafting through the compressor, which in both cases necessitated turning the exhaust gases through 90° or more in a confined space. The Teledyne Continental 600 SHP T72 engine, which was not taken through into production, did step up to concentric shafting however.

The weights of the engines mentioned so far are relatively high, in excess of .385 lb/SHP. This reflects an initial conservative design or rating approach. As an indication of progress it may be pointed out that the Allison C-20 weight ratio is 12% better than the C-18's. Its volume /SHP ratio is also improved by 20%. This Allison 250 C-20 was certificated in 1970. Fig. 4 gives major details. Compared with the earlier C-18 engine, mass flow is up 20% and pressure ratio up one ratio. The six axial plus one centrifugal stage compressor design of the C-18 has been retained.

Before discussing more recent designs, some remarks will be made regarding costs. Early hopes for lower price levels have not yet materialised so that turbines marketed to date have remained significantly more expensive than turbocharged piston engines. While work is going on to produce lower cost designs, notably at N.A.S.A. where efforts are under way to evolve a low cost small turbofan (reference 5), the piston engine continues to set a challenging standard.

Factors contributing to the higher cost of turbine engines are several. The turbine engine uses more expensive and difficult to handle raw materials, so that forging and casting costs, quite apart from machinability costs, are higher. Production technology has not to date solved the problem of producing at low cost the various complicated high temperature sheet metal weldments that proliferate in the aero gas turbine. Material, geometry and fabrication considerations make quality control expensive. Fuel pump, control and fuel nozzles cost two to three times the equivalent components of a typical 350 SHP turbocharged fuel injection piston engine. Compared with piston engines, turbine engine production runs for any given model are also smaller. This is apparent from small aircraft production statistics.

These, plus amortization of high development and start-up expenses, are some reasons why the cost progress of small aero gas turbines has been modest. Fig. 5 shows a typical cost breakdown by component group for the United Aircraft of Canada PT6B-16 and for quick comparison the corresponding weight breakdown. It is interesting that weight and cost percentages are similar, except, as one might expect, for the fuel systems group. Note that reduction gearing seems relatively inexpensive.

It is useful to mention three developments which have contributed to cost reduction. The first is the development of successful methods of casting integral compressor and turbine wheels, the second is casting of integral turbine nozzle rings with and without provision for cooling, and the third is centrifugal impeller machining based on methods which intimately combine machine motion and cutter geometry with the generation of aerodynamically and structurally efficient vane and hub profiles. Fig. 6 shows an impeller which is designed and machined in this manner from a titanium forging. It is hoped, one day, to use a titanium casting, thus reducing or eliminating passage machining. Casting of compressor and turbine wheels however entails weight and performance penalties. The casting process is limited by considerations of filling the thin trailing edges of the blades, which in turn implies working with lower aspect ratio, longer chord designs. The relatively thick trailing edges are not compatible with maximum aerodynamic efficiency. Long chords entail weight penalties.

Turning to more recent designs, Fig. 7 shows major data for the Garrett TSE 231 475 SHP engine which is currently in development. It features concentric shafting, which necessitates two intershaft bearings, according to illustration in reference 4. Compare this with no intershaft bearings on some large, modern two spool turbofan engines. This is the dynamics price for small scale coupled with high tip speed, radial components.

A characteristic of the new generation engines is higher compressor pressure ratio. The pressure ratio of the Garrett 231 engine is 8.6:1, and it is achieved by two stages of centrifugal compression following the example of the 331 turboprop engines. The pressure ratio of the Rolls Royce RS 360 is 12:1 and the design here follows recent Rolls Royce practice by the adoption of multispooling. The RS 360 has a 830 SHP take-off and a 900 SHP contingency rating, and it is aiming at very low fuel consumption, namely .49 lb/SHP/hr. (220 grams/PS/hr.). It must have been a difficult decision to adopt such a layout, i.e. three concentric shafts, on this small dimensional scale. There is little space available to design oneself out of development troubles in the bearing/shafting/oiling/sealing functions, and the complexity must inevitably make itself felt in production cost.

Another noteworthy feature of the TSE 231 is the single stage compressor turbine. It must have high stage loading by virtue of the elevated engine pressure ratio, and combining this with the small size, 4.3 lb/sec. (1.95 Kg/sec.), the resulting efficiency must be compromised. The decision was probably influenced by cost and by shaft dynamics considerations. The RS 360 features a total of four turbines, one for each compressor spool as might be expected and two power turbine stages. Incidentally, it is the first small aero engine equipped with a vapourising fuel system.

Although the UACL PT6 Twin Pac which went into production in 1970 is an 1800 SHP powerplant, each barrel of this engine has an output of 900 SHP and it may thus qualify for brief mention in this survey. Fig. 8 is a picture of this engine and Fig. 9 shows major data. Reference 1 lists a similar Turmastazou model of the same power.

Small engines so far have all employed uncooled turbine blading and the new generation engines are no exception. Indeed, cooling in these small sizes will be more difficult than cooling in large size engines. The requirements of economic use of cooling air, good durability, plus manufacture to reliable quality standards, will set particularly onerous tasks for the designer of small hardware. There are minimum wall thicknesses and minimum passage widths limits which are set by these considerations, and these, plus the sheer lack of space provide restrictions which will inevitably make for lower cooling effectiveness than in large size engines. Hence, the small engine will not benefit from cooling to the same extent as its bigger brothers. The PT6 Twin Pac and the TSE 231 (reference 1) operate at the 1900°F (1311°K) temperature level. The blade material employed by the former is IN 100 and the vanes are WI 52 but cooled. To illustrate the need for thicker than aerodynamically desirable geometry, the nominal trailing edge thickness of these vanes is .063" (1.6 mm) which is 12% of the throat opening.

As a last item in this section, helicopter engine reliability will be briefly reviewed. Fig. 10 is a résumé of the service experience of the Allison 250. Its time between overhaul (TBO) in commercial helicopters is 1125 hours as of December 1970. No data were received from Turbomeca. Though United Aircraft of Canada Limited has only turboprop statistics, these indicate that with continued development (5 million flying hours) typical airline reliability achievements, such as .16 chargeable premature removals per 1000 hours and .03 in-flight shutdowns per 1000 hours (1970 average for the Pratt & Whitney Aircraft JT3D) can be equalled. Admittedly the TBO of the airline engines is much higher. In an attempt to relate the arduousness of fixed versus rotary wing installations, we have compared service statistics for the Pratt & Whitney Aircraft JT12 jet engine with its JFTD12 helicopter counterpart. This is one of the few powerplants for which such a comparison should be possible. Though the comparison basis in terms of engine flying hours is rather limited (the JFTD12 hours in the Sikorsky Skycrane are 61,000 to date), these data indicate that there should be little engine-chargeable reliability difference between fixed wing aircraft and helicopters given appropriate product improvement efforts.

Surface Engines

Turning now to surface transportation use, a number of engines developed originally for aeronautical applications have been applied to sundry surface uses, though these have to date not been large in terms of numbers of engines sold. Some of the surface applications have been quite exotic, such as illustrated in Fig. 11, which shows a PT6 engine powering a successful piece of snow removal equipment. This remains a single example of such an installation inspite of its good record. The turbine engine here scores by way of high power density. This same reason, coupled with consequent vehicle design innovations, has also brought about some train applications of below 1000 SHP engines such as the ST6 powered Turbotrain shown in Fig. 12. An experimental train with a Garrett engine and using electric drive is in operation in the U.S.A. A Japanese train is also in operation, using derated Lycoming T53 engines. Also in existence is an Astazou powered version of the French Bertin tracked air cushion train.

Small gas turbines are a "natural" powerplant for small Air Cushion Vehicle lift and/or propulsion power, at least in non tracked designs. For tracked designs, the long term objective is the linear induction motor.

The gas turbine has not penetrated significantly as a powerplant for small marine craft. It must be concluded that price is the commercial obstacle since remarkable performance has been demonstrated and since appreciable space is saved.

There have been many prototype applications in road and off-the-road vehicles. The most extensive truck efforts are those by Ford. The most comprehensive automobile efforts were probably Chrysler's who tested 50 cars with a "fourth generation" engine (reference 6). Some laboratory work on subsequent generation engines is stated to be going on (reference 7). Special mention must also be made of Mr. Noel Penney and his Rover team who were the first to run a turbine powered car and who performed an important publicity service to the whole industry by successfully completing the vingt-quatre heures du Mans.

PART II - THE FUTURE

Preamble

In projecting future technical attainments two approaches are possible. The first consists of predictions based on assembling advances in major individual components. This approach is not followed here since components are the subject of subsequent specialised papers. The second and chosen approach consists of overall predictions based on extrapolation of past trends and on judgment. In order to provide a more valid overall perspective, we have canvassed the industry for its opinion, and not just relied on a one company or author personal judgment. To this end, a questionnaire was sent to all known major turbine companies in Western Europe and in North America active in the small engine field. The nine companies named in the Introduction and United Aircraft of Canada replied to this survey. It is regretted that Ford and General Motors, whose replies would have been particularly interesting, did not respond. The views of any one particular company will not be divulged, as anonymity was promised.

In devising such a questionnaire it is imperative to keep it simple, so as to get a reasonable response. Fig. 13 shows the questions asked in regard to helicopter engines. We did not question weight since forecasts of power to weight ratio are secondary in importance to the questions asked. Also, weight ratio tends to be a function of the thermodynamic design at which the questions are chiefly directed. Nevertheless, some comments on weight are included in the text.

Our requests were mostly addressed to the Directors of Engineering of the companies, and we must presume that the answers reflect an authoritative point of view. This is not intended to sound cynical; it is instructive to quote from one reply: "In some cases I have had to argue with company people competent for the particular problem. In cases where opinions were at variance, I answered in accordance with my own views."

The comments and opinions expressed in this part of the paper reflect the authors' own views and not those of United Aircraft of Canada Limited.

Following the previous order in this paper, helicopter engines will be discussed first, followed by the surface transportation scene.

Helicopter Engines of 1980

Results of the questionnaire will first be presented with brief remarks on the values shown. Overall technical comment will follow at the end. The statistical analysis methods used are described in the appendix.

Turning to the first question, namely specific fuel consumption to be expected in a reasonably priced 1980 production engine, we have the picture shown in Fig. 14. It will be seen that the improvement expected is from .63 lb/SHP-hr. to .52 (282 to 233 gr/PS-hr.), or 17% in the 300 - 500 SHP field, and from .60 to .48 (270 to 215) or 20% in the 700 - 1000 SHP field. As shown by the standard deviation numbers the spread of the forecasts was rather large.

The second question concerned turbine inlet temperature. Results are given in Fig. 15. Note also, by considering the standard deviation, the very high temperatures expected by some respondents. Material/coating systems would require 150 - 250°F (65 - 120°C) improvement to enable these temperatures to be met without cooling. Up to 100°F (38°C) is to be expected with some certainty, albeit at a cost penalty, but beyond this matters become problematical and cooling may have to be relied on.

The next question was designed to ascertain probable compressor pressure ratio. Fig. 16 gives the forecast. Judging by the small standard deviation for the 300 - 500 SHP value, there was remarkable certainty here. However, this may be coincidence, considering the degree of uncertainty associated with the 700 - 1000 SHP value. Both best numbers imply advanced multistage or multispool machinery.

Unanimity was achieved on the question whether heat exchangers would be used. The clear answer is no. This question was included since some years ago such a possibility was actively contemplated in the U.S. for military applications.

Fig. 17 illustrates the answers to the price trend question. The questionnaire invited chief reasons for the particular reply, but no one commented. Interestingly enough, though the percentage fuel consumption improvement was about the same for the two size fields considered, respondents were fairly sure that the small size price would be static, whereas, though the majority still considered prices would stay the same, they were more inclined to predict a price increase in the larger size. This apparent inconsistency may reflect the feeling that production rates in the lower SHP range would be higher. Perhaps also respondents meant to say that price will need to remain the same in the small size, but that some price increase in the larger sizes may be more acceptable. Further comment on price will follow.

Finally, as to durability and reliability, 37% of respondents consider it will stay at current standards and the remaining 63% expected significant improvements. In the light of the earlier comments on service statistics, this appears a reasonable expectation. However, the increased engine complexity required to lower fuel consumption will make this a more difficult task.

Back calculating from the most likely values for SFC, pressure ratio and temperature, and assuming a reasonable range of cycle pressure and cooling losses, expansion and gear efficiencies, and allowing a 3% guarantee margin, the .52 SFC for the 300 - 500 SHP size implies a total to static adiabatic compressor efficiency of 76% - 79% at an airflow of just under 3 lbs/sec (.136 Kg/sec) for 400 SHP, and the .48 SFC for 700 - 1000 SHP engines implies an efficiency of 78% - 81% at an airflow of just over 5.5 lbs/sec (2.5 Kg/sec) for 850 SHP. Allowing for Reynolds number effects due to size, both power ranges thus require nearly the same compressor quality. The required efficiency values correspond to the 7:1 pressure ratio compressors typical of today's production engines, i.e. compressors of 1960 original design vintage plus 10 years of development. Thus in another 10 years respondents expect a formidable advance, namely doubling of pressure ratio at constant efficiency, while maintaining adequate surge margin and distortion tolerance.

One of the new engines now being developed, the Garrett TSE 231, has a fuel consumption of .605 lb/SHP-hr. (270 gr/PS-hr.). Its two stage centrifugal compressor will have efficiency in the calculated range mentioned, but its pressure ratio at 8.6:1 is less than halfway between today's 7 and the expected 12 of 1980. And the rest of the way will be technically a much thornier road than the first half. Thus, the apparent trend is more modest than the expected. Similarly, in today's 700 - 1000 SHP engines, a good combination of pressure ratio and efficiency is 8:1 with 80% - 82% total static. Thus the expected step to 14:1 per Fig. 16, at nearly constant efficiency but lower flow (due to higher specific power) also represents a large step.

As regards turbines, the present day efficiency levels are likely within two points of the best that can be achieved with current stage loadings and mechanical and fabrication restrictions, and using practical tip clearances. Efficiency improvements can therefore be expected chiefly from increasing the number of stages. This appears to have been done to good effect on the Turmastazou for example, where the three stages of the single shaft Astazou are replaced by two gas generator plus two power turbine stages. But efficiency decrements will again be incurred due to blade cooling.

Gearbox and parasitic losses (bearings, fuel and oil pump drives, etc.) will not be reduced. Increased cycle pressure ratio will require detail design refinement to hold windage and leakage losses to today's values.

In predicting SFC for the 700 - 1000 SHP range, respondents were possibly influenced by knowledge of the RS 360 with its below .5 SFC objective and 1974 intended production date. They may also have been influenced by still lower SFC targets for somewhat larger, advanced technology engines. However these designs require increased complexity and hence a higher price which 58% of the respondents did not expect. This contradiction would be overcome to a degree if one considered the RS 360 SFC as more representative of the best that might be expected than of the most probable. When discussing fuel consumption it is also necessary to define terms. The authors' comments reflect the North American commercial and military meaning of SFC, which is that all production engines shipped must demonstrate no worse than the specification values. In practice, this requires that engines must be developed to a better average standard so as to go smoothly through production acceptance testing.

Manufacturers may not only differ in this sense with respect to the fuel consumption they quote in brochures, or which they have in mind when predicting future attainments. Other choices also affect brochure figures, such as acceleration considerations, desired maximum cruise rating at altitude (certainly a consideration with turboprops) or part load fuel consumption. For example, since single shaft engines operating at constant speed can be matched near the surge line where compressors generally exhibit best efficiency, it will be possible to quote lower SFC values than for an equivalent free shaft engine. Similarly, good part load fuel consumption will remain a continuing requirement for most helicopter engines, and high power SFC may be sacrificed to this end.

Let us now examine some price implications. One North American engine is offered for about $75/SHP, according to the technical press. Another has been offered for approximately $50/SHP, though this may have pertained to a special situation. Let us say then that a practical competitive price for 300 - 500 SHP engines of roughly .6 SFC quality (270 gr/PS-hr.) is $60/SHP. Now the answers to the price question indicated constant price expectation, but the answers to the SFC question showed an expected reduction to .52 lb/SHP-hr. (234 gr/PS-hr.). This seems contradictory in view of the greater design sophistication required. Applying estimated increases to the individual component groups in Fig. 5, and allowing for amortization of the larger Engineering development and tooling cost, a required price of $80 - $90/SHP in terms of 1970/71 $, North American costs would result. For 700 - 1000 SHP the price range may be slightly lower due to economies of scale. It would be interesting to have some valid studies from the helicopter industry as to whether an advanced technology engine warrants such high prices in a commercial situation. The authors' prejudgment of the outcome of such a study is that this price is higher than the commercial market would bear. Only a firm military quantity order at the outset of an engine project would encourage an engine manufacturer to embark on such a venture. This was indeed the case with the RS 360. But the prospects of such orders are less and less real as military budgets are shaved.

The forecast technique chosen really required a second cycle whereby the holders of the most extreme opinions are given a chance to modify their views. There was no time for this. Perhaps a second cycle would have moderated the degree of advance expected.

Although, as explained earlier, the questionnaire purposely did not enquire into weight, some comments are in order. The new engines under development, such as the RS 360 and TSE 231, have a weight/take-off or military (intermittent) power ratio of .36 lbs/SHP (.161 Kg/PS), which is at least 10% better than current engines in service. The PT6B-30 model, which is not yet certificated, but which is based on one power section of the production PT6 1800 SHP Twin Pac has a ratio of .30 (.134). Thus the new engines with more sophisticated compressors are not light. The picture will improve with higher turbine temperature, but radical improvements are not to be expected. The cost effectiveness of light weight, composite materials in small engines is questionable.

One finel comment. Today there are comparatively few small gas turbine powered commercial helicopters. To the end of 1970 there were about 1150 single and twin commercial machines powered by Allison in all Western countries (reference 8), and roughly half this number of commercial helicopters powered by all other engines (reference 9). Given the continuously increasing surface congestion, it seems certain that more use must be made of the air, particularly for vital services, so that, by 1980, we should see many single and particularly twin engined helicopters in public and business service. There should be good market opportunities for the company that makes the correct trade between engine quality and costliness.

Future Surface Transportation Engines

The survey questions concerning this field endeavoured not to repeat the technical rigor of the helicopter questionnaire. A broader approach was chosen.

Fig. 18 presents the results of the first question, concerning the percentage of 1980 new vehicle production that would be turbine powered. The only comment here is that the 15% penetration in the 100 - 250 SHP field must represent many hundred thousand engines. This pleasant forecast must be sparked by pollution considerations, but a doubter may ask: "If the turbine is necessary for pollution, why should it not have a much higher penetration? On the other hand, if the piston engine can get away with it, why will there be any turbines at all?" A good question. Is the answer that regulations in some places will be so exceedingly tight as to force all vehicles registered there to be turbine powered? But, at least in the U.S.A., the present trend appears to be towards uniform federal regulation, and away from local regulation (reference 10). Pollution, and a trend towards larger and faster vehicles, will put the case for the turbine on increasingly firm economic grounds, as evidenced by the higher penetration percentage expected in the larger sizes. We would like to refer to a comprehensive recent paper on the subject of turbine economics, reference 11. (Incidentally, economic studies are apt to assume equal driver hourly wages as between Diesels and the gas turbine. But Unions have been known to gear their demands to innovation).

Cost studies for quantity production turbines are being carried out in many places. The most definite report is one from Ford who will price their latest 450 SHP model 707 turbine the same as an equivalent Diesel. This Ford turbine is a free shaft, ceramic regenerator engine, with variable power turbine nozzles. It is not the earlier, more complex and very flat SFC design. The brochure gives full load fuel consumption as .4 lb/SHP-hr. (179 gr/PS-hr.) uninstalled, on a 60°F day. After 30,000 hours of dynamometer testing, the turbine is to enter production in August at 200 units for the last five months of 1971 (reference 12). This will be followed shortly by a 525 SHP engine (reference 13).

Fig. 19 shows the percent penetration in the off-highway field. The results are similar to the on-highway figures. This would appear reasonable, as the power ranges are the same and there is no special reason why the turbine should have an additional advantage in this equipment compared with on-highway trucks. Volume and weight economies are probably equally important in both types of equipment.

The replies to the heat exchanger question, as shown in Fig. 20, show complete uncertainty on this point in the small and the large size range, but a firmer expectation of heat exchangers in the mid-range. Can it be that the indefiniteness of the replies suggests serious doubts that a viable heat exchanger can be developed? (It should be noted however that the authors were unsuccessful in obtaining responses from the large U.S. auto manufacturers who have most experience with this device). Does the uncertainty of the replies in the 100 - 250 SHP light/medium bus and truck range suggest that heat exchanger engines may be too costly in this size? And does the uncertainty in the above 450 SHP super truck and heavy off-highway equipment field suggest that pollution control may force the issue here so that even higher SFC simple cycle engines will be acceptable? At any rate respondents vote 2:1 that the mid power range of 250 - 450 SHP engines will require exchangers. Presumably they are more certain that in this range a heat exchanger is essential for economic viability.

The responses are not very consistent and it is hoped that the discussion will be more enlightening. In the meantime, and considering the rising oil prices which will raise fuel expense, already the largest element of operating costs, it would seem that potential quantity manufacturers of small turbines for revenue vehicles must devote more millions to the continued development of heat exchangers.

Manufacturers having extensive experience with rotary ceramic regenerators seem to keep this experience very much to themselves. It is a pity that they were not lured into the open by inclusion of a specialist topic on the agenda of this conference. There are varying reports on progress in the area of seal wear, cost reduction achievements and matrix durability. One problem which intrigues us is long term carbon formation. Small particles are shot from the burner through the regenerator, where some of them will accumulate, and when this has turned to the cold side, these particles will be blasted back into the burner. Thus a carbon (or dust) loop is set up, with gradual accumulation even in a clean (and well filtered) engine, with consequent erosion. One wonders whether this will be a significant problem.

Fig. 21 again illustrates divided opinion in regard to means of achieving extended turbine blade life. Opinions are equally divided between low inlet temperature, better materials and cooling. As an example of an actual choice, the first production generation Detroit Diesel (GM) turbines are slated to use 1700°F (1200°K) rated inlet temperature (reference 14). So far as we know, the North American durability objective is 10,000 hours without engine removal. In reference 11 it was 700,000 Km. Let us briefly consider the challenge implied. Compared with aero engines, the truck, bus and off-highway turbine must

- spend a larger proportion of its life at elevated turbine temperatures, by virtue of a high average power requirement and maintenance of high temperatures even at part load in flat SFC engines;

- operate on fuel containing more sulphur;

- breathe high density and contaminated air;

- withstand frequent power setting changes;

- endure a relatively indifferent maintenance standard.

Considering these factors, and the much less than 10,000 hours time between overhaul demonstrated by small aero engines to date, it seems hardly warranted to assume a longer "block" durability than for Diesel engines, as is sometimes done in economic studies. Reliability and durability development, and driver and maintenance staff training will have to be thorough, otherwise the turbine will quickly earn itself a poor reputation. Not surprisingly, Fig. 22 shows that manufacturers are agreed that the heat exchanger and the turbines will require the most effort to achieve a good field record.

Finally, Fig. 23 shows the percent development costs that manufacturers expect to devote to the various main tasks required to market a product that will uphold their pre-turbine reputations. Those who have the best gasoline and Diesel engines can expect to spend the most development money. Perhaps this reputation difference also accounts for the standard deviations shown in Fig. 23. At any rate, nearly half the Engineering investment will be devoted to durability and reliability development.

The questionnaire did not address itself to the automobile field. The application of turbines here is dependent on the pollution situation and on whether or not Otto cycle powerplants can be developed clear of the laws. As is well known, GM recently acquired rights to the Wankel engine so as to increase their Otto cycle options. A heat exchanger is questionable because of first cost, so that, if pollution dictates it, the public will have to be prepared to buy considerably more fuel. But the gasoline engine will not die easily. If pollution legislation looks too stringent for it, there will be strong pressure to relax the law to just feasible requirements.

One less obvious problem with the auto gas turbine is high exhaust temperature. Compared with gasoline engines there will be less cooling in the exhaust pipe due to higher mass flow. The automobile exhaust design will thus have to take into account pedestrian hazards, effects on the ground (melting of tarmac, setting fire to grass) and effects on garage equipment and installations. Trucks should be less handicapped in these regards because of the greater freedom of exhaust routing and treatment, and naturally heat exchanger engines will have reduced exhaust gas temperatures.

Marine applications will follow the lead of road vehicles. For the rest, derivatives of aircraft turbines will continue to power those special devices, such as ACV's, where the turbine scores for much the same reasons as in aircraft.

CONCLUDING REMARKS

We have endeavoured to give a broad brush review of the current state of the helicopter turbine art, and of the current degree of turbine usage for surface transport. The latter is meagre, and helicopter usage has to date been largely confined to two turbine models. Lately, twinned engines of below 1000 SHP per power section have been introduced and these are seeing service in at least two larger machines, the Bell 212 and S58T commercially. Military applications have been paramount.

The survey forecast predicts improvement in helicopter engines SFC to .52 lb/SHP-hr. (233 gr/PS-hr.) by 1980 for the 300 - 500 SHP size, and to .48 (215) for the 700 - 1000 SHP size. However, apart from special and increasingly rare military situations, price considerations are likely to dictate more modest progress, since such low values imply too rapid and drastic improvement in simple components. More modest progress is also indicated by extrapolating achieved fuel consumption improvement rates. Weight improvements will result primarily from turbine inlet temperature increase, i.e. specific power. In very low fuel consumption engines weight gains will however be offset by the increasing complexity of the required aerodynamic machinery. Further development of electric and laser machining, casting, pressed powder, inertial welding and similar processes will open up some new design possibilities and bring about detail cost savings. But there are no signs on the horizon that point to really significant break throughs in production technology so that intrinsic manufacturing cost should not get appreciably better.

Reliability is a function of many factors, some of which are outside the design and development engineers' jurisdiction. From a purely technical point of view, improvements are no doubt possible, and indeed necessary to support practical helicopter usage.

The surface transportation future for the gas turbine will be heavily dependent on pollution considerations except perhaps in the above 450 SHP size range where economics are increasingly favoured by weight and space considerations. The technical challenges are formidable, and, except for the very largest manufacturers with captive markets, one may see a trend towards corporate cooperation in the development and manufacture of suitable engines. Inspite of these difficulties, several developments are close to production and the numerous efforts generally underway testify to the industry's determination and expectation that the problems will be overcome.

REFERENCES

1. P. Wilkinson, "Aircraft Engines of the World, 1970".

2. Rolls Royce communication, 13th October, 1970.

3. Col. Eugene B. Conrad, "70 Survivability". AHS-AIAA-UTA Symposium on Environmental Effects on VTOL Designs, November 1970, University of Texas.

4. Verti Flite, January 1970, Volume 16, No. 1.

5. Aviation Week and Space Technology, 1st June, 1970.

6. "History of Chrysler Corporation Gas Turbine Vehicles". Published by Engineering Office, Chrysler Corporation, January 1964, revised August 1966.

7. Chrysler communication, 12th October, 1970.

8. Based on Allison communication, 5th November, 1970.

9. Inferred from Jane's "All the World's Aircraft, 1969-70" and "Helicopter World", June 1970.

10. "Control of Air Pollution from New Motor Vehicles and New Motor Vehicle Engines". U.S. Department of Health, Education and Welfare. Notice of Proposed Rule Making, 29th June, 1970. Federal Register, Volume 35, No. 136.

11. Dir. Dr.-Ing. Dr.-Ing. E.h.B. Eckert, Stuttgart, Die Gasturbine als Antriebsmaschine fur das schwere Strassenfahrzeug. Motortechnische Zeitschrift, Volume 31, 1970, Nos. 5, 7 and 10.

12. Gas Turbine International, July - August 1970.

13. Ford communication, 15th October, 1970.

14. Detroit Diesel brochure. This shows a 280 and a 400 SHP engine, both having 1700°F (1200°K) rated inlet temperatures. For the 400 SHP engine it quotes mass flow of 5.2 lb/sec. (2.36 Kg/sec) and rated pressure ratio of 5.1:1. Pilot production inception is planned for mid-1971. Fuel consumption is not given.

APPENDIX

CORRELATION OF THE QUESTIONNAIRE DATA

Introduction

A total of ten questionnaires was completed and returned by various leading aero-engine manufacturers. In addition, questionnaires were completed by several persons within United Aircraft of Canada Limited. These internal replies were processed independently and the resulting consensus was included with the other replies.

Helicopter Engines

For question 1.1, part (a) (see Figure 13) the "lowest", "most probable" and "highest" values of predicted specific fuel consumption given in each reply were tabulated. Using a weighting factor of two for the "most probable" values and a weighting factor of one for the others, the mean specific fuel consumption and the standard deviation were calculated.

The mean was calculated according to the equation:

$$\bar{X} = \frac{\Sigma x}{n}$$

The standard deviation was calculated according to the equation:

$$\sigma = \sqrt{\frac{\Sigma x^2 - n\left(\frac{\Sigma x}{n}\right)^2}{n-1}}$$

Figure 14 shows the projected specific fuel consumption for the 300 - 500 SHP engine size range and for the 700 - 1000 SHP size range. The specific fuel consumption shown is the mean value as calculated above.

The projected turbine inlet temperatures and compressor pressure ratios are shown in Figures 15 and 16 respectively along with their standard deviations. The calculations were done using the same weighting factors as those used for part (a).

For question 1.2, the percentage of replies predicting each particular price trend for the two engine size classes was calculated and shown in Figure 17.

Surface Transportation Engines

For Figures 18 and 19, concerning anticipated turbine usage, the mean and standard deviation were calculated for each power range using the equations shown above.

The manner of presenting results for Figures 20 and 21 is self explanatory.

The method of analysis concerning the order of difficulty presented by various components, Figure 22, is illustrated by the following fictitious example:

The means and standard deviations of the numbers representing the order of difficulty were calculated and tabulated as shown:

Component	Mean	Std. Deviation	Order
A	1.21	1.10	1
B	3.57	2.37	4
C	2.50	1.81	2
D	2.50	3.71	3
E	4.71	1.27	5

From the "mean" column it can be seen that component A, since its mean is the lowest, is number one in the order. Also, components B and E will be order number 4 and 5 respectively. Components C and D show an equal mean. However the standard deviation for C is smaller, thus indicating a greater degree of certainty as to its difficulty. Hence it will be given a higher order number than component D.

For Figure 23, the mean and the standard deviation were simply calculated for each development category from the questionnaire replies.

SHAFT HORSEPOWER	
TAKE-OFF RATING	317 HP
MAXIMUM CONTINUOUS RATING	270 HP
SPECIFIC FUEL CONSUMPTION	
TAKE-OFF RATING	0.697 lb/HP-hr. (312 g/PS-hr.)
MAXIMUM CONTINUOUS RATING	0.706 lb/HP-hr. (316 g/PS-hr.)
PRESSURE RATIO	6.2:1
TURBINE INLET TEMPERATURE	1825°F (1269°K)
AIR MASS FLOW	3.0 lb/sec (1.36 kg/sec)
WEIGHT	139 lb. (63 kg)
DIMENSIONS	LENGTH 40.0 in. (1017 mm.)
	WIDTH 19.0 in. (483 mm.)
	HEIGHT 22.5 in. (572 mm.)

Fig.1 Allison 250 C-18 free turbine turboshaft characteristics

SHAFT HORSEPOWER	
TAKE-OFF RATING	550 HP
MAXIMUM CONTINUOUS RATING	550 HP
SPECIFIC FUEL CONSUMPTION	
TAKE-OFF RATING	0.72 lb/HP-hr. (322 g/PS-hr.)
MAXIMUM CONTINUOUS RATING	0.72 lb/HP-hr. (322 g/PS-hr.)
PRESSURE RATIO	5.2:1
AIR MASS FLOW	9.5 lb/sec (4.3 kg/sec)
WEIGHT	287 lb (130 kg)
DIMENSIONS	LENGTH 71.5 (1815 mm.)
	WIDTH 19.9 (507 mm.)
	HEIGHT 24.7 (627 mm.)

Fig.2 Turbomeca Artouste 3B single shaft turboshaft characteristics

THERMODYNAMIC SHAFT HORSEPOWER	
TAKE-OFF RATING	732 HP
MAXIMUM CONTINUOUS RATING	732 HP
THERMODYNAMIC SPECIFIC FUEL CONSUMPTION	
TAKE-OFF RATING	0.610 lb/HP-hr. (273 g/PS-hr.)
MAXIMUM CONTINUOUS RATING	0.610 lb/HP-hr. (273 g/PS-hr.)
MECHANICAL SHAFT HORSEPOWER	
TAKE-OFF RATING	690 HP
MAXIMUM CONTINUOUS RATING	690 HP
SPECIFIC FUEL CONSUMPTION (MECHANICAL)	
TAKE-OFF RATING	0.618 lb/HP-hr. (276 g/PS-hr.)
MAXIMUM CONTINUOUS RATING	0.618 lb/HP-hr. (276 g/PS-hr.)
PRESSURE RATIO	7.0:1
AIR MASS FLOW	6.5 lb/sec (3.1 kg/sec)
TURBINE INLET TEMPERATURE	1740°F (1222°K)
WEIGHT	269 lb (122 kg)
DIMENSIONS	MAX. DIAMETER 19 in. (483 mm.)
	LENGTH 59.5 in. (1515 mm.)

Fig.3 UACL PT6B-16 free turbine turboshaft characteristics

	250-C20	
SHAFT HORSEPOWER		
TAKE-OFF RATING	400 HP	
MAXIMUM CONTINUOUS RATING	346 HP	
SPECIFIC FUEL CONSUMPTION		
TAKE-OFF RATING	0.630 lb/HP-hr. (282 g/PS-hr.)	
MAXIMUM CONTINUOUS RATING	0.645 lb/HP-hr. (289 g/PS-hr.)	
PRESSURE RATIO	7.2:1	
AIR MASS FLOW	3.6 lb/sec (1.6 kg/sec)	
WEIGHT	155 lb (70 kg)	
DIMENSIONS	LENGTH	40.3 in. (1023 mm.)
	WIDTH	19.0 in. (483 mm.)
	HEIGHT	22.5 in. (572 mm.)

Fig.4 Allison 250 C-20 free turbine turboshaft characteristics

	% COST	% WEIGHT
INTAKE, COMPRESSOR AND DIFFUSER	35	36
BURNER, FUEL SUPPLY AND IGNITION SYSTEMS	19	12
TURBINES, EXHAUST AND POWER TURBINE SHAFT	26	26
ACCESSORY GEARBOX	11	13
REDUCTION GEARBOX	9	13

Fig.5 PT6B-16 cost/weight breakdown

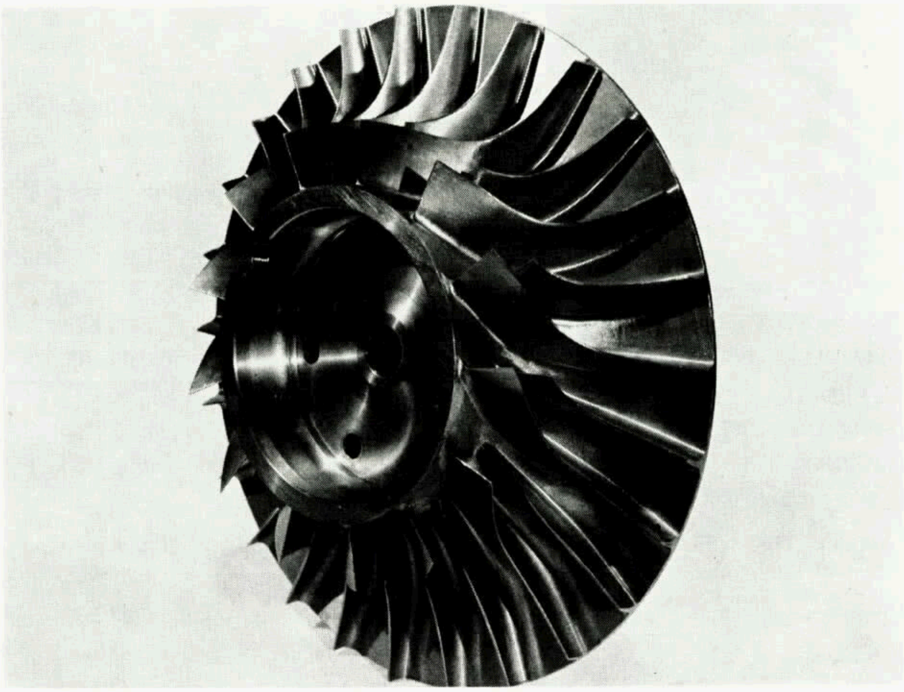


Fig.6 PT6 impeller

SHAFT HORSEPOWER	
TAKE-OFF RATING	474 HP
MAX.CONTINUOUS/NORMAL RATING	403 HP
SPECIFIC FUEL CONSUMPTION	
TAKE-OFF RATING	0.605 lb/HP-hr. (271 g/PS-hr.)
MAX.CONTINUOUS/NORMAL RATING	0.619 lb/HP-hr. (277 g/PS-hr.)
PRESSURE RATIO	8.6:1
TURBINE INLET TEMPERATURE	1900°F (1312°K)
AIR MASS FLOW	4.3 lb/sec (1.95 kg/sec)
WEIGHT	171 lb (77 kg)
DIMENSIONS	DIAMETER: 15.2 in. (386 mm.)
	LENGTH: 41.0 in. (1041 mm.)

Fig.7 Garrett Airesearch TSE 231-1 free turbine turboshaft characteristics

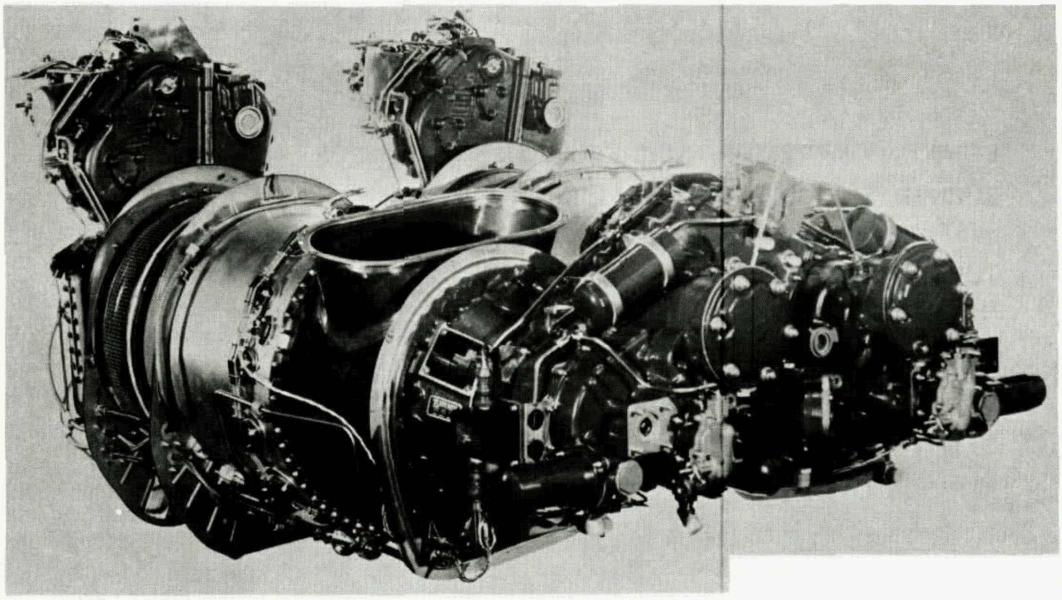


Fig.8 PT6T-3 Twin-Pac

		TOTAL OUTPUT	SINGLE POWER SECTION
SHAFT HORSEPOWER			
TAKE-OFF RATING		1800 HP	900 HP
MAXIMUM CONTINUOUS RATING		1600 HP	800 HP
SPECIFIC FUEL CONSUMPTION			
TAKE-OFF RATING		0.595 lb/HP-hr.	(266 g/PS-hr.)
MAXIMUM CONTINUOUS RATING		0.599 lb/HP-hr.	(268 g/PS-hr.)
PRESSURE RATIO		7.2:1	
AIR MASS FLOW (Per Power Section)		6.59 lb/sec	(3.0 kg/sec)
WEIGHT		617 lb	(281 kg)
DIMENSIONS	LENGTH	65.3 in.	(1659 mm.)
	WIDTH	44.4 in.	(1128 mm.)
	HEIGHT	31.6 in.	(803 mm.)

Fig.9 UACL PT6T-3 twin turboshaft characteristics

TOTAL FLIGHT HOURS	1,860,000
PREMATURE REMOVAL RATE	0.8/1000 hrs.
IN-FLIGHT SHUT-DOWNS (250-C18 ONLY)	0.19/1000 hrs.

NOTE: REMOVAL AND SHUT-DOWN RATES ARE BASED ON ENGINE RELATED EVENTS ONLY.

Fig.10 Allison 250-C18/T63 operational experience in helicopter applications through December 1970

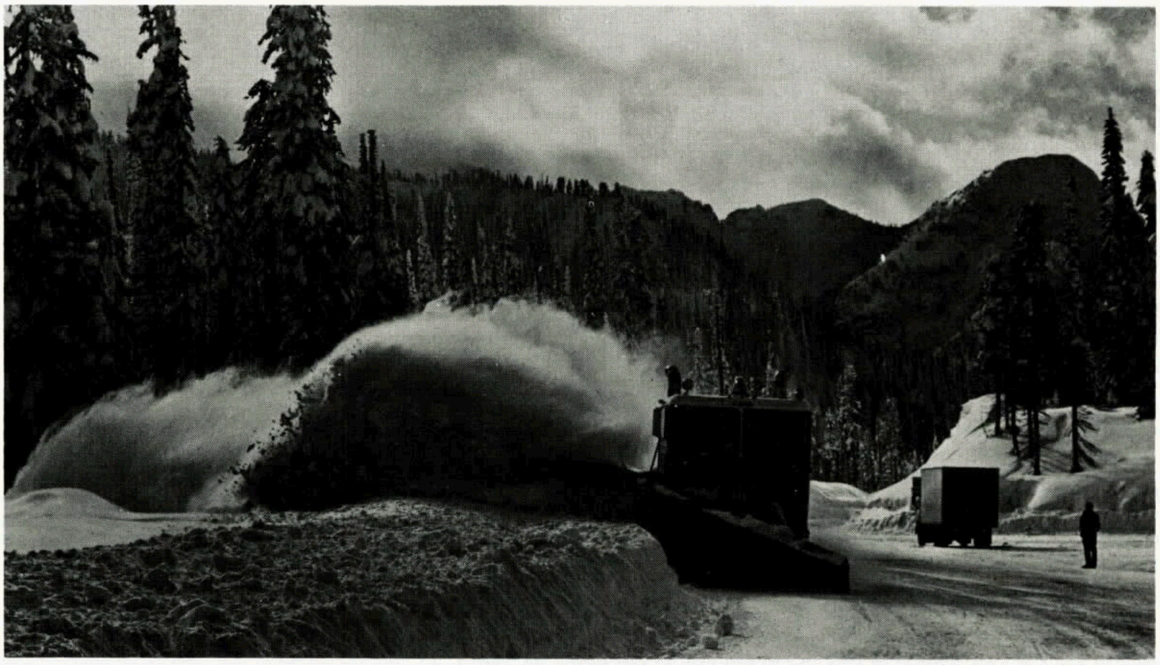


Fig.11 ST6 powered snowplow

Fig.12 ST6 powered CNR Turbo Train

TECHNICAL FORECASTS: CONSIDERING NOT ONLY THE PROBABLE PROGRESS IN TECHNOLOGY, BUT ALSO PRACTICAL MARKET PRICE RESTRAINTS, PLEASE ANSWER THE FOLLOWING QUESTIONS TO THE BEST OF YOUR JUDGMENT.

1. 1980 PRODUCTION TURBOSHAFT ENGINES FOR HELICOPTERS:

ENGINE POWER RANGE (SHP)

(TAKE-OFF, SEA LEVEL, STATIC, 15°C)

300 to 500 — 700 to 1000

1.1 ENGINE CHARACTERISTIC

	(UNITS)	LOWEST	MOST PROBABLE	HIGHEST	LOWEST	MOST PROBABLE	HIGHEST
a. SPECIFIC FUEL CONS.	( )						
b. TURBINE INLET TEMP.	( )						
c. PRESSURE RATIO	( )						

	YES	NO	YES	NO
d. IS AN EXHAUST HEAT EXCHANGER LIKELY?				

1.2

	INCREASE MEASURABLY	REMAIN THE SAME	DECREASE MEASURABLY	INCREASE MEASURABLY	REMAIN THE SAME	DECREASE MEASURABLY
MOST LIKELY PRICE TREND (IN TERMS OF 1970 $/HP)						

CHIEFLY WHY?

1.3

	SIGNIFICANTLY GREATER	APPROXIMATELY THE SAME
RELIABILITY AND DURABILITY TREND (RELATIVE TO CURRENT STANDARDS)		

Figure 13

	300 – 500 SHP	700 – 1000 SHP
SPECIFIC FUEL CONSUMPTION	0.52 lb/HP-hr. (233 gr/PS-hr.)	0.48 lb/HP-hr. (215 gr/PS-hr.)
STANDARD DEVIATION OF REPLIES	0.08 lb/HP-hr. (36 gr/PS-hr.)	0.06 lb/HP-hr. (27 gr/PS-hr.)

Fig.14 Projected specific fuel consumption for 1980 production helicopter turboshaft engines

	300 – 500 SHP	700 – 1000 SHP
TURBINE INLET TEMPERATURE	2048°F 1393°K	2089°F 1416°K
STANDARD DEVIATION OF REPLIES	195°F 108°K	219°F 122°K

Fig.15 Projected turbine inlet temperature for 1980 production helicopter turboshaft engines

	300 – 500 SHP	700 – 1000 SHP
PRESSURE RATIO	12	14
STANDARD DEVIATION OF REPLIES	0.7	3.8

Fig.16 Projected compressor pressure ratio for 1980 production helicopter turboshaft engines

TREND	PERCENT OF REPLIES	
	300 – 500 SHP	700 – 1000 SHP
INCREASE	14%	36%
REMAIN SAME	65%	58%
DECREASE	21%	6%

Fig.17 Projected price trend for 1980 production helicopter turboshaft engines referred to 1970 $/HP

ENGINE POWER RANGE	100 – 250 SHP	250 – 450 SHP	450 – 700 + SHP
VEHICLE APPLICATION	BUS, LIGHT & MEDIUM TRUCKS	HEAVY TRUCKS	"SUPER" TRUCKS
PERCENT TURBINE POWERED	15%	32%	59%
STANDARD DEVIATION OF REPLIES	15.2%	21.8%	30.1%

Fig.18 Percent of 1980 production highway vehicles powered by gas turbine engines

ENGINE POWER RANGE	100 – 250 SHP	250 – 450 SHP	450 – 700 + SHP
VEHICLE APPLICATION	HEAVY TRACTORS, LIGHT – MEDIUM DUMP TRUCKS	HEAVY SCRAPERS MEDIUM DUMP TRUCKS	HEAVY DUMP TRUCKS
PERCENT TURBINE POWERED	11%	24%	47%
STANDARD DEVIATION OF REPLIES	7.9%	16.3%	26.3%

Fig.19 Percent of 1980 production off-highway vehicles powered by gas turbine engines

ENGINE POWER RANGE	100 – 250 SHP	250 – 450 SHP	450 – 700 + SHP
REPLIES FOR HEAT EXCHANGERS	47%	63%	53%
REPLIES AGAINST HEAT EXCHANGERS	53%	37%	47%

Fig.20 1980 production land vehicle gas turbine engines equipped with heat exchangers

METHOD	% OF REPLIES
MODEST TURBINE INLET TEMPERATURE	39%
NOVEL MATERIALS AND COATINGS	37%
HIGH DEGREE OF VANE AND BLADE COOLING	24%

Fig.21 Most likely means of ensuring long-life turbine components

COMPONENTS ARE LISTED IN DECREASING ORDER OF DIFFICULTY

HEAT EXCHANGER
TURBINE
BURNER
FUEL SYSTEM
BEARINGS
OTHERS
COMPRESSOR

Fig.22 Order of technical difficulty in achieving a maintenance and reliability record equal to or better than the heavy truck diesel

DEVELOPMENT GOAL	% OF DEVELOPMENT COST	STANDARD DEVIATION OF REPLIES
ADEQUATE FUEL CONSUMPTION	27%	12.0%
LOW EMISSION AND NOISE	7%	5.7%
RELIABILITY AND DURABILITY	41%	18.2%
LOW COST DESIGN FOR PRODUCTION	25%	12.7%

Fig.23 Development cost breakdown for A 200–500 SHP truck turbine engine

CYCLES DE TURBINES A GAZ

par

P. ALESI - R. LAURENS

Direction Technique

S.N.E.C.M.A. - VILLAROCHE

La puissance d'une turbine à gaz est définie par quatre paramètres : le débit d'air, le taux de compression, la température entrée turbine et le rendement des éléments (slide 1).

Le débit est directement lié à la taille de la turbomachine tandis que le taux de compression est une variable de cycle. La température devant turbine est aussi une variable de cycle, mais sa valeur est limitée, à matériau donné, par la taille de la machine à cause des problèmes de refroidissement. Les rendements de compression et de détente sont fonction de la taille de chaque étage donc du cycle puisque les hauteurs de passage sont d'autant plus petites que le taux de compression est élevé.

La puissance d'une turbine à gaz est donc directement liée à son cycle thermodynamique et à sa taille. Il s'ensuit qu'une turbine à gaz pourra être dite petite, non seulement par son niveau de puissance mais aussi par son cycle thermodynamique.

Cependant, si l'influence de paramètres tels que le débit, le taux de compression et la température devant turbine sur les performances d'un turbomoteur est bien connue, il peut être utile ici de rappeler l'effet des rendements élémentaires (slide 2).

Ces courbes montrent la variation de la consommation spécifique d'une turbine à gaz en fonction du taux de compression pour plusieurs valeurs du rendement polytropique du compresseur. Les courbes à rendement constant présentent un minimum pour une valeur du rapport de pressions d'autant plus grande que le rendement est élevé.

Avec un premier compresseur de rapport 6/1, on obtenait les performances représentées par le point P6. Deux autres compresseurs ont été essayés respectivement à 10/1 et 12/1 de taux de compression. Les niveaux de consommation obtenus sont indiqués par les points P10 et P12. On constate une augmentation de C_s de 46 % quand on passe de 6 à 12 de rapport de pressions alors qu'à rendement constant, on obtiendrait une diminution de C_s de 19 %. Ces résultats montrent l'importance des rendements élémentaires sur les performances des turbines à gaz et ils s'expliquent par la diminution de la hauteur de passage donc du rendement en fonction d'un taux de compression croissant.

Grâce à des études systématiques sur les formes de veine des compresseurs et des turbines axiaux, il a été possible de définir les profils géométriques de plusieurs machines (slide 3).

Chaque veine a été calculée en utilisant les critères de dimensionnement en vigueur pour les compresseurs axiaux et en supposant pour chaque étage une élévation de température totale de 37°C (67 °F).

De plus, pour apprécier l'effet d'échelle en même temps que l'effet de cycle, trois valeurs du débit d'air ont été envisagées : 10 ; 5 et 1 kg/s.

Le rendement d'une grille est essentiellement fonction des pertes de profil et des pertes d'extrémité. Une part importante de ces pertes peut être évaluée en considérant l'influence du nombre de Reynolds calculé sur une dimension caractéristique de la grille, par exemple sa corde (slide 4).

Dans ce cas, nous écrirons que le nombre de Reynolds est égal au produit de la vitesse moyenne par la corde divisé par la viscosité dynamique.

Chaque compresseur étudié étant parfaitement défini géométriquement et thermodynamiquement, il a été possible de définir le nombre de Reynolds caractérisant chaque étage.

Pour chiffrer l'influence du nombre de Reynolds sur le rendement d'étage, nous avons choisi une courbe de variation réalisant un compromis entre des études théoriques qui conseillent une loi en puissance 1/6 ou 1/7 et des résultats d'essais qui présentent une variation beaucoup plus forte (slide 5).

Cette courbe représente les écarts de rendement de compression polytropique en fonction du nombre de Reynolds calculé dans la roue mobile. Cet écart est nul pour une valeur du Reynolds correspondant au 1er étage d'un compresseur axial de 10 kg/s de débit d'air choisi comme référence et dont le rendement mesuré au banc partiel se trouve en parfait accord avec les rendements des machines existantes de cette taille. Les écarts lus à gauche de cette valeur de base sont à déduire du rendement de référence afin d'obtenir la valeur correspondant au nombre de Reynolds de l'étage.

Cependant, dans l'évaluation des pertes d'extrémité, il existe un phénomène qui n'est pas lié à la valeur du nombre de Reynolds, c'est l'influence sur le rendement des pertes dues aux jeux en bout d'aube (slide 6).

Cette planche présente une statistique de jeux en bout d'aube portant sur plusieurs moteurs S.N.E.C.M.A. La tolérance est donnée en fonction de la hauteur de l'aube, mais dépend aussi des dilatations donc des matériaux, de la température de fonctionnement, de la forme de la veine et de la technologie du moteur.

Ces considérations expliquent la disparité des points représentatifs, mais il apparaît tout de même que la valeur absolue du jeu en bout d'aube diminue quand la hauteur de l'aube diminue. Cependant, il est clair qu'on ne pourra descendre au-dessous d'une certaine limite qui semble sur nos exemples se situer autour de 0,5 mm.

La combinaison de l'effet de Reynolds et de l'effet de jeu au sommet a permis de mettre en évidence les valeurs relatives des rendements des différents compresseurs étudiés (slide 7).

Quand le nombre d'étages augmente, le nombre de Reynolds varie peu car, bien que les dimensions et vitesses caractéristiques diminuent rapidement, la viscosité dynamique varie de la même facon et équilibre l'effet d'échelle.

La perte de rendement correspondante est pratiquement constante en fonction du nombre d'étages, mais il faut remarquer qu'un résultat un peu différent aurait pu être enregistré si la longueur caractéristique définissant le Reynolds avait été choisie autrement. Par exemple, en utilisant la hauteur de l'aube ou le rayon hydraulique, l'influence du nombre d'étages aurait été plus marquée mais n'aurait pas traduit aussi bien les pertes de profil.

Cependant, en tenant compte de l'effet des jeux en bout d'aube, on obtient une variation sensible du rendement qui diminue de 3 points quand le compresseur passe de 5 à 11 étages.

L'influence de la tolérance au sommet des aubes est également nette si on compare entre eux 3 compresseurs à 5 étages dimensionnés respectivement pour 10 - 5 et 1 kg/s de débit d'air (slide 8).

Ici, l'effet du nombre de Reynolds et l'effet de jeu viennent s'ajouter pour abaisser le niveau de rendement.

De plus, bien que l'énergie spécifique nécessaire à l'entraînement des 3 compresseurs soit la même, puisque l'élévation de température a été prise constante dans chaque étage, le taux de compression correspondant varie suivant la valeur du rendement, c'est-à-dire diminue quand le débit d'air diminue.

Un travail analogue de définition thermodynamique et de dimensionnement a été effectué sur les turbines associées aux différents compresseurs étudiés en les combinant à deux niveaux de température entrée turbine 1 000 et 1 200°C.

Contrairement aux compresseurs, le nombre de Reynolds calculé dans une roue de turbine est peu significatif du rendement de la grille.

Pour apprécier l'effet de taille, nous avons choisi d'utiliser une corrélation entre les rendements de compresseurs et de turbine basée sur des études théoriques de variation du rendement de turbine (slide 9).

La courbe est unique pour deux températures différentes. D'une part, l'augmentation de masse volumique qui accompagne l'augmentation de température favorise le rendement à cause de l'effet d'échelle. D'autre part, la réintroduction dans la veine d'un débit de refroidissement de plus en plus grand vient abaisser le niveau de rendement.

La combinaison de ces deux effets donne une corrélation unique pour 1 000°C et 1 200°C, mais au-delà, l'effet de détérioration du rendement dû au refroidissement l'emporterait sur l'effet de taille.

Les travaux précédents ont permis de calculer une gamme de moteurs entièrement définis par une géométrie de veine et des rendements élémentaires bien déterminés.

Les performances puissance et consommation spécifique reflètent l'influence des rendements et du cycle et mettent particulièrement en évidence l'effet de la taille de la machine (slide 10).

On constate une augmentation très importante de la consommation spécifique quand le débit d'air, donc la taille, diminue.

La valeur absolue de la puissance est surtout sensible au débit et à la température d'entrée turbine. Bien que la puissance spécifique présente un optimum en fonction du taux de compression, donc du nombre d'étages, on s'aperçoit que la puissance réelle est peu sensible au rapport de pressions. En effet, l'augmentation du nombre d'étages s'accompagne d'une diminution de rendement qui vient atténuer l'effet bénéfique d'une haute valeur du taux de compression.

Les différents moteurs étudiés couvrent une gamme de 200 kW à 2 000 ou 2 500 kW suivant la température devant turbine, et il est intéressant de voir quelles sont les régions où le mode de compression par compresseur centrifuge serait préférable aux compresseurs axiaux.

Dans ce but, nous avons porté, sur un réseau, les valeurs des hauteurs de passage sortie compresseur (slide 11).

On constate que les hauteurs augmentent avec la puissance mais qu'elles dépendent aussi directement du nombre d'étages de compression. Par exemple, on s'apercevra qu'une machine à 5 étages et 200 kW a même hauteur de passage qu'une turbine à 11 étages produisant 2 000 ou 2 500 kW.

On peut aussi, sur ce réseau, essayer de définir les domaines d'intérêts des compresseurs axiaux et centrifuges, en déterminant une hauteur au-dessous de laquelle il serait plus intéressant, du seul point de vue rendement, d'utiliser un compresseur centrifuge plutôt qu'un axial. De toute façon, il semble que cette limite existe puisqu'il est rare de trouver des aubes de compresseurs d'une hauteur inférieure à 1/2 pouce, valeur que nous avons indiquée à titre d'exemple sur les courbes.

Il faut cependant noter qu'il n'est pas notre intention de dire qu'il n'y a pas ou qu'il ne devrait pas y avoir de compresseurs centrifuges au-dessus de cette limite et de compresseurs axiaux en-dessous. Notre but est simplement d'essayer de définir une zone d'intérêt pour chaque type de compresseurs en relation avec leurs valeurs de rendement.

Cependant, si l'évaluation de la hauteur de sortie du dernier redresseur donne des renseignements intéressants sur les performances des petites turbines à gaz, il n'est pas possible d'utiliser ce paramètre pour séparer les petites turbomachines des autres, car il ne fait pas intervenir la température devant turbine. Nous avons donc choisi comme paramètre caractéristique la hauteur du distributeur de turbine qui, en plus du débit et de la pression, tient compte de la température maximale du cycle (slide 12).

Dans ces réseaux, d'une manière analogue à ce qui vient d'être fait pour les compresseurs, on pourrait séparer l'intérêt des turbines axiales et centripètes, et c'est pourquoi l'on trouvera une limite fixée ici à 16 mm (5/8") qui semble représenter la hauteur minimum admise sur un distributeur de turbine axiale.

On peut également utiliser ce paramètre pour essayer de distinguer les petites turbines à gaz des autres turbomachines. On constate alors que la puissance n'est plus un facteur suffisant puisqu'il existe des machines à puissances élevées de 2 500 kW pour lesquelles se posent des problèmes de petites machines et puisque l'on trouve des moteurs de 500 kW sans problèmes de taille particuliers.

En fait, il est clair que le distinguo petites turbines à gaz ne saurait se faire en prenant comme seul critère le niveau de puissance car il ne correspond pas à une limite de rendement ni à une limite de taille. On peut même ajouter que les problèmes rencontrés dans les réalisations des petites turbines sont les mêmes que rencontrent les motoristes qui construisent un moteur de 10 kg/s comprimés à 10/1 ou 20 kg/s comprimés à 20/1.

Nous proposons donc d'utiliser un paramètre tel que la hauteur du distributeur de turbine qui, directement dépendant du débit réduit à l'entrée des turbines, donne une bonne appréciation de la véritable taille du turbomoteur.

Les dimensionnements qui viennent d'être faits étaient basés sur le cycle nominal du moteur donnant les performances maximum. A charge partielle, le cycle thermodynamique se déforme par une diminution conjointe de la température entrée turbine du rapport global de pressions et du débit d'air traversant la machine.

La température entrée turbine est un facteur déterminant de la taille du turbomoteur. Au point nominal de dimensionnement, un accroissement de T.E.T. s'accompagne, à puissance égale, d'une diminution de la taille car, malgré la diminution simultanée des rendements, la puissance spécifique augmente. Par exemple, le débit d'air nécessaire à une turbine de 1 500 kW est de l'ordre de 5 kg/s à 1 200° C de T.E.T. mais devient de 7,5 kg/s à 1 000°C devant turbine.

La considération des régimes partiels va mettre en lumière un autre avantage d'une forte température devant turbine (slide 13).

Pour un turbomoteur assez fortement comprimé, sans échangeur, on constate que la température entrée turbine du point nominal, a une influence importante sur le comportement à régime partiel. La consommation spécifique reste d'autant plus basse que la température T.E.T. est plus forte. Pour un moteur dimensionné à 800°C, la consommation spécifique double à 25 % de la puissance maximum. Cette augmentation n'est que de 50 % pour un moteur dimensionné à T.E.T. = 1 200°C et moins de 40 % pour une température de 1 600°C.

Cette influence reste du même ordre pour un turbomoteur moins comprimé mais fonctionnant avec un échangeur récupérateur de la chaleur des gaz d'échappement (slide 14). Les augmentations de consommation spécifique à puissance partielle sont également très liées à la température nominale devant turbine. On peut relever qu'à 25 % de la puissance maximum, les accroissements de consommation réduite sont de 100 % - 60 % et 40 % pour respectivement 800°C - 1 200° C et 1 600°C de T.E.T. nominal.

Il faut également noter que ces courbes démontrent qu'à charge partielle, un turbomoteur à cycle simple se comporte aussi bien qu'un turbomoteur à cycle à échangeur, ou aussi mal suivant le point de vue où l'on se place. On retiendra cependant qu'une haute température entrée turbine permet d'avoir une courbe de consommation assez plate, avantage très recherché pour tous les moteurs thermiques et dans le domaine où les caractéristiques des turbines à gaz sont traditionnellement très critiquées.

Après avoir vu l'influence du cycle thermodynamique sur les régimes partiels, il paraît intéressant d'apprécier l'effet de la taille de la machine. Cette influence a été introduite en considérant les performances à trois niveaux de rendement qui, comme on l'a vu plus haut, sont directement liés à la taille du moteur (slide 15).

L'avantage d'un bon rendement apparaît alors nettement mais son importance n'est pas aussi sensible que la variation de température entrée turbine. L'influence du rendement peut se chiffrer à 10 % d'écart de consommation spécifique pour 3 points de rendement au 1/4 de la puissance maximum.

Pour un turbomoteur travaillant avec un échangeur, le résultat est pratiquement identique (slide 16).

Comme ces courbes sont assez rapprochées et que les trois rendements couvrent pratiquement toute la plage de débit des petites turbines à gaz, on peut conclure que l'influence de la taille est faible sur les performances à puissance partielle. En particulier, l'avantage qu'on est en droit d'attendre d'une haute valeur de la température entrée turbine sera valable également pour les petits turbomoteurs, et c'est de l'amélioration des techniques de refroidissement des turbines que viendront non seulement de bonnes caractéristiques de compacité mais aussi d'excellentes performances à régime réduit.

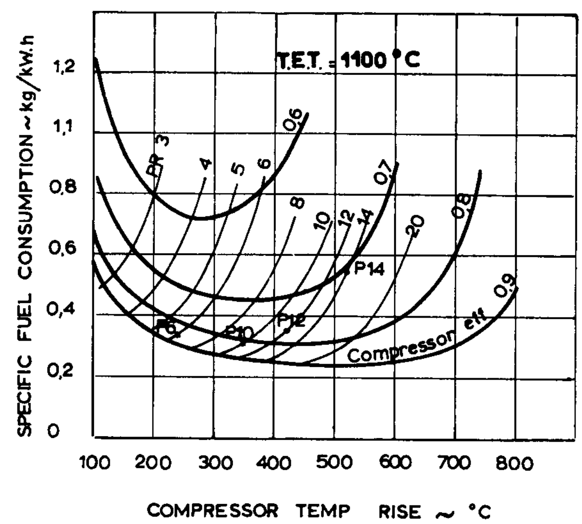
(1)

POWER is defined by AIRFLOW
OVERALL PRESSURE RATIO
TURBINE INLET TEMP
COMPONENTS EFFICIENCIES

AIRFLOW	as fonction of	SIZE
OVERALL PRESSURE RATIO	as fonction of	CYCLE
TURBINE INLET TEMP	as fonction of	CYCLE + SIZE
COMPONENTS EFFICIENCIES	as fonctions of	CYCLE + SIZE

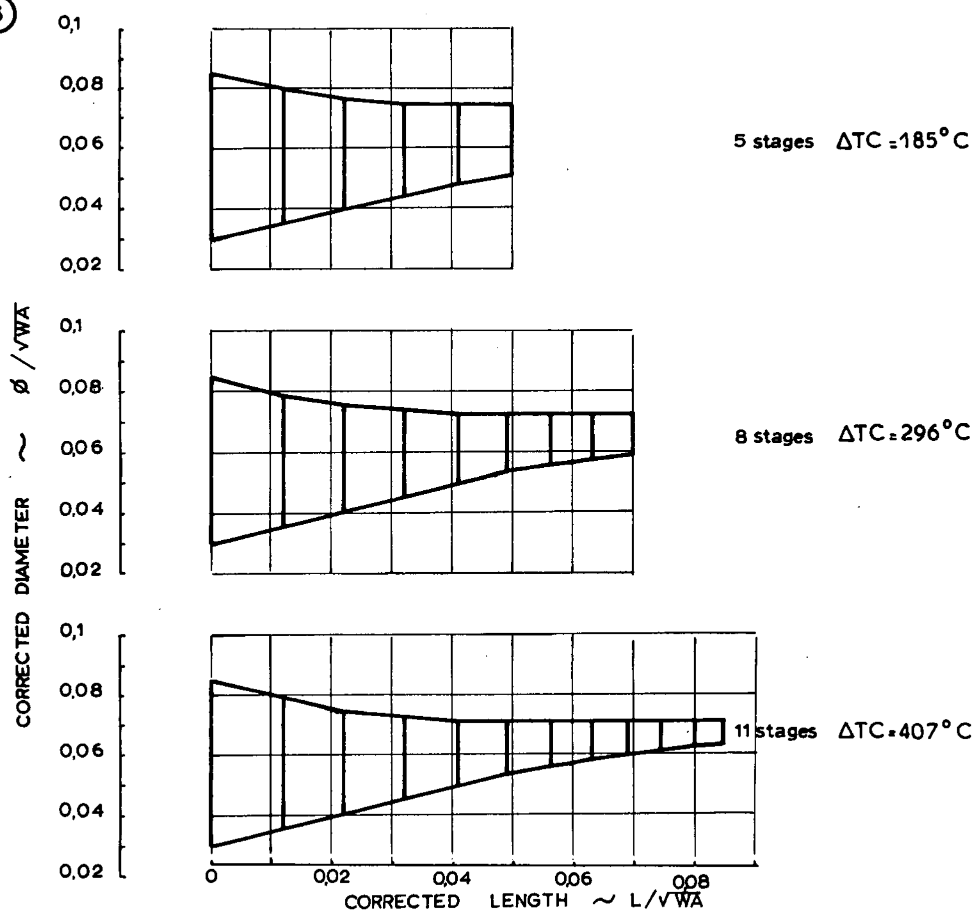
COMPONENT EFFICIENCY EFFECT

(2)

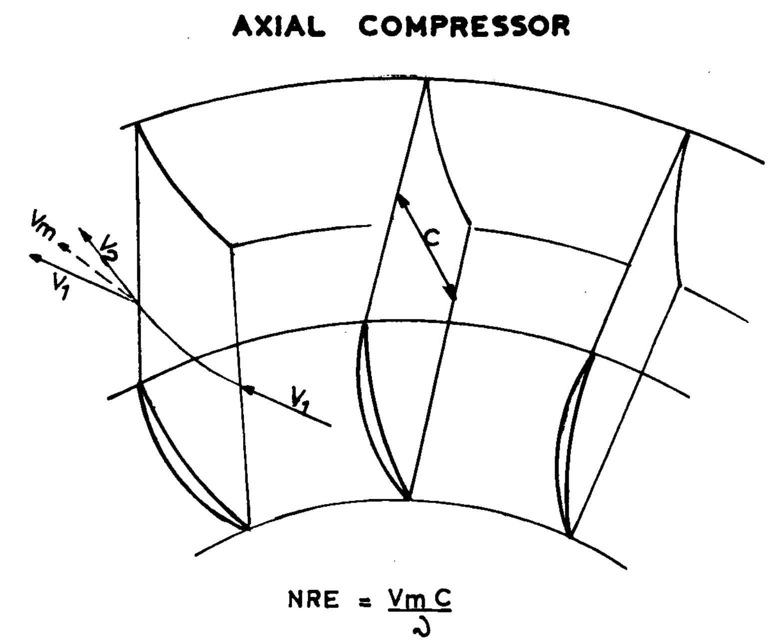


AXIAL COMPRESSOR GEOMETRY

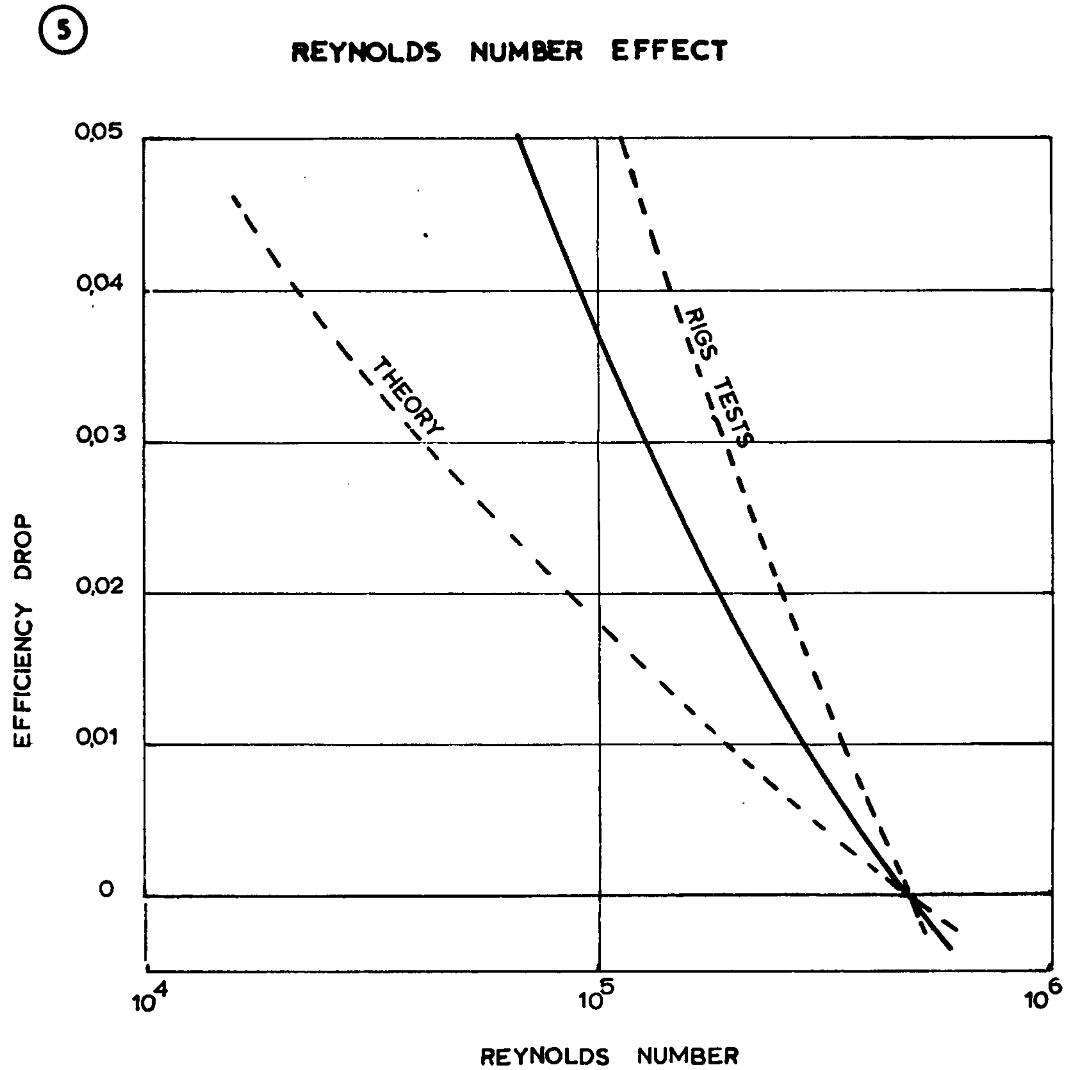
(3)



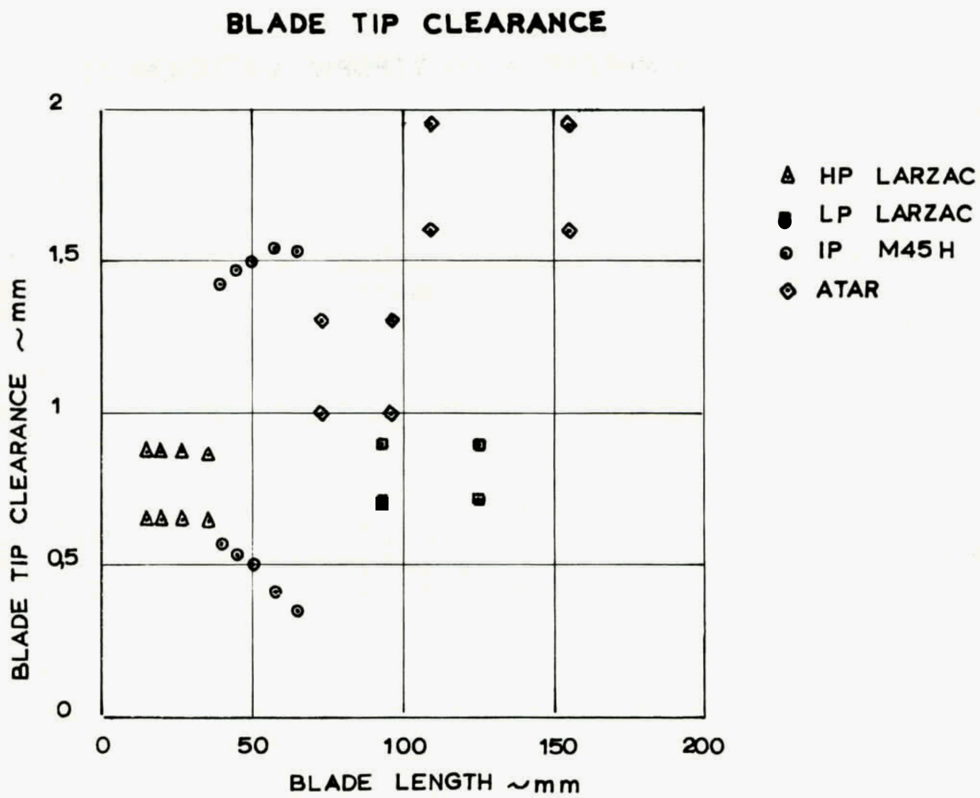
4


AXIAL COMPRESSOR
Vm
V2
V1
C
V1
NRE = Vm C / ν

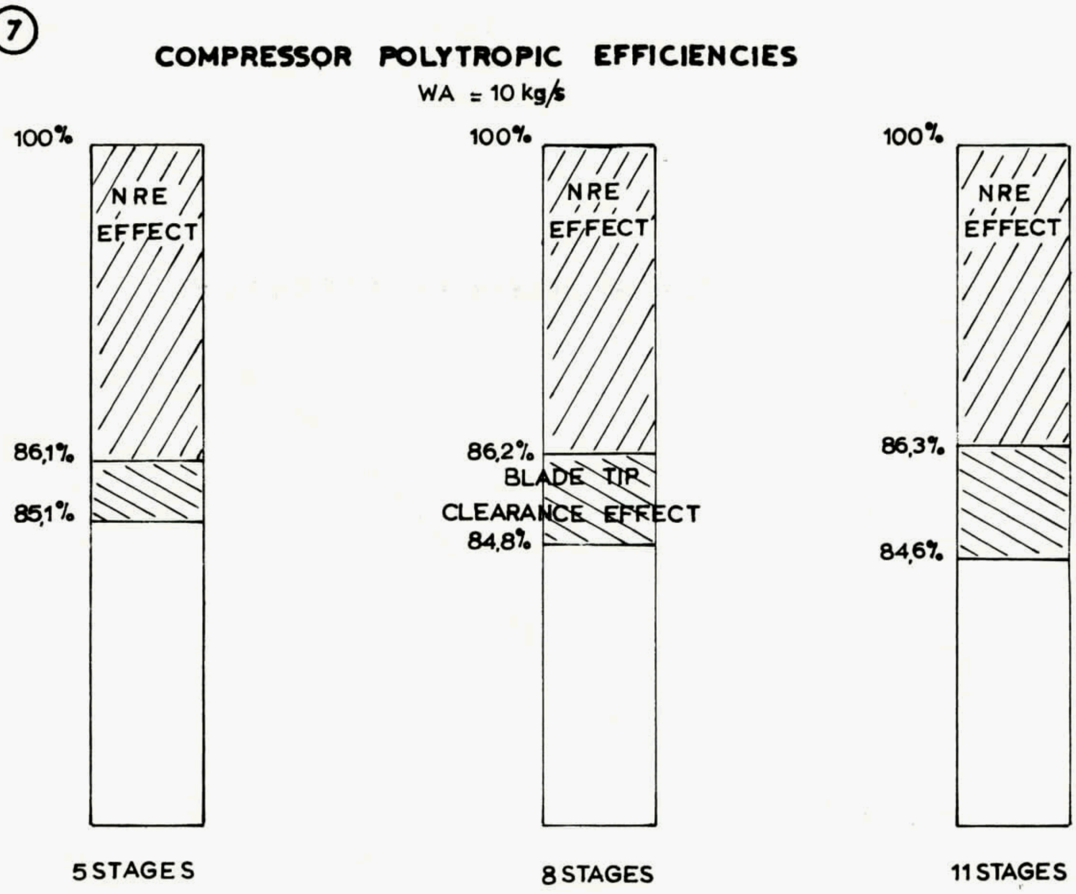
5


REYNOLDS NUMBER EFFECT
0,05
0,04
0,03
0,02
0,01
0
EFFICIENCY DROP
THEORY
RIGS TESTS
10^4
10^5
10^6
REYNOLDS NUMBER

⑥

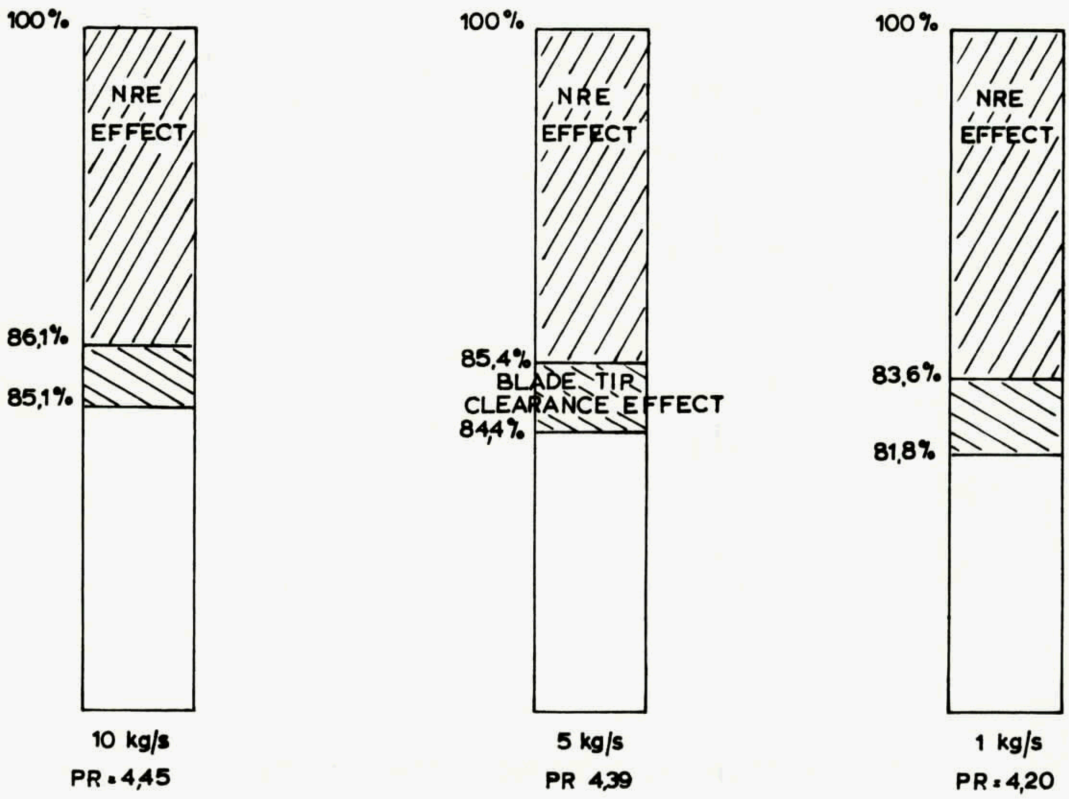

BLADE TIP CLEARANCE
BLADE TIP CLEARANCE ~mm
2
1,5
1
0,5
0
0
50
100
150
200
BLADE LENGTH ~mm
HP LARZAC
LP LARZAC
IP M45 H
ATAR

⑦


COMPRESSOR POLYTROPIC EFFICIENCIES
WA = 10 kg/s
100%
NRE EFFECT
86,1%
85,1%
5 STAGES
100%
NRE EFFECT
86,2%
BLADE TIP CLEARANCE EFFECT
84,8%
8 STAGES
100%
NRE EFFECT
86,3%
84,6%
11 STAGES

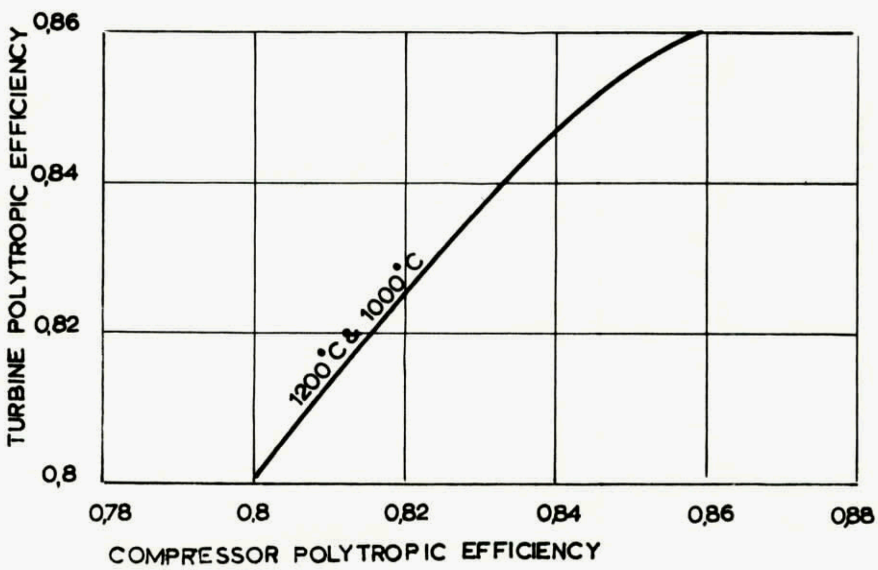
COMPRESSOR POLYTROPIC EFFICIENCIES

8

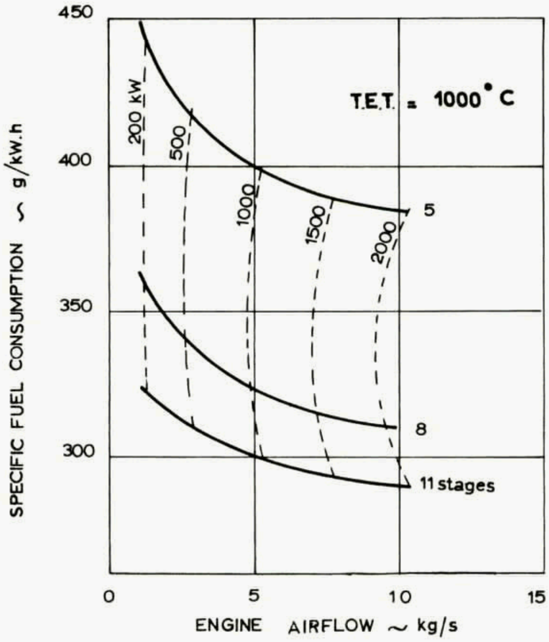


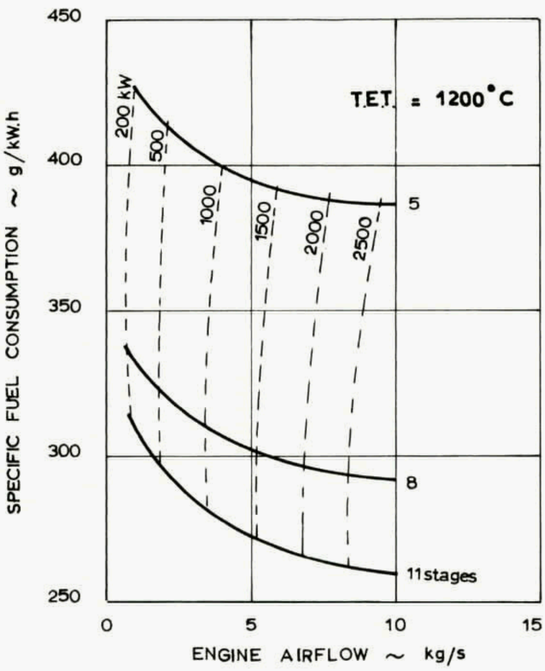
9

TURBINE POLYTROPIC EFFICIENCIES

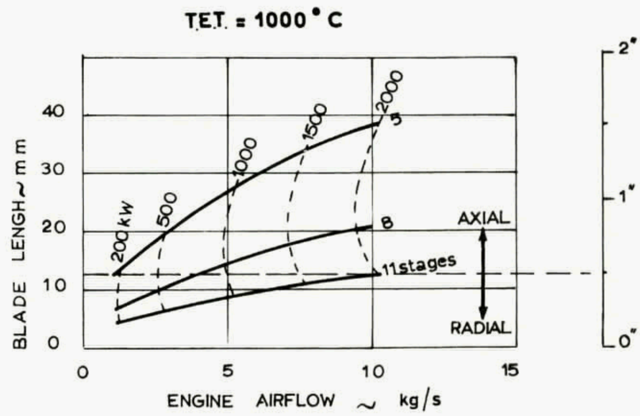


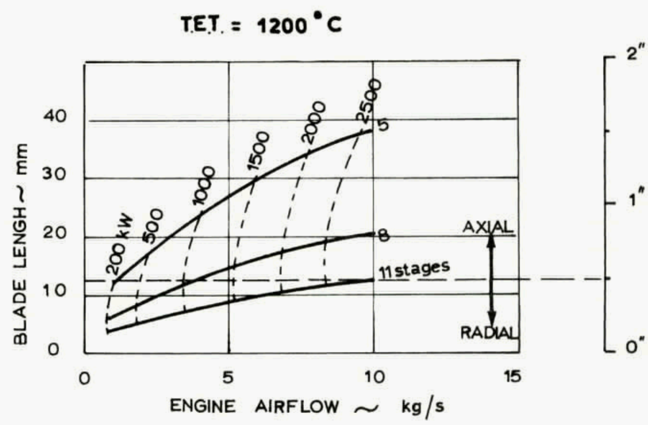
(10) GAS TURBINE PERFORMANCE

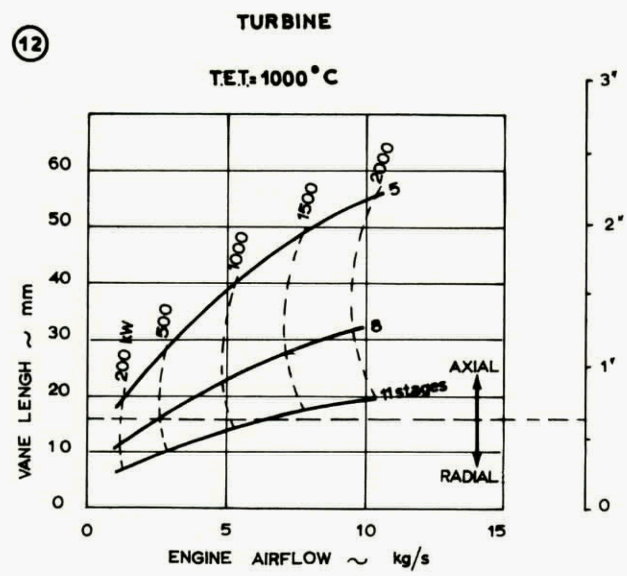




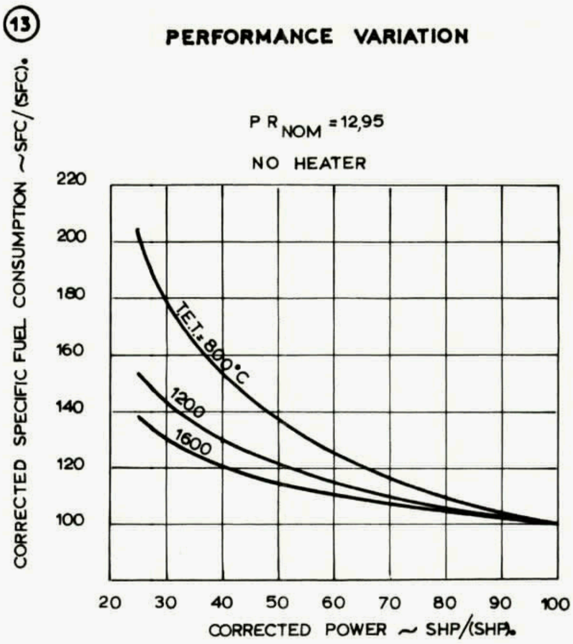
(11) COMPRESSOR



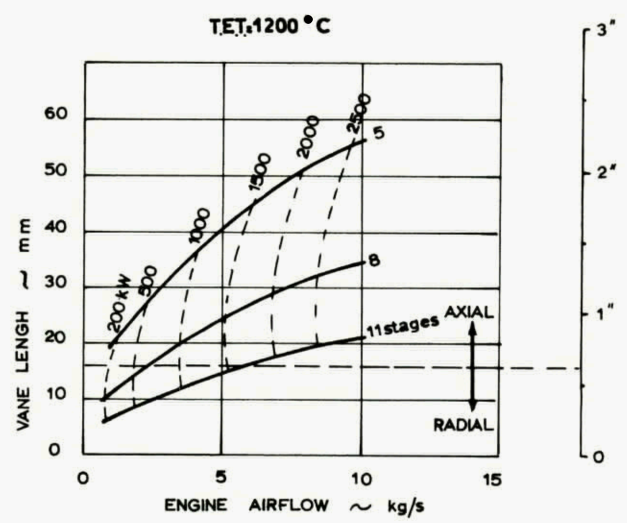




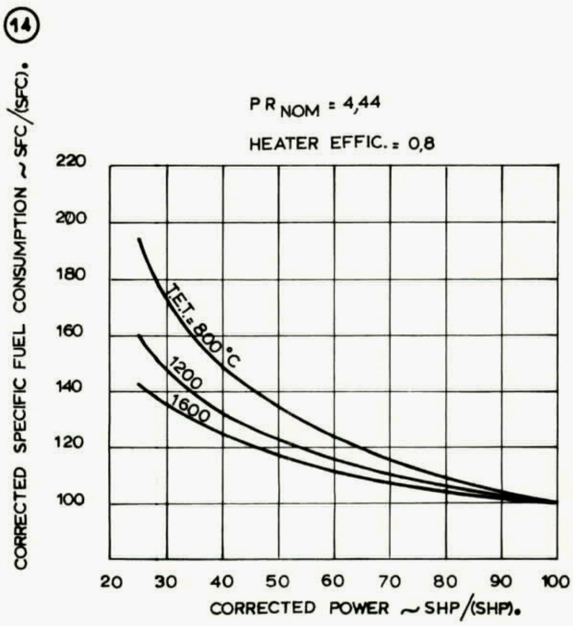
12
TURBINE
T.E.T.= 1000 °C
VANE LENGH ~ mm
ENGINE AIRFLOW ~ kg/s
200 kW
500
1000
1500
2000
5
8
11 stages
AXIAL
RADIAL



13
PERFORMANCE VARIATION
P R NOM = 12,95
NO HEATER
CORRECTED SPECIFIC FUEL CONSUMPTION ~ SFC/(SFC)₀
CORRECTED POWER ~ SHP/(SHP)₀
T.E.T.= 800 °C
1200
1600

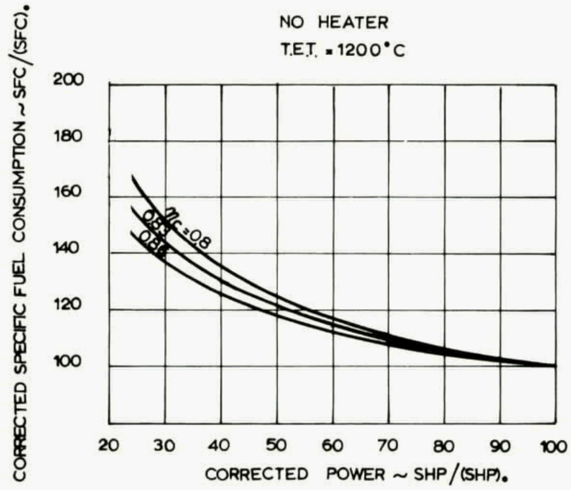


T.E.T.=1200 °C
VANE LENGH ~ mm
ENGINE AIRFLOW ~ kg/s
200 kW
500
1000
1500
2000
2500
5
8
11 stages
AXIAL
RADIAL

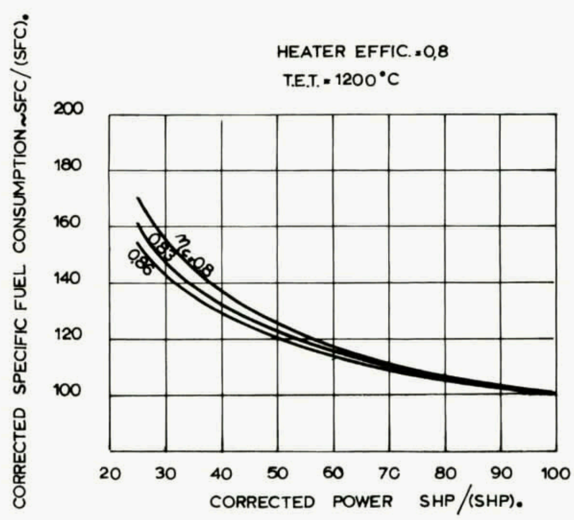


14
P R NOM = 4,44
HEATER EFFIC. = 0,8
CORRECTED SPECIFIC FUEL CONSUMPTION ~ SFC/(SFC)₀
CORRECTED POWER ~ SHP/(SHP)₀
T.E.T.= 800 °C
1200
1600

15 **PERFORMANCE VARIATION**



16



ANALYSIS OF SMALL GAS TURBINE ENGINE COMPONENTS

by

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SUMMARY

Aircraft gas turbines are to be developed for lowest weight and smallest volume; therefore they are built without utilization of the exhaust heat but for high pressure ratios. For vehicle gas turbines, however, the specific fuel consumption is the determining factor and therefore the heat exchanger is an essential component of the engine.

For small gas turbine engines cooled turbine blades can only be used to a limited extend. In certain cases higher efficiencies can be expected with radial turbines than with axial turbines having unfavorable aspect ratios.

Two-shaft engines (having a free power turbine) compete with single-shaft engines; auxiliary-attachments (hydraulic torque converter or hydrostatic transmission) render the single-shaft engine feasible to be used for traction purposes.

ANALYSE DES COMPOSANTS DES PETITES TURBINES A GAZ

RESUME

La réalisation de turbines à gaz destinées à la propulsion aérienne doit être guidée par un souci de poids et d'encombrement minima; elles sont donc conçues pour fonctionner sans récupérateur de chaleur, mais à des rapports de pression élevés. Dans le cas des turbines à gaz pour véhicules terrestres, cependant, la consommation spécifique est facteur déterminant et, par conséquent, l'échangeur de chaleur constitue un élément essentiel du moteur.

Pour les petites turbines à gaz, l'utilisation des aubes refroidies est limitée. On peut, dans certains cas, obtenir de meilleurs rendements avec des turbines radiales qu'avec des turbines axiales présentant un allongement défavorable.

Les moteurs à arbre double (avec turbine libre) rivalisent avec les moteurs à arbre unique; certains dispositifs auxiliaires (convertisseur de couple hydraulique ou transmission hydrostatique) permettent d'utiliser le moteur à arbre unique en traction.

STRUCTURE

I. SURVEY ON CYCLE-ANALYSIS

Small gas turbines for the propulsion of helicopters on the one hand and of vehicles on the other are subject to different requirements; these requirements substantially determine the design of the gas turbines and their components:

Aircraft turbines,

whether used as turbomotors in helicopters or as turboprops in sports or executive planes, will primarily be designed for low weight and small volume, i.e. high specific power at favorable specific fuel consumption.

Vehicle turbines,

at least those in commercial trucks must be designed to achieve the lowest fuel consumption possible at low production costs. The fuel consumption should be in the range of competing Diesel engines. Weight and volume are of minor importance.

From these different requirements great differences result in gas turbine design. Aircraft turbines are always built without waste-heat utilization in a heat exchanger, because during normal flight times (in general less than 2 - 4 hours) the reduction in fuel consumption does not compensate for the additional weight of a heat exchanger (plus additional airframe weight). In vehicle turbines, however, the heat exchanger is an essential part of the gas turbine.

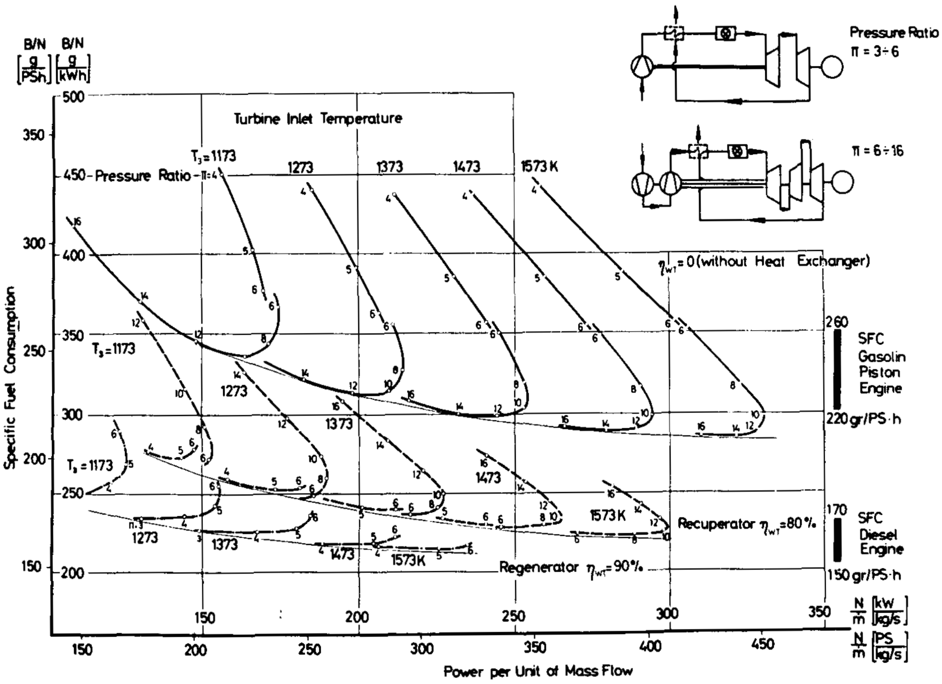
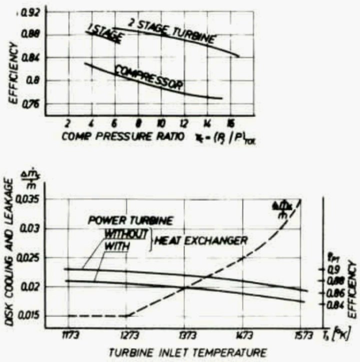


Fig. 1 Performance-Diagram for small Gas Turbines

Fig. 1 shows the results of cycle process calculations for gas turbines without and with heat exchanger. It shows the specific fuel consumption versus power output per kg/s air mass flow for various turbine inlet temperatures. Up to pressure ratios π= 6 the compressor as well as the compressor turbine are assumed to have a single stage; for higher pressure ratios, however, both are assumed to be of multistage design (see sketch in Fig. 1 above on the right). No turbine cooling is taken into account. It will be discussed later. Only a small amount of lost air for disc and casing cooling and for sealing is included. Assumed values for efficiencies and pressure losses, on which the calculations are based represent the to-day's standard and may be taken from Fig. 1a.

Gas turbines without heat exchanger require high pressure ratios in order to achieve an optimum specific fuel consumption, while gas turbines with heat exchanger achieve an optimum fuel consumption at relatively low pressure ratios. With heat exchanger the results are valuable for gas turbines with a recuperative heat exchanger of 80 % effectiveness and with a regenerative heat exchanger of 90 % effectiveness; the latter also has an assumed leakage rate proportional to the compressor pressure ratio.



constant parameters

Pressure losses	ΔP/P
Inlet	0.04
combustion chamber	0.035
heat exchanger air side	0.035
heat exchanger gas side	0.06
discharge (with a heat exchanger)	0.02
discharge (without a heat exchanger)	0.04

Efficiencies	η
mechanical efficiency gas generator	0.99
gearing efficiency power turbine	0.96

INA sea level

Fig. 1a Assumed values for efficiencies and pressure losses

The specific power and with it the engine size are primarily determined by the turbine inlet temperature T_3. The influence of T_3 on the specific fuel consumption, however, is merely small at temperatures above 1400 to 1500°K.

In Fig. 1 you can also see (on the right) the specific fuel consumption of present reciprocating engines, both the Otto gasoline engine and the Diesel engine. The specific fuel consumption of an Otto gasoline engine can already be matched by a gas turbine with T_3 of approximately 1273°K at a high pressure ratio without a heat exchanger. However, in order to approach the specific fuel consumption of a Diesel engine, the application of a heat exchanger as well as turbine inlet temperatures above 1273°K are absolutely necessary.

The above calculations are based on component efficiencies to be expected in 500 to 600 hp range gas turbines. Small gas turbines are considerably more influenced by absolute engine size than large engines. With decreasing power the Reynolds numbers of bladings decrease, i.e. the losses increase; in addition, the relative roughness and especially the clearance losses increase. In a calculation it is very difficult to consider these effects in their entirety.

Fig. 2, therefore, shows the dependency on the specific fuel consumption of present small gas turbines. The upper curve pertains to aircraft engines without heat exchanger, the lower curve pertains to some vehicle engines with heat exchanger. Although there are differences in pressure ratio and turbine inlet temperature between the represented engines, the specific fuel consumption can readily be seen as a function of the power output. In the power range above 600 hp the curve for aircraft turbines is considerably less sloped, i.e. the influence of the absolute structural size decreases more and more. Below 600 hp, however, the specific fuel consumption increases considerably with decreasing power output. At 200 hp the specific fuel consumption is about 50 % higher than at 600 hp. The same applies to turbines with heat exchanger. However, it must be noted that small turbines with an output of about 150 to 250 hp are generally competing with Otto carburator engines only, while turbines with higher output must compete with the more economical Diesel engines.

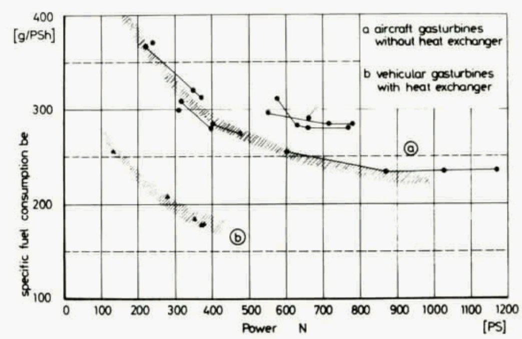


Fig. 2 Influence of Power Output on specific Fuel Consumtion

II. DISCUSSION OF ENGINE COMPONENTS

1. Compressors

The optimum compressor ratios for small gas turbines without heat exchanger are about π = 8-12 according to Fig. 1. These values are below the optimum values of large gas turbines and jet engines (π > 20) because of the lower component efficiencies. The compressor design is determined by the small air mass flows discussed here ($\dot{m} \approx 1$ - 2.5 kg/s) and the small geometric flow areas at the compressor outlet, which result from the above mentioned pressure ratios. Therefore multistage axial compressors, generally found in large engines, cannot be expected to have higher efficiencies than centrifugal compressors. Moreover, the production of multistage axial compressors is more expensive because of the close tolerances. The following compressor designs are therefore suitable:

a) Combined axial-radial-compressors with a single or multistage LP-axial compressor and a HP-radial stage.
The following turbo engines and turbo-prop engines are of the above design:

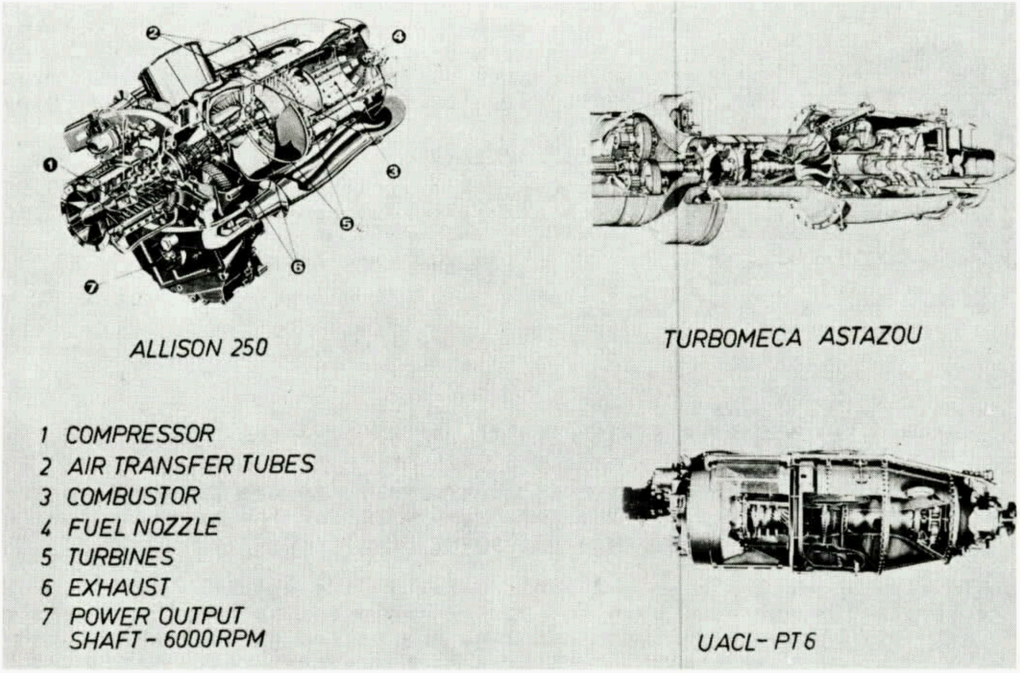


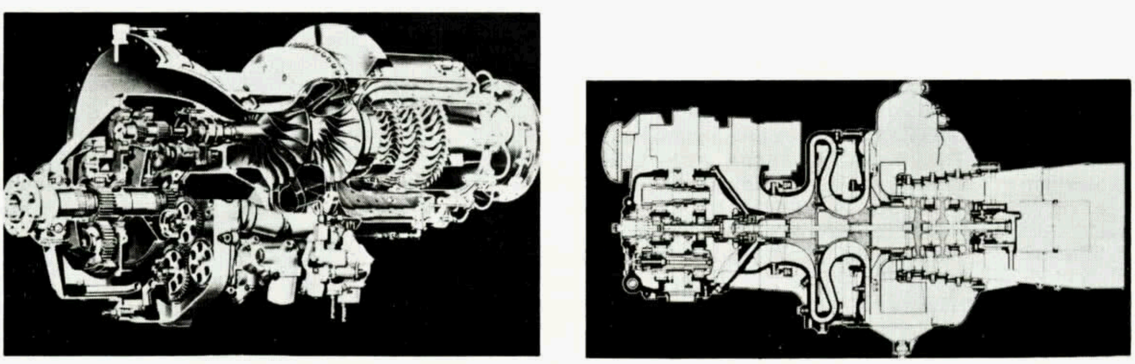
Fig. 3 Turbo Engines with Combined Axial-Radial Compressor

<u>ALLISON 250</u> (Fig.3a)
Compressor data: $\dot{m}$ = 1.54 kg/s
π = 7
Design A6/R1
(6 axial stages / 1 radial stage)

<u>TURBOMECA-Astazou</u> (Fig. 3b)
Compr. data: $\dot{m}$ = 3.3 kg/s
π = 8.1
Design A2/R1

<u>UACL-PT6</u> (Fig. 3c)
Compr. data: $\dot{m}$ =3.0kg/s
π = 7
Design A3/R1

or b) Two stage centrifugal compressors with reversing blades between the stages. This type is primarily used by GARRET/AIRESEARCH. For example



GARRETT / AIRESEARCH T76 MTU 6022-A 3

Fig. 4 Turbo Engines with two stage Radial Compressor

GARRET/AIRESEARCH T 76 (Fig. 4a) and the turbo engine MTU 6022-A3 (Fig. 4b)

Compressor data: $\dot{m}$ = 2.77 kg/s
π = 8.6
Design R2

Compr. data: $\dot{m}$ = 1.95 kg/s
π = 6.6
Design R2

Which of the two designs finally will be applied depends primarily on the know-how of the company with regard to a certain compressor design. The two stage centrifugal compressor generally has a greater structural weight than the mixed compressor and above all it requires good control of the flow conditions in the duct between the two stages. The combined axial-radial compressor, however, is of a relatively complicated design and probably more expensive than the two-stage radial, as long as the axial LP-compressor cannot be limited to one transonic compressor stage. If axial and centrifugal compressors are designed to optimum conditions (for example to optimum specific speed) then, in general, a S-shaped duct between the compressors becomes essential (see Fig. 3b). However, if axial- and centrifugal compressors are directly connected (as for example, in turbo engine UACL-PT6, Fig. 3c), the design of both compressors must include certain compromises. The axial part of the compressor rotates with lower than optimum circumferential speed and therefore requires more stages, the centrifugal part will have an unfavorable hub and/or diameter ratio both affecting the efficiency.

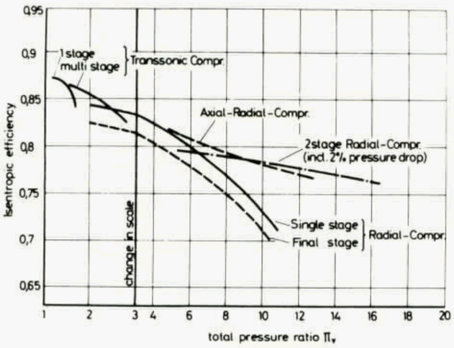


Fig. 5 Compressor isentropic Efficiencies versus Total Pressure Ratio

Fig. 5 gives the estimated efficiencies for the particular compressor types. In case of the centrifugal compressor it is distinguished between the single stage and a HP- or final stage, since the latter will always run under less favorable intake conditions and therefore will have a slightly lower efficiency. The reversing blades required for a two stage centrifugal compressor has been taken into consideration by a loss of 2 % of the total pressure behind the first stage.

The figure shows that specific ranges can be assigned to each type . At low pressure ratios, which are of no interest in this context, the single- and multistage transonic compressor reveals the best efficiencies. At pressure ratios up to about π = 5 , a single stage centrifugal compressor would be chosen. At higher pressure ratios the combined compressor appears to be most efficient. At pressure ratios beyond π = 9 - 10 the combined axial-radial compressor is finally surpassed in efficiency by the two stage centrifugal compressor. In the range of π = 8 - 12, which is of most interest here, the combined compressor and the two stage centrifugal compressor are of practically equal efficiency. At π= 8 the single stage centrifugal compressor has already a 2 to 2 1/2 points lower efficiency and it continues to drop rapidly as pressure ratio increases.

In comparison to the state of art in compressor design the efficiencies for the single stage centrifugal compressor, represented here, may appear somewhat low. However, these values are presumably closer to values attainable in an engine than mere test stand values.

Since the mid-sixties great efforts have been made in developing the single stage centrifugal compressor for pressure ratios of π= 10 and more. Extremely high pressure ratios cause high Mach numbers in the outer area of the inducer inlet (relative velocity) as well as at the impeller discharge or the vaned diffuser inlet respectively. By means of inlet guide vanes the inlet relative Mach number may be reduced by prewhirl in direction of rotation, but simultaneously for a given pressure ratio the absolute Mach number of the impeller discharge velocity increases, but in less proportions.

Fig. 6 shows the dependency of these two velocities on prewhirl angle for pressure ratios 3 ./. 10. In order to simplify calculations not the Mach numbers of the velocities but the Laval numbers (with corresponding total temperatures i.e. relative total temperature at the inlet and absolute total temperature at the discharge) have been chosen to stand for ordinate and abscissa respectively. Included is the mass flow parameter. The Laval number of the relative inlet velocity decreases rapidly at already small prewhirl angles (α_o= 10° - 20°) without increasing the Laval number of the impeller discharge velocity too much. At very high flow parameters there is a minimum of the inlet velocity at α_o= 10°. Greater prewhirl angles will increase both (again in case of the relative inlet velocity and further in case of the absolute discharge velocity).

The optimum prewhirl angle for minimum inlet relative velocity therefore is a function of pressure ratio and air mass flow respectively. High Mach numbers at the vaned diffuser inlet require special considerations in the design of the entry part of

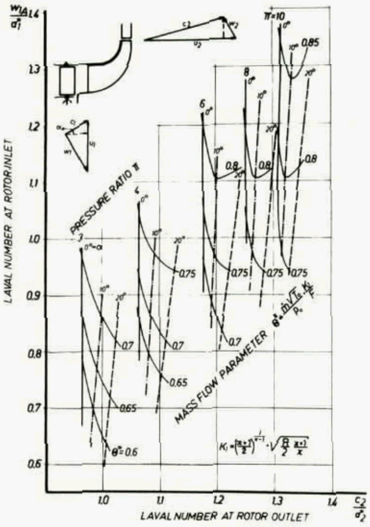


Fig. 6 Inlet relative Laval Number versus Exit absolute Laval Number having Prewhirl Angle as Parameter

$$\Theta^* = \frac{\dot{m}\cdot\sqrt{T^o}}{p^o}\cdot\frac{K_1}{F} = \frac{\varrho\cdot c}{(\varrho\cdot c)_{La=1}}$$

$\dot{m}$	=	mass flow	[kg/s]
T^o	=	total temperature	[°K]
p^o	=	total pressure	[N/m²]
F	=	flow area ⊥ c	[m²]
K_1	=	$\left(\frac{\varkappa+1}{2}\right)^{\frac{1}{\varkappa-1}}\cdot\sqrt{\frac{R}{2}\cdot\frac{\varkappa+1}{\varkappa}}$	$\left[\sqrt{\frac{J}{kg\cdot{}^oK}}\right]$
R	=	gas constant	$\left[\frac{J}{kg\cdot{}^oK}\right]$
$\varrho\cdot c$	=	product (density×velocity)	
$(\varrho\cdot c)_{La=1}$	=	product (dens.×vel.) at La=1	
c^*	=	$\sqrt{2\cdot\frac{\varkappa}{\varkappa+1}\cdot R\cdot T^o}$	
La	=	c/c^*	

the diffuser vanes. V-shaped and U-shaped leading edges were examined but without the desired success. The UACL achieved good results with the design of the pipe diffuser in connection with impellers having an extremely large number of blades at the impeller discharge. The diffuser channels are formed by cylindrical and conical bores (see Fig. 7); the leading edges result from the bores intersecting at a certain angle. For good operation of the diffuser a uniform influx is of utmost importance; it can be achieved by a low aerodynamic load on the blades. Especially near the impeller discharge additional splitter-blades are required, which, however, will be subject to high centrifugal stresses.

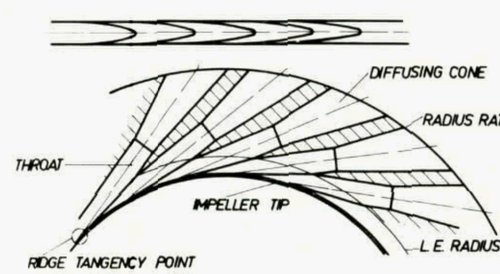


Fig. 7 UACL Pipe Diffuser

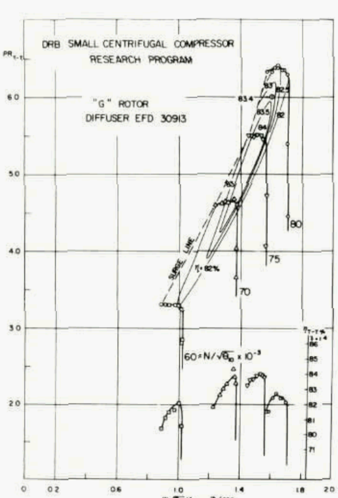


Fig. 8 Performance Characteristics of a Radial Compressor with Pipe Diffuser

Fig. 8 shows the characteristics of a UACL-compressor, which has been designed for a pressure ratio of about $\pi = 6$. (It must be noted that the efficiencies were determined by use of total pressure probes at the pipe diffuser outlet, i.e. $M \approx 0.15$; therefore approximately 2.0 points must be subtracted compared with the common definition.)

Single stage centrifugal compressors with very high pressure ratios certainly do have advantages in construction and in weight, but for high pressure ratios they offer relatively steep and narrow characteristics as compared with two stage compressors (see Fig. 9). Because of the narrow diffuser caused by the high pressure ratio the characteristics of the single stage compressor are narrowed down especially at lower speeds; in doing so the single stage compressor chokes at considerably lower air mass flow. The operating line, which has been included in Fig. 9, is therefore entering the area of poor efficiencies, which in turn increases the fuel consumption at low part load as well as the required idling speed. These effects can be corrected only to some extend by means of variable inlet guide vanes. A diffuser with variable cross section would be more efficient, however, it seems technically unfeasible in case of the pipe diffuser.

Rotor and blades for the LP-compressor - the axial as well as the radial compressor - are generally made from AL-forgings. The rotor of the HP-stage will have to be made from Titanium because of temperatures above 550 to 600°K. Since the HP-stage requires a very small discharge width metal-cutting will be necessary.

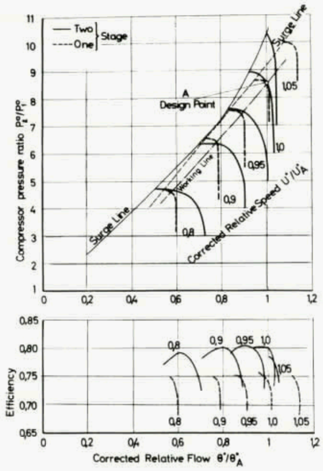


Fig. 9 Comparison of the Characteristics of a One- and a Two Stage Radial Compr.

The compressor pressure ratios for gas turbines with heat exchanger (vehicle turbines) are about $\pi = 4$ ./. 5; these values apply to optimum and generally realized compressor pressure ratios. In this range the single stage centrifugal compressor is dominant. Despite the very small output of these realized turbines (150 - 350 hp, in a special case 600 hp) compressor efficiencies of about 80 % and more are present state of art. The small pressure ratio requires a relatively small number of blades and allows more simple blade configurations, thus permitting a less expensive production.

2. Turbine

The dominating turbine design is that of a multi-stage axial turbine; This applies also to small engines, with the exception of gas turbines with extremely small output (below 150 hp) where single stage radial turbines are used. Gas turbines with a free power turbine will generally have a single stage gas producer turbine for pressure ratios up to about $\pi = 6$ (in exceptional cases even up to $\pi = 8$). Beyond this value the gas producer turbine will have two stages, because of efficiency and strength. The number of stages of the free power turbine depends on the turbine inlet temperature T_3 and the specific power $N/\dot{m}$. Up to values of about $N/\dot{m} = 350$ [PS/kg/s] the power turbine may have a single stage. For higher specific power two stages will have to be used. In turboprop engines for long flight times a two stage power turbine may already be appropriate even at a smaller specific power. In this case, efficiency i.e. fuel consumption and engine weight must be correlated. Single-shaft engines with the above mentioned pressure ratios have in general, three stage turbines.

The specific power and with it the cross section of a gas turbine depend primarily on the turbine inlet temperature. Fig. 10 shows the chronological development of the maximum turbine inlet temperature for large aircraft engines since 1950. The temperature increase can partly be attributed to the development of better materials - however at 10°K/year it is relatively small - and partly be attributed to the introduction and improvement of blade-cooling by which since 1960 a temperature increase of about 20°K/year was achieved. The big leap forward was therefore achieved by the development of blade-cooling. The following cooling methods are primarily applied (Fig. 11):

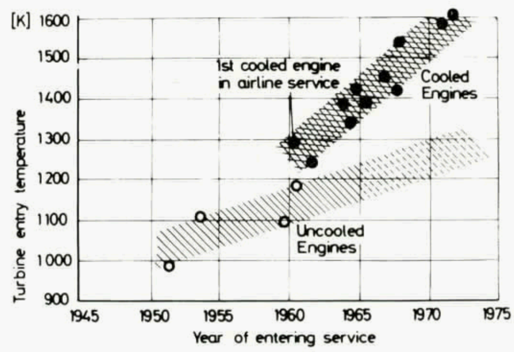


Fig. 10 Development of Turbine Inlet Temperatures for Aircraft Gas Turbines (according to Rolls-Royce)

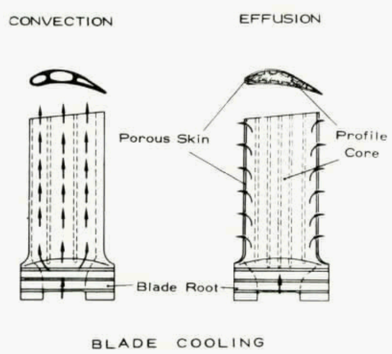


Fig. 11 Scheme of Convection- and Effusion cooling

<u>Convection cooling:</u> Cooling channels are within the blade. Cooling air is supplied at the blade root, passes through these cooling channels and is bled at the blade tip (rotor blades) or in the area of the trailing edge (nozzle blades) respectively.

<u>Effusion cooling:</u> A grooved blade core is covered with a porous skin (for example a high temperature wire gauze). The cooling air is bled at the entire surface, thus covering the blade with an air film.

The convection cooling already applied in German jet engines during WW II. has reached since then quite a very advanced state of art technology, Effusion cooling is more efficient, i.e. it requires less cooling air, but it is still in the experimental stage. In both cooling systems the cooling air is flowing back into the main gas stream; however, the mixing creates a loss, which rapidly increases with the fraction of cooling air of the total air flow.

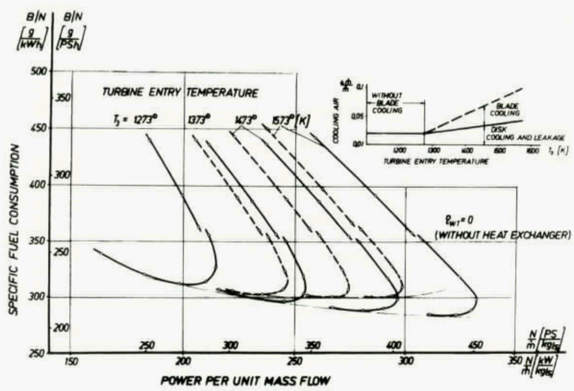


Fig. 12 Influence of Blade-Cooling on the specific Power

In Fig. 12 you can see the influence of blade-cooling on the specific power of the engine. The curves "without cooling air" correspond to the curves in Fig. 1. Needed cooling air depends on the cooling system and - in connection with convection cooling on the configuration of the cooling channels - the admitted material temperature and the cooling air temperature; primarily, however, it depends on the turbine inlet temperature. Only the latter has been considered in the calculations. With blade-cooling the specific power decreases and the specific fuel consumption increases as compared to the "uncooled turbine"; minimum consumption is obtained at a turbine inlet temperature of about $T_3 = 1450^oK$ for convection cooling (at about $T_3 = 1600^oK$ for effusion cooling respectively . But effusion cooling is not discussed here. It will be scarcely applied because of manufacturing problems). These temperatures constitute the economical limits for the cooling methods, at least with the assumptions for the component efficiencies and needed cooling air, etc. of this discussion. With a further increase of turbine inlet temperatures the specific fuel consumption starts to increase again. Despite this fact, aircraft engines are often designed for higher inlet temperatures, i.e. above the economical minimum fuel consumption, since the specific power still increases and thus structural size and weight decrease as the temperature increases.

However, the small gas turbine gains only very little by the development of cooling techniques. Application of convection cooling requires at least a turbine having an output of about 500 hp. But even above this limit the effect expected is smaller than in large engines. The smaller the blade then not only the arrangement of cooling channels becomes more difficult, but also the ratio of cooling inner surface to heat absorbing outer surface becomes more and more unfavorable.

Fig. 13 shows, for example, the ratio of the radial coolant flow area to profile area as a function of the profile chord length for a certain profile (according to an analysis of McBridge, Norris and Perugi). In the Figure, the smallest wall thickness between cooling holes or between cooling hole and profile contour is quite optimistically assumed to be 0.28 mm. Below a certain profile length, which depends on the profile shape and the thickness ratio , the cross section for the cooling air will become zero and therefore cooling will be impossible. The relatively long uncooled trailing edge area must also be considered, at least in the case of the here given profile shape with its aerodynamically favorable thin trailing edge. Thick trailing edges improve the possibilities of cooling the rear part of the blade, but because of the mixing- and impulse losses thick trailing edges have greater losses in the wakes behind the blades.

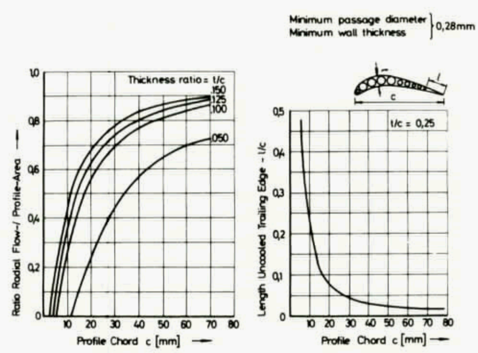


Fig. 13 Possible Radial Coolant Flow versus Profile Chord Length (according to Mc Bridge, Norris and Perugi)

Small gas turbines, which are used at the present time, do generally still have uncooled turbine blades. Their maximum turbine inlet temperature is about $T_3 = 1325^oK$. Modern manufacturing techniques even permit production of relatively small uncooled blades with satisfactory profile accuracy.

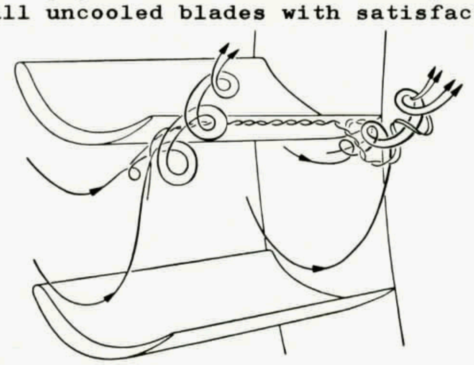
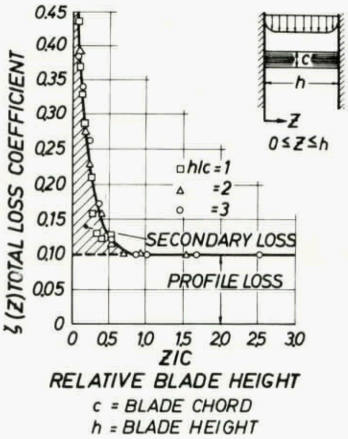


Fig. 14 Secondary Flows in a Cascade

The next generation of small gas turbines, however, is to be designed for turbine inlet temperatures of about $T_3 = 1425 - 1475^oK$ and will therefore need cooled blades. Relatively large profile length and thickness will be required for arranging the cooling channels, as already has been said. However, in connection with the short blade height this will result in an unfavorable aspect ratio h/l with considerable secondary losses.



In a flow through a curved blade channel (Fig. 14) there is a pressure difference between the pressure side and the suction side of the channel. The poor energy boundary-layer flow near hub and casing is thereby forced towards the suction side of the channel causing secondary vortexes. The intensity of these secondary vortexes depend on

profile curvature and/or flow deflection,
thickness of boundary layer at the cascade inlet
and profile length.

In case of relatively slender but long blades, secondary flows have an influence on the outer borders of the blade channel only (Fig. 15). At a blade aspect ratio of $h/l < 1$ the secondary flow areas meet each other and secondary flow losses form the largest portion of the total losses of a cascade.

Fig. 15 Influence of Aspect Ratio on Secondary Flows (according to SCHLICHTING and DAS)

Fig. 16 shows the great dependency of the cascade loss coefficient according to measurements by SCHLICHTING and DAS.

With a favorable design of the bordering walls (hub and casing) - see Fig. 17 - the intensity of the secondary flow can be influenced by a built up of a radial pressure gradient and by flow acceleration in axial direction (reduction of cross section). In addition careful profiling is required under consideration of the boundary layer built up along the profile contour. If both measures are considered, a considerable improvement in efficiency can be anticipated, even for small aspect ratios (and relatively low Reynolds numbers).

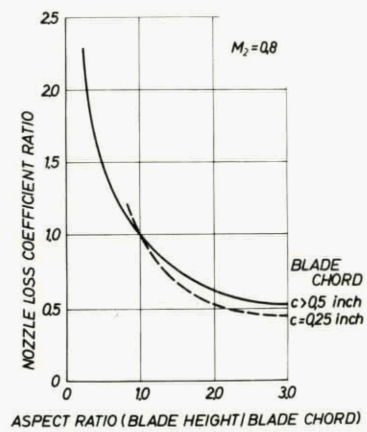


Fig. 16 Cascade Loss Coefficient versus Blade Aspect Ratio

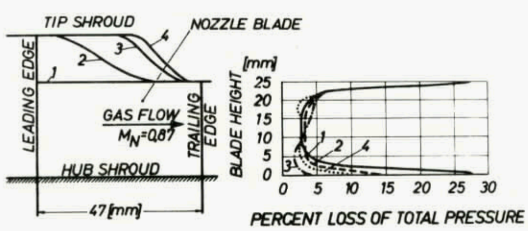


Fig. 17 Influence of Shroud Contour on Pressure Loss in a Cascade (according to Russian Investigations)

Fig. 18 gives a survey on the distribution of the losses of a conventional turbine stage and the losses which can be anticipated in an "advanced stage".

(According to estimates of the VON KARMAN INSTITUTE)

The same - above mentioned - design considerations apply to the turbine part of <u>vehicle gas turbines</u>. Because of the smaller pressure ratios in gas turbines with heat exchanger the gas producer turbine as well as the power turbine, in general, do have one stage only. Variable nozzles in front of the power turbine (Fig. 19) offer an efficient means for improving the part load behavior. The practical solution, however, in designing such variable nozzles seems very difficult, because of the high temperatures and the different expansion of outer and inner casing etc. On the other hand this solution presents aerodynamically considerable advantages. Fig. 20 shows the efficiency and the relative air mass flow versus angular position of variable power turbine nozzles. In order to accelerate the gas producer, the power turbine nozzles are opened by about 20 degrees, thus increasing the specific gas flow by almost 20%. Simultaneously the head split changes in favor of the gas producer turbine and therefore acceleration of the gas producer improves. In the part load range the cross section of the

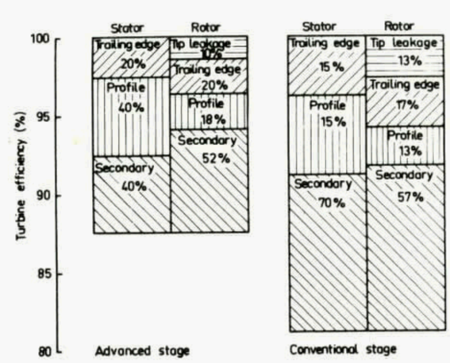


Fig. 18 Partition of Losses for Small Turbine Stages

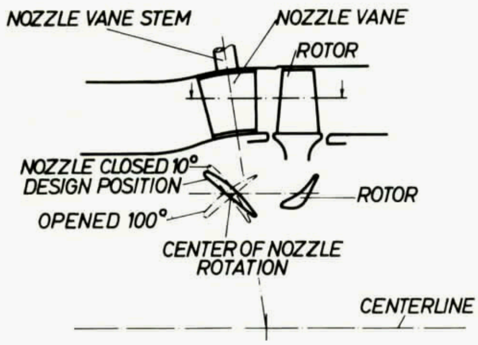


Fig. 19 Scheme of variable Nozzles for a Power Turbine

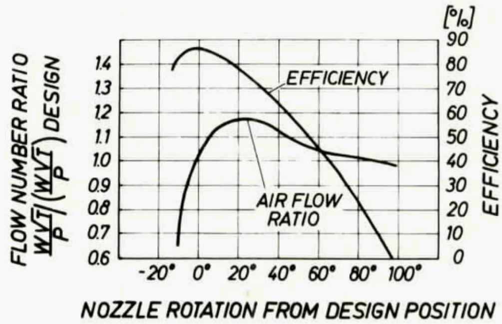


Fig. 20 Efficiency and relative Air Mass Flow versus Angular Position of the Nozzles

nozzles is made smaller, airflow is reduced and turbine inlet temperature increased; the thermal efficiency improves considerably despite a reduction of the power turbine efficiency caused by increasing incidence losses. If the position of the nozzles is over 90°, the rotor blades are subject to a strong negative incidence and thus the rotor is decelerated by the gas flow; if necessary a reverse movement will be possible.

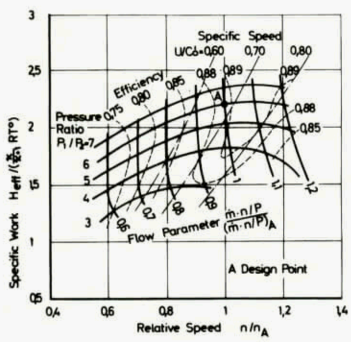


Fig. 21 High Pressure Ratio Radial Turbine Performance Map

Radial turbine

Hitherto radial turbines have been applied at low power and relatively low pressure ratios only. The new generation of small gas turbines with high pressure ratios and increasing inlet temperatures will lead to a very short blade height in an axial turbine having considerable decreases in efficiency, at least in the first turbine stage. This opens a new range of application for the radial turbine, the more as new aerodynamic achievements permit the design of radial turbines with good efficiencies up to high pressure ratios. Fig. 21, for example, shows the relative characteristics of such a radial turbine for high pressure ratio and small air mass flow. The radial turbine combines small cross sections at the turbine inlet and large outlet cross sections and its efficiency is less sensitive to clearances between rotor and casing. At high pressure ratios radial blades are common and necessary and an increasing blade thickness towards the hub does not cause any aerodynamic losses. In general the rotor outlet with its long blades does not require cooling. The rotor inlet, however, with its high temperatures, is not subject to high mechanical stresses. The maximum stress concentration is normally found near the rotor hub - similar to the centrifugal compressor rotor - where excellent cooling is possible. Therefore radial turbine rotors permit very high circumferential speeds. The UACL developed a radial turbine rotor with cooled blades (Fig. 22); in this case cooling air is fed into the rotor at the hub passes through the hollow blade and is bled at the suction side

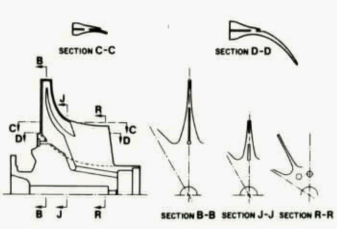


Fig. 22 Sketch of a Radial Turbine Impeller with Convection Cooling (UACL)

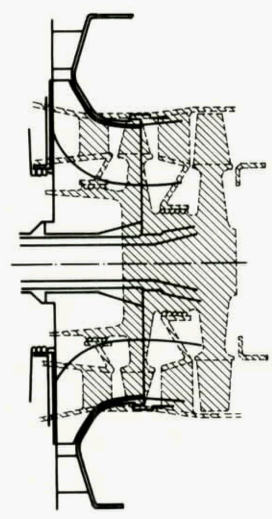


Fig. 23 Comparison between a Radial Turbine and a Two Stage Axial Turbine of same Output

of the blade. The rotor is designed to a turbine inlet temperature of 1530°K and a pressure ratio of $\pi = 5 : 1$; in addition, air cooled turbine nozzles are used. However, up to this date no test results have been published.

Radial turbines have a larger outer diameter than axial turbines of the same output (Fig. 23), but they will have a good duct to a reverse flow combustion chamber as well as to a single can type combustion chamber; in many cases the radial turbine will be shorter than the perhaps necessary too stage axial turbine of the same output. But if there is a power turbine behind the radial turbine, then a relatively long duct is required between these turbines and the radial turbine offers no longer an advantage in structural weight. The (uncooled) radial turbine, however, is not only cheaper than the two stage axial turbine but also lighter in weight and often more efficient. Therefore an increasing attention to-day will be paid to the radial turbine in the design of small gas turbines.

Materials

For the hot parts of gas turbines generally materials on the base of high-temperature Nickel- and Cobalt alloys are used. In small gas turbines it is strived for the use of integral cast rotors because of cost reasons, even though rotors with forged discs and fitted blades have been built down to a power of 100 hp. The increase in turbine inlet temperature due to the development of better materials within the last decade was about 10°K/year - as already has been mentioned. The material temperatures of modern aircraft engines are already up to 80% of the melting temperature. Nickel- and Colbalt alloys therefore will not allow - without cooling - too much further increase in permitted material temperature. For high temperature melting metals, such as Niobium and Tungsten and their alloys, the problem of oxidation is not yet solved. An additional draw-back for the application of these materials for rotors is the high specific weight and the unfavorable tearing length, respectively.

Ceramic materials offer new possibilities. Despite a research work of more than one decade Aluminium-oxide proved to be too sensitive on thermal shocks. Better expansibilities show Silicon carbide and Silicon nitride. With these materials turbine inlet temperatures of more than 1500°K can be expected. Other advantages are the relative low raw-material costs. However, a production method for series production has still to be developed. Research work on ceramic materials for turbines is mainly done at big automobile companies with regard to a vehicle gas turbine being reasonable in price as well as competitive. As an example Fig. 24a shows turbine nozzles made of Silicon nitride; Fig. 24b shows a turbine inlet housing made of Silicon carbide. These are developments of the Advanced Technology Department at FORD MOTOR Company. Similar experiments are known to be made at the English Company LEYLAND GAS TURBINES LTD.

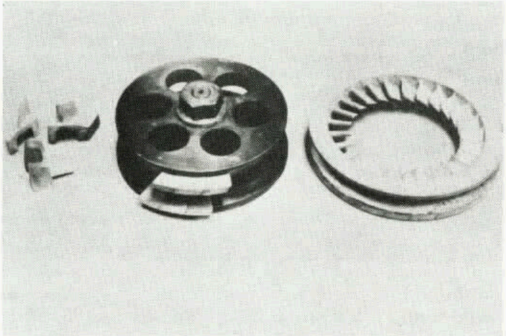


Fig. 24a Turbine Nozzles made of Silicon Nitride (according to FORD MOTOR Company)

Fig. 24b Turbine Inlet Housing made of Silicon Carbide (according to FORD MOTOR Company)

3. COMBUSTION CHAMBER

The combustion chamber of a small gas turbine should have an axial over-all length as short as possible in order to get a most compact design with compressor and turbine close together and with a short and sturdy gas generator in consideration of its high speed. The outer diameter of the combustion chamber on the other hand is of minor importance since at least the final stage of the compressor is built as a radial type, and therefore a sufficient front surface for the combustion chamber is available.

There are three main types of modern combustion chambers:

- Reverse-Flow-Annular Combustor
- Radial/Axial Annular Combustor
- Can Type Combustor

The common Straight-Through Annular Combustor almost exclusively used with large turbine engines requires a larger axial over-all length than the types as mentioned above, and is therefore of mean importance for small gas turbines. The engine GE-T58 with its straight through annular combustor seems to be an exception to the rule but with its power output of 1200 to 1800 hp it is already beyond the power range of the small gas turbines being under consideration.

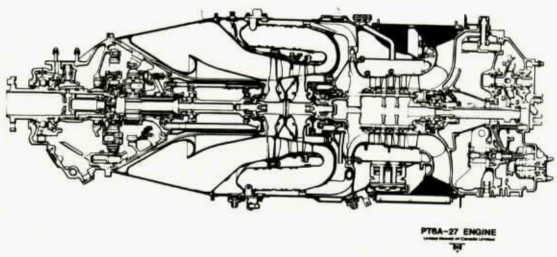


Fig. 25 Reverse Flow Annular Combustion Chamber in UACL PT6 Engine

The Reverse-Flow-Annular Combustor is the most commonly used type for aircraft engines of small power output (Fig. 25). Owing to the 180° reverse flow before entry into the turbine a sufficient mixing zone length in spite of the short axial over-all dimension is available; this is favorable, in view of a good temperature distribution but it increases the pressure loss in comparison with a straight through flow. It should be noted, however, that the combustion or burning process must be completed prior to the entry of the gases into the elbow. A further prerequisite is a good mixing with the dilution air in order to avoid local overheating on the elbow outer wall.

The thermal combustion chamber load is generally low with reverse-annular chambers because of the relatively large structural volume; nevertheless, there are problems especially in cooling of the inner wall of the flame tube because the cooling air flows along the flame tube outer wall and around the dome and is heated up as well as dropped in pressure. The duct boundary opposite to the flame tube inner wall is the turbine casing and it also radiates heat to the incoming air.

When the turbine is of the axial type there is often - in spite of the relatively large flame tube volume - only a small flame tube height with an unfavorable i.e. large ratio of the flame tube surface to the flame tube volume. More favorable in this case would be a radial turbine with regard to the design of the combustor which permits a larger flame tube height. Furthermore, the flow at the flame tube exit would have to be bent by only 90° whereas the second 90° bend would be within the radial turbine rotor.

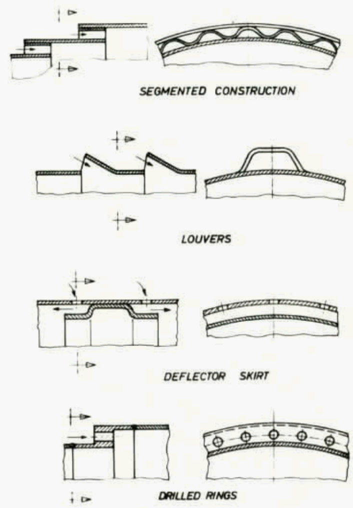


Fig. 26 Examples for Cooling Air Inlets at Flame Tubes

An only convective cooling of the flame tube walls is - beside other points also because of the low air velocities in the outer and inner air-duct -mostly not satisfying, therefore the flame tube walls must be protected by means of an air flow or "veil". Fig. 26 illustrates design versions for the supply of cooling air. The distance between the individual metal sheets to-day is commonly provided by means of fins or corrugated material. With increasing turbine inlet temperature not only the combustors but also the cooling air proportion is getting smaller. Therefore, also with small gas turbines it seems necessary to turn to machined distance rings (Fig. 26) because they permit a very accurate calibration of the cooling air supply - independent of temperature deviations. The development is heading to an effusion cooling system whereby the flame tube is made of ceramic or metallic porous material. Similar to the blade cooling the necessary cooling air consumption for the effusion cooling is approximately 1/3 of the film cooling. At present, however, the development testsare still in the first phase.

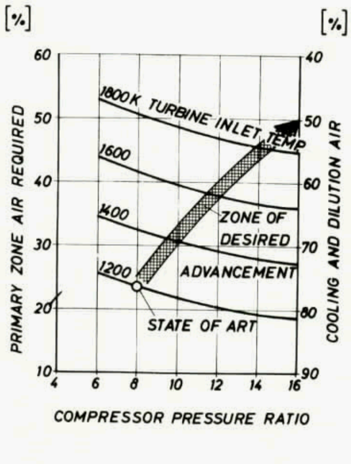


Fig. 27 Distribution of Air Flow in a Combustion Chamber

In the primary i.e. in the actually burning zone of the combustor a nearly stoichiometric combustion occurs. In the adjacent mixing zone cooling and dilution air are added in order to lower the temperature to the scheduled turbine inlet temperature. With increasing turbine inlet temperature the proportion of the primary air with respect to the whole air flow is also rising (Fig. 27) and this results in a shortage of air for cooling and dilution purposes. The designs of combustor dome and combustion zone respectively, are gaining more and more importance. Moreover, great efforts should be made to design the cooling faces as small as possible. Turbine inlet temperatures above $T_3 = 1550°K$ will make necessary the use of effusion cooling - as is necessary for

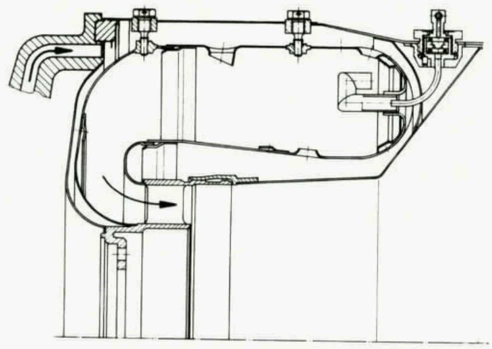


Fig. 28 Combustion Chamber of KHD T112 Small Engine

blade cooling - because the needed cooling air supply for film cooling is no longer sufficient.

The fuel supply into the combustion zone can take place both by means of fuel pressure atomization - this method still being economical with the pressure ratios of small gas turbines - and by atomization by means of pressurized air, whereby it is difficult, however, to build correspondingly small air atomizer nozzles.(The only known use of this type is with the APU Solar T-G2T-12 "Titan"). With regard to fuel pressure atomization both Simplex Swirl Atomizers and Dual Orifice Swirl Atomizers are in use whereby the last type has two separate supply lines and during idling and low load operates as Simplex Atomizer. For a sufficient uniform temperature distribution with reverse annular combustors 10 to 12 orifices are necessary. The fuel spray or mist should leave the orifice in the shape of a cone angle as large as possible. However, it should be noted that at a small flame tube height the danger of wetting the inner wall by fuel is given which might cause local overheating and sooting.

An alternative method therefore is the pre-vaporization of the fuel in vaporizer tubes, located in the combustion zone of the flame tube whereby the fuel is already heated. Fig. 28 illustrates the fuel vapour combustor of a small gas turbine of only 150 hp rated power. Provided are 12 rectangular bent vaporizer tubes with tangential fuel flow exit. The maximum fuel mass flow is 54.0 kg/h, the idle or no-load flow is 8.0 kg/h. Notice should be taken of the very small flame tube height; it would be very difficult to develop a fuel pressure atomization system for these conditions.

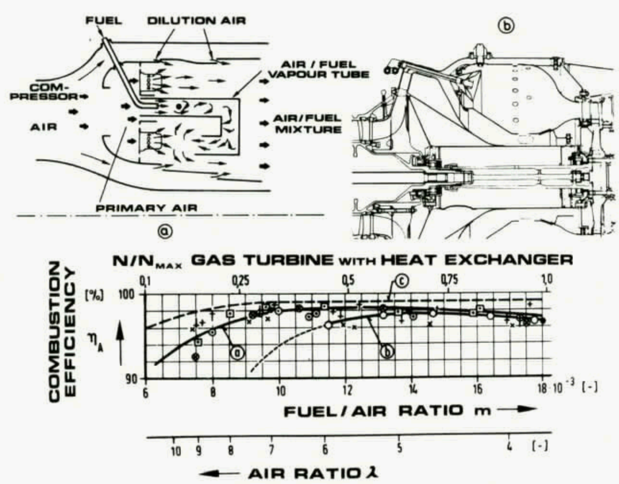


Fig. 29 Combustion Efficiencies of different Types of Combustion Chambers (according to B. Eckert)

Vaporizer combustors moreover have a broader operation range between the weak and rich extinction limits then combustors with fuel pressure atomization. Fig. 29 delineates a comparison between the combustion efficiency of a vaporizer combustor with fuel pressure atomization. However, one should not overlook the drawback of the vaporizer with regard to the poor ignition capacity at low temperatures i.e. when practically no vaporizing exists.

The Radial-Axial Combustor with rotating injection has been developed primarily by TURBOMECA for their small gas turbines (Fig. 30) and has been licence built and improved by CONTINENTAL. It also is used with the Mini-Jet engine MTU 6012. This type has a short axial over-all length too and can be well matched with the outer diameter of a radial compressor.

The fuel is supplied through the hollow engine shaft whereby only small injection pressures are necessary. Owing to centrifugal forces the injection pressure rises up to 140 kg/cm^2 on its way to the exit holes on the atomizer wheel whereby a good atomization is granted. The holes in the atomizer wheel generally are not filled with fuel, and therefore can be designed relatively big so that no danger exists for clogging by fuel pollution.

Thermically endangered with this combustor type again is the flame tube inner wall, because the total required air supply passes through the hollow turbine vanes whereby it is heavily heated up. At the same time, however, these vanes are intensively cooled so that a good temperature distribution in circumferential direction is not of great importance and relatively high turbine inlet temperatures can be permitted.

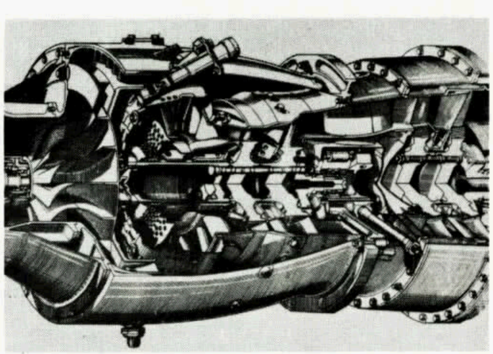


Fig. 30 Radial-Axial Combustion Chamber with rotating Injection (System TURBOMECA)

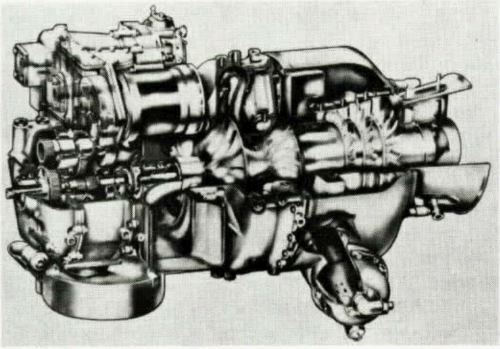


Fig. 31 Gas Turbine MTU 6022 with tangential mounted Can Type Combustion Chamber

The load factor of the TURBOMECA combustion chamber is (relatively) high for small gas turbines, nevertheless with the engines known as yet only convection cooling of the combustor walls is provided. With rising turbine inlet temperature, however, it will not be possible to do without a film cooling or a similar method. A critical point furthermore is the sealing between combustor walls and the shaft in the area of the atomizer wheel. As this sealing is located in the primary zone, the stability of the flame can be affected by uncontrollable leakage losses.

Can-type combustors present the smallest flame-tube surface to be cooled at a given combustor volume thus being the most favorable type with regard to cooling. But the tangentially mounted can-type combustor requires a - mostly voluminous - turbine inlet spiral which requires cooling too. The flow from the compressor to the flame tube is mostly asymetric and therefore is the source of local thermal and aerodynamic problems. This type of tangentially mounted combustors therefore is loosing importance for aircraft engines. Fig. 31 shows the small gas turbine MTU 6022 as an example.

A really interesting application of the can-type combustor you can see in the ALLISON engine T63 (Fig. 32). In this case the combustor, a so-called "cup" combustor, is arranged in longitudinal direction of the engine. The gas flow is not bent after entrance to the flame tube. In- and outgoing flows are nearly axis-symmetric and the combustor is easily accessible for inspection purposes and removable without disassembling of the engine. The fuel supply is provided by a single, centrally located fuel nozzle, therefore no problems by clogging or during mass flow control arise.

These advantages in the design of the combustor, however, have their price:

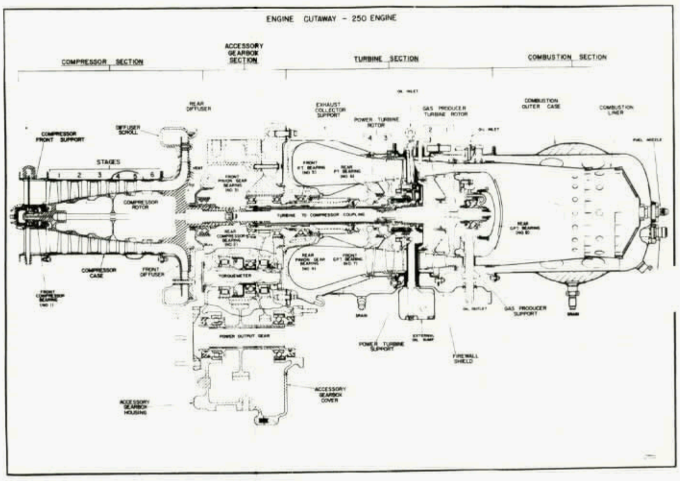


Fig. 32 ALLISON T 63 with Single Can Type Combustion Chamber

- the missing thrust compensation between compressor and turbine. (For both therefore thrust bearings of high load capacity are necessary.)
- an unfavorable exhaust gas exit in the middle of the engine and there is no residual thrust to be utilized.
- and a relatively expensive power output in the middle of the engine which certainly permits power output shafts to the front as well as to the rear side of the engine but in any case long driven shafts are necessary.

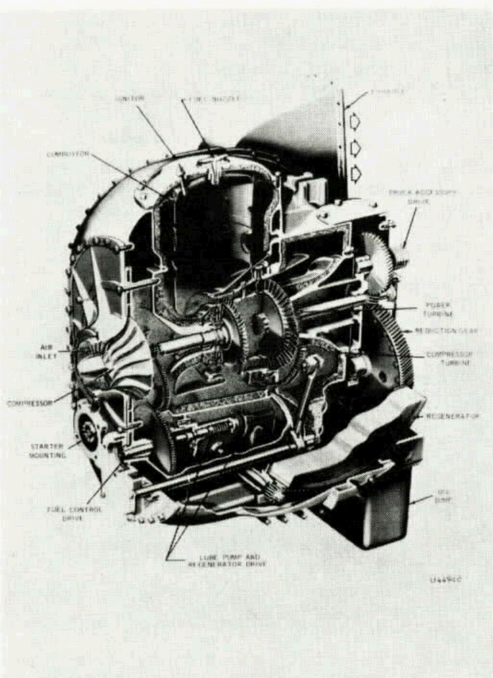


Fig. 33 Arrangement of the Combustion Chamber in the Vehicle Gas Turbine FORD 707

Combustion chambers for vehicle turbines are exclusively designed as can-type combustors whereby in most cases a single can combustion chamber is used. Its arrangement with regard to compressor and turbine mainly depends on type and design of the heat exchanger. Mostly a central arrangement above the engine shaft is used. Fig. 33 shows the arrangement of the combustion chamber of a vehicle gas turbine FORD 707. The cooling of the flame tube wall in vehicle gas turbines is particularly difficult because only pre-heated air from the heat exchanger is available for cooling purposes. The heat resistance especially because of the many load changes in a vehicle gas turbine is therefore a vital factor for its life.

4. HEAT EXCHANGER

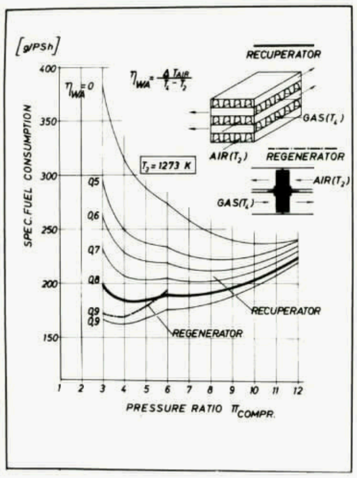


Fig. 34 Specific Fuel Consumption versus Pressure Ratio. Parameter:Heat Exchanger Effectiveness

The specific fuel consumption of a gas turbine can decisively be reduced by applying heat energy from the exhaust gas to preheat the compressed air, thus reducing the temperature rise in the combustion chamber. So far heat exchangers have not been used in aircraft engines because they would increase the engine weight by 3 or 4 times and would considerably enlarge the assembly space. Individual trials to develop heat exchangers which by no means enlarge the front surface of the engine have been initiated but did not get further than testbed experiments. Especially for small gas turbines for helicopters the flight times are not long enough so that the reduction in fuel consumption would compensate for the weight increase of the engine (including the frame part).

For vehicle gas turbines the heat exchanger is an essential if not the most important element of the power plant. Weight and volume of the engine are less important than the fuel consumption. Fig. 34 shows the dependency of the specific fuel consumption on the heat exchanger effectiveness η_{WA} and the compressor pressure ratio $\pi_{Compr.}$

There are two types existing (refer to Fig. 34)

recuperators in which air and exhaust gas is lead in separate channels whereby a convective heat transfer takes place, and

regenerators in which a porous disc (or drum) is alternatively flown through by hot gas and by compressed air thus absorbing heat from the hot gas and delivering it to the compressed air, respectively.

The two types are bound to certain limitations. Weight and volume of a recuperator rise decisively (with it also the price) with increasing heat exchanger effectiveness and therefore an effectiveness of η_{WA}=0.75-0.80 should represent approximately the highest limit. Regenerators can be built having a reasonable pressure loss and reasonable dimensions with an effectiveness of $\eta_{WA} \approx 0.90$. With increasing power, however, big disc or drum sizes would result so that the range of application in respect to gas turbine installation into a vehicle is restricted to about 350 - 400 hp. The air and gas sides of the regenerator are separated from each other on the rotating disc by sealing elements whereby - irrespective of existing pressure losses which also occur in the recuperator - additional leakage losses will result by overflowing of compressed air to the exhaust gas-side. The leakage losses are nearly proportional to the compressor pressure ratio and limitate the applicable compressor pressure ratio to values below π= 6. Thanks to Messrs. CORNING GLASS WORK a ceramic disc material has been developed, called "Cercor". Fig. 35 gives an example of a Cercor heat exchanger disc for a vehicle gas turbine.

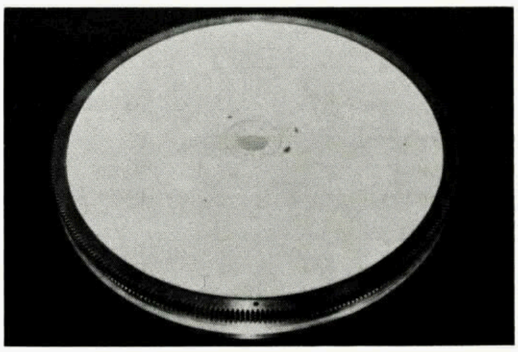


Fig. 35 Cercor Heat Exchanger Disc for a Vehicle Gas Turbine

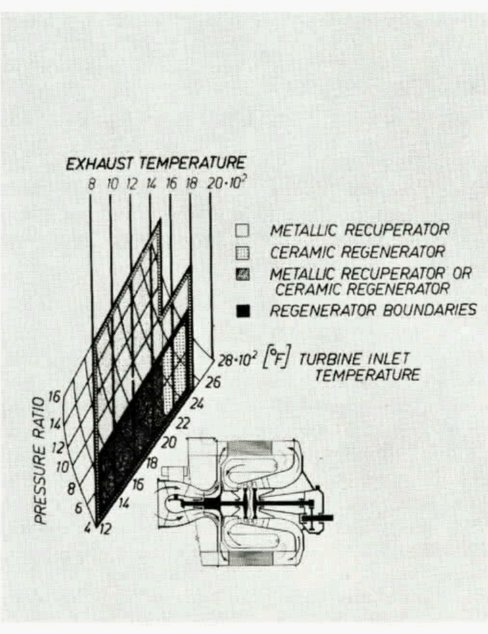


Fig. 36 Temperature and Pressure Boundaries for different Types of Heat Exchangers (according to Gas Turbines)

If you add also the temperature ranges for which the various heat exchanger types can be applied, then you get Fig. 36, the front page of the journal "Gasturbine" published in September/October 1966. Very high turbine inlet temperatures can only be governed by using ceramic materials as regenerators whereby the compressor pressure ratio is limited to values below π= 8; The heat exchanger casing and the drive of the rotating disc still represent some unsolved cooling problems. Recuperators made of metal are restricted to turbine exhaust temperatures below 1100°K, however, they allow high power output and high compressor ratios.

Most vehicle gas turbines are designed for a power output below 400 hp (with the exception of one US Army contract, where a power of 600 hp was required). In vehicle gas turbines the Cercor-Regnerator is prevailing almost without exception; in some cases the metallic regenerator is used. In Fig. 37 you can see the 150 hp vehicle gas turbine with Cercor-Regenerator of Messrs. Rover-Leyland. Because of the very low power a radial compressor turbine is used in this case, too. A subsequent version having 350 hp, however, is equipped with an axial compressor-turbine (and axial power turbine). An outsider among the modern vehicle turbines the 300 hp turbine SOLAR-B from INTERNATIONAL HARVESTER CO. is shown in Fig. 38. It is designed with an annular combustion chamber and with a recuperator having an effectiveness of 78 %; The used annular heat exchanger design presents space savings and favorable flow conditions. This gas turbine has - regardless the higher performance - a radial compressor turbine, too.

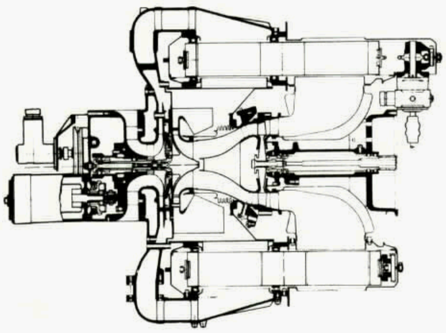


Fig. 37 150 hp Vehicle Gas Turbine with Cercor Regenerator (ROVER-LEYLAND)

5. DESIGNS AND OUTPUT DEVICES

For aircraft gas turbines there are available both the single shaft engine and the multi-shaft engine with free power turbine.

The single-shaft engine having the load mechanically rigid connected to the gas turbine shaft during operation is especially suited for constant speed drives (e.g. helicopter rotor or industrial application: electric generator). Load variations are only obtained by changing the turbine inlet temperature, the engine thus immediately reacts to load changes. The engine in its design is simpler than the multi-shaft engine.

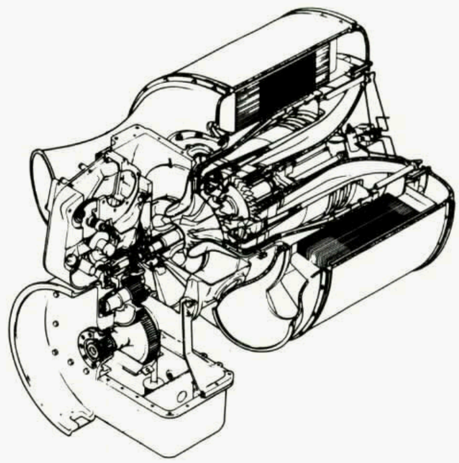


Fig. 38 300 hp Vehicle Gas Turbine SOLAR-B with Annular Recuperator (International Harvester)

The gas turbine rotor in most cases has two bearings only and compared to the multi-shaft gas turbine one turbine stage often can be saved.

These advantages nevertheless are confronted with disadvantages in operation: The single-shaft engine delivers almost no torque below 50 % of the rated speed (full load speed).

Fig. 39 gives an illustration of the torque characteristics versus drive speed for a single-shaft engine as well as for a gas turbine with free power turbine.

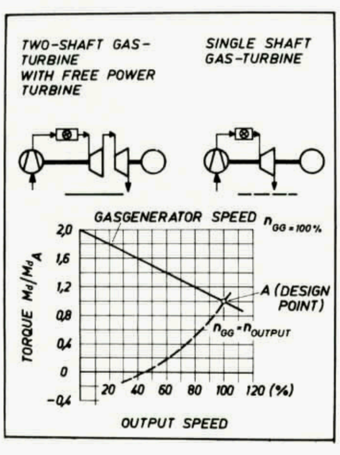


Fig. 39 Comparison of Torque Characteristics of Single-Shaft and Two-Shaft Gas Turbines

To start the single-shaft engine a clutch must be operated between the load and the gas turbine. This can be done by using centrifugal clutches as well as hydraulically controlled clutches. Otherwise the load of the driven engine must be reduced to a large extent, (as e.g. by switching off the induction current of electric generators or by throttling at the inlet of pumps and compressors).

The range of speed changes under load conditions is very limited (approximately 80 to 100 % of the rated speed). Fig. 40 demonstrates the performance map, i.e. the power range versus the output speed for a gas turbine with and without free power turbine respectively. In case of a single-shaft engine the available power drops rapidly with decreasing speed. The limits are given by the max. turbine inlet temperature and - in many cases at lower speed - by the compressor surge line. Performance higher than full load can be obtained without a rise of the turbine inlet temperature in the speed range above 100 %. Thus the engine can be overloaded without increasing strain due to temperature. Overspeeds at constant power lead to lower turbine inlet temperatures, so that less thermal strain is favorable for the additional mechanical stress.

For the gas turbine with free power turbine a connection between the gas generator and the free power turbine only exists by gas forces. The load is mechanically connected only to the power turbine. Gas generator speed and power turbine speed are

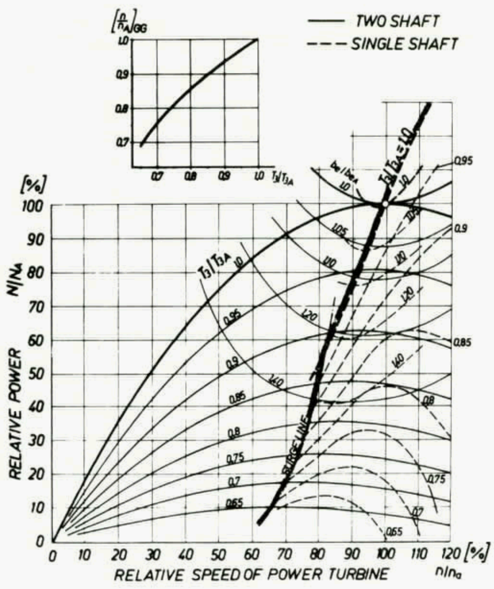


Fig. 40 Comparison of Power Characterisitics and specific Fuel Consumption of a Two-Shaft and a Single-Shaft Gas Turbine

independant from each other. Torque output and output speed can freely be selected between zero and full load. In the performance map (Fig. 40) the lines of constant gas generator speeds and constant turbine inlet temperatures are in first approach parabolas having the vertexes at the speeds of best efficiencies of the power turbine ($U/C_0 \approx$ constant). Along each of the indicated parabolas $T_3/(T_3 = 100\%)$ = constant the power delivered from the gas generator to the power turbine is constant and the variable output results from the speed dependent efficiency of the power turbine.

Load changes - also at constant output speed - result in a speed change (and temperature change) of the gas generator and therefore take place with a certain delay in time. The gas generator is to be designed with lowest possible moments of inertia i.e. small wheel radii - this specially applies to vehicle turbines which are subjected to very frequent and quick load changes. Where aerodynamic aspects (for instance optimum efficiencies) require larger diameters, the efficiency and the fuel consumption, respectively and the acceleration behavior should carefully be balanced.

For vehicle gas turbines the two-shaft version with free-power turbine can currently be considered to be the standard type. The low pressure ratio meets conveniently the requirement for a low moment of inertia for the gas generator. A heat exchanger of the regenerative or recuperative type is used to improve the specific fuel consumption. A variable head split between compressor turbine and power turbine by means of varaiable nozzles in front of the power turbine results in a more economical fuel consumption in the part load range, and a very smooth consumption curve versus load can be attained. Fig. 41a demonstrates the comparison of the specific fuel consumption for a vehicle turbine with and without variable nozzles. It should be noted that the improvement in fuel consumption in the part load range is due to the interaction of nozzle variation and heat exchanger. Due to the reduction of the flow area in the power turbine nozzles both the turbine inlet temperature as well as the turbine exhaust temperature i.e. heat exchanger inlet temperature are increased and thus the heat recovery in the heat exchanger is improved. In an engine without heat exchanger the variable nozzles are significantly less effective with regard to the part load fuel consumption.

An enlargement of the throat area of the power turbine nozzles changes the head split in favor of the compressor turbine so that more power for acceleration of the gas generator is available.

Fig. 41b shows the gas generator speed characteristic versus the time for a turbine with and without variable nozzles. For a turbine of 250/350 hp-class using variable nozzles the acceleration times of the gas generator can be expected to be only 0.8 - 1.2 sec. from idle to rated speed. Adjusting the nozzles beyond 90° the blades of the power turbine are subject to a strong negative incidence and thus a retardation - if necessary a reverse rotation - is possible (Fig. 41c). A limitation of the power for braking in some cases is given by the heat exchanger, since in the case of braking there is hardly a temperature drop in the power turbine; therefore, when braking, the permissible turbine inlet temperature is governed by the temperature resistance of the heat exchanger - in case of a regenerator the temperature resistance of the sealing strips. However, braking results in additional fuel consumption.

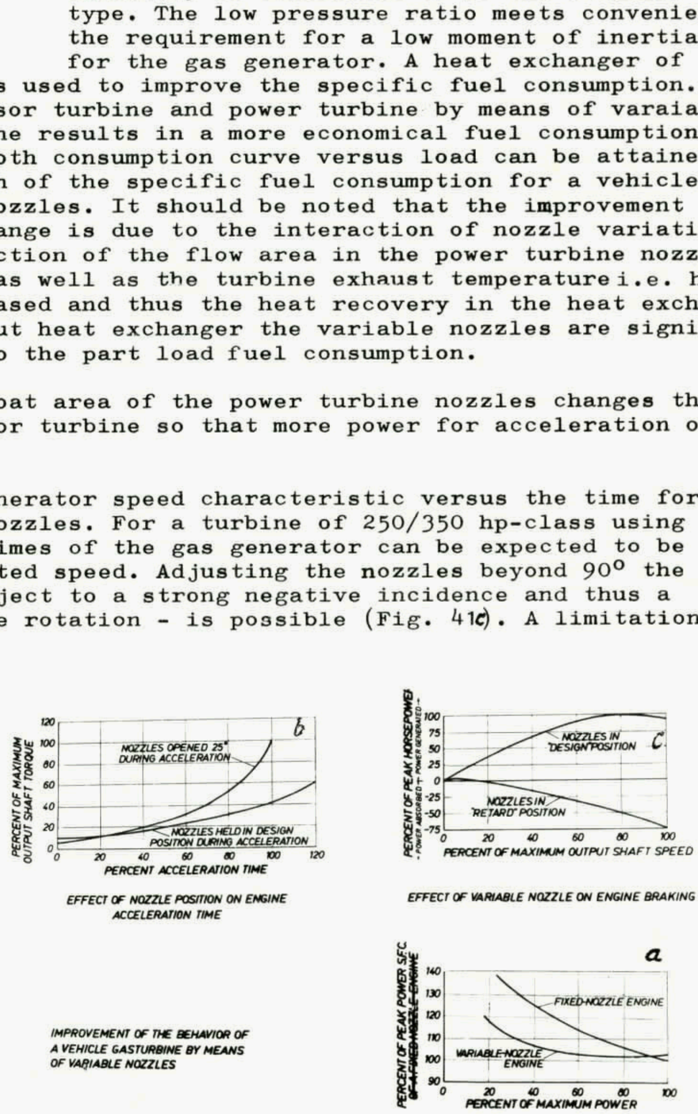


Fig. 41 Improvement of the Behavior of a Vehicle Gas Turbine by Means of variable Nozzles

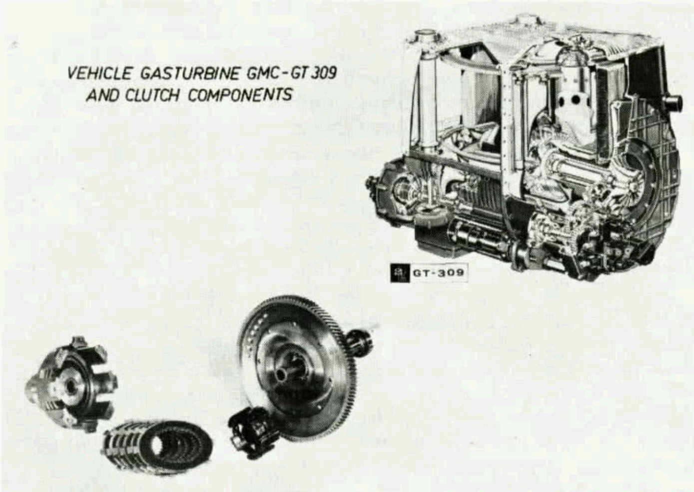


Fig. 42 Vehicle Gas Turbine GMC-GT309 and Clutch Components

The variable nozzles for the power turbine are used to-day by most of the vehicle turbine manufacturers as e.g. ROVER/LEYLAND - FORD - CHRYSLER - SOLAR etc. Instead of variable nozzles GENERAL MOTORS CORP. (G.M.C.) uses a clutch with controlled contact pressure (Fig. 42) between compressor turbine and power turbine. Thus a scheduled speed-dependant part-load is transferred from compressor turbine to the power turbine aiming for keeping the turbine inlet temperature also in the part-load range as high as possible. Clutch and turbines are designed in such a way that even at full load a certain torque is still transferred. To accelerate the gas generator the clutch is disengaged and the total compressor turbine power is available. For braking and at overspeed of the power turbine the clutch will be engaged and the compressor input (less the cold turbine power) is used for braking purposes. In this case braking does not cost any fuel; the braking effect could be increased by using a by-pass downstream the compressor, however, in any case is very dependant on the speed and sufficient at higher speeds only.

The steady slip of the clutch during operation means wear and increase of the evolving heat due to the power loss. Since, (according to Turunen and Collman) in the common operation range only relative small power at small speed differences will be transferred there should not be any mechanical problems.

The gas turbine with free power turbine meets the requirements of motive power for vehicles having an increasing torque at decreasing speed. Contrary to the piston engine the vehicle can start moving without using a clutch and a high stall torque reduces the necessary number of transmission gears. Two other designs should be considered here, that are used to increase the stall torque above the already mentioned value of approximately twice the rated torque.

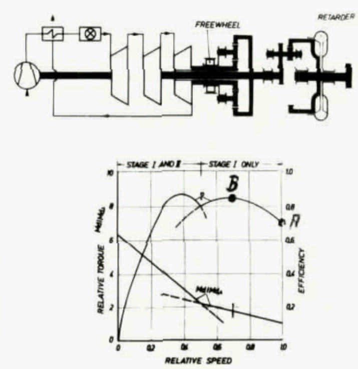


Fig. 43 Scheme of VOLVO Vehicle Gas Turbine and its Torque Characteristic

The VOLVO vehicle gas turbine (Fig. 43) in its design is a two-shaft gas turbine. The power turbine has - due to experiences with hydraulic torque converters - two rotors in line which are connected to each other through a freewheel planetary gear. In the high output speed range (> 50 % speed) the second rotor is running free without any power output. At lower output speeds the freewheeling blocks and the second rotor is engaged, now deviates the flow and reduces the exit swirl of rotor I. The torque of rotor II increases the output torque (Fig. 43). Since the full-speed point is already in the range of decreasing efficiency of the power turbine - the power turbine at full speed therefore aerodynamically runs at overspeed - a stall torque of 6 times the full-speed torque was expected. If referred to point B, in other words referred to the point of best efficiency of the power turbine the torque ratio to stall is only 4 times; the disadvantage is that in the normal operation range (n/n_{max} = 60-100%) the second, free running rotor represents a considerable drag and thus reducing the total efficiency of the power turbine.

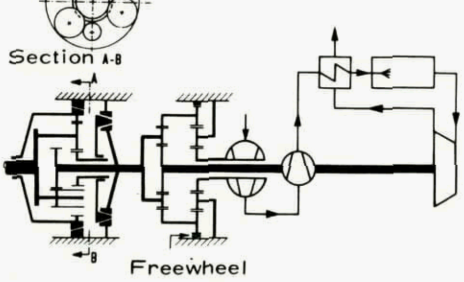


Fig. 44 Scheme of Differential-Gas-Turbine with Split Compressor (according to Hryniszak and Jacobsen)

An other possibility to increase the ratio of stall torque to rated torque to the factor 4 is given by the differential gas turbine with split compressor (Fig. 44). In this engine the gas generator is connected with the sun wheel and the supercharging compressor is coupled with the outer wheel of a planetary gear. The planet carrier is connected with the output shaft. At normal working conditions the augmenting compressor produces only a relatively low pressure ratio. With decreasing output speed (speed of the planet carrier) the speed of the augmenting compressor increases (at the same time an increasing part of the gas generator power is transferred to the supercharging compressor) thus mass flow and total pressure ratio increases. The gas generator therefore is being super-charged. Under normal conditions for the differential gear and with possible speed limits for the augmenting compressor torque ratios of approximately 4 can be expected. The tests have shown, however, that for matching the augmenting compressor and the main compressorvariable diffusor vanes would be necessary for both compressors; this would result in lower efficiencies for both compressors and in a remarkable increase in price for the power plant. The gas turbine with free power turbine to-day is predominant in the application for vehicle traction, notwithstanding the more sophisticated design compared to the single-shaft turbine andthe necessary variable parts being highly strained due to temperature and wear. A similar operation range as is possible with a free power turbine could be attained by coupling a hydraulic torque converter to the simpler single-shaft engine. The single-shaft engine works as a generator with mechanical power output for the hydraulic torque converter. For extending the operation range and for improving the part-load consumption a torque converter with variable guide vanes would have to be used. The vane geometry of the torque converter should be varied in such a way that the pump driving torque does not essentially increase with decreasing output speed i.e. decreasing ratio $\nu = n_T/n_P$ because only a narrow speed range $[0.90<(n/n_o)_{GT}<1.10]$ of the gas turbine is available; therefore, a speed "reduction" of the prime mover should be avoided. Fig. 45 shows the characteristics of a torque converter with variable guide vanes, these points of view being considered in its design. To have an efficiency as high as possible in the most frequently used operation ranges (e.g. of a heavy truck) the design point should not be located in the mesh-point $M_T/M_P = 1.0$ with the guide vanes having normal position $S/S_o = 100$ % - the increase in pump torque with decreasing output speed would not be permissible anyhow - but in the mesh point $M_T/M_P = 1.25$. With a stall/design torque ratio of approximately 2.95 / 1.25 = 2.36 the hydraulic torque converter in any case would need the application of an additional multistage variable gear in order to attain sufficient stall torque.

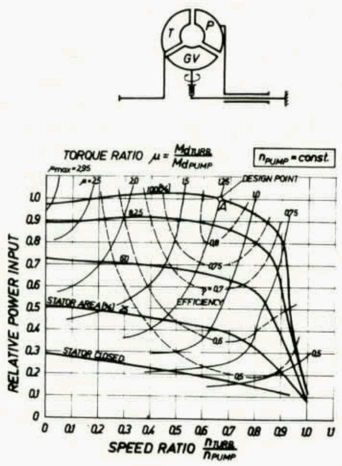


Fig. 45 Characteristics of Hydraulic Torque Converter with Variable Guide Vanes

In the discussed horsepower range hydraulic torque converters with variable guide vanes have maximum efficiencies of approximately 82 - 84 %. Taking into account the additional loss of variable nozzles in a two-shaft gas turbine (approx. 2 - 4 %) then the single-shaft engine being coupled with a variable torque converter will have a worse consumption by about 14 - 16 % over the total operation range i.e. with present turbine inlet temperatures approximately 25 - 30 g/hp h.

Hydrostatic gears in general do have lower efficiencies than hydraulic torque converters. Good efficiencies over a wide speed range can be achieved according to a proposal of Messrs. JONSSON Corp. by a useful combining of mechanical power gears (planetary gear) with a hydrostatic pump/motor set; the stall/rated torque ratios are thereby about 4 : 1. The proposed gear is a version of the Dual Mode Hydromechanical Transmission for gas turbines as described by R.H. Guedet and J.E. Louis.

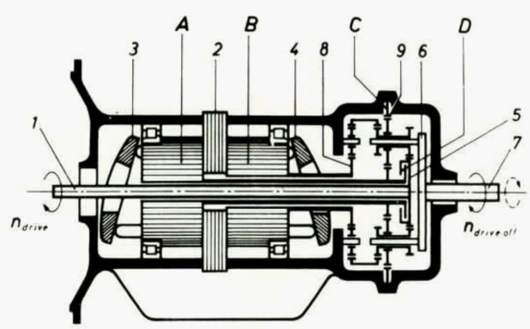


Fig. 46 Hydro-Differential Transmission JONSSON Corp.

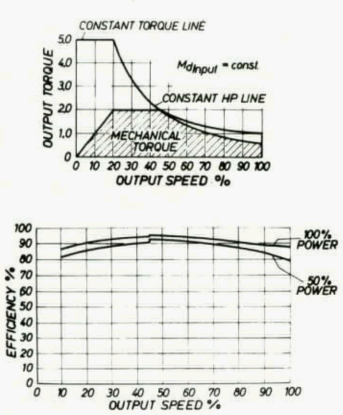


Fig. 47 Basic Performance Characteristics of hydrodifferential Transmission

Fig. 46 illustrates in a scheme the design of the hydrostatic gear according to JONSSON. The units A and B are identical and coaxially arranged hydr. swash plate displacement pumps (piston pumps) which operated alternately as pump or motor.

The pumps are mounted back to back on the control pad 2 therefore minimal mass flow ways exist. The driven shaft coupled with the gas turbine is connected with the sun wheel 5 of a two-stage planetary gear. The planet carrier 6 on the other hand is coupled with the output shaft 7.

D and C are controllable clutches. D permits to directly connect the shaft of the unit B with the driven shaft 1. C fixes or releases the outer wheel 9. With the planetary gears through gear wheel 8 the unit A is connected. 3 and 4 are controllable swash-plates which can be operated in the range of $\pm$ 18°.

By shifting the swash-plates 3 and 4 and alternately engaging the clutches D and C all operation phases of n_{output} = 0 or reversal run, respectively, up to n_{output} = n_{driven} can be demonstrated. The practical realizable ratio of stall output torque to input torque is approximately 4 : 1. Fig. 47a illustrates the combination of the output torque (at constant input torque). Since only a part of the power is transmitted by the hydrostatic drive (max. approximately 55 % - in average approximately 25 - 30 %) the relative low efficiency of the hydraulic unit is not of decisive importance and as demonstrated in Fig. 47b total efficiencies of more than 90 % over a broad speed range can be expected.

III. CONCLUSIONS

The development of small gas turbines calls for the solution of a series of special problems. High turbine inlet temperatures are in this case too of vital importance. Cooled turbine blades, however, can only be applied to a limited extent. Even at higher horse-power output higher efficiencies can be expected in certain cases for radial turbines, than with axial turbines having unfavorable aspect ratios.

Aircraft gas turbines are built without heat recovery for reasons of saving weight and volume. High compressor pressure ratios are necessary to attain favorable specific fuel consumption and therefore multi-stage compressors or combined compressors become necessary. In vehicle gas turbines heat exchangers will be used because of the demand for lowest fuel consumption possible. Recuperators and regenerators have their special range of application whereby for outputs up to approximately 400 hp the regenerator - specially the ceramic regenerator will show better data. To improve the part-load consumption a variable head split between compressor turbine and power turbine will be necessary. This can be achieved by a mechanical power transmission between the two turbines (clutch) or by changing the nozzle geometry (variable nozzles).

Two-shaft engines have an increasing torque at decreasing output speed and an output speed being independant of the gas generator; but at load changes it will react with a certain delay.

Single-shaft engines operate in a narrow speed range only and therefore are especially suited for driving purposes at constant speeds. Load changes are obtained by temperature changes only and practically without time delay. By using suitable accessory units i.e. hydraulic torque converters or hydro differential gears the single-shaft engine will get characteristics similar to those of the two-shaft engine and therefore will be applicable for similar purposes.

ACKNOWLEDGEMENT

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REFERENCES

Compressor

(1) Baljé, O.E.: A Study on Design Criteria and Matching of Turbomachines
ASME - Transact. (Journ. Eng. and Power) Jan. 1962, page 83 / 114

(2) Baljé, O.E.: Loss and Flow Path Studies on Centrifugal Compressors
ASME - Paper, No. 70-GT-12

(3) Groh, F.G. / Wood, G.M. / Kulp, R.S. and Kenny, O.P. :
Evaluation of a High Hub/Tip Ratio Centrifugal Compressor
ASME - Paper, No. 69-WA/FE-28 (1969)

(4) Howard, J.H.G. and Lennemann, E.:
Measured and Predicted Secondary Flows in a Centrifugal Impeller
ASME - Paper, No. 70-GT-55 (1970)

(5) Kenny, D.P.: A Novel Low Cost Diffuser for High Performance Centrifugal Compressors
ASME - Paper, No. 68-GT-38 (1968)

(6) Kenny, D.P.: Supersonic Radial Diffusers
ASME - An. Gas Turbine Conf. (1970)

(7) Morris, R.E. and Kenny, D.P.: High Pressure Ratio Centrifugal Compressor for Small Gas Turbine Engines
AGARD - Prop. and Energ. Panel - 31st Meeting (1968)

(8) Rodgers, G.: Variable Geometry Gas Turbine Radial Compressors
ASME - Paper 68-GT-63

(9) Smith, V.J.: A Review of the Design Practice and Technology of Radial Compressor Diffusers
ASME - Paper, No. 70-GT-116

COMBUSTION CHAMBER

(10) Anderson, R. and Wagner, D.: Advanced High Temperature Combustor Research for Small Gas Turbine Engines
SAE - Paper 680446 (1968)

(11) Beer, H. / Schlott, B. and Schlader A.F.: Dampffilmkühlung in Flugtriebwerks-Brennkammern.
ZFW 18 / 1970, H.8., page 286/90

(12) Bell, L.E.,jr.: Design Requirements of Small High Temperature Rise Combustors
SAE - Paper 680447 (1968)

(13) Durrant, F.: The Control of Atmospheric Pollution from Gas Turbine Engines
SAE - Paper 680347 (1968)

(14) Scarfe, R.G.M.: A Brief Review of Small Engine Combustion Systems and Problems
GTCC - Combustion Sub-Committee, Paper No. 264 (1968)

(15) Seifferlein, Th.: Die Verbrennung in Flugstrahltriebwerken
MTZ 19 (1958) Nr. 7, page 259/62

(16) __________ : A Look at Burners for Small Gas Turbines.
SAE - Journal (June 1962), page 64/70

TURBINE

(17) Amann, Ch. and Sheridan, D.C.: Comparison of Some Analytical and Experimental Correlations of Axial Flow Turbine Efficiency
ASME - Paper, No. 67-WA/GT-6 (1967)

(18) Amann, Ch. A and Dawson, D.W.: The Power Turbine and its Diffuser
SAE - Paper, No. 690032 (1969)

(19) Baljé, O.E. and Binsley, R.L.: Axial Turbine Performance Evaluation
ASME - Transact. (Journ. Engng. and Power) October 1968, P.341/360

(20) Bayley, F.J. and Turner, A.B.: The Transpiration - Cooled Gas Turbine
ASME - Paper, No. 70-GT-56 (1970)

(21) Chauvin, J. / Papailiou, K. and Burrows, L.: Aerodynamic Problems in Cooled Turbine Blading Design for Small Gas Turbine
AGARD - Prop. and Energ. Panel - 36th Meeting (1970)

(22) Dunham, J. and Came, P.M.: Improvements to the Ainley-Mathieson Method of Turbine Performance Prediction
ASME - Paper, No. 70-GT-2 (1970)

(23) Dunham, J. and Smith, D.J.L.: Some Aerodynamic Aspects of Turbine Design
AGARD - Prop. and Energ. Panel - 36th Meeting (1970)

(24) Gabel, R.M.: Feasibility Demonstration of a Small Fluid-Cooled Turbine at 2.300 F
SAE - Paper, No. 690034 (1969)

(25) Helms, H.E. and Emmerson, C.W.: Analysis and Testing of Air-Cooled Turbine Rotor and Stator Blades
ASME - Paper, No. 65-WA/GTP-10 (1965)

(26) Lombardo, S. and Moskowitz, S.L.: Experience with Transpiration Cooled Blades
AGARD - Prop. and Energ. Panel - 36th Meeting (1970)

(27) McBridge, J.W. / Norris, R.H. and Perugi, A.H.: Some Factors in High Temperature Gas Turbine Design - Size, Working Fluid and Operational Effects
ASME - 11th Gas Turbine Conf. (1966)

(28) McLean, A.F.: The Application of Ceramics to the Small Gas Turbine
ASME - Paper, 70-GT-105 (1970)

(29) Mitchell, R.W.St. and Ogale, V.A.: Gas Turbine Blade Cooling - Retrospect and Prospect
ASME - Paper No. 67-WA/GT-9 (1967)

(30) Moskowitz, S.L. and Schober, T.E.: Design and Test of a Small Turbine at 2500 F with Transpiration Cooled Blading
SAE - Paper No. 690035 (1969)

(31) Okapuu, U and Calvert, G.S.: Cooled Radial Turbine for High Power-To-Weight Applications
AJAA - Paper No. 69-524 (1968)

(32) Okapuu, U. and Calvert, G.B.: An Experimental Cooled Radial Turbine
AGARD - Prop. and Energ. Panel - 36th Meeting (1970)

(33) Rahnke, Chr.J.: The Variable-Geometry Power Turbine
SAE - Paper 690031 (1969)

(34) Rogo,C.: Experimental Aspect Ratio and Tip Clearance Investigation on Small Turbines.
SAE - Paper 680448 (1968)

(35) Watanabe, J. / Ariga, J. and Mashimo, T.: Effect of Dimensional Parameters of Impellers on Performance Characteristics of a Radial-Inflow Turbine
ASME - Paper No. 70-GT-90 (1970)

(36) Wood, H.J.: Current Technology of Radial-Inflow Turbines for Compressible Fluids
ASME - Transact. (Journ. of Engng. f. Power) Jan. 1963, P.72/83

SMALL GAS TURBINE - GENERAL FEATURES

(37) Darfeuil, A.: Application des petites turbines à gaz à des usages industriels.
La Technique Moderne, Sept. 1967

(38) Markham, B.G.: Reliability in Aero Engines
The Chartered Mechan. Eng., Jan. 1967

(39) Merz, J.: Graphische Ermittlung des Betriebsverhaltens einer Gasturbine
MTZ 29 (1968) No. 12, page 505/509

(40) van der Nuell, W.T.: Gasturbinen im Wettstreit mit Kolbenbrennkraftmaschinen
MTZ 29 (1968) No. 5, page 216/20

(41) van der Nuell, W.T.: Fluggasturbinen kleiner und mittlerer Leistung
MTZ 31 (1970) No. 5, page 211/215

(42) Paul, C.H.: The Development of Small Gas Turbines for Aircraft Auxiliary Power
Royal Aeronaut. Soc. (March 1970)

(43) Szydlowski, J.: Développement en France des petits turbines à gaz - Applications industrielles
Entropie No. 11 (Sept./Oct. 1966)

(44) Vogts, J.: Die Gasturbine als Antrieb in Schienenfahrzeugen
Glas. Ann. 93 (1969) No. 5, page 141/49 and 163

(45) Vogts, J.: Eine neue Hilfsgasturbine für die Luftfahrt
MTZ 31 (1970) No. 5

VEHICLE GAS TURBINE ENGINES

(46) Eckert, B.: Der heutige Entwicklungsstand der Kraftfahrzeugturbine
Automobil-Industrie No. 1 / 1964

(47) Eckert, B.: Hat die Automobile-Gasturbine eine Chance
ATZ 69 (1967) No. 9 / 12

(48) Eckert, B.: Die Gasturbine als Antrieb für das schwere Straßenfahrzeug
MTZ 31 (1970) No. 5

(49) Gimmler, J.: Aufbau und Auslegung des Föttinger-Wandlers für Kraftfahrzeuge
Automobil-Ind. 1970, No. 3, page 97/104

(50) Gühne, H.P.: Donaldson - Luftfilter für Gasturbinen
MTZ 30 (1969) No. 10

(51) McLean, A.F.: Case for the Single-Shaft Vehicular Gas Turbine Engine
Lecture: Techn. Advances in Gas Turbine Design - Paper 16 (1969)

(52) Marwood R.M. and Prueser, W.H.: Potential Performance of Gas Turbine Power Plants
SAE - Paper No. 650715 (1965)

(53) Montgomery, J.E. / Schelp, H.R. and Furst D.G.: 200 ./. 300 HP Gas Turbine Engine Family for the US-Army.
SAE - Paper No. 795 A (1964)

(54) Mund, M.G. and Gühne, H.P.: Gas Turbines-Dust-Air Cleaners: Experience and Trends
ASME - Paper 70-GT-104

(55) Mund, M.G. and Gühne, H.P.: Abmessungen und Betriebskennwerte moderner Gasturbinenluftfilter
MTZ 31 (1970) No. 5-7-10

(56) Penny, N.: Rover Case History of Small Gas Turbines
SAE - Paper 634 A (1963)

(57) Phillips, P.A. and Spear, P.: Design, Development and Application of Vehicle Gas Turbine Engines
Proc. Inst. Mech. Engrs. 1966/67, Volume 181 Pt2A, No. 5, page 149/172

(58) Singer, R.B. and Beck, N.J.: The Economics of Turbine Power Plants for Trucks
SAE - Paper 690548 (1969)

(59) Turunen, W.A. and Collman J.S.: The GENERAL MOTORS Research
GT- 309 Gas Turbine Engine
SAE - Paper No. 650714 (1965)

(60) Wakefield, R.: 4-Wheel-Drive by Planetaries and Hydrostatics-
The Jonsson System
Road and Track, Vol. 19, No. 12 (August 1968) page 86/89

(61) White, J.W.: Advanced Army Components Technology Program
SAE - Paper No. 650707 (1965)

INDUSTRIAL AND TECHNOLOGICAL PROBLEMS OF SMALL GAS TURBINES FOR HELICOPTERS AND GROUND TRANSPORT

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SUMMARY.

After considering why a small engine needs to rotate fast, and be made of integral rather than built up parts, some of the consequent vibratory problems are discussed with the conclusion that methods of introducing damping into the system are required.

Fuel system limitations due to dirt being the same size for big and small engines limit the use of scaled down large engine designs. Contamination of compressors by foreign objects is similarly more pronounced.

A number of workshop problems special to small size are considered and shown to respond to the use of suitable techniques.

Finally a glance at some of the costs which don't scale indicate proportionately high launching costs.

It was not altogether by accident that gas turbines evolved during the early phases of their development at the size they did. Undoubtedly, it was because aircraft were the size they were, and had the drag they had, that the requirement existed to develop the early gas turbines at a certain size. Fortunately, this was a good size from which to develop the later engines and some problems were easier to overcome because of the size. Had the requirement been for what we now call small engines, and small engines which run well throttled back, gas turbines would not have been so successful.

GENERAL CONSIDERATIONS.

The most significant parameter in a turbine engine is Mach Number, for it is this which is the basis of compressor and turbine operating characteristics. The speed of sound through a gas is not in the least affected by the size of the engine, so Mach Number doesn't scale. (Diagram 1) This means in turn that blade speeds must be kept the same in terms of velocity and so a small engine must rotate as much faster as the linear scale of the mean height of the blading. (Diagram 2).

The thermodynamic cycle also does not scale. For a given thermodynamic efficiency pressure ratio and temperature are fixed.

If one took a 10,000 lb. thrust engine and said lets make it a 1,000 lb. thrust engine, one could merely take a slice of the engine such that one had one tenth of the annulus height all the way through. (Diagram 3). This would give satisfactory Mach Numbers and clearly the blading has adequate length to generate and expand the pressure ratio. The resultant small annulus is almost all losses, boundary layers etc. with proportionately more heat and mechanical losses. (Diagram 4.) It is the attempt to improve upon this situation which makes the small engine a particular technological challenge.

To restore the hub tip ratio of the annulus, or the ratio of the height of the blading to its diameter, we must reduce the mean height, which forces us into higher rotational speeds. (Diagram 5) Higher rotational speeds produce more whirling problems, which can most readily be attacked by reducing axial length. Although this would come automatically from scaling length as well as diameter, most processes used in the gas turbine require the same length. (Diagram 6) Burning fuel in less axial length is a particular problem, because of course flame speed doesn't scale, the velocity of air in the combustion space must be the same so that the time to burn and the length of the flame tend to be non-scaleable. In order to reduce length between bearings of this high speed shafting one can alter the layout and the flame tube is one of the most convenient to move, and one can see widely divergent solutions to the problem of re-siting the flame tube. (Diagram 7.)

One has arrived at a design from these considerations which rotates fast, has probably somewhat less favourable hub tip ratio in its blading but is essentially smaller in all dimensions looked at on a cross section. (Diagram 8)

TOLERANCES.

Of course, tolerances on dimensions don't scale, and one finds that the variations on blade profiles and annulus heights are a bigger proportion on small engines so that one has a series of options from using different techniques, as watchmakers do (expensive) to down grading the nominal or average overall engine performance. If one has scaled length then axial tolerances are also proportionately more critical. But axial length for the basic processes has probably not been changed, and axial tolerances will be unchanged. The standard of tooling used for making the parts has a direct impact on the consistency of shape that can be maintained, and so the small high efficiency engine demands a relatively high tooling expenditure. This, surprisingly, may not increase the product price because the better tools can produce more cheaply as well as more consistently. But the tooling cost adds to the capital at risk in the launching cost.

Distortion of some degree occurs in practically any engine part after it has run, and it is, of course, useless to make a part more accurately, off expensive tooling if it distorts to a significant extent in the engine. Turbine clearances are always particularly important in high pressure ratio engines and care must be taken at the design stage to ensure that circularity, planarity, concentricity will be maintained when hot, even at the expense of some other desirable feature like weight.

INTEGRAL COMPONENTS.

The most obvious technique to try to use to minimise cost and tolerance penalties is to make components integral rather than built up out of a number of separately made pieces. This can be done either by machining from lumps or casting. Typical examples are the cast compressor on the Allison 250 engine or the integral turbine blades and disc on Nimbus. A fundamental problem of this type of construction is the very high Q or response to excitation. A bell which gives a decent ring is invariably made from one piece, because too much damping is introduced at an interface (Diagram 9). There is plenty of energy in a turbine engine at all sorts of frequencies which can ring any bell like structure. The integral component is, therefore, very susceptible to fatigue failures unless these sources are completely understood. In my experience we are a long way from understanding all these sources. A complication of an integral structure is that it has mechanical coupling between blades and disc which can produce a whole range of resonant frequencies of equal significance because the individual blades are not frequency matched due to small variations in size. (Diagrams 10 and 11.) Resonances occur not just at the individual blade frequencies but at a complex permutation of the individual blade frequencies dependent upon the mechanical coupling. This can occur in the first, second etc. families of disc modes and bell modes. Mathematical models of these complex patterns can now be constructed, and their relevance can be confirmed by the use of laser operated holograms. Although these techniques are not yet far enough developed to be used by the designer, they have already explained the lack of success in using strain gauge techniques for trouble shooting on development engines. Although one can take steps to reduce the important interferences, to reduce the band of frequencies, a better line is, in my view, to put damping into the system, and I would expect to see big progress in this area during the forthcoming decade. (Diagrams 12,13 and 14).

An important area to give detail attention is the mechanical coupling between a known source of excitation such as a gear mesh and an integral component. In this case the solution must isolate the source axially as well as torsionally if there is any suspicion of an umbrella mode. Allowing adjacent parts to rattle as a means of absorbing energy has to be treated with care, for wear due to fretting can be dangerous. The thin section of the small engine can tolerate less wear than big ones. The usual anti-fret treatments can be used, but it is important not to remove the damping characteristics in overcoming a wear problem.

We saw why small engines have to rotate fast, this means that excitation frequencies are similarly high. Some fatigue problems can be considered in terms of amplitude x frequency = constant for a given life. (Diagram 15). So we can see that a much smaller amplitude is important. If the amplitude is governed at all by machining imperfections, boundary layer thickness, aerodynamic or mechanical damping then the problem is more difficult in the small amplitude, high frequency case. If one is relying on stick-slip friction damping, it may be very difficult to get enough amplitude to be in the slip regime before a dangerous resonance has evolved.

FASTENERS

Fasteners are a limitation to the design both because of the human skills involved (one is using engine fitters and not watch makers) and the standard workshop equipment and techniques are not readily adapted to using screw threads much below 10 UNF. This is not a fundamental law of nature, but a practical point. One is better compromising the design slightly to use larger bolts in wider flanges, than to involve your own workshops, but more importantly your customers', in specialist techniques. This could well change. I have seen a small agricultural engine using screw sizes conventionally considered only suitable for model or instrument makers, and it works.

But the biggest gain comes from trying not to have fasteners. I have just discussed some of the problems of major components which are made integrally, nevertheless using brazes, spot welds, beamwelds or bonding must be exploited on the small engine as much as possible even at the expense of some complication in overhaul shop techniques.

FUEL SYSTEM

The size of dirt particles in the fuel taken into the tanks or generated by the tanks, pipes, pumps etc. is no different for big and small engines. The practical filtration pass size cannot be lowered without making the fuel filter disproportionate to the engine. Thus the smallest tolerable hole is fixed. In the passages feeding fuel into the engine,one can either have a smaller number of admission points,or use lower pressures. Viscosity is just the same and pressure needed to break up the flow into a spray is no different, so the pressure can't be lower. (Diagram 16) A solution to this is to use a different sort of system, and we have been fortunate in the RS.360 in having had the vaporiser system already developed elsewhere. (Diagram 17) This system doesn't depend upon pressure to break up the jet. Other solutions have been to feed fuel into the main shaft and use rotational effects to produce the spray, which is a very effective way of making a spray. (Diagram 18). This is a common feature on Turbomeca engines. Starting up either of these two systems has to be done by a torch sprayer, and the small engine approaches the point where more than one torch provides enough fuel to overheat the engine during a start. (Diagram 19) With a single torch one has more difficulty in getting propogation, and in achieving high reliability of first time starts than in a 10,000 lb. thrust engine where one can afford to have 5-10 torches, starting combustion at several parts around the can. (Diagram 20) A convenient solution for the very small size is to use an external combustion chamber. (Diagram 21)

In the control system of a hydro-mechanically controlled engine, the limiting hole size and clearance result in substantially as large a system, irrespective of engine size, with the associated weight and cost penalty.

CONTAMINATION

I mentioned the problem of fuel filtration. The same basic problem exists for all other contaminants whether it be grass, hailstones, gravel, bricks, spanners, or any of the curious things which operators manage to feed to engines, they are as big in absolute size and, therefore, proportionately bigger for small engines. Some things, like large birds, just can't get into small engines because they won't get down the ducting, but this is not much consolation. The small engine maker must recognise that foreign object damage is going to hurt him more than his big brother. Erosion protection of blading with todays techniques is more dependent on contaminant size, such that one needs the same thickness of coating. On a large fan blade this thickness will not affect its aerodynamics, whereas it might represent half the thickness of a small blade. Typical applications of the small engine has one compensating advantage here, in that it is not impractical to filter the whole engine flow of a small shaft engine. There are some losses, and some weight penalty. It can be and has been done, and the Donaldson Tube is a particularly convenient tool for doing this, and with today's techniques filtration of the whole engine flow when in particularly bad environments is the best way to go. (Diagramm 22) Small engines are inclined to go into vehicles which get into very intimate contact with dirty environments and have excellent stirring devices like helicopter rotors or tank tracks. High performance sophisticated equipment needs protection in these conditions. If one must match to such an environment as the most important criterion of the design, then clearly one would choose a less efficient, thicker, more rugged design of aerodynamic component than when one is looking for the best half power fuel consumption or lowest weight per horse power. The customer and designer must work closely together to ensure that the stated requirements are really what is needed.

MECHANICAL LOSSES.

Mechanical losses in shafting,bearings and windage are almost insignificant in a large engine. The designer need hardly consider the number of bearings etc. as having a direct impact on power output. We have seen the need for small engines to go to higher speeds, and must note that churning losses go up as $N^{3/2}$, at the same time as the output power is made smaller for the smaller engine. Tolerances of dimensions on bearings can't be much different, which restrains one from scaling rolling element sizes down as far as one would like or else one loses capacity due to mismatch of elements and surface finish. These combine to give proportionately much higher, and now significant, losses which limit the advantages which can be taken of more complex cycles or one achieves less than the expected gain in thermodynamic efficiency.

The restraints mentioned earlier usually result in the diameter not being scaled down fully and the annulus height for the working fluid being narrower and further out than ideally, which in turn means that windage is becoming significant, and must be considered as part of the compromise in reaching the design.

POWER OFF TAKE.

Some services are scaleable. A bigger passenger aircraft requires more air for cabin conditioning than a small one, the energy for operating control surfaces are higher, anti-icing is over a bigger surface. But the electrical load to operate pilots instruments, radio aids and so on are the same and the power off take looks disproportionately large on the small engine. This is no particular difficulty except that the accessory drive mechanism is not fully scaled down and imposes some weight and size limitations, and the abstraction of significant power has an effect on the gas generator section which must be watched particularly on multi-shaft engines. As power off takes rotate at the same speed e.g. a generator, the gearing down from the high speed, small shaft is more complex. If as much load as possible is put on to the power turbine stages of a shaft power engine, the problem will be that much smaller.

ACCESSORIES.

Accessories are similar to power off takes. They don't have to be any bigger on a 30,000 lb. engine than on a 1,000 lb. engine. If accessory development can produce smaller lighter units, big engines will take advantage of them as quickly as small ones. The need for smaller accessories, perhaps using quite different techniques, is greater on small engines because they may currently cost and weigh as much as 10% of the small engine and the restraint which they are imposing today is, therefore, very significant. The current expansion in small engine activity will, undoubtedly, put pressure on accessory development. The evolution of the helicopter from a vehicle which one man just had enough arms and legs to fly towards one where the pilot has time to do something else as well, and be multi-engined, produces a demand for much more sophisticated control systems which will do some of the work for him. These control systems are larger than required on simpler but larger fixed wing engines. Tremendous advances have been, and are being, made in electronics such that an engine mounted electronic control system is now quite practical. The reliability of first generation electronic systems was appalling, resulting in the movement of the control system away from the engine and the placing of more reliance upon mechanical systems. This trend must reverse, and I'm sure that we will see more and more of the engine control and monitoring moved from the pilot to electronics.

INSTRUMENTATION.

One clearly cannot use large instruments when developing small engines because they will affect the answer one is trying to find. For instance putting a large engine pressure rake into the air flow will produce significant blockage. Hanging large vibration transducers on a small component will alter its resonance characteristics. The only solution here is to use watchmakers techniques and face up to the need to have specialist skills capable of handling delicate equipment. Fortunately, great steps forward in miniaturising this type of equipment have been made and, apart from being expensive, this has substantially overcome the technical problems. The high frequencies involved make a lot of standard equipment unsuitable and one must be prepared to make up one's own, or invest in new equipment.

ACCELERATION RATES

These need not be a problem. The light rotors of the small engine give a basically quick response engine. Helicopter applications demand quick response and as long as adequate surge and fuel margins are provided, two smaller engines should be better than one equivalent larger one.

MATERIALS

Although basically the same materials can be used, the very thin sections demanded on small components mean that in some materials the grain size is of the same order as the section, and then the properties are adversely affected. The techniques of casting thin sections demand special attention, and almost always this has been successful. Small engine activity will put pressure on casting technology and we can expect to see progress here.

MANUFACTURING AND HANDLING DAMAGE.

Although one tries to avoid making delicate pieces, the other restraints make it inevitable that small engine components will be more sensitive to damage both during manufacture and when they are being handled in stores and supply circuits. With the advent of cheap soft plastics this damage can be avoided without much cost, but it does take conscious action to produce the necessary discipline amongst people used to dealing with larger components.

INSPECTION PROBLEMS.

These are not a major limitation. Some special equipment is needed particularly to deal with parts made by integral techniques. One example, pioneered by G.E. as far as I know, is the use of water flow as a means of establishing nozzle guide vane capacity more cheaply and accurately than dimensional measurements. A common small engine problem is that the inspector cannot get his fingers inside a bore. Not very expensive air gauge equipment can be inserted, or expensive light or even laser reflection techniques may be justified for production runs. For many such problems a re-think of the fundamental objective throws up a number of unconventional ways of achieving the same end, often better, which become practical in the small size application.

BENCH TESTING.

The small engine in itself requires no more or less testing hours to prove its integrity. The amount of testing will be determined more by the extent to which it departs from already proven design concepts and the demands of the installation. We have noted the trend to more sophisticated control systems and these will obviously add to the difficulty of testing and also to the sophistication of the test bed itself.

In the case of the RS.360 in the WG.13, the engine has been designed to be an integral part of the mass system coupled to the rotor, in order to ease the rotor design problems. This has added a further complication to the test bed design in order to simulate the nodding motion which will be imposed by the rotor.

MODULARITY.

Although in service modularity is equally advantageous to any size of engine, small engines show an additional advantage. As the fitters are the same size only two men may be able to work on a complete engine at any one time without getting in each other's way. A small engine which can be stripped quickly into a number of major units, can then be worked on by a large number of men thus escaping from a penalty of increased turn round times and even showing an advantage if taken far enough. 'In Service' modularity depends not just upon the physical design capability, but the performance margins available for mismatch of components, and the degree of variability of components. The remarks about the need for more sophisticated tooling are even more applicable when one is looking for trouble free interchangeability of major module assemblies. Even a small discontinuity in annulus line which would be inconsequential to a large engine will be important for the small modular engine.

COSTS WHICH DON'T SCALE.

When considering the parameters like cost per horse power to see whether the small engine manufacturer has done a good job it is as well to remember some of the costs which don't scale.

Marketing. - Fairly obviously, you can't produce smaller sales brochures, do less market research, go on shorter trips. You can afford to carry fewer people, which is probably a good thing, and/or you must sell in larger numbers to spread the costs, which is, undoubtedly, a good thing to do.

Performance. - The calculation of engine cycles, analysis of results, optimisation trouble shooting are, alas, no easier because of small size.

Drawings. - Although the total length of lines drawn does scale, the difficulty and time of drawing them is independent of size. Some small components have to be drawn 2 times or 5 times full size to be intelligible, as well as full size, which makes it more difficult rather than less so.

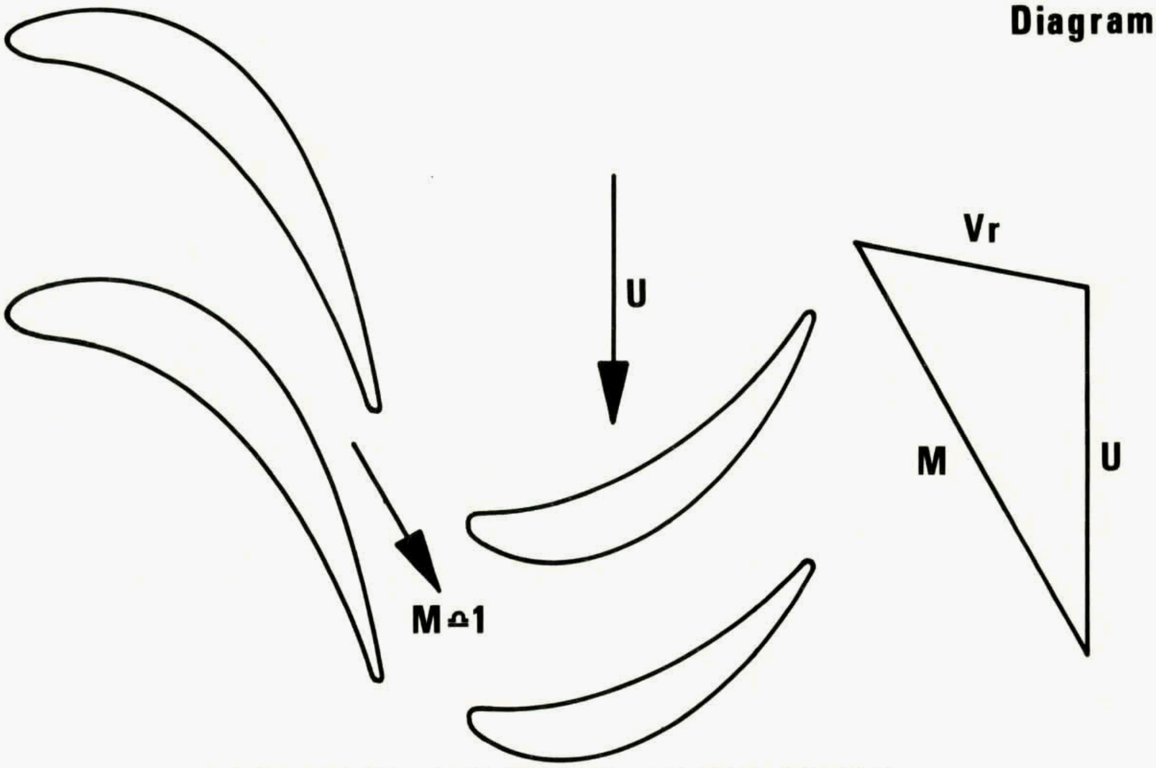
Testing observations and analysis. - The small engine is much easier to put on a test bed, and won't use as much fuel, but for a given technological standard, the number of observations and extent of analysis occupy just as much time. We have been unable to find a breed of small engineers who will agree to small salaries to go with small engine.

Product Support. - Service statistics, spares, supplies, field service engineers, all the supporting services are unaffected by size itself.

Launching Costs. - Add to these the remarks about better tooling, sophisticated instrumentation and inspection and one can see that there are a number of factors preventing the scale down of costs.

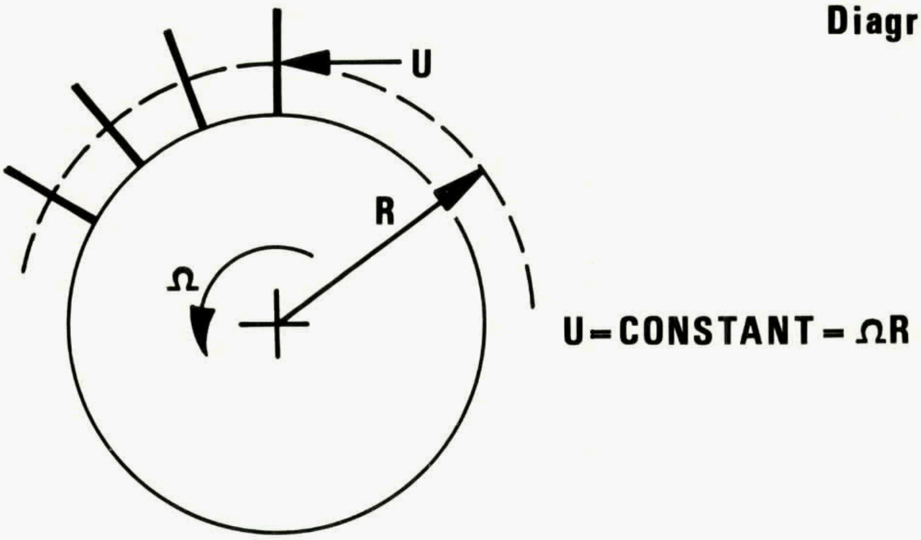
Despite these disadvantages, the small cost small size turbine engine is needed, and industry will provide it when given time to adapt.

Diagram 1



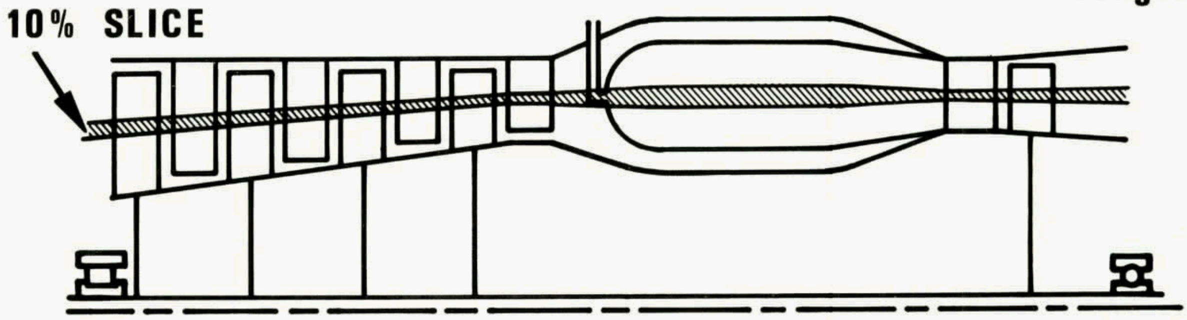
VELOCITY TRIANGLES DON'T SCALE

Diagram 2



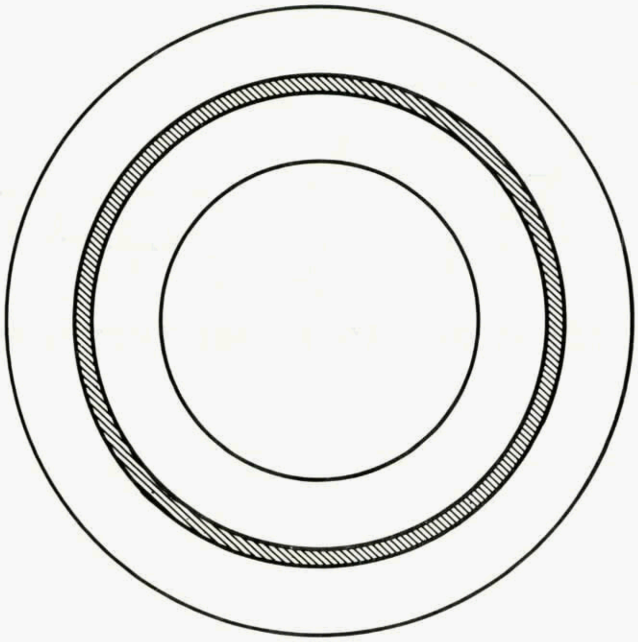
ROTATIONAL SPEED GOES UP AS RADIUS DECREASES

Diagram 3



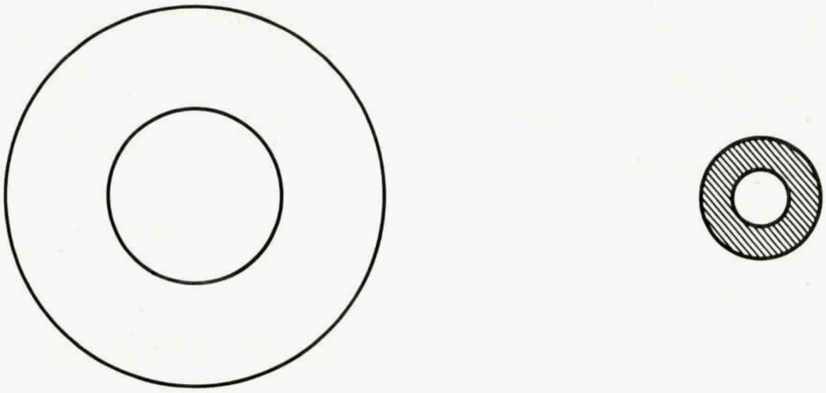
TAKING A SLICE THROUGH A 10000 lb ENGINE

Diagram 4



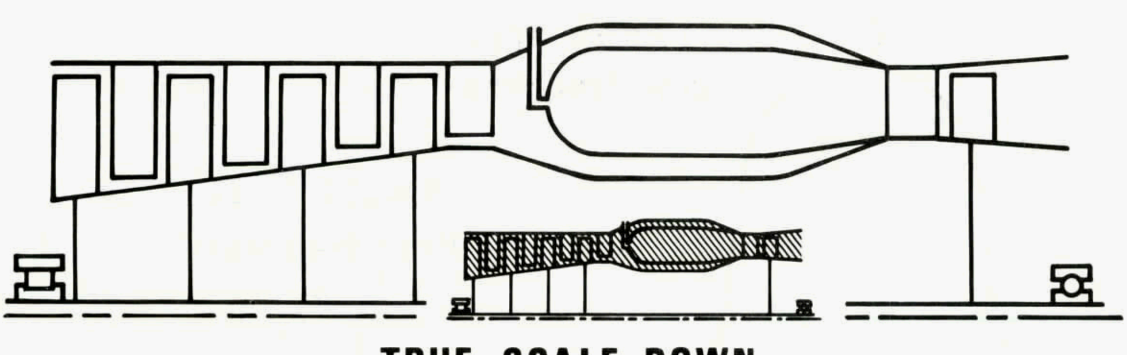
CROSS SECTION OF A 10% SLICE

Diagram 5



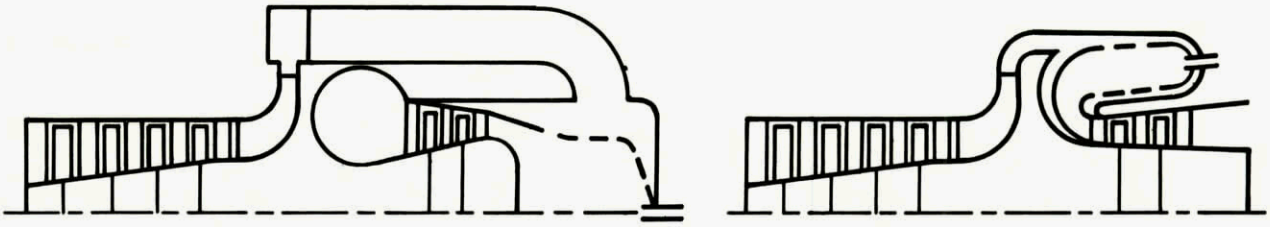
PRESERVING ANNULUS HEIGHT/DIAMETER RATIO

Diagram 6



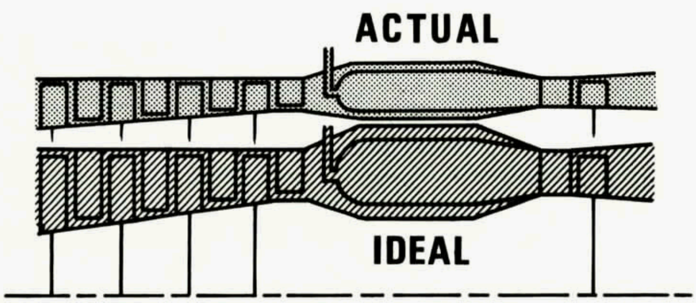
TRUE SCALE DOWN

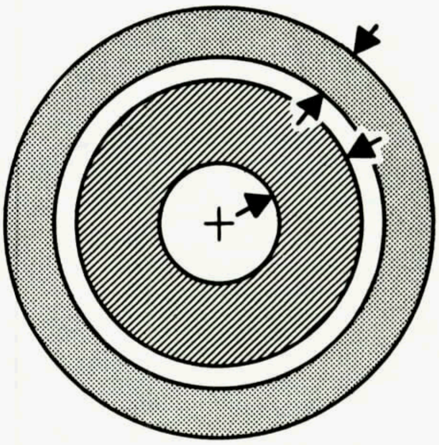
Diagram 7



DIVERSE SOLUTIONS TO FLAME TUBE PROBLEM

Diagram 8





EQUAL ANNULUS AREAS

Diagram 9

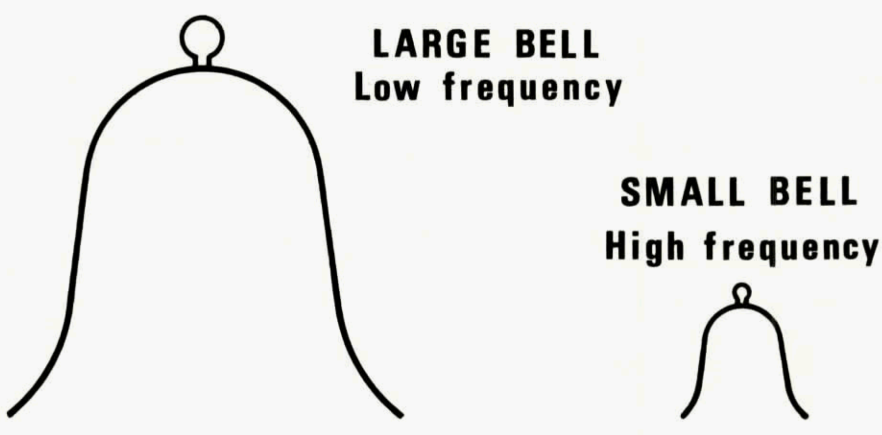
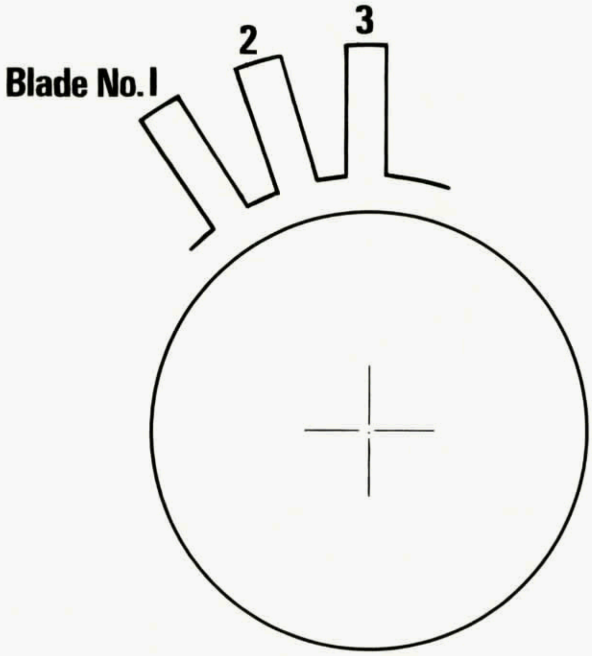


Diagram 10



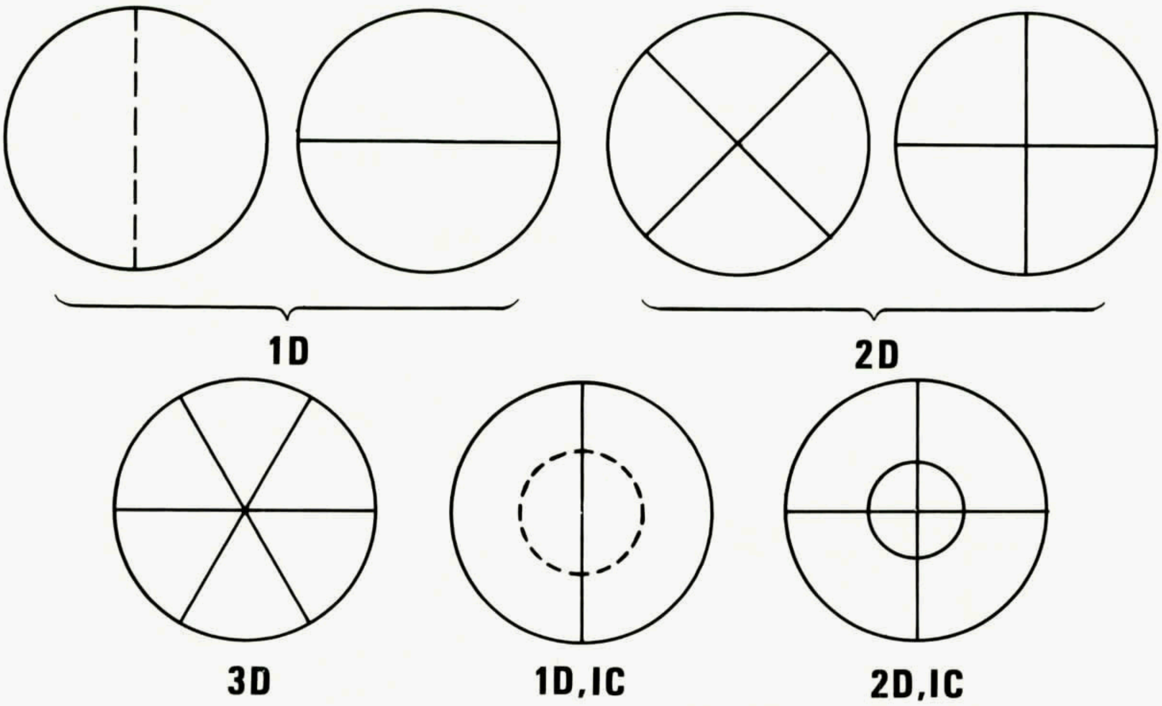
INTEGRALLY BLADED DISCS

Blade 1 has frequency f_1

Blade 2 has frequency f_2

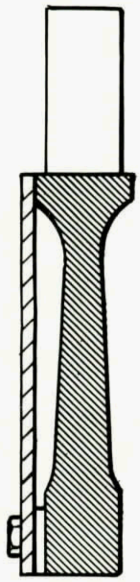
Blade 3 has frequency f_3 etc.

Diagram 11

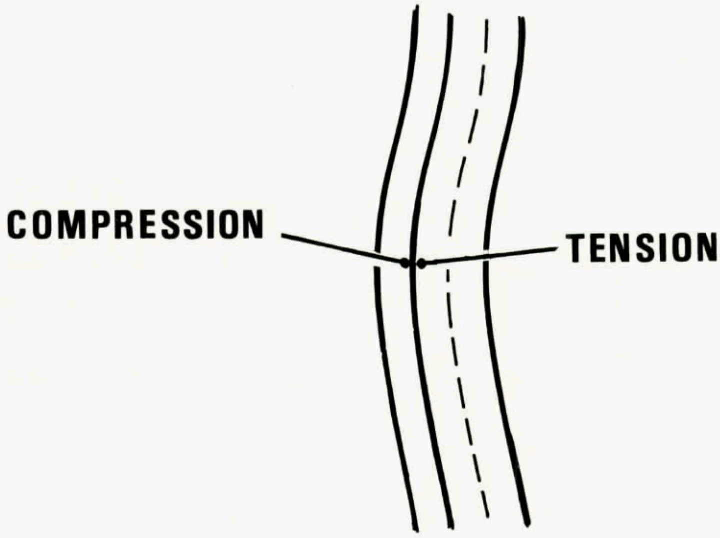


DIAMETRAL MODES

Diagram 12

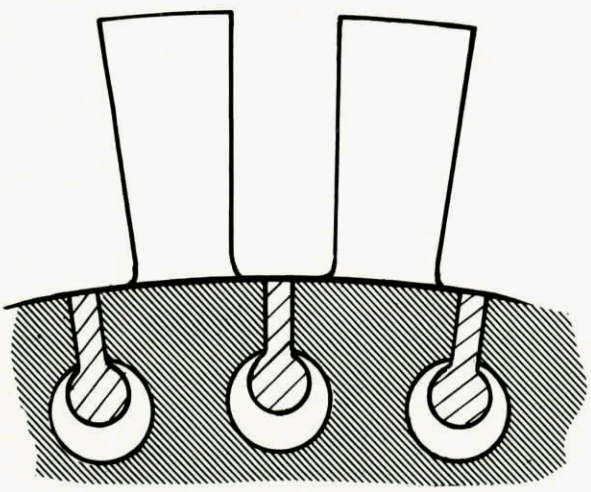


DAMPER PLATE ON AN INTEGRALLY BLADED DISC

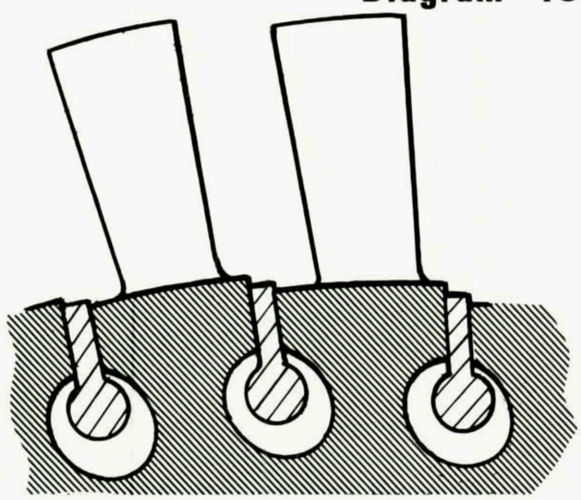


SMALL MOVEMENTS DAMPING 1D MODE

Diagram 13

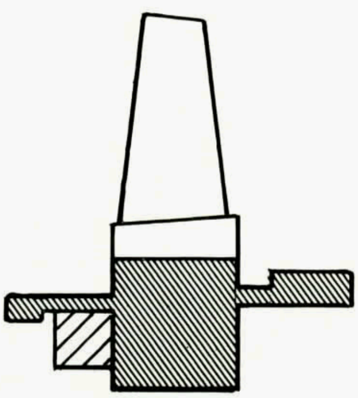


KEYHOLE DAMPER BETWEEN BLADES

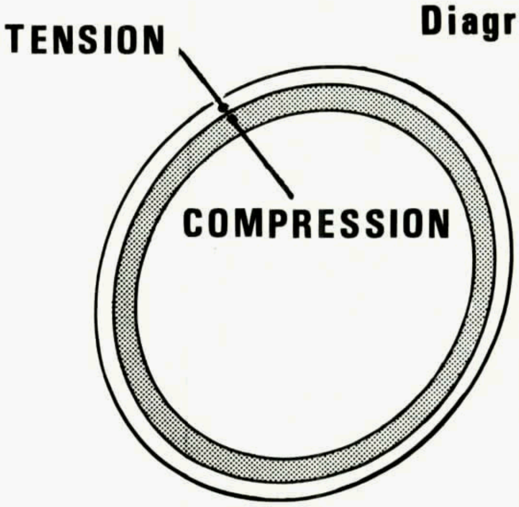


RELATIVE MOVEMENT IN FIRST FLAP

Diagram 14



TRAM WHEEL DAMPER UNDER DISC RING



DAMPING IN A BELL MODE

Diagram 15

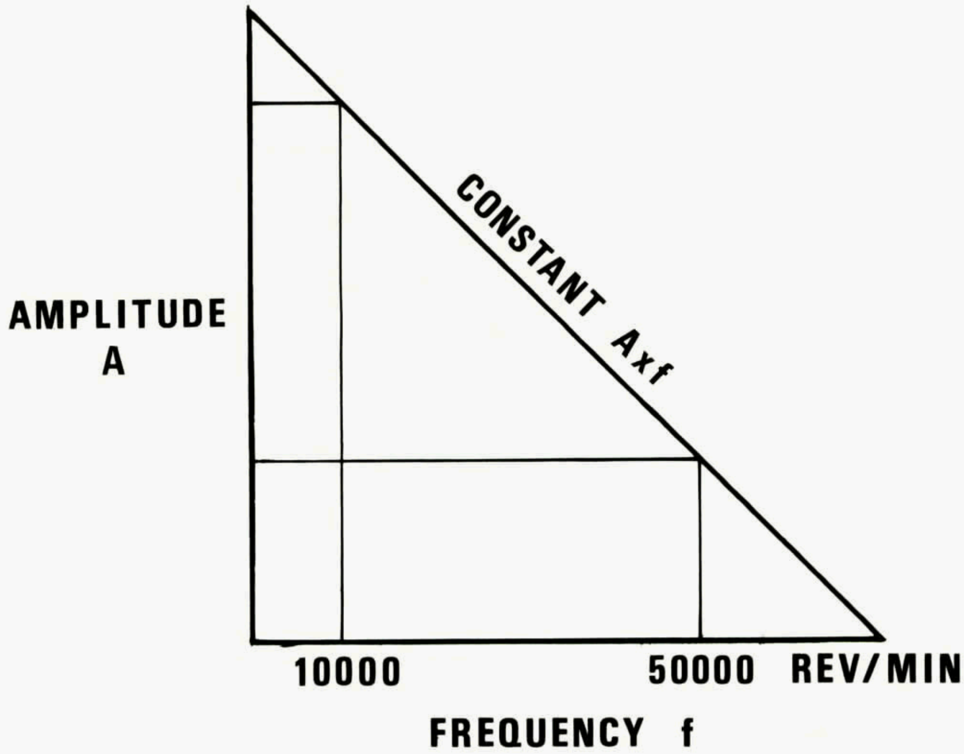
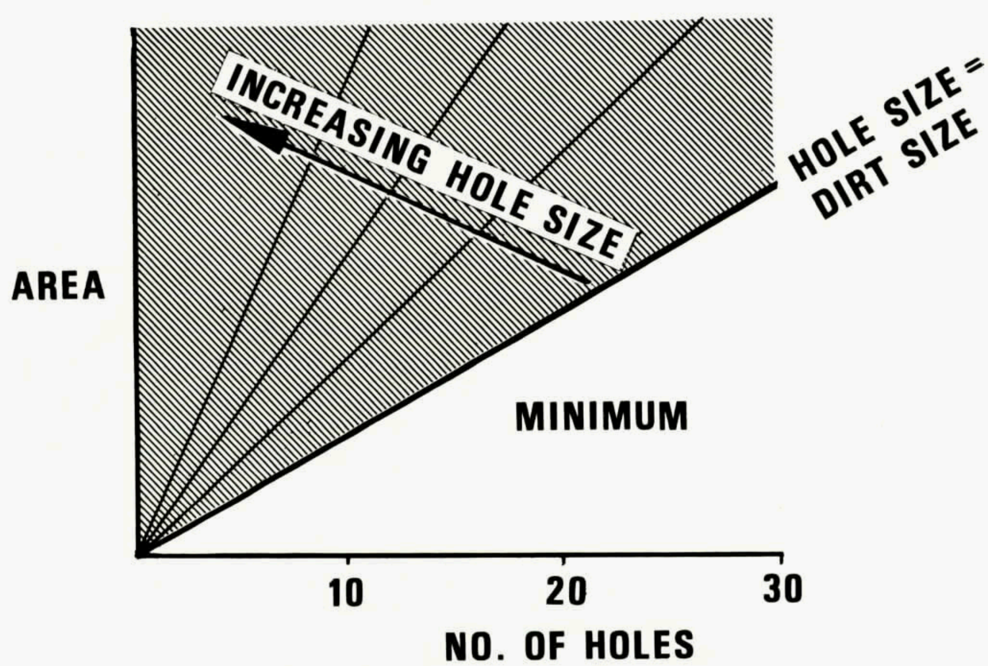


Diagram 16



SPRAY BURNER FEED LIMITATIONS

Diagram 17

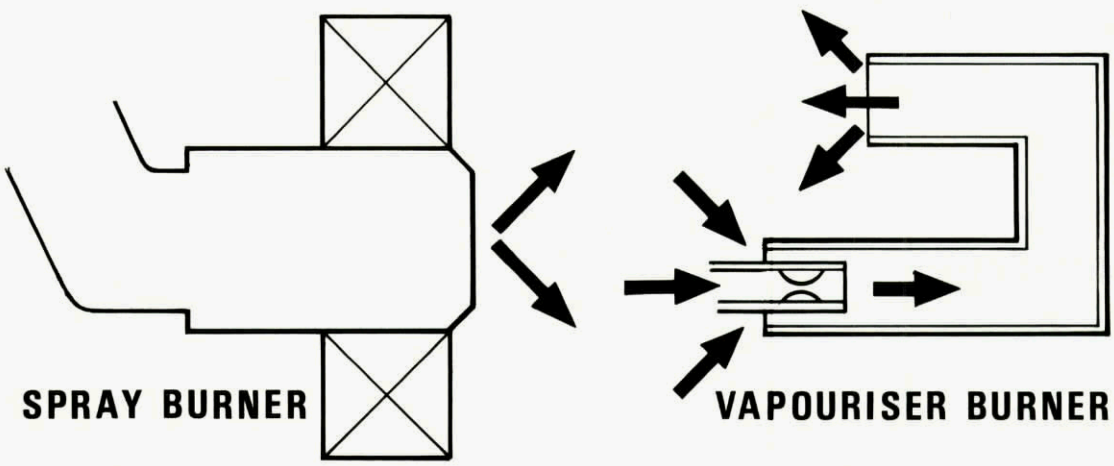


Diagram 18

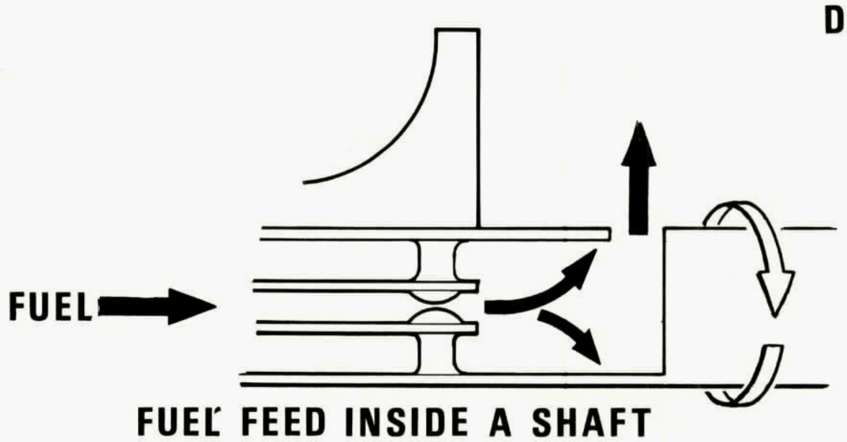
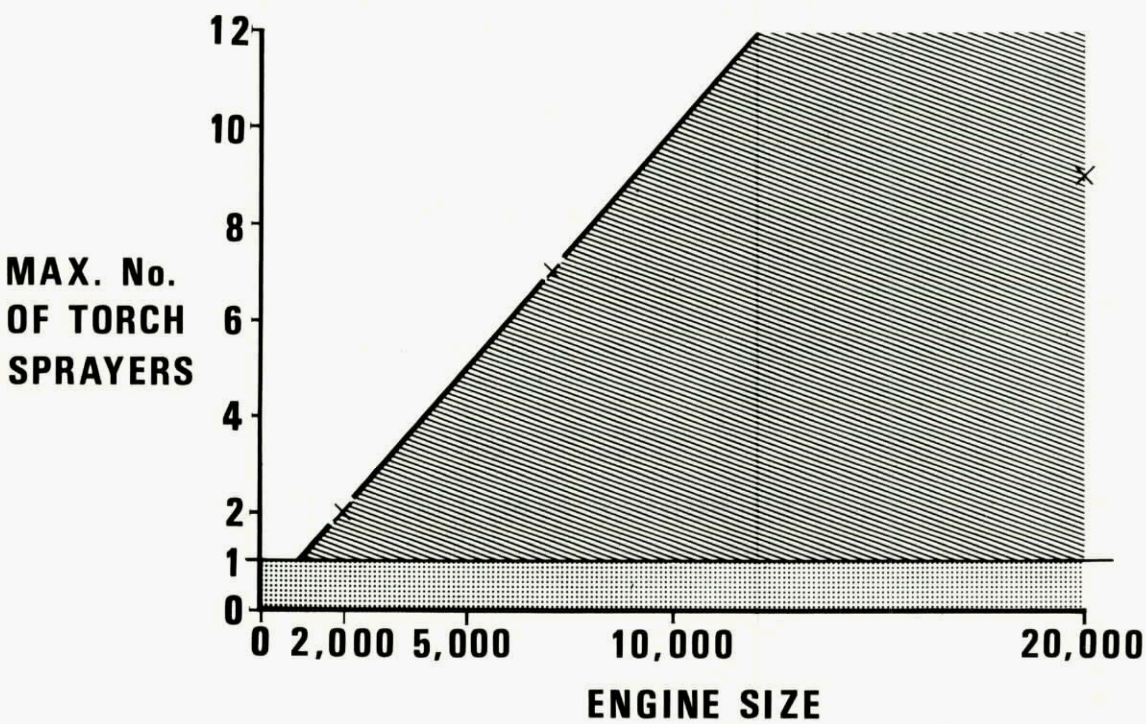


Diagram 19



LIMITING NUMBER OF TORCH SPRAYERS

Diagram 20

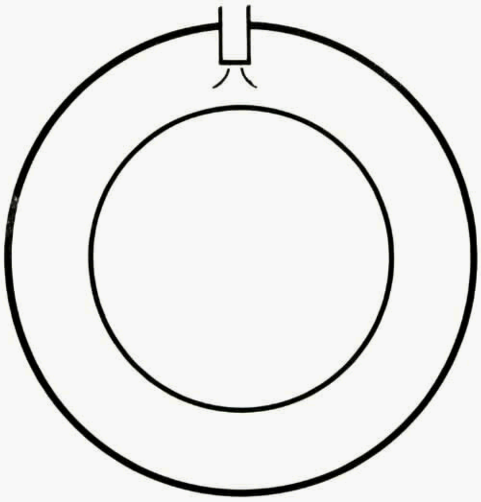
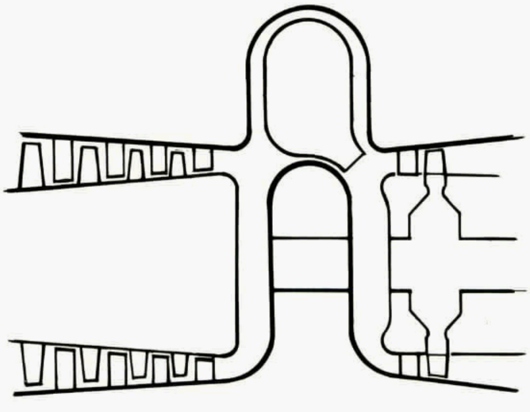
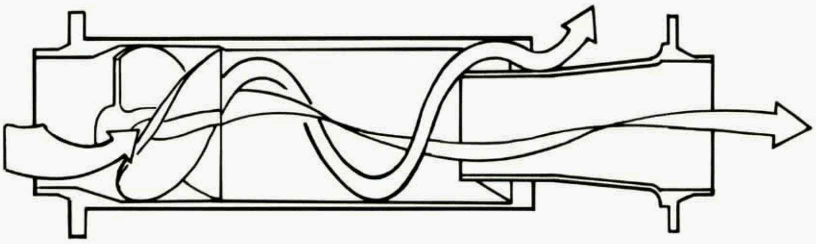


Diagram 21



SINGLE COMBUSTION CHAMBER

Diagram 22



DONALDSON TUBE

APPLICATION TO POWER GENERATION

André L. JAUMOTTE, Recteur de l'Université de Bruxelles

Sommaire : L'article présente un aperçu des applications des turbines à gaz de faible puissance (inférieure à 500 kW) tant dans les domaines aéronautique qu'industriel et spatial. Le premier chapitre envisage les avantages et inconvénients des turbines à gaz par rapport au moteur Diesel.

Le deuxième chapitre traite de l'utilisation des petites turbines pour la production combinée d'énergie électrique et d'énergie thermique; outre les caractéristiques thermodynamiques, quelques exemples d'applications industrielles illustrent le système à énergie totale.

Le troisième chapitre enfin donne un aperçu des possibilités offertes par l'emploi des turbines à gaz dans la recherche spatiale, notamment en ce qui concerne la production de l'énergie nécessaire à bord des engins d'exploration.

1. La production de puissance auxiliaire et de secours

Les turbines à gaz de faible puissance sont de plus en plus utilisées comme sources d'énergie auxiliaire ou de secours mais également comme source d'énergie de base dans de nombreuses applications où elles remplacent avantageusement les moteurs Diesel ou même les moteurs électriques. Le premier domaine d'application est évidemment l'aéronautique :
- démarreurs embarqués pour les réacteurs, permettant aux avions d'être indépendants de la puissance disponible au sol, ce qui se justifie par le nombre croissant d'aéroports visités;
- alimentation électrique et conditionnement d'air des avions, soit uniquement au sol, soit continuellement;
- alimentation en huile des commandes permettant d'effectuer les vérifications avant vol des servo-commandes; d'assister le pilotage et d'assurer la descente de l'avion en cas de panne des réacteurs.

Un avantage particulier des A.P.U* embarqués est leur niveau sonore nettement plus faible que les groupes de puissance au sol utilisés habituellement, et cela tant pour le personnel au sol que pour les passagers de l'avion (7).

A titre d'exemple pour le Lockheed Tristar, l'A.P.U. amorce avant chaque décollage le démarrage des moteurs et fonctionne en permanence pendant chaque vol; l'alternateur de secours qu'il entraîne est couplé avec les trois alternateurs montés sur chacun des moteurs principaux (R B 211-22).

Parmi les autres domaines d'application, on trouve :
- propulsion de camions, engins de travaux publics, voitures, trains, navires, aéroglisseurs;
- groupe de production d'énergie électrique de campagne, en général transportable par un ou deux hommes, utilisé pour les hopitaux mobiles, les télécommunications et les commandes de tir;
- générateur électrique de précision assurant une grande fiabilité et stabilité, utilisé pour alimenter les ordinateurs de la troisième génération, le groupe assure également le conditionnement d'air de l'installation (2);
- entraînement de pompe de chargement de réserve pour les navires pétroliers, justifié par un faible taux d'utilisation;
- production de force motrice pour les cabines de téléphériques automobiles utilisées pour la construction de digues dans le cadre du plan Delta (Hollande) (3).

Il existe également des projets de turbines à gaz en cycle fermé couplées à un réacteur nucléaire constituant des groupes électriques autonomes (4).

Les principaux avantages de la turbine à gaz qui justifient ces utilisations sont :
- sa masse spécifique et son encombrement nettement inférieurs à ceux des moteurs Diesel (cf. fig. 1) ce qui permet une grande mobilité ou des frais d'installation très réduits;
- ses possibilités d'adaptation à des applications très diverses :
  - production de puissance à l'arbre, à vitesse variable ou constante;
  - production d'air sous pression;
  - production d'air et de gaz chauds (conditionnement d'air);
- l'utilisation d'une gamme de carburants plus large que les moteurs Diesel correspondants (gaz naturel, butane, propane, Diesel oil) et possibilité de changer de carburant en fonctionnement en cas de nécessité, pour autant que l'injection ait été prévue pour la substitution d'un combustible à l'autre;
- une durée de vie importante combinée avec une simplicité de fonctionnement et des frais d'entretien réduits;
- l'absence de circuit de refroidissement ce qui réduit les installations auxiliaires;
- le niveau de vibration faible

Les caractéristiques de la turbine qui lui sont moins favorables sont :
- sa consommation spécifique supérieure à celle des moteurs à piston pour les faibles puissances (cf. fig. 2) mais qui tend à s'en rapprocher (6);
- sa sensibilité à la température ambiante qui réduit sa puissance et son rendement lorsque la température augmente;
- son niveau sonore assez élevé et qui actuellement ne peut être réduit que par des silencieux à l'aspiration et à l'échappement avec réduction du rendement;
- son prix d'achat encore relativement élevé mais qui diminue progressivement grâce à l'augmentation de la production;
- sa vitesse de rotation élevée qui nécessite souvent un réducteur de vitesse pour l'adapter aux machines réceptrices existantes. Le développement des convertisseurs de fréquence statique permet d'envi-

* Auxiliary Power Unit

sager de produire de l'électricité à fréquence élevée et donc de coupler l'alternateur directement à la turbine en supprimant le réducteur de vitesse.

On trouvera au tableau 1 les valeurs caractéristiques pour quelques machines.

La turbine à gaz, malgré son rendement inférieur, présente des avantages, qui finalement la rendent compétitive vis-à-vis du moteur Diesel dans de nombreuses applications et même dans certains cas vis-à-vis du moteur électrique.

L'augmentation de la production et le développement de machines réceptrices mieux adaptées élargissent encore ses possibilités.

TABLEAU 1

Nota : $1 \frac{\text{kJ comb}}{\text{kJ ut}}$ = 3600 kJ comb/kWh = 860 kcal/kWh = 2546 B.T.U/H.P.h

Exemple (N = vitesse nominale en tr/min)	Puissance nominale [kW]	Consommation spécifique $\left[\frac{\text{kJ combustible}}{\text{kJ utiles}}\right]$	Masse spécifique [kg machine/kW nominaux]
Microturbo - Noelle 002 groupe de démarrage embarqué N = 50.000	59 à 1800 tr/min du démarreur	13	0,407 sans démarreur 0,583 avec démarreur
- Emeraude . démarrage au sol du réacteur de l'avion Concorde . entraînement du relais d'accessoires soit au sol soit en vol lorsque le moteur principal est en panne . alimentation en huile de la commande d'embrayage N = 47.000	125	7,6	0,455 avec démarreur
- Athos IV . démarrage des réacteurs . applications industrielles (puissance à l'arbre ou air comprimé) N = 53.000	74 à 2600 tr/min du démarreur	20,6	0,414 sans démarreur 0,606 avec démarreur
- Jaguar . démarrage d'avion par air comprimé (plusieurs démarreurs alimentés par un turbo générateur) N = 54.000	38,3 à 6000 tr/min du démarreur	16,2	0,888 sans démarreur 1,2 avec démarreur
- Cyclone . groupe de servitude au sol pour démarrage électrique des réacteurs N = 50.000	9	51	5,1 sans accessoires ni chariot
- Gevaudan . générateur électrique embarqué . fourniture éventuelle d'air comprimé . autre formule : démarrage du Rolls-Royce Vipertt et générateur de courant continu N = 50.000	30	19,8	1,2 avec génératrice
- Sermel - Turbomoteur T M S 60 N = 45.000	184	8	0,19 turbine seule 0,3 avec réducteur de vitesse 45.000 - 8000 tr/min
- Turbomoteur T M S 30 N = 55.000	110	8,55	0,27 turbine seule 0,435 avec réducteur de vitesse 55.000 - 8000 tr/min
- "Titan" Solar . hélicoptères individuels . groupe électrogène	44	9	0,570 avec réducteur de vitesse
- Motoren and Turbinen Union, München GmbH M T U 6022 - A S	184	5,58	0,53 turbine seule

2. Concept d'Energie Totale

2.1. Introduction

Nous limitant aux tubines de faible puissance, nous traitons le cas où le combustible est le gaz naturel. La turbine à gaz peut fonctionner au fuel oil lourd mais à des conditions restrictives :
- soit ne pas dépasser une température maximale de 650°C;
- soit utiliser des fuel oils lourds répondant à des conditions limites quant au contenu des cendres en certains éléments comme le vanadium, éventuellement avec des additifs.

Dans les deux cas, le fuel oil doit être réchauffé jusqu'à la température qui lui assure la viscosité la plus appropriée à l'injection et la turbine doit être périodiquement nettoyée. Pendant le fonctionnement, la puissance et le rendement diminuent. L'ensemble de ces conditions restrictives est acceptable dans certains cas particuliers; il ne l'est pas pour des installations industrielles de puissance modeste.

Celles-ci deviennent compétitives dans la mesure où le combustible Diesel Oil ou gaz naturel peut leur être fourni à un prix qui n'est pas trop éloigné de celui de la calorie fuel oil lourd. Cette circonstance se présente parfois pour la calorie gaz naturel.

2.2. Production simultanée d'énergie et de chaleur

2.2.1. Fonctionnement au point nominal

L'utilisation considérée a été caractérisée par le rapport entre la puissance à l'arbre produite et la puissance calorifique utile soit P_a/P_t.

Pour représenter les aspects thermiques de la production combinée d'énergie et de chaleur, on a employé:
- le taux d'utilisation de l'énergie du combustible τ, définie comme le rapport entre les sommes d'énergie mécanique et thermique utiles, et l'énergie contenue dans le combustible :

$$\tau = \frac{P_a + P_t}{P_c}$$

- la consommation d'énergie calorifique par unité d'énergie mécanique produite x .
  La consommation de chaleur affectée à la production d'énergie mécanique est définie par la différence entre la consommation totale d'énergie venant du combustible P_c et la consommation qui serait requise pour produire la même puissance calorifique utile P_t dans une chaudière normale, soit P_{co}.
  On a donc $x = \frac{P_c - P_{co}}{P_a}$

A titre d'exemple, nous prendrons la centrale de l'Usine Turboméca de Bordes (Basses Pyrénées - France) avec trois turbines Turboméca (ASTAGAZ 3, ASTAGAZ 12 et BASTANGAZ 6) alimentées au gaz naturel (fig. 3). Les caractéristiques de ces turbines au point de fonctionnement nominal sont les suivantes :

TABLEAU 2

	ASTAGAZ 3	ASTAGAZ 12	BASTANGAZ 6
P_a (kW)	350	400	550
Consommation spécifique (kcal/kWh)	4200	3830	4350
Débit d'air (kg/s)	2,55	2,87	4,69
Température d'échappement T_4 (°C)	450	450	425

Nous effectuons le calcul de P_t dans quelques hypothèses de températures à la sortie de l'échangeur (T_5) : 240°C, 180°C, 150°C et 120°C. On a :

$$P_t = \dot{m}_{air}\,(i_4 - i_5) = \dot{m}_{air}\,c_{p\,air}\,(T_4 - T_5)$$

$$= \dot{m}_{eau}\,(i_7 - i_6) = \dot{m}_{eau}\,c_{p\,eau}\,(T_7 - T_6)$$

Nous admettrons que l'eau entre dans l'échangeur à 78°C et le quitte à 115°C. Le débit d'eau est déduit de l'équation ci-dessus.

Pour le calcul de P_{co}, on a admis un rendement de chaudière de 0,8. On a donc : $P_{co} = \frac{P_t}{0,8}$
Le tableau suivant résume les résultats du calcul :

TABLEAU 3

	T_5	120°	150°	180°	240°
P_t (kW)	AST 3	880	815	700	565
	AST 12	990	915	785	635
	BAST 6	1510	1370	1160	930
$\dot{m}$ (m³/h)	AST 3	21,5	19,9	17,1	13,8
	AST 12	24,2	22,3	19,2	15,3
	BAST 6	37	33,6	28,4	22,8

T A B L E A U 3 (suite)

	T_5	120°	150°	180°	240°
P_a/P_t (kWh/m^3)	AST 3 AST 12 BAST 6	16,3 16,5 14,9	17,6 17,9 16,4	20,5 20,8 19,4	25,4 25,8 24,1
τ (%)	AST 3 AST 12 BAST 6	71 78 74	68 73 69	61 66 61	53 58 53
x	AST 3 AST 12 BAST 6	1,74 1,34 1,63	2,08 1,60 1,94	2,38 2,00 2,41	2,89 2,47 2,93

Il est illustré par la figure 4 où l'on a porté P_t en fonction de la température à la cheminée, la figure 5 où l'on a porté le débit d'eau $\dot{m}$ en fonction de cette même température.

Les figures 6 et 7 donnent respectivement le taux d'utilisation de l'énergie du combustible et la consommation d'énergie calorifique par unité d'énergie mécanique en fonction du rapport P_a/P_t. Ces deux figures font clairement apparaître l'intérêt de la production simultanée d'énergie mécanique et de chaleur par turbine à gaz de petite puissance.

2.2.2. Fonctionnement en charge partielle

On a traité le cas d'une installation avec chaudière de récupération échappant à 150°C, grâce aux caractéristiques données par le constructeur (Turboméca - Bastangaz 6). Le débit de gaz est quasi constant, mais la température d'échappement diminue fortement. La figure 8 donne le rapport de la chaleur récupérée à la chaleur récupérée au point de fonctionnement nominal P_t/P_{tN}, ainsi que le taux d'utilisation de l'énergie τ, en fonction du rapport de la puissance mécanique à la puissance mécanique au point nominal P_a/P_{aN}. On constate que le rapport de la chaleur récupérée diminue proportionnellement au rapport de la puissance à l'arbre produite.

2.3. Production simultanée d'électricité et de froid

Nous envisagerons le cas où la chaleur contenue dans les gaz d'échappement sert de source de chauffage au bouilleur d'une machine frigorifique à absorption. Le coefficient d'effet frigorifique pratique est le rapport entre la quantité de froid recueillie à l'évaporateur et la quantité de chaleur fournie pour chauffer le bouilleur; dans les conditions réelles $\varepsilon = Q_{EV}/Q_B$. Le constructeur américain York Corporation donne comme valeur $\varepsilon = 0,55$ pour une machine à absorption d'ammoniac dont la température de source chaude est 115°C (Bouilleur), la température intermédiaire 30°C (Condenseur) et la source froide 0°C. (Evaporateur). En figure 9 on a porté la quantité de froid produite P_o (en fg/h) en fonction de la température à la cheminée pour les trois mêmes turbines.

2.4. Production simultanée d'électricité, de chaleur et de froid

Prenons comme exemple l'usine de l'American Meter's Fullerton (Californie). Quatre turbines produisent l'électricité requise, la chaleur des gaz brûlés est récupérée pour alimenter les systèmes de conditionnement d'air et de chauffage.

Deux des quatre turbines Solar utilisent un cycle régénératif (préchauffage de l'air dans un récupérateur de chaleur); ce qui réduit de 12 % leur consommation en gaz naturel. L'ensemble de l'équipement électrique produit au maximum 728 kW. De jour, trois turbines fonctionnent normalement et produisent 500 kW; de nuit, une seule turbine fournit les 100 kW nécessaires.

Le système à énergie totale doit remplir quatre conditions :
- le système électrique doit produire 700 kW à 60 cs et sous 12 kV pour être compatible avec les installations existant préalablement;
- le conditionnement d'air de l'usine doit pouvoir maintenir la température à 2° près dans 35.000 m^3 de locaux. Une capacité de 270 tonnes d'eau froide à 7°C est suffisante pour les besoins de l'usine, y compris le prérefroidissement de l'air à l'entrée du compresseur;
- il faut produire de l'eau chaude pour chauffer les locaux; l'échangeur la fournit à 60°C;
- il est nécessaire de prévoir pour les développements futurs une capacité supplémentaire en eau chaude et en eau froide.

Pour satisfaire ces quatre conditions, l'installation à énergie totale est divisée en six groupes principaux (figure 10) :
1) génération de puissance : quatre turbines Solar T-350 entraînent des alternateurs. Le diagramme de charge électrique détermine le nombre de turbines en fonctionnement. Les alternateurs sont triphasés 480 V - 60 cs.
2) distribution de l'énergie électrique :tous les alternateurs sont connectés à un seul transformateur élévateur de tension 480 V/12 kV.
3) récupérateur de chaleur : tous les gaz brûlés sont conduits vers un échangeur Besler, avec possibilité de by-pass, et enfin s'échappent à l'atmosphère en traversant un silencieux. L'ordinateur règle les débits de chaudière et de by-pass en fonction de la charge thermique. L'eau chaude est stockée dans un réservoir sous pression et alimente d'une part les locaux, d'autre part la machine à absorption, l'ordinateur réglant l'ensemble au moyen de vannes et de thermostats.
4) machine frigorifique : un modèle à absorption York E-28 est associé à une tour de refroidissement qui rejette à l'atmosphère la chaleur d'absorption et de condensation. L'eau froide alimente six unités de conditionnement d'air, deux prérefroidisseurs de l'air à l'entrée des compresseurs et les système de refroidissement de toutes les pompes à eau chaude.
5) système de chauffage et de refroidissement de l'usine : six unités de conditionnement d'air fournissent l'air frais aux locaux. Le chauffage est produit par l'eau chaude provenant du récupérateur

ainsi que par des brûleurs à gaz auxiliaires.

6°) l'ordinateur: avec ses circuits de mesure, de contrôle et de protection, il assure automatiquement les séquences de synchronisation et de mise en parallèle des alternateurs, la bonne marche des chaudières, de la machine frigorifique, de la tour de refroidissement, des diverses pompes à eau chaude, à eau froide, à condensat, ainsi que des circuits de relais.

Les caractéristiques de la turbine Solar T-350 au point de fonctionnement nominal sont les suivantes :

TABLEAU 4

P_a = 190 kW
P_c = 1350 kW
Taux de compression = 3,8
Débit d'air = 1,82 kg/s
Température d'échappement = 540°C

Les caractéristiques thermiques de l'installation combinés sont les suivantes (pour une turbine):

P_t = 780 kW et P_{co} = 980 kW

On en déduit $\frac{P_a}{P_t}$ = 0,244

τ = 72 %

x = 1,95

3. Applications spatiales

3.1. Sources de puissance auxiliaire pour cabines spatiales

3.1.1. Introduction

Dans ce chapitre, nous nous limiterons à la conversion thermodynamique de l'énergie par turbomachines. D'autres systèmes sont envisageables, mais :

- l'énergie chimique impose immédiatement d'emporter des masses prohibitives;
- la conversion thermoionique ou thermoélectrique nous donne du courant continu qui doit être converti en courant alternatif. Ceci signifie un poids supplémentaire et une diminution du rendement.

En fait, la conversion thermodynamique par turbomachines semble être le seul bon système dès que l'on a besoin de plus de quelques kW . Ce système de conversion pose deux problèmes majeurs : le choix du cycle de travail, et celui de la source de puissance.

3.1.2. Cycle de travail

Nous nous trouvons de prime abord devant deux système différents : le cycle de Rankine et le cycle de Brayton. Dans l'espace, le seul système de réjection de la chaleur est de disposer d'une grande surface travaillant par rayonnement. Plus la température à laquelle on rejette la chaleur non convertie est élevée, plus petit pourra être le radiateur utilisé, pour pourvoir au même transfert de chaleur. Le cycle de Rankine apparaît ainsi à première vue comme le plus intéressant, car il permet un rejet de la chaleur à une température plus élevée que ne le permet le cycle de Brayton.

Cependant, une analyde détaillée nous montre différents avantages du cycle de Brayton [12], [15]. Nous citerons pour exemples :

- le fluide de travail est un gaz inerte et ne nous fait pas rencontrer tous les problèmes liés, dans le cycle de Rankine, à l'emploi des métaux liquides;
- le choix dès pressions peut être fait indépendemment de celui des températures;
- de considérables progrès ont été réalisés sur les turbomachines d'aviation, où les machines composantes sont très semblables à celles utilisées dans le cycle de Brayton. Ceci a conduit à une nette amélioration du rendement et à une haute fiabilité;
- les progrès technologiques incessants autorisent à élever continuellement la température maximum du cycle, ce qui augmente le rendement;
- des progrès dans le dessin des compresseurs, des turbines et des échangeurs de chaleur conduisent aussi à des améliorations de rendement;
- le choix judicieux du fluidè de travail peut améliorer lui aussi l'efficacité globale de la conversion. A ce point de vue, l'Hélium semble être le gaz le plus favorable bien que d'excellents résultats aient été obtenus avec de l'Argon ou avec des mélanges Hélium-Xénon.

3.1.3. Sources de puissance

Généralement, lorsque l'on a à étudier un nouveau système d'apport de puissance auxiliaire pour une cabine spatiale, trois buts principaux peuvent être visés :

- la recherche du poids minimum;
- la recherche du rendement maximum;
- la recherche d'une aire de rejet de la chaleur minimum.

Ces facteurs sont mutuellement exclusifs. Le premier est certainement le plus important mais le troisième l'est également, surtout en ce qui concerne l'apport de puissance aux unités de petite taille.

3.1.3.1. Energie solaire (13)

L'énergie solaire n'est pas très dense (140 W/cm^2 au voisinage de la Terre) et ne peut pourvoir aux besoins de puissance d'une longue exploration spatiale loin du Soleil. D'autre part, les missions circomterrestres imposent un passage cyclique dans l'ombre de la Terre, ce qui nécessite d'embarquer un moyen de stockage de l'énergie. Le poids du système s'en troove considérablement augmenté. D'un autre point de vue, la puissance électrique maximum disponible à la sortie d'un système de conversion utilisant l'énergie solaire comme source de puissance est limitée par l'aire des collecteurs de cette énergie. Pour atteindre un rapport poids/puissance acceptable, ces collecteurs requièrent des miroirs de concentration. Ceci conduit à des problèmes technologiques supplémentaires. De plus, les cellules collectrices se dégradent avec le temps, à cause de l'espace environnant.

Nous voyons donc que l'énergie solaire ne nous procurera la puissance auxiliaire nécessaire que dans une gamme limitée à environ 10 kWe , et sera essentiellement valable pour les missions de courte durée.

3.1.3.2. Energie nucléaire [13], [17]

Deux possibilités se présentent :
- l'énergie nucléaire sous forme isotopique qui ne pourra suppléer qu'aux très faibles besoins de puissance;
- le réacteur nucléaire qui ne connaît pratiquement pas de limite en ce qui concerne la puissance disponible. Le poids spécifique d'un tel système de conversion est approximativement indépendant de la puissance. Le seul problème lié à l'emploi d'un réacteur nucléaire est la nécessité de blindages appropriés destinés à protéger des radiations dangereuses, mais qui augmentent le poids de l'ensemble.

3.1.3.3. Conclusion

Chaque système est bien adapté à une gamme de puissance, comme le montre la figure 11.

3.2. Quelques réalisations à l'étude

Nous examinerons deux projets, le premier utilisant l'énergie solaire comme source de puissance, le second utilisant un réacteur nucléaire.

3.2.1. Utilisation de l'énergie solaire comme source de puissance [18]

Le système que nous présentons a été étudié depuis 1963 au Lewis Research Center de la NASA, pour pourvoir aux besoins en énergie électrique d'une station spatiale habitée, en orbite terrestre. Cet engin devrait être équipé de trois modules du type de celui dont nous parlons. La figure 12 nous donne un aperçu schématique du système de conversion.

L'Argon, avec une masse moléculaire de 40, a été choisi comme fluide de travail. L'aire du collecteur d'énergie solaire est environ 76 m^2. Le poids total du système s'établit à 790 kg.

Pour atteindre ces performances, la température d'entrée de turbine a été fixé à 1090°K. La sélection de cette température maximum a été faite en tenant compte des caractéristiques du matériau choisi pour le stockage de l'énergie thermique. Ce dernier choix s'est effectué de manière à trouver le meilleur compromis possible entre un rendement élevé et les exigences de résistance du matériau. L'utilisation du fluorure de lithium, dont le point de fusion se situe aux alentours de 1100°C, a été décidée. Avec ces réglages, le rendement de la conversion se situe à 0.25.

3.2.2. Utilisation d'un réacteur nucléaire comme source de puissance [16]

Ce système est à l'étude chez AiResearch, sous un contrat de la North American Rockwell Corporation. Son but est d'apporter la puissance auxiliaire nécessaire à un grand laboratoire spatial habité. Les contraintes imposées étaient les suivantes :
- puissance : 100 kWe
- source de chaleur : réacteur nucléaire à hydrure de Zirconium, refroidi au NaK
- surface maximum disponible pour le rejet de la chaleur : 655 m^2
- durée de vie du système : 10 ans.

La configuration du système de conversion est un peu compliquée : on dispose de deux réacteurs, chacun d'eux étant couplé à quatre modules de conversion indépendants. Chaque module se compose d'un compresseur, d'une turbine et d'un alternateur, et est capable de fournir 25 kWe . Cette disposition permet d'obtenir 200 kWe si le besoin s'en fait sentir, et présente de plus une grande souplesse et une haute fiabilité.

Le fluide de travail utilisé dans les turbomachines de conversion est un mélange de xénon et d'hélium de masse moléculaire 40. Le débit massique est de 1.60 kg/s. La température de sortie du réacteur -donc la température maximum de l'installation- est environ 865°K. D'autres essais ont été faits à des températures différentes (920°K et 976°K), mais le rendement de la conversion est apparu constant, égal à 0.22. On pourrait penser dès lors qu'il vaudrait mieux sélectionner la température la plus haute possible dans le but de diminuer l'aire du radiateur. Mais il faut encore tenir compte du dernier critère : la durée de vie requise de 10 ans. Différentes investigations sur le réacteur ont prouvé que 865°K n'étaient pas loin d'être la température maximum compatible avec cette exigence.

3.3. Perspectives d'avenir

Les plus importantes possibilités de développement sont tributaires des nouvelles solutions technologiques que nous découvrirons. Diverses expériences sont en cours dans différents domaines, tels :
- l'utilisation de vitesses de rotation élevées;
- l'utilisation des paliers à gaz;
- la réduction des importantes pertes secondaires qui se manifestent surtout à cause des dimensions très réduites des machines.

Mais un problème peut être considéré comme le point le plus important des investigations actuelles : l'élévation de la température maximum du cycle. De nos jours, ce problème a son importance dans toute turbine à gaz car il a une influence directe sur le rendement, mais il apparaît comme plus vital encore dans les applications spatiales où une température élevée permet en outre de réduire l'aire de réjection de la chaleur non convertie.

Deux nouvelles solutions ont été proposées :
- la première consiste à utiliser certains métaux à la place des superalliages traditionnels. Nous pensons par exemple au Molybdène, dont la température de fusion est de 2620°C. L'emploi de pareilles substances est permis par le fait que nous travaillons en atmosphère réductrice, de par l'usage même des fluides de travail sélectionnés (hélium, gaz inertes). Mais un problème majeur s'oppose à cette solution séduisante : les difficultés rencontrées lors de l'usinage et de la mise en oeuvre de tels métaux.
- la seconde solution préfère se tourner vers l'emploi de céramiques pour remplacer les superalliages [14]. Au point de vue résistance mécanique, l'usage de ces matériaux pose trois gros problèmes :
  - la charge mécanique statique dont les effets peuvent être réduits par un dessin adéquat, de façon à faire travailler les parties en céramiques en compression plutôt qu'en traction;

- les chocs mécaniques. Heureusement, les critères de résistance aux impacts pour une petite turbine ne sont pas aussi sévères que pour les gros turbomoteurs d'avions. De plus, dans le cas particulier des sources de puissance auxiliaire pour applications spatiales, le fait de travailler avec des cycles fermés nous prémunit contre l'ingestion de corps étrangers. Le seul danger qui subsiste est le détachement d'une partie abîmée de l'intérieur de la machine. Cette possibilité existe d'ailleurs dans toute turbine.
- en fait, le seul critère sévère à appliquer aux matériaux céramiques est celui de la charge centrifuge dans les rotors. Dans le cas de l'emploi du métal, les risques de ruptures proviennent de la charge cyclique excursionnant sous et au-dessus de la limite élastique, donc par fatigue. Ce n'est pas le cas avec les céramiques qui sont des matériaux fragiles, n'admettant aucune déformation. En conséquence, la contrainte doit toujours s'y trouver sous la limite élastique.

  En ce qui concerne les céramiques, on constate une forte dépendance entre la résistance et la densité, comme il apparaît à la figure 13 où nous avons porté le rapport entre la résistance à la traction et la densité (la résistance spécifique) en fonction de la température, pour des superalliages et pour des matériaux céramiques. Cette figure montre l'avantage que l'on peut retirer de l'emploi de céramiques très denses, surtout aux hautes températures. Ces types de céramiques sont en fait les seules compatibles avec le critère de la charge centrifuge. A l'heure actuelle, elles ne peuvent être obtenues que par compression à chaud, procédé inacceptable d'un point de vue économique.

Une résistance acceptable à l'environnement et aux chocs thermiques peut s'obtenir avec des céramiques du système lithium-alumine-silicate. Ces matériaux peuvent être mis en oeuvre par moulage, procédé relativement bon marché.

Pour conclure, nous dirons que l'emploi des céramiques possède un avenir prometteur, en particulier pour les parties fixes. Mais dès qu'il s'agit des rotors, nous devons nous tourner vers du carbure de silicium ou du nitrure de silicium à haute densité, deux matériaux trop chers à l'heure actuelle pour trouver une application pratique.

Bibliographie - Chapitre 1

[1] P.A. Pitt - "Small Gas Turbines in Industrial Applications" ASME Paper No.66-GT-110 - 10 pages
[2] Robin Mackay - "The Gas Turbines as a Source of Continuous Precise Power" ASME Paper No 69-GT-20 11 pages
[3] J.A.K. Konings & L. Erlebach - "Aircraft-Type Gas Turbines Drive 16 Independent Cableway cars to be Used Under the "Delta Plan" (The Netherlands)" ASME Paper No 70-GT-13 15 pages
[4] John R. Hoffmann & Ernest G. Fehev - "150 kWe Supercritical closed Cycle System" ASME Paper No 70-GT-89 11 pages
[5] H. Lanoy - "Nouvelle petite turbine à gaz de 50 à 80 ch pour hélicoptères individuels, plates-formes volantes, groupes électrogènes, etc" DOCAERO No 64 - Septembre 1960 - pages 27 à 30
[6] J. Chauvin - "Perspectives et Problèmes du développement des Turbines à Gaz de Faible puissance" Revue de la S.R.B.I.I. N° 4 avril 1970 - pages 235 à 249
[7] C.H. Paul - "The Development of Small Gas Turbines for Aircraft Auxiliary Power" The Aeronautical Journal of the Royal Aeronautical Society Vol. 74 October 1970 - pages 797 à 805

Chapitre 2

[8] R. Ecabert - "Installations combinées gaz-vapeur avec production simultanée d'énergie électrique et de vapeur de fabrication" Revue Technique Sulzer 1963, N° 2 - pages 75 à 80
[9] A. Jaumotte et J. Hellemans - "Utilisation de la turbine à gaz alimentée au gaz naturel dans les installations indestrielles de production combinée de vapeur et d'électricité" Réunion de Vienne 1965 de la Commission de la Turbine à gaz de la F I P A C E 11 pages
[10] D. Hale - "Turbine Total Energy system provides total power for heavy manufacturing Plant" America, Gas Journal April 1965 - pages 47 à 54
[11] "1967 Gas Turbine Catalog" - Gas Turbine Publications Inc. - page S-14

Chapitre 3

[12] P. Barthélémy, C. Chauvineau & G. Dievlot - "Thermodynamique du cycle de Brayton pour applications spatiales" papier présenté au Sixième Colloque AGARD, Cannes, 16-20 mars, 1964 - Agardograph 81, pp. 181-205 - Gordon & Breach Science Publishers, London, 1967.
[13] B. Lespinasse - "Les sources de puissance dans l'espace" Conférence faite à la deuxième Ecole d'Eté du CERS-ESRO, Oxford, 1964 - Technologie des véhicules spatiaux, ESRO SP-12, Vol. IV : Sources de puissance dans l'espace, pp. 2-12, décembre 1966.
[14] Mc Lean - "The Application of Ceramics to the Small Gas Turbine" ASME paper No 70-GT-105, 9 pages
[15] J. Mason - "Working gas selection for the Closed Brayton Cycle", paper presented at the Sixth AGARD Colloquium, Cannes, March 16-20, 1964 - Agardograph 81, pp. 223-252 - Gordon & Breach Science Publishers, London, 1967.
[16] E.A. Mock - "Closed Brayton cycle system optimization for undersea, terrestrial and space applications" Von Karman Institute for Fluid Dynamics, Ledtures Series n° 24, pp. 58-72.
[17] F.K. Pittmann and J.D. Lafleur Jr - "Nuclear Energy Source Limitations" paper presented at the Sixth AGARD Colloquium, Cannes, March 16-20, 1964 - Agardograph 81, pp. 5-24, Gordon & Breach Science Publishers, London, 1967.
[18] B.T. Resnick - "The Closed Brayton Cycle for Space Power . an Assessment" ASME paper n° 66-GT/CLC 10, 8 p.

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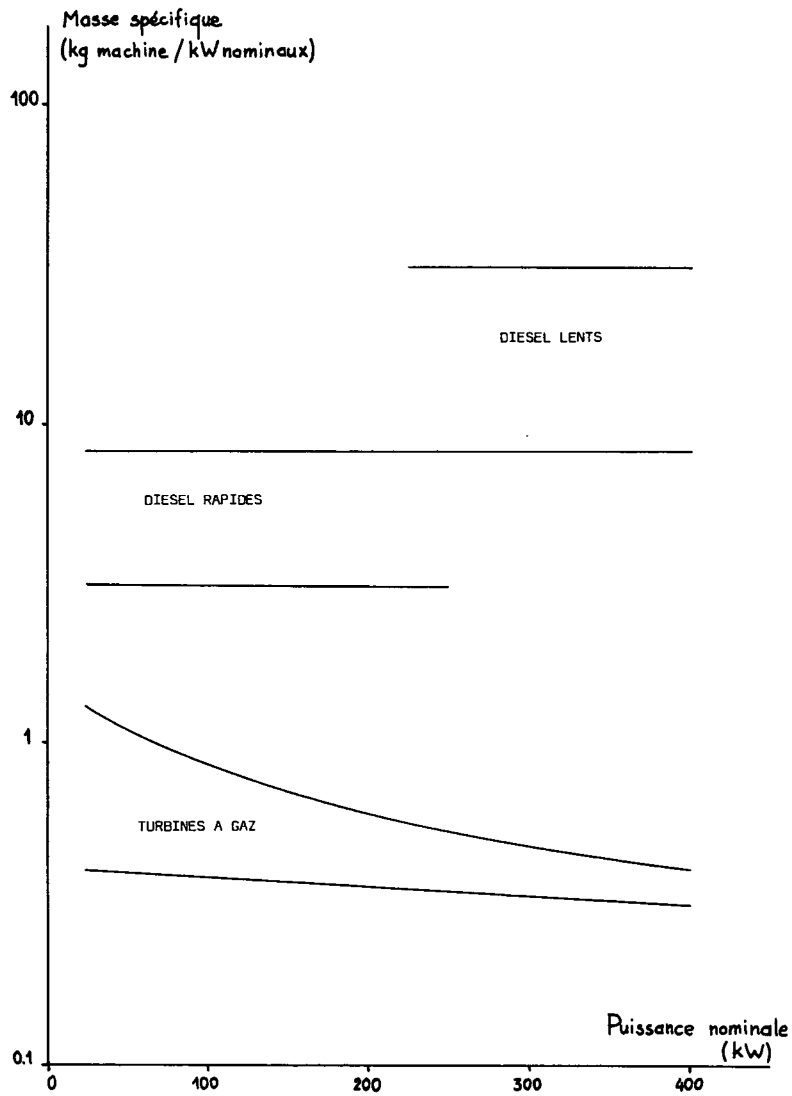

FIG.1 - COMPARAISON DE LA MASSE SPECIFIQUE DES TURBINES A GAZ ET DES MOTEUR DIESEL EN FONCTION DE LA PUISSANCE NOMINALE

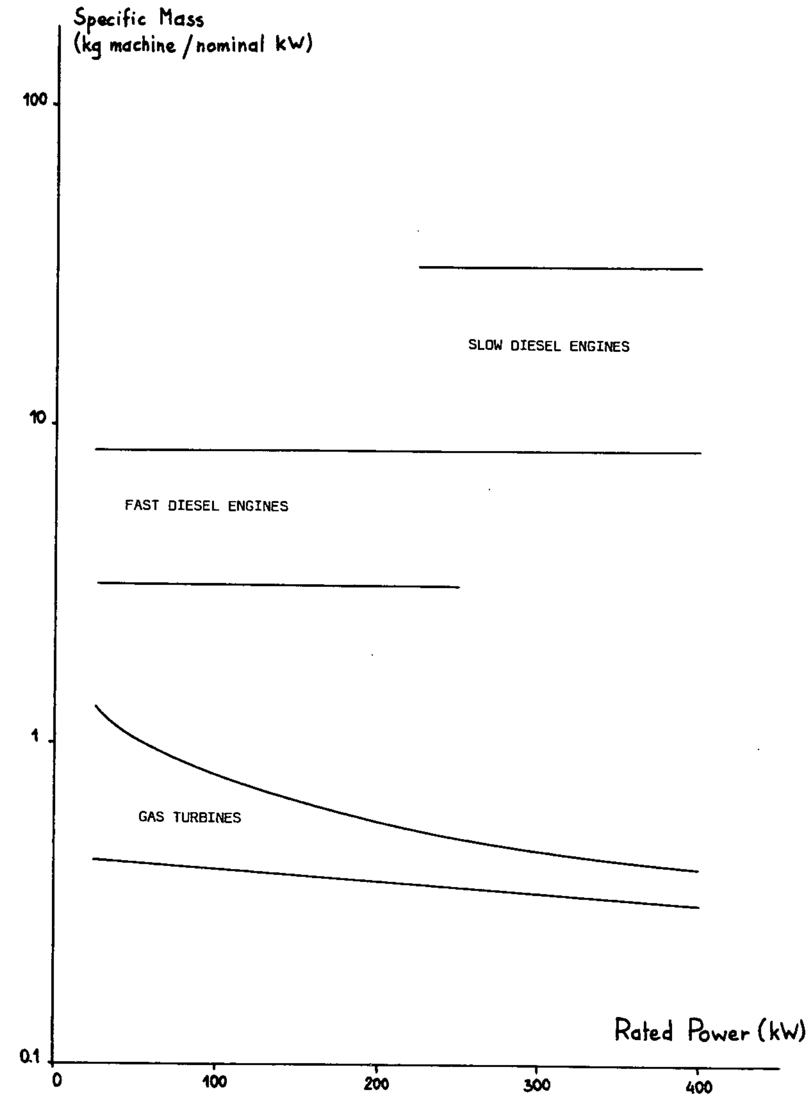

FIG. 1 - COMPARISON OF GAS TURBINE AND DIESEL ENGINES SPECIFIC MASSES AS A FUNCTION OF RATED POWER

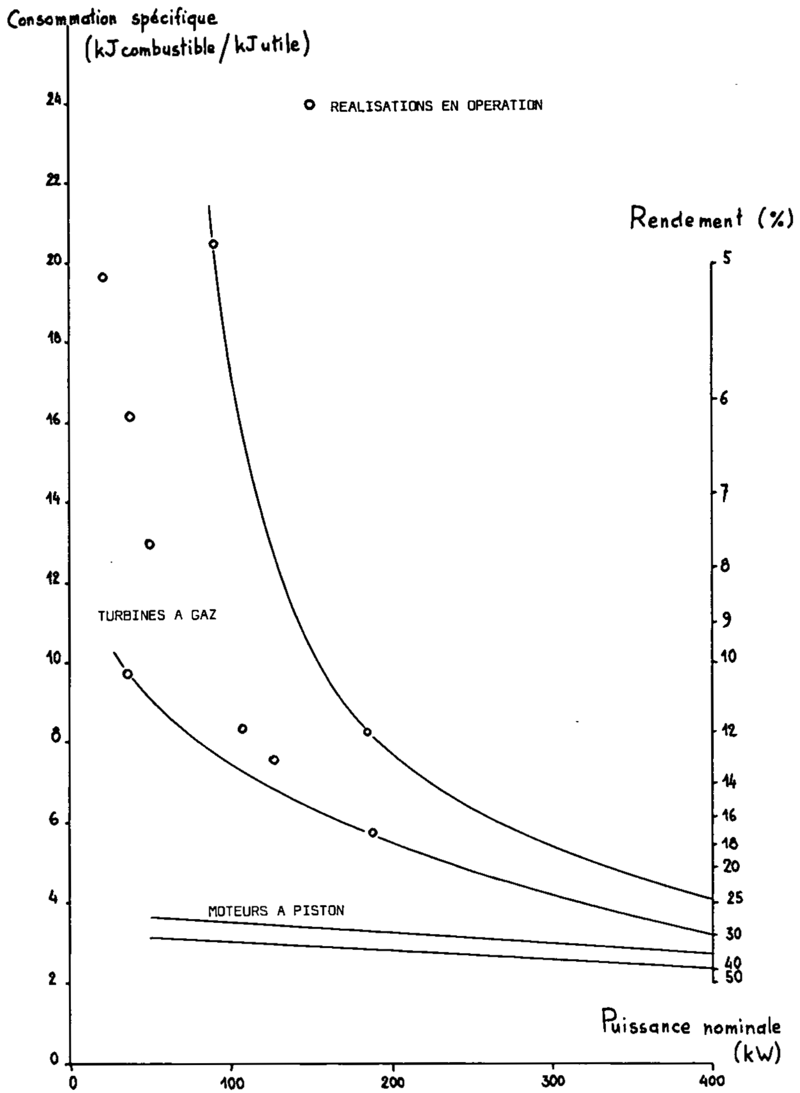

FIG. 2 - COMPARAISON DE LA CONSOMMATION SPECIFIQUE DES PETITES TURBINES A GAZ ET DES MOTEURS A PISTON EN FONCTION DE LA PUISSANCE NOMINALE

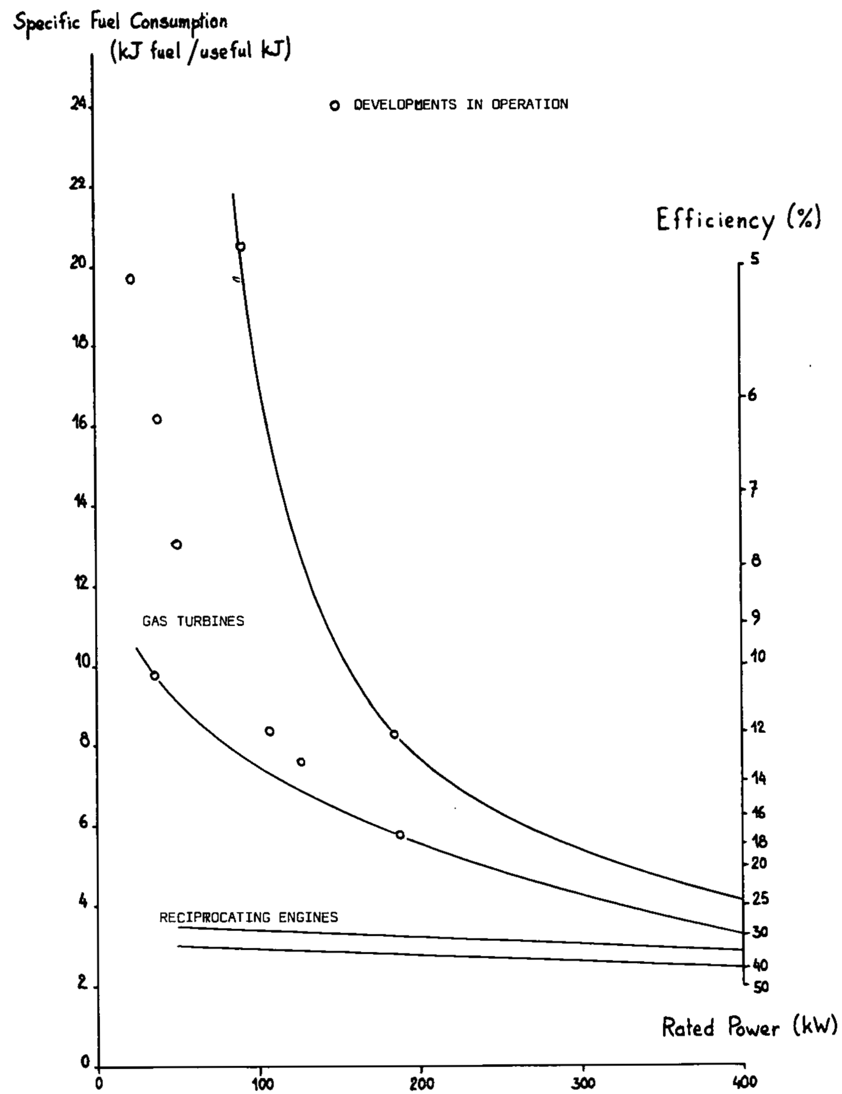

FIG. 2. COMPARISON OF SMALL GAS TURBINE AND RECIPROCATING ENGINE SPECIFIC FUEL CONSUMPTION AS A FUNCTION OF RATED POWER

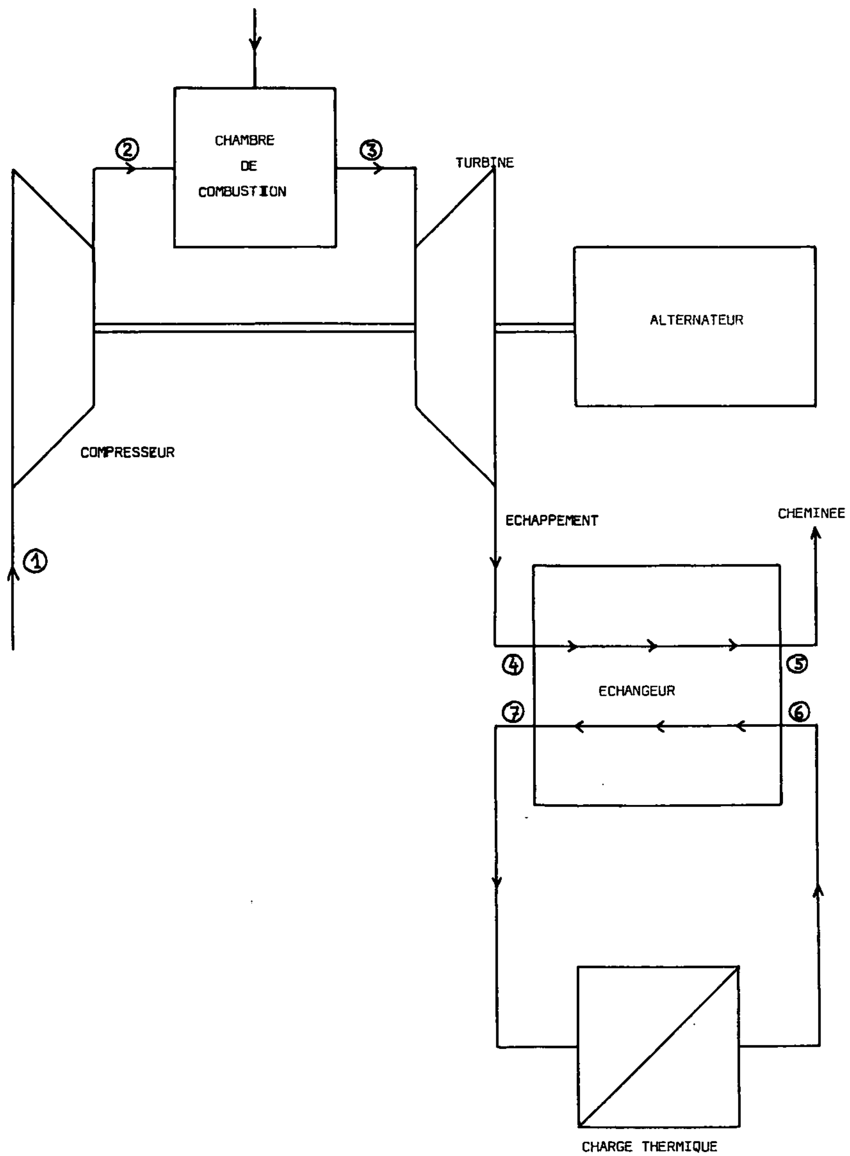

FIG. 3 - TURBINE A GAZ AVEC CHAUDIERE DE RECUPERATION

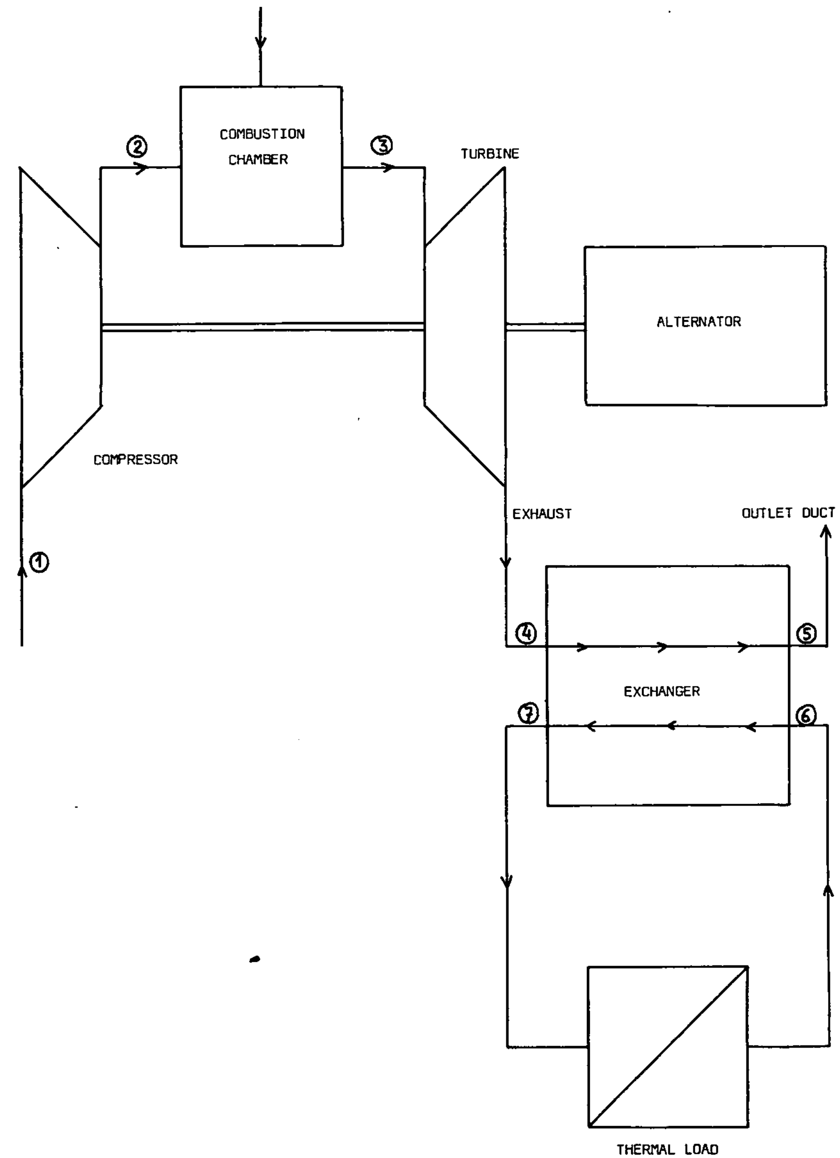

FIG. 3 - GAS TURBINE WITH REGENERATING BOILER

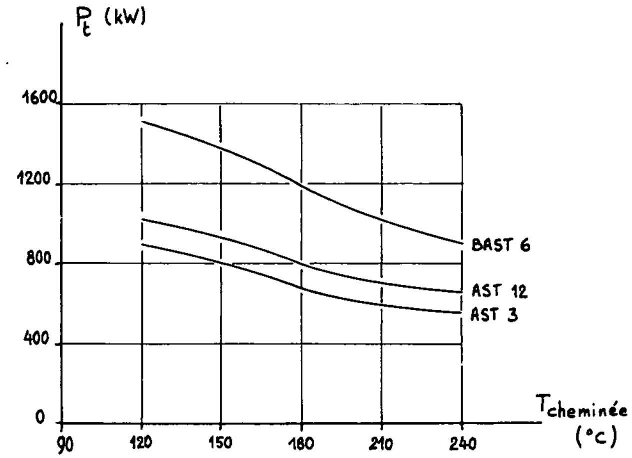

FIG. 4 - PUISSANCE CALORIFIQUE UTILE EN FONCTION DE LA TEMPERATURE A LA CHEMINEE

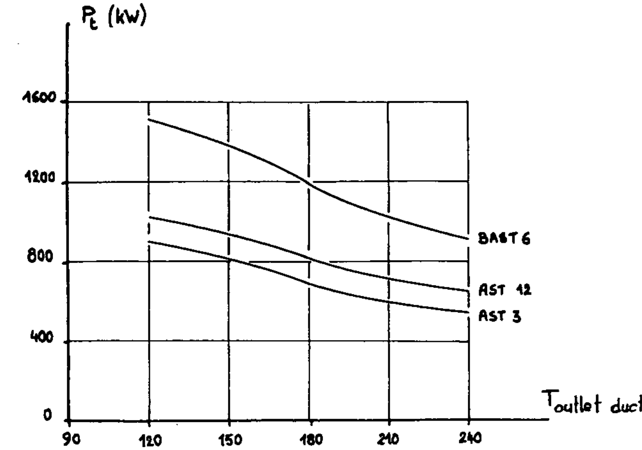

FIG. 4 - USEFUL THERMAL POWER AS A FUNCTION OF THE TEMPERATURE AT THE OUTLET DUCT

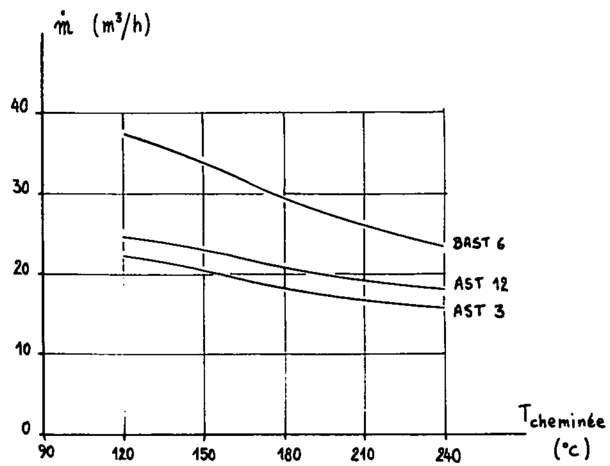

FIG. 5 - DEBIT D'EAU CHAUDE A 115°C EN FONCTION DE LA TEMPERATURE A LA CHEMINEE

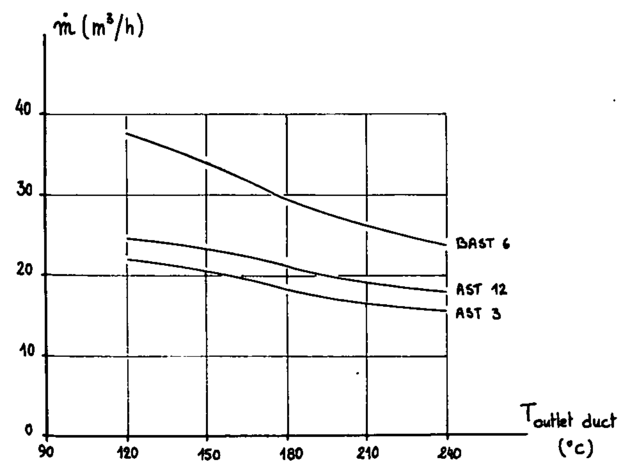

FIG. 5 - HOT WATER FLOW RATE AT 115°C AS A FUNCTION OF THE TEMPERATURE AT THE OUTLET DUCT

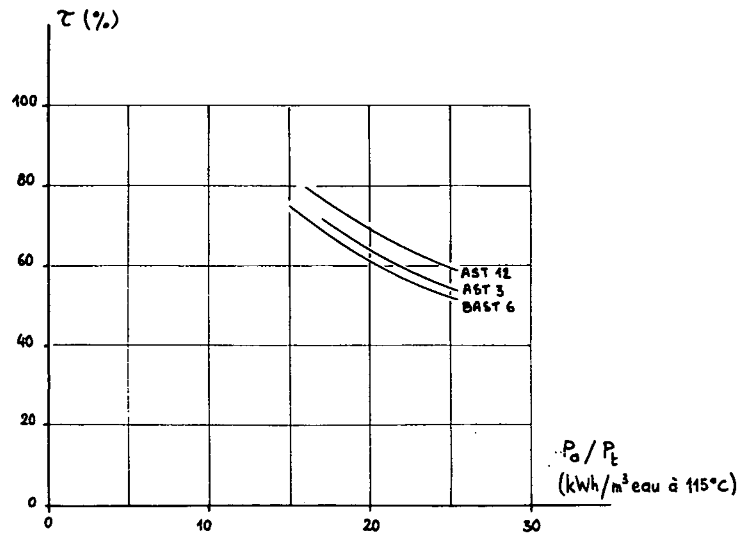

FIG. 6 - TAUX D'UTILISATION DE L'ENERGIE DU COMBUSTIBLE
EN FONCTION DE L'UTILISATION

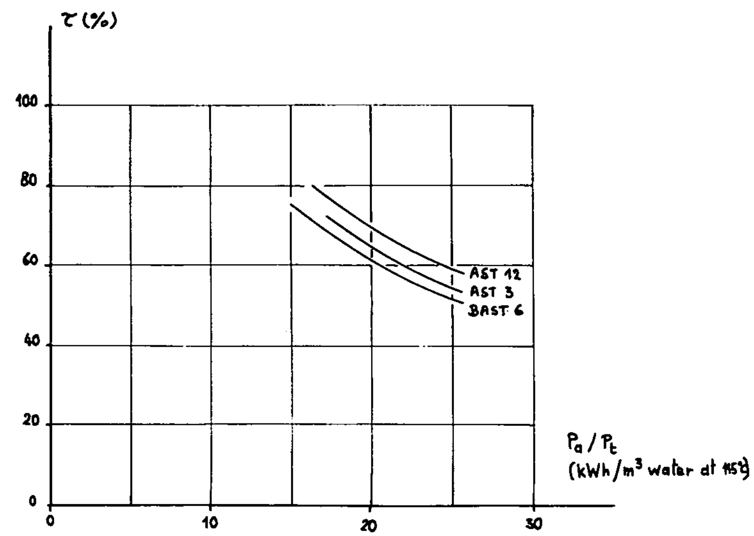

FIG. 6 - RATE OF USE OF FUEL ENERGY AS A FUNCTION OF USE

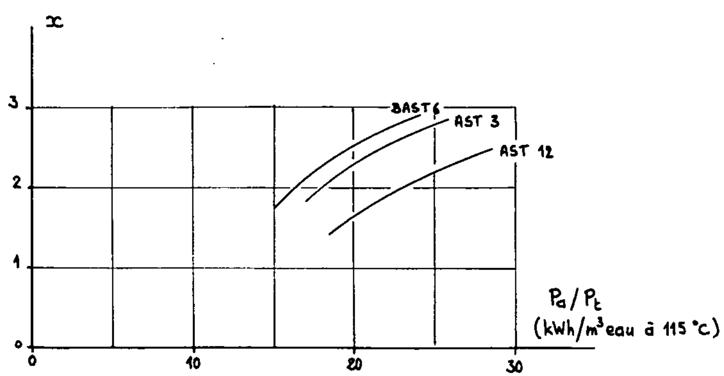

FIG. 7 - CONSOMMATION D'ENERGIE CALORIFIQUE PAR UNITE D'ENERGIE
MECANIQUE PRODUITE, EN FONCTION DE L'UTILISATION

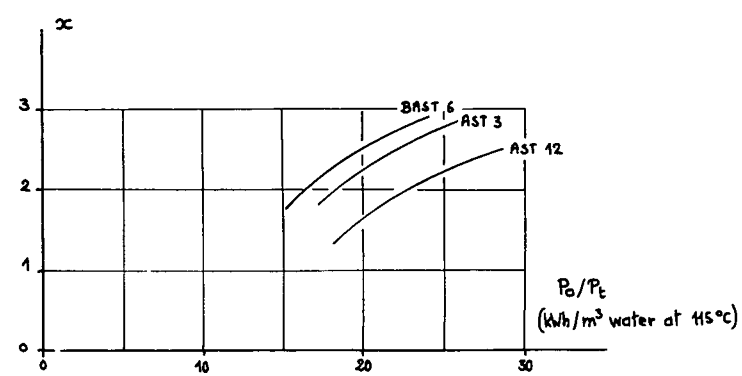

FIG. 7 - THERMAL ENERGY CONSUMPTION PER UNIT OF MECHANICAL
ENERGY PRODUCED, AS A FUNCTION OF USE

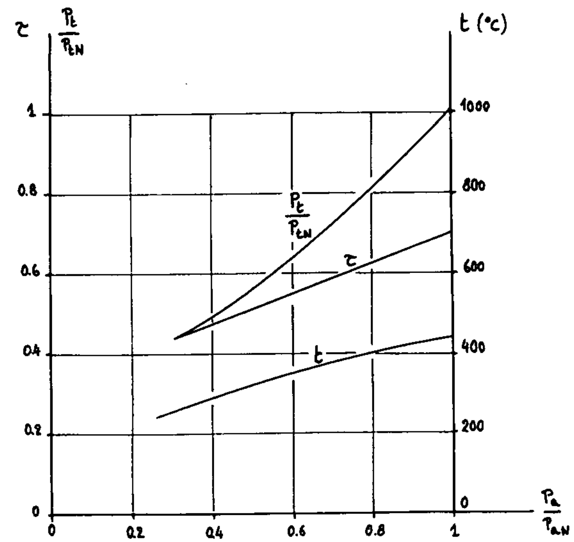

FIG. 8 - FONCTIONNEMENT EN CHARGE PARTIELLE DE LA TURBINE BASTANGAZ 6

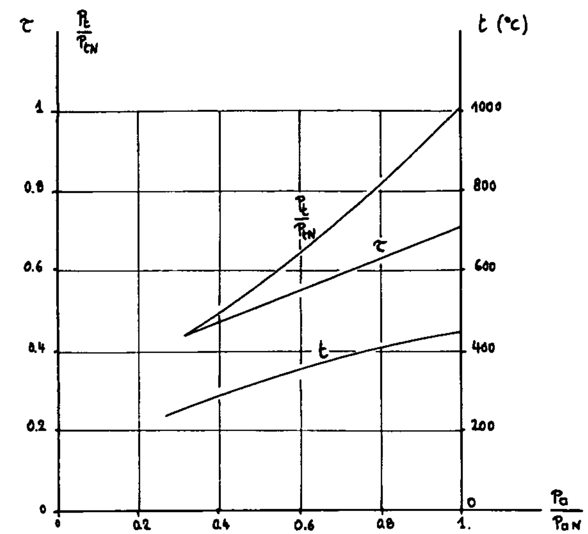

FIG. 8 - PART-LOAD OPERATION OF THE BASTANGAZ 6 TURBINE

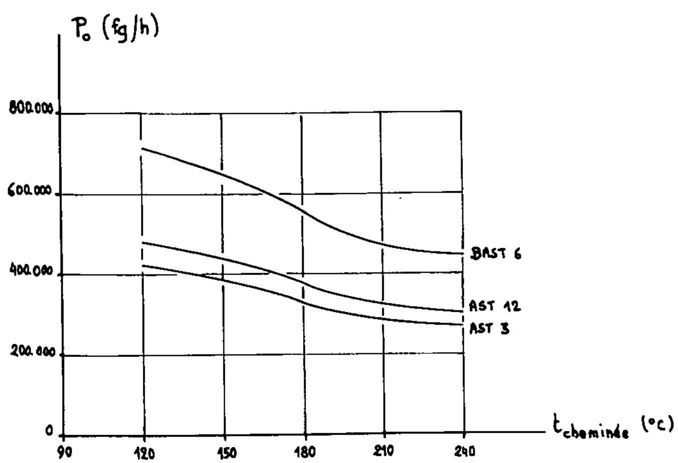

FIG. 9 - PRODUCTION DE FROID EN FONCTION DE LA TEMPERATURE A LA CHEMINEE

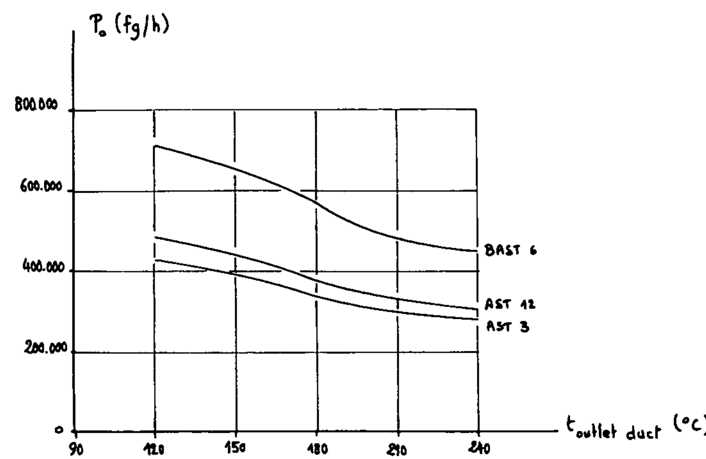

FIG. 9 - PRODUCTION OF COLD AS A FUNCTION OF THE TEMPERATURE AT THE OUTLET DUCT

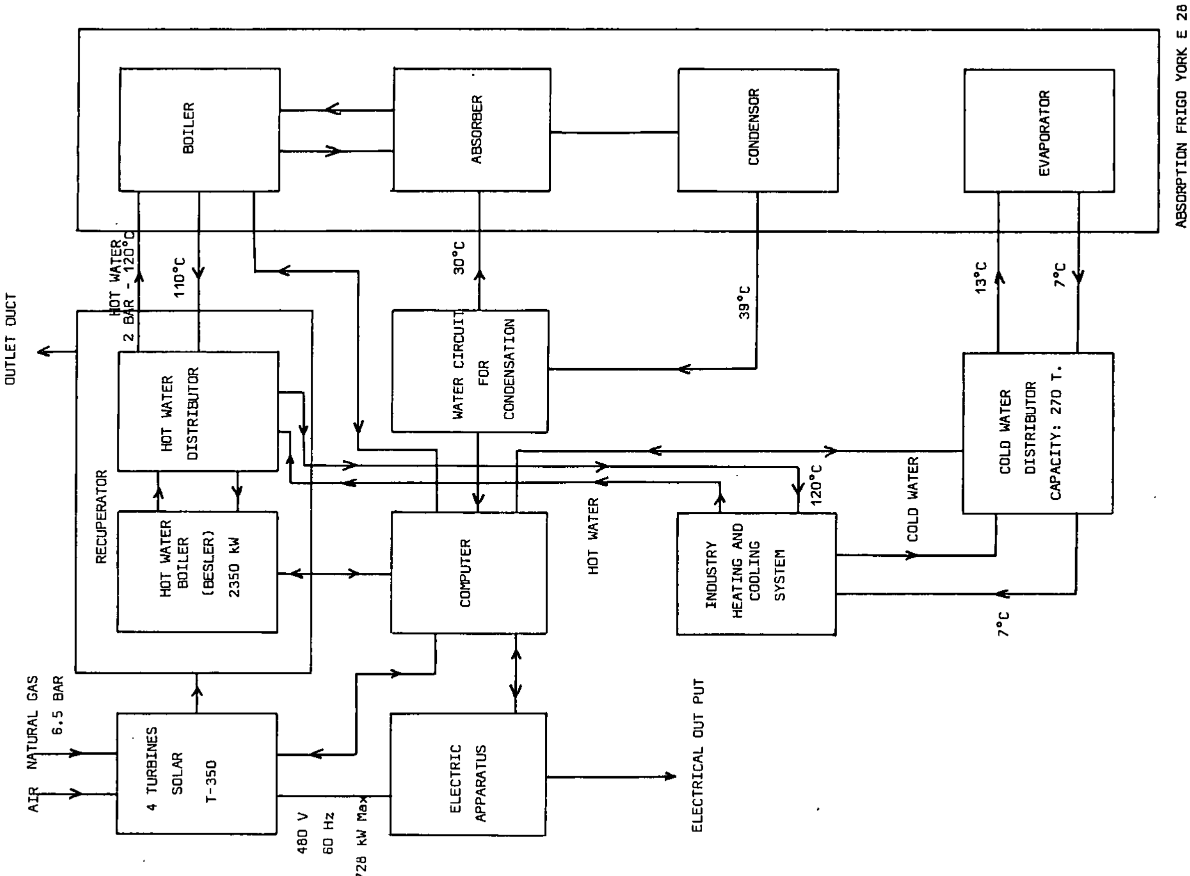

FIG. 10 - TOTAL ENERGY SYSTEM

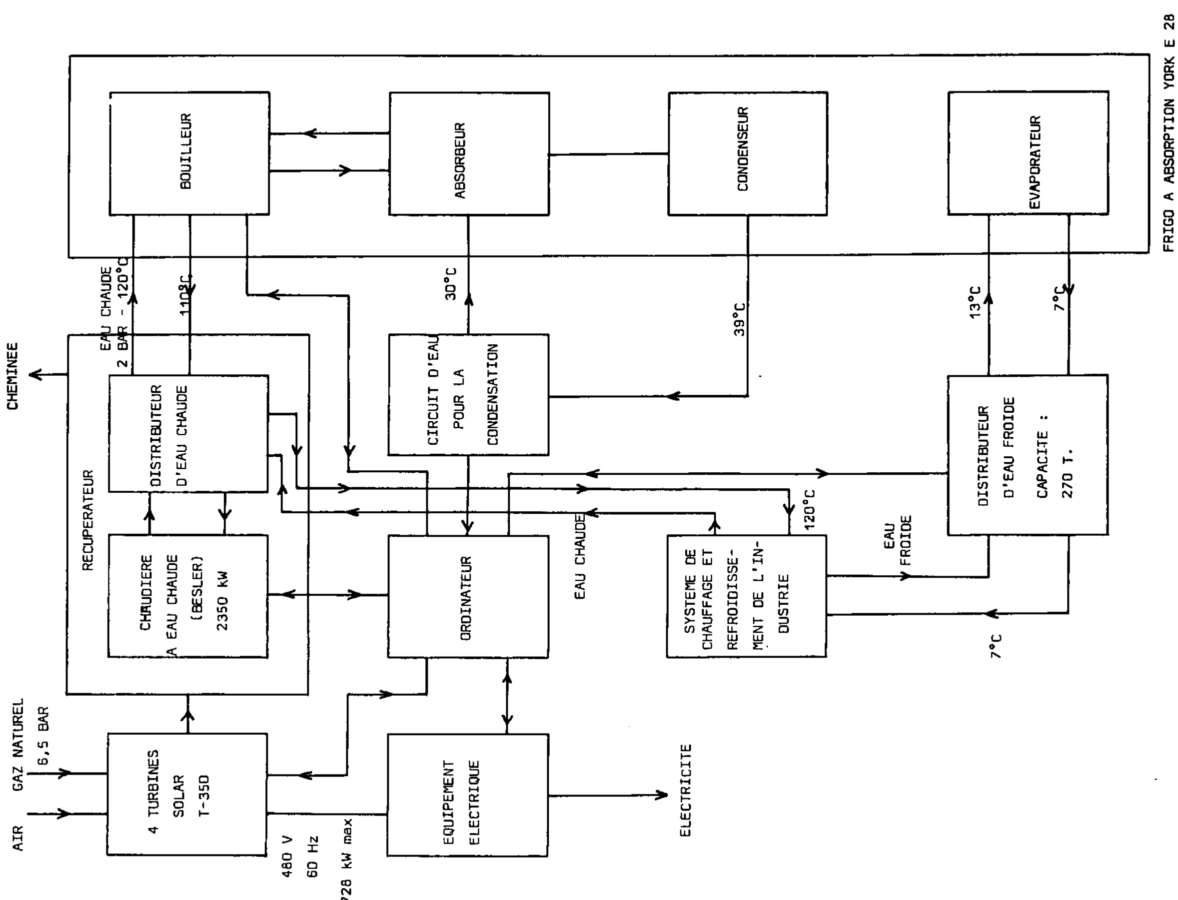

FIG. 10 - SYSTEME A ENERGIE TOTALE

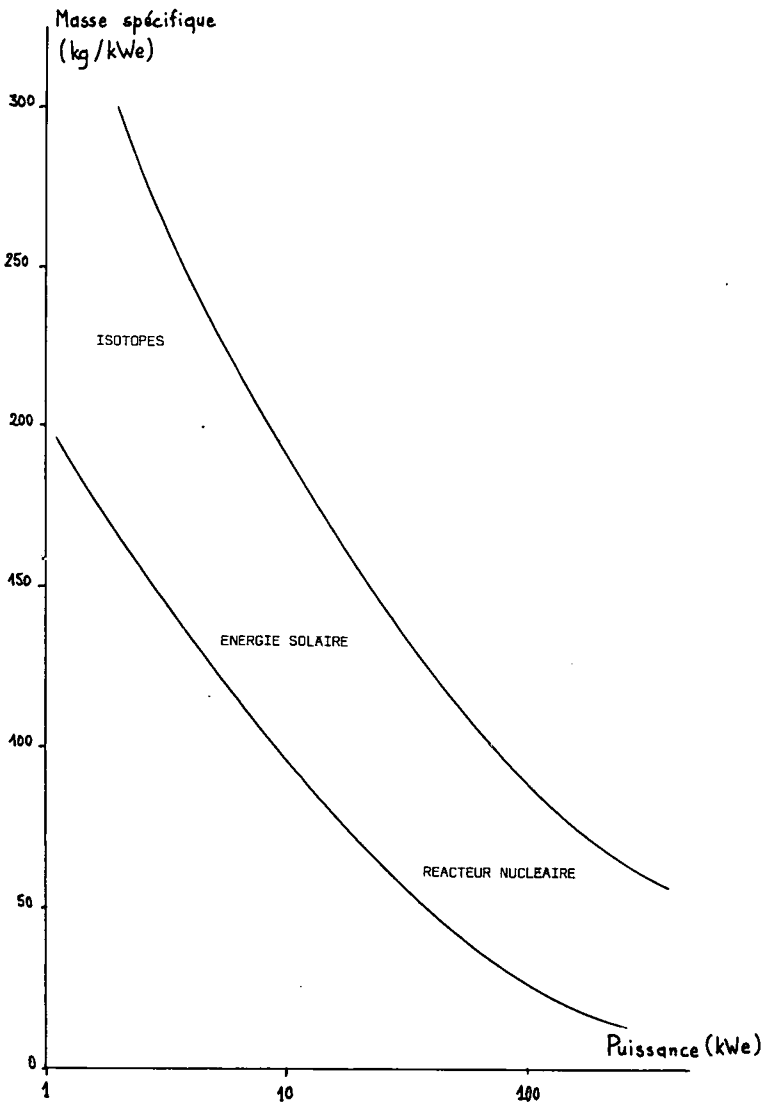

FIG. 11 - MASSE SPECIFIQUE D'UN SYSTEME DE CONVERSION A CYCLE DE BRAYTON EN FONCTION DE LA PUISSANCE . EXTRAIT DE (23)

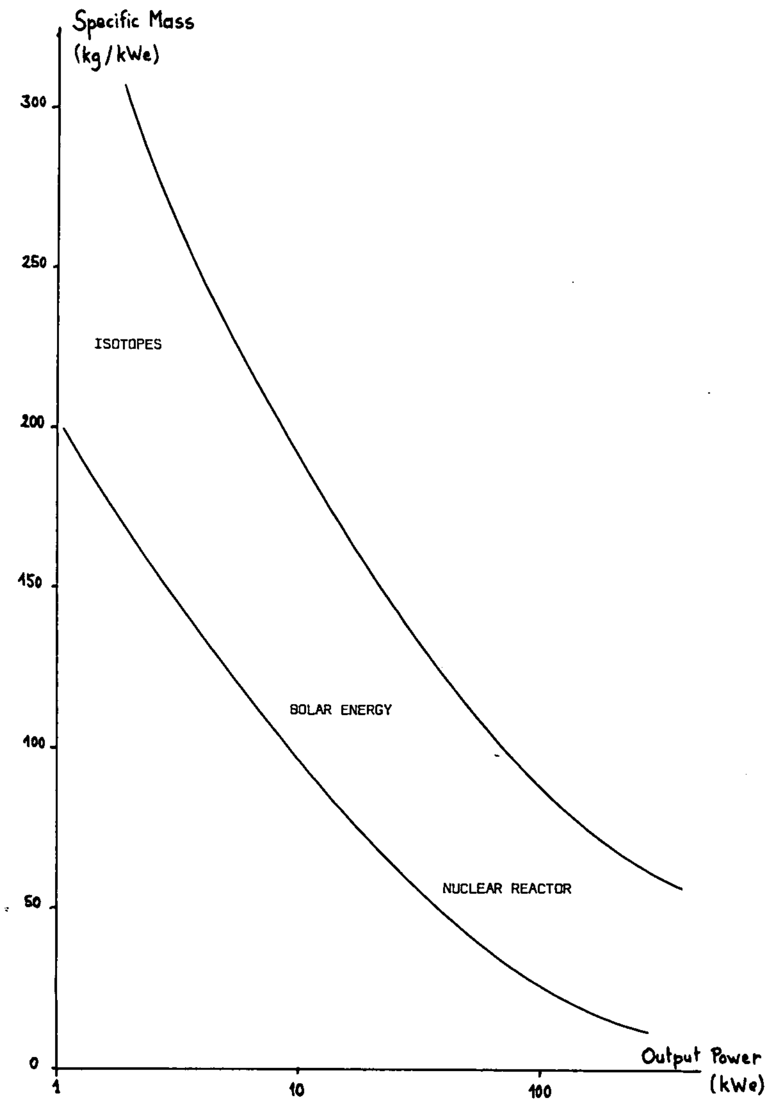

FIG. 11 - SPECIFIC MASS OF A BRAYTON CYCLE CONVERSION SYSTEM AS A FUNCTION OF THE ELECTRIC POWER SUPPLIED . FROM (23)

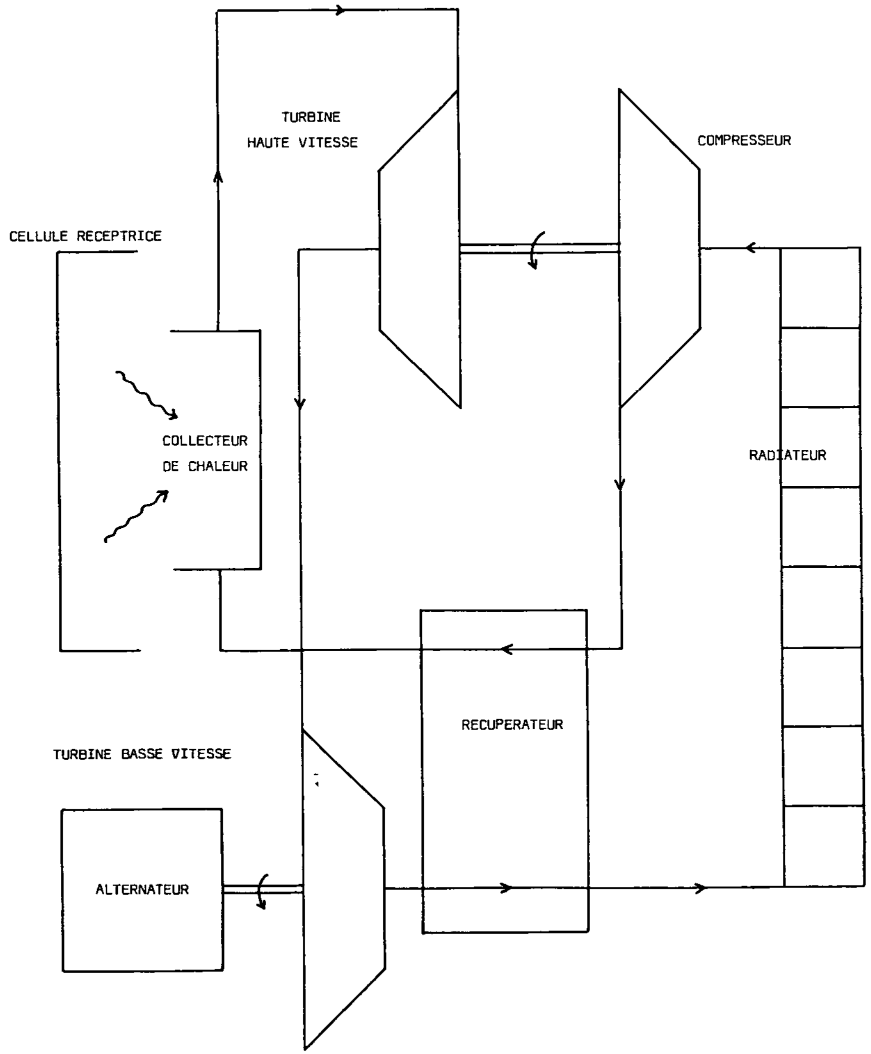

FIG. 12 - SYSTEME DE CONVERSION A CYCLE DE BRAYTON UTILISANT L'ENERGIE SOLAIRE EXTRAIT DE (23)

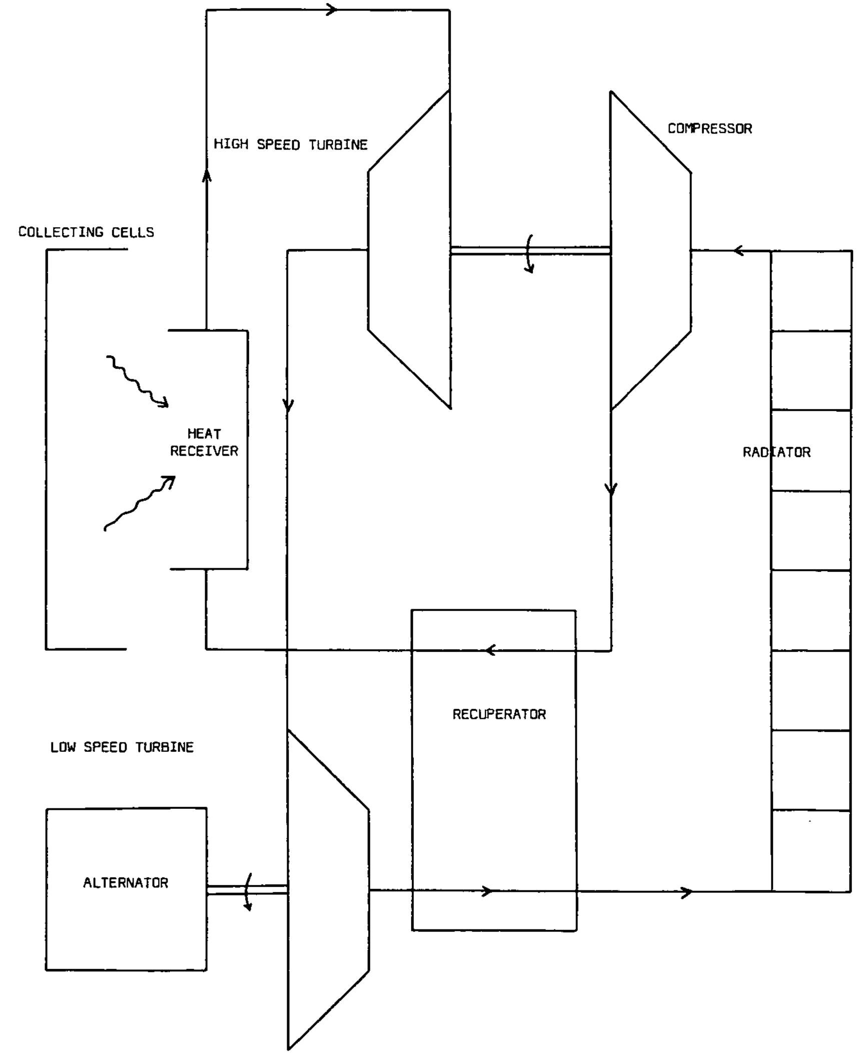

FIG. 12 - BRAYTON CYCLE CONVERSION SYSTEM USING SOLAR ENERGY. FROM (23)

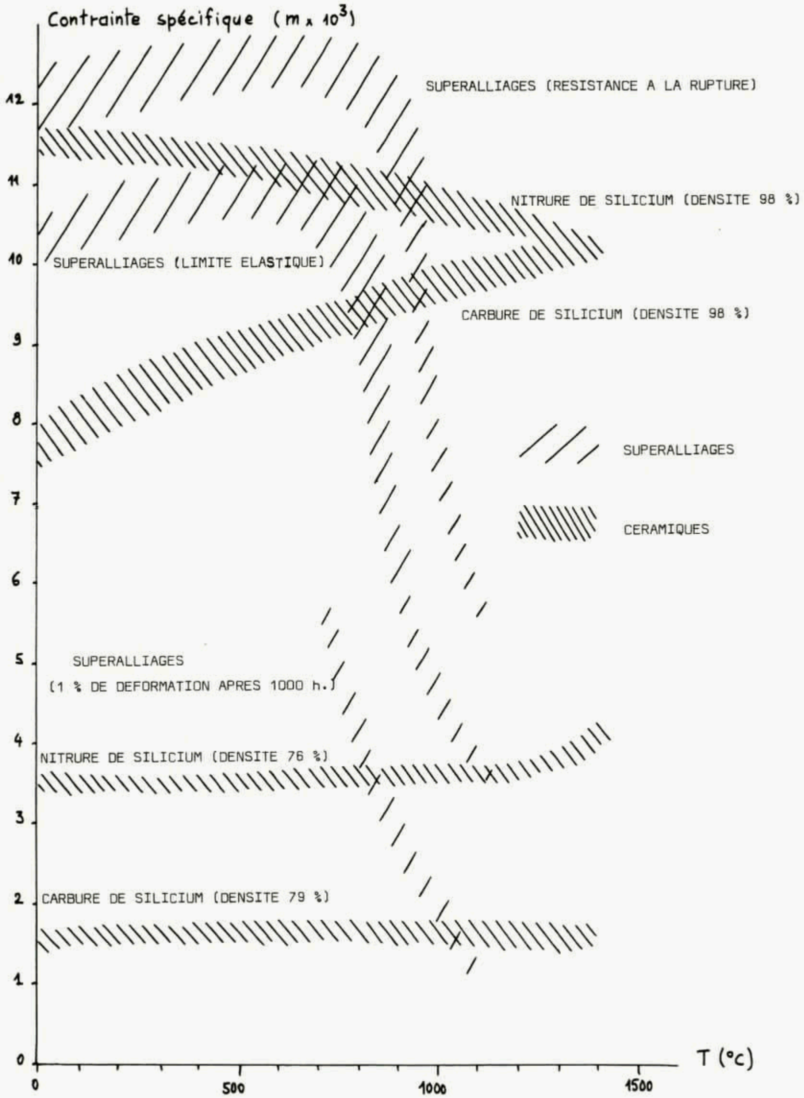

FIG. 13 - CONTRAINTE SPECIFIQUE EN FONCTION DE LA TEMPERATURE POUR LES SUPERALLIAGES ET LES CERAMIQUES. Extrait de (19)

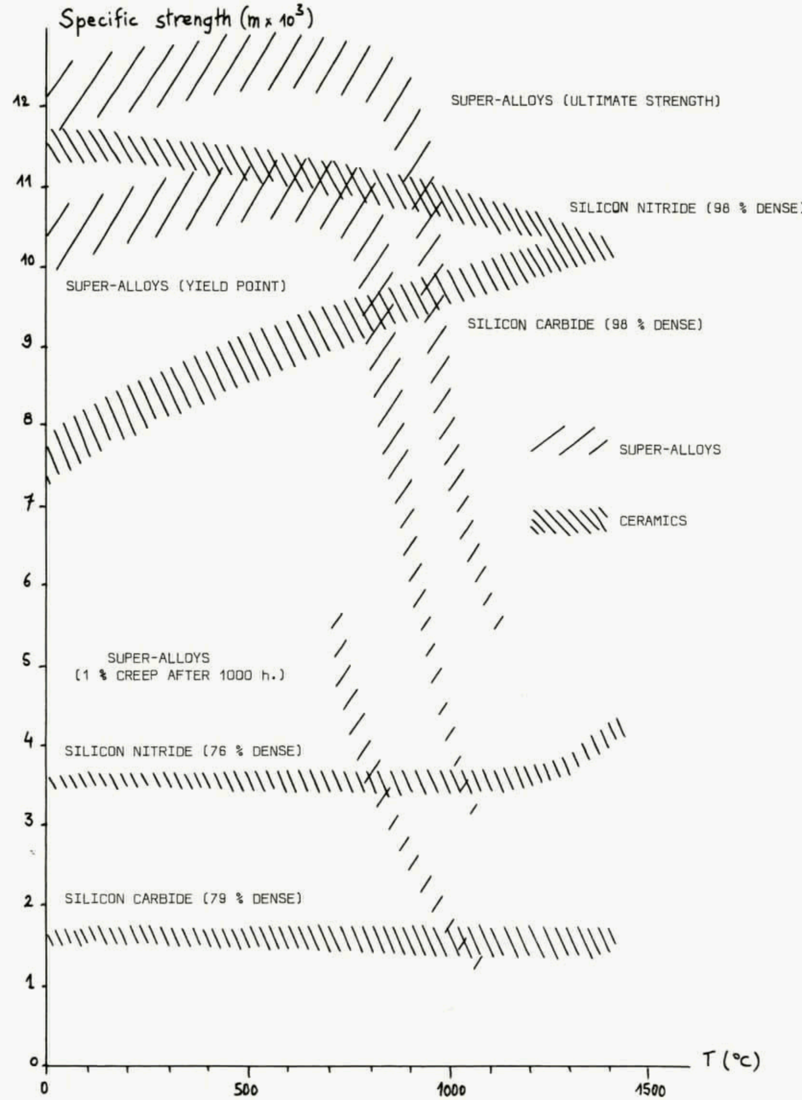

FIG. 13 - SPECIFIC STRENGTH AS A FUNCTION OF TEMPERATURE FOR SUPER-ALLOYS AND CERAMICS. FROM (19)

APPLICATION TO POWER GENERATION

André L. JAUMOTTE, Recteur de l'Université de Bruxelles

Summary : The applications of gas turbines of low power (below 500 kW) in the fields of aeronautics, industry and space are reviewed. In the first chapter, the advantages and drawbacks of gas turbines in comparison with Diesel engines are discussed.

The second chapter deals with the use of small turbines for the combined production of electric and thermal energy. The thermodynamic characteristics of the total energy system are described and a few examples of industrial applications given.

The third chapter outlines the possibilities offered by the use of gas turbines in space research, especially as regards the production of the energy required on board exploration vehicles.

1. Auxiliary and Emergency Power Production

Low power gas turbines are increasingly utilized as auxiliary or emergency energy sources, as well as main energy sources in many applications where they advantageously replace Diesel engines or even electric motors. The main field of application is, of course, that of aeronautics:

- airborne starters for jet engines, enabling aircraft to be independent of ground power supplies; this is justified by the increasing number of airports visited;
- aircraft electric supply and air conditioning, either on the ground only, or continuously;
- power hydraulic control systems, thus permitting pre-flight servo-control checks assisted piloting and aircraft descent in case of engine failure.

One of the particular advantages of airborne Auxiliary Power Units is their noise level, markedly lower than ground based power units in current use; this is an advantage both to ground personnel and the aircraft passengers (7).

As an example, in the case of the Lockheed Tristar, the Auxiliary Power Unit initiates the engine start prior to each take-off and operates throughout flight; the emergency alternator which it drives is connected with the three alternators mounted on each of the main engines (R B 211-22).

Among the other fields of application, one can mention:

- the propulsion of trucks, heavy construction machines, cars, trains, ships, air cushion vehicles, etc.
- electric generation units, usually transportable by one or two men, for use in mobile hospitals, telecommunications and firing control systems;
- accurate electric generator units insuring high reliability and stability, used to supply third generation computers, providing also the air conditioning of the facility (2);
- powering of reserve loading pumps on board tanker ships justified by the low utilization rate;
- production of propulsive power for self-propelled telepheric vehicles as used for dike building in Holland (Delta Programme) (3).

Closed cycle gas turbines powered by nuclear reactor, making up self-contained electric power generation units, are also planned. (4).

The main advantages of the gas turbine which justify these uses are :

- its low specific mass and bulk, markedly lower than those of Diesel engines (cf. fig. 1), which allow either high mobility or very reduced installation costs;
- its possibilities of adaptation to extremely varied applications:
- production of shaft power at variable or constant speed;
- production of pressurized air;
- production of hot air and gases (air conditioning);
- use of a larger range of fuels than corresponding Diesel engines (natural gas, butane, propane, Diesel oil) and possibility of fuel switching if necessary while the engine is operating provided the injection system was designed to allow this;
- long life time associated with simple operation and reduced maintenance cost;
- the absence of a cooling system, which simplifies the auxiliary components;
- low vibration level.

The less favourable characteristics of the turbine are the following :

- its specific fuel consumption, which is greater than that of piston engines at low power ratings (cf fig. 2) but which tends towards being competitive as the power increases (6);
- its sensitivity to the ambient temperature, with power and efficiency reducing as the temperature increases.
- its rather high noise level, which can only be reduced, for the time being, by intake and exhaust

mufflers, resulting in loss of efficiency;

- its cost, which is still rather high, but decreases progressively as the scale or production increases;
- its high speed of rotation, which often necessitates a speed reducing gear to adapt it to existing driven machines. Due to the progress achieved in the field of static frequency converters, it is possible to contemplate the production of high frequency electricity, and therefore to connect the alternator directly with the turbine and do without speed reducing gears.

Table 1 shows characteristic values for a few machines.

In spite of its lower efficiency, the gas turbine offers advantages which, finally, make it competitive with the Diesel engine for many applications and even, in some cases, with the electric motor.

Increased demand and production, together with the development of machines better adapted for drive by gas turbines, will extend the range of applications of the gas turbine.

TABLE 1

Note : $1 \frac{\text{kJ fuel}}{\text{useful kJ}}$ = 3600 kJ fuel/kWh = 860 kcal/kWh = 2546 B.T.U/H.P.h

Example (N = rated speed expressed in rpm)	Rated Power (kW)	Specific Fuel Consumption $\left[\frac{\text{kJ fuel}}{\text{useful kJ}}\right]$	Specific Mass (kg machine/rated kW)
Microturbo - Noelle 002 . airborne starting unit N = 50,000	from 59 to 1800 rpm of the starter	13	0.407 without starter 0.583 with starter
- Emeraude . Starting on the ground of Concorde jet engine . driving of fail safe devices in case of failure of the main engine . oil supply of clutch control N = 47.000	125	7,6	0.455 with starter
- Athos IV . jet engine starting . industrial applications (power at the shaft or compressed air) N = 53,000	74 at 2600 rpm of starter	20.6	0.414 without starter 0.606 with starter
- Jaguar . aircraft starting by compressed air (several starters supplied by a turbo-generator) N = 54,000	38.3 at 6000 rpm of the starter	16.2	0.888 without starter 1.2 with starter
- Cyclone . ancillary ground based unit for electric starting of engines N = 50,000	9	51	5.1 without accessories or carriage
- Gevaudan . airborne electric generator . possible supply of compressed air . other concept: starting of Rolls-Royce Vipertt and direct current generator N = 50,000	30	19.8	1.2 with dynamo
Sermel - Turbine-engine T M S 60 N = 45,000	184	8	0.19 turbine alone 0.3 with speed reducing gear 45,000 - 8000 rpm
- Turbine-engine T M S 30 N = 55,000	110	8.55	0.27 turbine alone 0.435 with speed reducg. gear 55,000 - 8000 rpm
- "Titan" Solar . individual helicopters . generating plant	44	9	0.570 with speed reducing gear
Motoren and Turbinen Union Munchen GmbH M 1 U 6022 - A S	184	5.58	0.53 turbine alone

2. Total Energy Concept

2.1. Introduction

The discussion will be limited to low power turbines, operating with natural gas or fuel. The gas turbine can operate on heavy fuel oil, provided that :

- either a maximum temperature of 650°C be not exceeded;
- or the heavy oil fuels used meet restrictive requirements as regards the amount of some elements, such as vanadium, contained in the ash, possibly with additives.

In both cases, the fuel oil must be heated up to the temperature ensuring the most appropriate viscosity for injection purposes and the turbine must be periodically cleaned. During the operation, the power and the efficiency decrease. These overall restrictive conditions are acceptable in some specific cases; they are not in the case of industrial facilities of modest power.

The latter become competitive in so far as they can be supplied with Diesel oil or natural gas fuel at a price relatively close to that of the heavy fuel oil per calory. This requirement is sometimes met by the natural gas calory.

2.2. Simultaneous Heat and Energy Production

2.2.1. Operation under Design Conditions

The utilization considered may be characterized by the ratio of the shaft power produced to the useful thermal power, that is P_a/P_t.

To represent the thermal aspects of the combined production of heat and energy, one can use :

- the energy utilization rate of the fuel τ, defined as the ratio of the useful amounts of mechanical and thermal energy to the energy contained in the fuel:

$$\tau = \frac{P_a + P_t}{P_c}$$

- the consumption of thermal energy per unit of mechanical energy produced x.
  The heat consumption related to the production of mechanical energy is defined by the difference between the overall consumption of the energy derived from the fuel P_c, and the consumption which would be required to produce the same useful thermal power P_t in a normal boiler, that is P_{co}.

  Therefore, $$x = \frac{P_c - P_{co}}{P_a}$$

As an example, one can consider the generating station of the Turbomeca Plant at Bordes (Basses Pyrénées, France), with three Turbomeca turbines (ASTAGAZ 3, ASTAGAZ 12 and BASTANGAZ 6), supplied with natural gas (fig. 3). The characteristics of these turbines when operating under design conditions are as follows:

TABLE 2

	ASTAGAZ 3	ASTAGAZ 12	BASTANGAZ 6
P_a (kW)	350	400	550
Specific consumption (kcal/kWh)	4200	3830	4350
Air flow rate (kg/s)	2,55	2,87	4,69
Exhaust temperature T_4 (°C)	450	450	425

Calculating P_t for a few assumed temperatures at the exchanger outlet (T_5): 240°C, 180°C, 150°C and 120°C, gives

$$P_t = \dot{m}_{air}\,(i_4 - i_5) = \dot{m}_{air}\; c_{p\,air}\,(T_4 - T_5)$$

$$= \dot{m}_{water}(i_7 - i_6) = \dot{m}_{water}\; c_{pwater}(T_7 - T_6)$$

Assuming that water gets into the exchanger at 78°C and out at 115°C, the water flow rate can be deduced from the above equation.

To calculate P_{co}, a boiler efficiency of 0.8 has been assumed.

Therefore: $P_{co} = \frac{P_t}{0.8}$

The following table summarises the results of the calculations.

TABLE 3

	T_5	120°	150°	180°	240°
P_t (kW)	AST 3 AST 12 BAST 6	880 990 1510	815 915 1370	700 785 1160	565 635 930
$\dot{m}$ (m^3/h)	AST 3 AST 12 BAST 6	21,5 24,2 37	19,9 22,3 33,6	17,1 19,2 28,4	13,8 15,3 22,8
P_a/P_t (kWh/m^3)	AST 3 AST 12 BAST 6	16,3 16,5 14,9	17,6 17,9 16,4	20,5 20,8 19,4	25,4 25,8 24,1
(%)	AST 3 AST 12 BAST 6	71 78 74	68 73 69	61 66 61	53 58 53
x	AST 3 AST 12 BAST 6	1,74 1,34 1,63	2,08 1,60 1,94	2,38 2,00 2,41	2,89 2,47 2,93

The results are illustrated in figure 4, where P_t has been plotted as a function of temperature in the outlet duct, and figure 5 where the water flow rate m has been plotted as a function of the same temperature.

Figures 6 and 7 show respectively the utilization rate of the energy provided by the fuel, and the thermal energy consumption per unit of mechanical energy, as a function of ratio P_a/P_t. These two figures bring out clearly the advantage offered by the simultaneous production of mechanical and thermal energy by a low-power gas turbine.

2.2.2. Partial Load Operation

The case has been considered of a facility equipped with a regenerating boiler, with an exhaust temperature of 150°C, on the basis of the characteristics provided by the manufacturer (Turbomeca - Bastangaz 6). The gas flow rate is almost constant; however, the exhaust temperature decreases considerably. Figure 8 shows the ratio of the heat recovered to the heat recovered when the operation takes place under design conditions: P_t/P_{tN}, as well as the energy utilization rate Υ, as a function of the ratio of the mechanical power to the mechanical power under design conditions P_a/P_{aN}. Note that the recovered heat ratio decreases proportionately to the ratio of the power delivered at the shaft.

2.3 Simultaneous Production of Electricity and Cold

Looking at the case when the heat contained in the exhaust gases is used as a heat source for the boiler of an absorption refrigerating machine, the practical refrigerating effect coefficient is the ratio of the quantity of cold collected at the evaporator to the quantity of heat provided to heat up the boiler; under real conditions $\varepsilon = Q_{EV}/Q_B$. The American firm York Corporation gives $\varepsilon = 0.55$ for an ammonia absorption machine with a heat source temperature of 115°C (Boiler) an intermediate temperature of 30°C (Condenser) and a heat sink of 0°C (Evaporator). On figure 9, the quantity of cold produced P_o (in fg/h) has been plotted as a function of the temperature in the outlet duct for the same three turbines.

2.4 Simultaneous Production of Electricity, Heat and Cold

Taking as an example the American Meter's Fullerton plant in California, four turbines produce the electricity required and the heat of the burnt gases is recovered to supply the air conditioning and heating systems.

Out of the four Solar turbines, two use a regenerating cycle (pre-heating of the air in a heat-recovery exchanger); this reduces the natural gas consumption by 12%. The electric equipment produces a maximum of 728 kW. In the day-time, three turbines operate normally and produce 500 kW; in the night-time, a single turbine produces the necessary 100 kW.

The total energy system must meet four requirements:

- the electric system must produce 700 kW at 60 cs and less than 12 kV to be compatible with the facilities previously existing.
- the air conditioning system of the plant must be capable of maintaining the same temperature, to within 2°C, throughout the 35,000 square metres of the premises. A capacity of 270 tons of cold water at 3°C is sufficient for the needs of the plant, including the pre-cooling of the air at the compressor intake.
- hot water must be produced for the purpose of heating up the premises; the exchanger provides

it at 60°C.

- for future developments, an additional hot and cold water capacity is contemplated.

To fulfill these four requirements, the total energy facility is divided into six main groups (figure 10):

1) power generation : four Solar T-350 turbines drive alternators. The electric charge diagram determines the number of turbines in operation. The alternators are triphase 480 V - 60 cs.

2) electric energy distribution: all the alternators are connected with a single step-up transformer 480 V/12 kV.

3) heat recovery device : all the burnt gases are driven towards a Besler exchanger, with a controllable by-pass, and are finally exhausted into the atmosphere through a silencer. The computer regulates the boiler and by-pass flow rates according to the thermal load. The hot water is stored in a tank under pressure and supplies the premises, on the one hand, and, on the other hand, the absorption machine; the computer controls the overall system by means of valves and thermostats.

4) refrigerating machine :an absorption model York E-28 is associated with a cooling tower which rejects the absorption and condensation heat to the atmosphere. The cold water supplies six air conditioning units, two air pre-coolers at the intake of the compressors, and the cooling systems of all the hot water pumps.

5) heating and cooling system of the plant: six air-conditioning units supply cool air to the premises. The heating is produced by the hot water from the regenerator, as well as by auxiliary gas burners.

6) the computer : by means of its sensing, control and protection circuits, it automatically insures the synchronization and parallel setting sequences of the alternators, the smooth operation of the boilers, the refrigerating machine, the cooling tower, the various hot and cold water and condensation pumps, as well as of the relay circuits.

The characteristics of the Solar 350-T turbine, when operating under design conditions, are as follows :

TABLE 4

P_a = 190 kW
P_c = 1350 kW
Compression rate = 3.8
Air flow rate = 1.82 kg/s
Exhaust temperature = 540°C

The thermal characteristics of the combined facility are the following (for one turbine) :
P_t = 780 kW and P_{co} = 980 kW

Therefore $\frac{P_a}{P_t} = 0.244$

$\Upsilon = 72\%$

$x = 1.95$

3. Applications to the Space Field

3.1 Auxiliary power sources for spacecraft

3.1.1. Introduction

In this chapter, the scope will be confined to the thermodynamic conversion of energy by turbomachines. Other systems can be contemplated, however:

- chemical energy imposes prohibitive masses;

- thermoionic or thermoelectric conversion provides us with direct current which has to be converted into alternating current. This means additional weight and reduced efficiency.

In fact, thermodynamic conversion through turbomachines seems to be the only adequate system when more than a few kW are needed. This conversion system poses two major problems: the choice of the working cycle and that of the power source.

3.1.2. Working cycle

Two different systems immediately present themselves: the Rankine cycle and the Brayton cycle. In space, the only means of heat rejection consists in having a large radiating surface available. The higher the temperature at which the non-converted heat is rejected, the smaller may be the radiator used to insure the same rate of heat transfer. Therefore, at first sight, the Rankine cycle seems more interesting, as it permits heat rejection at a higher temperature than the Brayton cycle. However, a detailed analysis reveals various advantages in the Brayton cycle (17), (20). For example:

- the working fluid is an inert gas, which does not raise the problems due to the use of liquid metals in the Rankine cycle;
- pressures and temperatures can be chosen independently;
- considerable progress has been achieved in the field of aviation turbine engines, where the components are extremely similar to those used in the Brayton cycle. This has led to a marked increase in efficiency and to high reliability;
- continuing technological progress is permitting a continuous increase of the cycle maximum temperature, which improves the efficiency;
- progress in compressor, turbine and heat exchanger designs also result in improvements in efficiency;
- a judicious choice of the working fluid can also improve the overall conversion efficiency. From this point of view, Helium seems to be the most useful gas, although excellent results have been obtained with Argon and Helium-Xenon mixtures.

3.1.3. Power sources

As a rule in considering a new auxiliary power system for a spacecraft, three main objectives are aimed at :

- minimum weight;
- maximum efficiency;
- minimum heat rejection area.

The above factors are mutually exclusive. The first one is undoubtedly the most important; however, the third one is also important, especially as regards the supply of power to small-size units.

3.1.3.1. Solar energy (18)

Solar energy is not very dense (140 W/cm^2 in the vicinity of the Earth) and cannot meet the power requirements of a long space exploration far from the Sun. On the other hand, circumterrestrial missions impose cyclic passages in the shadow of the Earth; therefore airborne energy storage facilities are necessary, which results in considerably increased weight. On the other hand, the maximum electric power available at the outlet of a conversion system using solar energy as the power source is limited by the area of the devices collecting this energy. To achieve an acceptable weight/power ratio, these collectors require focussing mirrors. This creates additional technological problems. Moreover, the collecting cells deteriorate with time, due to exposure to the space environment.

The deduction is, therefore, that solar energy can only provide the necessary auxiliary power within a limited range (approximately 10 kWe) and is adapted only to short-term missions.

3.1.3.2. Nuclear energy (18), (22)

Two possibilities exist :

- nuclear energy in an isotopic form, which can only meet very low power requirements;
- the nuclear reactor, which can provide practically unlimited power. The specific weight of such a conversion system is substantially independent of the power. The only problem inherent in the use of a nuclear reactor is the necessity for appropriate armour plating, for protection against dangerous radiations, which increases the overall weight.

3.1.3.3. Conclusion

Each system is well adapted to a given power range, as shown by figure 11.

3.2 Projects being studied

Two projects are discussed, the first uses solar energy as the power source, the second a nuclear reactor.

3.2.1. Utilization of solar energy as power source (23)

The system described here has been studied since 1963 at the NASA Lewis Research Center to meet the electric energy requirements of a manned space station orbiting around the Earth. This vehicle would be equipped with three modules of the type dealt with here. Figure 12 gives a schematic idea of the conversion system.

Argon (molecular weight: 40) was chosen as the working fluid. The area of the solar energy collector is approximately $76m^2$. The overall weight of the system is 790 kg.

To satisfy the performance requirements, the turbine inlet temperature was set at 1090°K. This maximum temperature was selected taking into consideration the characteristics of the material chosen as the hint sink. The latter was chosen to arrive at the best possible compromise between high efficiency and the material property requirements. It was decided to use lithium fluoride, the melting point of which is approximately 1100°C. With this system a 0.25 conversion efficiency is achieved.

3.2.2. Utilization of a nuclear reactor as power source (21)

This system is being studied by AiResearch, under a North American Rockwell Corporation contract. Its objective is to provide a large manned space laboratory with the necessary auxiliary power. The specifications are as follows:

- power : 100 kWe
- heat source : Zirconium hydride nuclear reactor, cooled with NaK.
- maximum surface available for heat rejection : 655 m^2
- system life-time: 10 years.

The configuration of the conversion system is somewhat complex: there are two reactors and each is connected with four independent conversion modules. Each module is composed of a compressor, a turbine and an alternator and can produce 25 kWe. This configuration makes it possible to obtain 200 kWe in case of need and, in addition, offers high flexibility and reliability.

The working fluid used in the conversion turbomachines is a mixture of xenon and helium with a molecular weight of 40. The mass flow rate is 1/60 kg/s. The reactor outlet temperature - therefore the maximum temperature of the installation is approximately 865°K. Other tests have been made at different temperatures (920°K and 976K) however, the conversion efficiency appeared to be constant at 0.22. It may be considered, therefore, that the highest possible temperature should be selected in order to reduce the radiator area. However, the last requirement - a ten year lifetime - also has to be taken into consideration. Various investigations on the reactor have proved that 865°K is not far from being the maximum temperature compatible with the lifetime requirement.

3.3 Future Prospects

The most important development possibilities are dependent upon new technological solutions to be realized. Experiments are in progress in various fields, such as:

- the use of higher rotation speeds;
- the use of gas bearings;
- the reduction of the considerable secondary losses which are mainly due to the very small sizes of the machines.

One factor, however, may be regarded as the most important goal in present investigations: raising the maximum temperature of the cycle. This problem is important in all gas turbines as it directly affects efficiency; however, it appears still more vital in space applications where a high temperature also permits a reduction of the rejection area necessary for unconverted heat.

Two new solutions have been proposed :

- the first one consists in using certain metals instead of conventional super-alloys. For instance, Molybdenum, the melting point of which is 2620°C, is being considered. The use of such materials is rendered possible by the reducing atmosphere in which they would work, due to the working fluids selected (helium, inert gases). However, major problems impede this attractive solution : the difficulties encountered in the production machining of such metals.

- the second solution envisages the use of ceramics in place of conventional super-alloys (19). From the mechanical strength viewpoint, the use of such materials poses three big problems:
  - the static mechanical loads, the effects of which may be reduced by adequate design, in order that the ceramic parts be subjected to compressive rather than tractive stresses;
  - mechanical shocks. Fortunately, impact resistance criteria are not as stringent for small turbines as for large aircraft turbine engines. Besides, in the particular case of auxiliary power sources for space applications, closed cycle operation removes the risk of the ingestion of foreign bodies. The only remaining hazard is the possibility of a damaged part breaking off inside the machine. Of course, such a possibility exists for any turbine;
  - in fact, the only stringent criterion to be applied to ceramic materials is that of the centrifugal load in the rotors. In the case of metals, bursting results from the oscillating cyclic load around the elastic limit, therefore, from fatigue. Such is not the case with ceramics which are brittle non-deformable materials. Consequently, the stress must always be kept below the elastic limit.

As far as ceramics are concerned, close dependence is observed between resistance and density, as shown on figure 13, where the tensile strength/density ratio (specific strength) is plotted as a function of temperature, for super-alloys and ceramic materials. This figure shows the advantage likely to be derived from the use of very dense ceramics, especially at high temperatures. In fact, these types of ceramics are the only ones compatible with the centrifugal load requirement. For the time being, they can only be obtained by hot-compression techniques, an unacceptable type of process from a financial viewpoint.

Acceptable resistance to environmental conditions and thermal shocks can be obtained with ceramics of the lithium-alumina-silicate type. Such materials can be developed by casting, a relatively inexpensive process.

In conclusion, it can be said that the use of ceramics is promising for the future, especially as regards fixed components. However, as far as rotors are concerned, the choice appears limited to high density silicon carbide or nitride, two materials which are too expensive at the present time for practical application.

Bibliography - Chapter 1

(1) P.A. Pitt - "Small Gas Turbines in Industrial Applications" ASME Paper No. 66-GT-110 - 10 pages
(2) Robin Mackay - "The Gas Turbines as a Source of Continuous Precise Power" ASME Paper N° 69-GT-20 11 pages
(3) J.A.K. Konings & L. Erlebach - "Aircraft-Type Gas Turbines Drive 16 Independent Cableway cars to be Used Under the "Delta Plan" (The Netherlands)" ASME Paper N° 70-GT-13 15 pages
(4) John R. Hoffmann & Ernest G. Fehev - "150 kWe Supercritical closed Cycle System" ASME Paper N° 70-GT-89. 11 pages
(5) H. Lanoy - "Nouvelle petite turbine à gaz de 50 à 80 ch pour hélicoptères individuels, plates-formes volantes, groupes électrogènes, etc." DOCAERO N° 64 - Septembre 1960 - pages 27 à 30.
(6) J. Chauvin - "Perspectives et Problèmes du développement des Turbines à Gaz de Faible puissance" Revue de la S.R.B.I.I. N° 4 avril 1970 - pages 235 à 249
(7) C.H. Paul - "The Development of Small Gas Turbines for Aircraft Auxiliary Power". The Aeronautical Journal of the Royal Aeronautical Society Vol. 74 October 1970 - pages 797 à 805.

Chapter 2

(8) R. Ecabert - "Installations combinées gaz-vapeur avec production simultanée d'énergie électrique et de vapeur de fabrication" Revue Technique Sulzer 1963, N° 2 - pages 75 à 80
(9) A. Jaumotte et J. Hellemans - "Utilisation de la turbine à gaz alimentée au gaz naturel dans les installations industrielles de production combinée de vapeur et d'électricité" Réunion de Vienne 1965 de la Commission de la Turbine à gaz de la F I P A C E 11 pages
(10) D. Hale - "Turbine Total Energy system provides total power for heavy manufacturing Plant" America, Gas Journal April 1965 - pages 47 à 54
(11) "1967 Gas Turbine Catalog" - Gas Turbine Publications Inc. - page S-14
(12) J. Sawyer - "Gas Turbine Engineering Handbook" Gas Turbine Publications Inc. 1966, 552 pages
(13) "Symposium on the Gas Turbine" Technische Hogeschool-Delft, 18 mai 1965.
(14) "Total Energy Gas and the Gas Turbine" A one-day symposium held at the technological University of Delft - The Netherlands, 23 Sept. 1966.
(15) R.W. Stuart Mitchell - Conference 1967 - Managing Energy - The Total Energy Concept" The Combustion Engineering Association, 48 pages.
(16) R.V. Hoffmann - "Gas Turbine for Total Energy" Mechanical Engineering, April 1968, p. 92 à 98.

Chapter 3

(17) P. Barthelemy, C. Chauvineau et G. Dieulot - "Thermodynamique du cycle de Brayton pour applications spatiales" papier présenté au Sixième Colloque d'AGARD, Cannes, 16-20 Mars, 1964 - AGARDograph 81, pp. 181-205 - Gordon & Breach Science Publishers, London, 1967.
(18) B. Lespinasse - "Les sources de puissance dans l'espace" Conférence faite à la deuxième Ecole d'Eté du CERS-ESRO, Oxford, 1964 - Technologie des Véhicules spatiaux, ESRO SP-12, Vol. IV: Sources de puissance dans l'espace, pp. 2-12, décembre 1966.
(19) McLean - "The Application of Ceramics to the Small Gas Turbine" ASME paper N° 70-GT-105, 9 pages
(20) J. Mason - "Working gas selection for the Closed Brayton Cycle" paper presented at the Sixth AGARD Colloquium, Cannes, March 16-20, 1964 - AGARDograph 81, pp. 223-252, Gordon & Breach Science Publishers, London, 1967.
(21) E.A. Mock - "Closed Brayton cycle system optimization for undersea, terrestrial and space applications" von Karman Institute for Fluid Dynamics, Lectures Series N° 24, pp. 58-72.
(22) F.K. Pittmann and J.D. Lafleur Jr. - "Nuclear Energy Source Limitations" paper presented at the Sixth AGARD Colloquium, Cannes, March 16-20, 1964 - AGARDograph 81, 5-24, Gordon & Breach Science Publishers, London; 1967.
(23) B.T. Resnick - "The Closed Brayton Cycle for Space Power : an Assessment" ASME paper N° 66-GT-CLC, 10, 8 p.

DEVELOPPEMENTS FUTURS DES PETITES TURBINES A GAZ

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INTRODUCTION

Ayant reçu la mission de faire une recherche prospective du propulseur optimal pour les véhicules de combat j'ai acquis une connaissance générale des différentes sources d'énergie mobiles. J'ai analysé les qualités opérationnelles de chacune d'elles et recherché les limites spécifiques de chaque architecture. A la lumière de cette analyse j'ai approfondi l'examen de solutions susceptibles de reculer ces limites.

Bien entendu mon activité a porté autant sur les moteurs Diesel que sur les turbines à gaz, ce qui m'autorise à oser un pronostic sur l'issue de la compétition qui s'est engagée entre ces deux sources d'énergie. Je ne me sens pas compétent pour ajouter quoique ce soit aux exposés précédents sur l'évolution des techniques actuelles. Je me contenterai donc de dire en quoi elles sont limitées et m'étendrai plus longuement sur quelques formules originales.

Avant d'envisager l'évolution future de la turbine à gaz je voudrais retracer rapidement l'historique du développement qui l'a conduite à son état actuel. Initialement la turbine à gaz a séduit par sa grande simplicité mécanique. Malheureusement cette simplicité se payait par des mauvaises performances qui la rendait difficilement utilisable. Cependant la nécessité d'abandonner l'hélice pour accroître la vitesse des avions lui donne un essor spectaculaire après la deuxième guerre mondiale.

Il faut noter au passage que la turbine s'impose alors pour son effet réacteur, c'est-à-dire sa faculté d'accélérer un fort débit d'air dans un faible maître couple. La recherche d'un bon rendement thermopropulsif vint beaucoup plus tard avec les réacteurs à double flux à grand taux de dilution. On s'aperçoit alors que pour des vitesses de croisière de l'ordre de 1000 Km/Heure on a intérêt à séparer la fonction propulsion et la fonction génération d'énergie qui se trouvaient liées dans les réacteurs monoflux. Cette évolution a fait progresser le rendement thermique des générateurs de gaz aux prix d'une complexité toujours croissante. On peut dire aujourd'hui que l'hélice a été remplacée par une soufflante et que le moteur à pistons a été remplacé par une turbine à gaz.

Si en 1945 on avait su que les avions commerciaux de 1970 seraient propulsés par des grandes soufflantes on aurait sans doute continué à développer des moteurs à pistons pour entraîner ces soufflantes. En effet on ne pouvait prévoir à cette époque les immenses progrès faits depuis pour ramener la consommation spécifique des turbines au niveau de celle des moteurs Diesel. Forte de ces performances la turbine a presque totalement éliminé le moteur à pistons du domaine aéronautique et tente maintenant de s'implanter dans le domaine terrestre et naval.

Il est probable que la partie sera beaucoup plus difficile à gagner.

En propulsion aérienne la turbine fonctionne dans des conditions qui lui sont particulièrement favorables :

- Puissance unitaire moyenne élevée
- Légèreté particulièrement appréciée
- Pression totale amont supérieure à pression totale avale
- Basse température ambiante
- Pas de filtration ni d'insonorisation
- Puissance et régime peu variable en croisière
- Très peu de fonctionnement à régime partiel

Les conditions de fonctionnement pour des utilisations au sol sont généralement très différentes :

- Puissance unitaire moyenne faible
- Compacité plus appréciée que la légèreté
- Pression totale amont inférieure à pression totale avale
- Température ambiante très variable
- Filtration et insonorisation presque toujours exigées
- Variations brutales de l'appel de puissance
- Fonctionnement à charge partielle fréquent

Il faut aussi constater qu'en gagnant ces performances au service de l'aéronautique la turbine à gaz y a perdu sa vertu originelle : la simplicité.

EVOLUTION DES PETITES TURBINES DE CONCEPTION CLASSIQUE

L'augmentation du rendement de cycle implique les variations simultanées de plusieurs paramètres. La diminution de la taille de la machine fait apparaître des incompatibilités entre ces variations. Par exemple, plus le rendement de compression est élevé plus le taux de compression optimal est élevé. Or une augmentation du taux de compression entraîne une réduction de la veine fluide et une dégradation du rendement de compression. De même, une augmentation du taux de compression doit s'accompagner d'une augmentation de la température devant turbine. Or la réduction de veine rend plus difficile le refroidissement des aubages.

Ces effets d'échelle, dont on pourrait citer beaucoup d'autres exemples, compromettent l'accès des turbines à gaz aux petites puissances, à moins de complications importantes : récupérateurs de chaleur, dispositifs de couplage entre turbine de travail et turbine du générateur de gaz, aubages à calage variable etc ... Tous ces dispositifs contiennent en eux les difficultés de mise au point et les sources de pannes qu'on avait espéré abandonner avec le moteur à pistons.

Il n'est pas douteux que les ingénieurs viendront à bout des difficultés actuelles, mais il ne faut perdre de vue que les succès industriels s'appuient généralement sur la solution technique la plus simple pour atteindre un objectif. C'est pour cette raison qu'à mon avis la turbine industrielle sera simple ou ne sera pas. Pour les puissances qui nous intéressent ici, le générateur de gaz pourrait être constitué d'un compresseur centrifuge supersonique et d'une turbine centripète refroidie. Les niveaux de pression, de température et de rendement qu'on saura atteindre avec ces éléments simples fixeront l'étendue des utilisations possibles. Je doute néanmoins qu'on puisse jamais atteindre simplement, c'est-à-dire sans récupérateurs de chaleur, les performances requises pour la propulsion des véhicules terrestres. En effet, de par leur conception, les petites turbomachines ont un rendement médiocre au point nominal et surtout aux charges partielles.

CAUSES DE LA LIMITATION DU RENDEMENT DES PETITES TURBINES A GAZ

On peut espérer atteindre dans l'avenir des taux de compression de 12 à 15 en un seul étage. En régime supersonique un rendement adiabatique supérieur à 0,75 semble peu réaliste. Le seul moyen de diminuer la consommation spécifique est alors d'augmenter la température devant turbine. Or les progrès sur la tenue à la température seront de plus en plus lents et difficiles et toujours accompagnés de techniques de refroidissement coûteuses, peu souhaitables sur les petites machines. Les petites machines, seront particulièrement handicapées par la détérioration des rendements de compression et de détente due à l'augmentation du taux de compression et de la vitesse de rotation.

Pour concilier simplicité et performances au point nominal, la limitation est imposée par la température admissible devant la turbine.
A charge partielle les performances des turbines sont toujours moins bonnes qu'au point nominal. Le pompage des turbo-compresseurs impose de diminuer le rapport de pression en même temps que le débit d'air.
Pour diminuer la puissance ont doit donc diminuer soit la température maximale du cycle, soit le rapport de pression, soit les deux. Dans tous les cas on diminue le rendement du cycle. Enfin, au ralenti une turbine consomme au moins 25% de sa consommation maximale, quand le moteur Diesel en consomme 2%. Le récupérateur de chaleur à haute efficacité apporte une solution qu'on paie très cher en complexité et en encombrement.

Je me propose d'examiner ci-après, la faisabilité de conceptions nouvelles ne mettant pas en oeuvre des échanges thermiques entre air et gaz.

FAISABILITE D'ARCHITECTURES DE CONCEPTIONS NOUVELLES

- Générateur de gaz à chambres de combustion tournantes

Une conception particulièrement bien adaptée à la propulsion terrestre est une turbine à chambres de combustion tournantes.
Le cycle avait été proposé par le physicien WALTER NERNST (1) au début du siècle mais avait été jugé inexploitable avec les matériaux disponibles à l'époque. Les progrès réalisés depuis sur les matériaux composites, les métaux réfractaires, les techniques de refroidissement m'ont conduit à reconsidérer la question.

- Principe du fonctionnement

Un générateur de NERNST est schématisé par un tube ABCD coudé en forme de U (figure 1) tournant autour de l'axe AD.
L'air est admis en E, comprimé par centrifugation en AB, brûlé à pression constante en BC et partiellement détendu en CD pour sortir en S avec l'énergie utilisable du cycle, qui sera recueillie dans une turbine de travail par exemple.

Pour entretenir la rotation du tube il suffit de vaincre les frottements des paliers et du fluide ambiant sur la surface extérieure du rotor : en effet, le moment cinétique emprunté par le fluide au rotor dans la branche AB lui est restitué dans la branche CD. Si de plus, l'entrée d'air et la sortie de gaz sont axiales le couple aérodynamique, appliqué au rotor, est nul.

La figure (2) représente le cycle sur un diagramme pression-volume entre les isobares P_1 et P_2 et les polytropes AB et EC.

Si U est la vitesse périphérique de la partie CD du rotor, le rapport de compression est donné sur la figure (3) pour deux valeurs du rendement polytropique de compression. A partir de U = 600 m/s les rapports de pression sont conséquents. Notons que la vitesse U n'est limitée que par la résistance du rotor puisque le nombre de Mach relatif dans le tube est indépendant de U.

Ainsi schématisé le rotor est symétrique et il faudra amorcer l'écoulement dans un sens à l'aide d'un aubage d'entrée. Une fois amorcé, l'écoulement s'entretient par différence de masse spécifique entre les colonnes AB et CD.placées dans le champ de gravitation centrifuge.
Si le système tourne à vitesse constante (et on choisira la vitesse maximale permise par les matériaux) le rapport de pression est fixé à sa valeur maximale.
Le débit de carburant fixera la température maximale du cycle T_3 et par suite la température T_4 de sortie du générateur. Pour une section de sortie donnée,nous verrons que le débit d'air de la machine dépend de T_3.
Pour éviter les frottements sur la surface externe, le rotor tournera dans un carter sous vide partiel. L'accélération dans la zone de combustion est égale à $\frac{U^2}{R}$, R étant le rayon du rotor.
On réduira donc les contraintes sur les pièces chaudes en augmentant le rayon du rotor.

EVALUATION DES PERFORMANCES

- Rendement de compression

Ce rendement a été évalué par application de lois établies pour les compresseurs centrifuges classiques. Les principales causes de perte dans un compresseur centrifuge sont les frottements sur le voile fixe du rouet, la distorsion de l'écoulement due au "glissement" de l'écoulement, et le mauvais rendement du diffuseur. Ces causes de perte sont toutes éliminées dans un générateur de NERNST :

Le canal est fermé et la vitesse relative peut être choisie faible pour réduire les frottements internes. Le glissement est annulé en périphérie quand la vitesse radiale s'annule (annulation de l'accélération de Coriolis); grâce au grand allongement radial il reste faible tout au long du canal. Le diffuseur est supprimé puisque toute la compression est assurée par augmentation d'enthalpie statique.

Compte-tenu de tout ce qui précède le rendement polytropique de compression est évalué à 0,93.

- Rendement de détente dans le générateur

Pour les mêmes raisons le rendement de détente est évalué à 0,95.

- Rendement du cycle

On s'approche ici du cycle parfait où le rendement thermique est indépendant de la température, et ne dépend que du rapport de pression. A vitesse périphérique égale et à rendement égal le rapport de pression d'un compresseur centrifuge classique est beaucoup plus élevé que celui d'un générateur de NERNST. La figure (3) donne le rapport de pression en fonction de la vitesse périphérique pour un compresseur classique ($\eta_c = 0,78$) et pour un générateur de NERNST.

Il faut noter qu'au delà de 500 m/s le rendement des compresseurs classiques est dégradé par les ondes de choc qui s'établissent dans le diffuseur. La vitesse périphérique du générateur de NERNST est limitée uniquement par la résistance du rotor. Il est évident qu'on la prendra aussi élevée que possible, puisque le rendement de compression n'en dépend pas. Une vitesse de 800 m/s doit pouvoir être atteinte avec l'acier si la structure est bien conçue. Les planches 4-5-6 donnent la consommation spécifique en fonction de la vitesse périphérique et des rendements $\eta_c = \eta_t$ pour une turbine de travail classique, de rendement polytropique 0,86. Notons le point suivant :

U= 800 m/s $P_2/P_1 = 11,5$ $\eta_c = \eta_t = 0,95$ $T_3 = 800°C$ $C_s = 170$ g/ch/h et $W_s = 240$ ch/Kg/s

Ces performances dépassent les meilleures machines actuelles.

ADAPTATION

Pour avoir le meilleur rendement du cycle à toutes les charges on choisira le fonctionnement du générateur à vitesse de rotation constante. Cette solution a en outre l'intérêt d'être la plus simple et d'annuler le temps de réponse à un appel de puissance. Le rapport de pression est alors invariable.

Deux solutions sont possibles pour moduler la puissance :

1) Maintenir constante la température maximale T_3 et faire varier le débit d'air qui est proportionnel à la puissance. Cette solution donne les meilleures consommations aux régimes réduits. Elle implique nécessairement :

- que le générateur ne pompe pas aux très faibles débits (de l'ordre du dixième du débit nominal). Ce point est sûrement acquis grâce à l'absence de capacité statorique sous pression.

- que la turbine de détente accepte une variation de débit réduit de 1 à 10 environ ce qui est exclus. On choisira donc la deuxième solution.

2) Moduler la température maximale du cycle T_3. La figure (7) donne la variation du rapport du débit réduit sortant au débit réduit entrant dans le générateur en fonc tion de T_3. On voit que entre 900°C et 450°C ce rapport varie peu, il croît de 0,6 à 0,85, alors que la puissance passe de W à 0,17 W. Ceci montre que si le distributeur de la turbine de travail est critique le débit qui traverse le générateur décroît de 1 à 0,7, ce qui correspond à une très faible désadaptation de la roue d'entrée. En effet, celle-ci aura une vitesse tangentielle en bout d'aube inférieure à 100 m/s. Sur toute la hauteur de l'aube la vitesse axiale sera très supérieure à la vitesse tangentielle. Quand cette vitesse varie de V à 0,7 V le triangle desvitesses se déforme très peu et l'incidence reste faible (figure 8)

Cette solution très simple puisqu'elle s'accommode de géométrie fixe partout, donne d'excellentes performances aux charges partielles (voir planche 9). La consommation spécifique passe de 1 à 1,26 quand la puissance passe de 1 à 0,2.

- <u>Cycle optimal</u>

En résumé, on cherchera à réaliser une turbine constituée d'un générateur de NERNST tournant à vitesse périphérique constante 800 m/s $\frac{P_2}{P_1}$ = 11,5 avec une température maximale de 900°C, alimentant une turbine de travail classique à géométrie fixe. On recherchera $\eta_c = 0,93$
$\eta_t = 0,95$ pour le générateur
$\eta_t = 0,86$ pour la turbine de travail

les performances seront alors :

Au point nominal : W_s = 240 KW/Kg/s = 325 ch/Kg/s
C_s = 230 g/KW/h = 170 g/ch/h
Au 1/5 de la puissance : C_s = 290 g/KW/h = 215 g/ch/h

- <u>Formes constructives</u>

Nous avons vu que les performances d'un générateur de NERNST sont liées à l'obtention de grandes vitesses périphériques. Le problème est donc de réaliser une enceinte tournant à 800 m/s qui sera le siège d'une combustion.

Deux structures ont été étudiées jusqu'à présent :

1) <u>Rotor à chambres de combustions axiales</u> :
Les figures 10 et 11 montrent une coupe perpendiculaire à l'axe et une coupe méridienne. L'aubage d'entrée et l'aubage de sortie sont reliés par 6 ou 8 canaux comportant chacun une chambre de combustion. Ces canaux sont délimités par une voûte extérieure travaillant à la traction et par une voûte intérieure travaillant à la compression. Ces voûtes sont suspendues à des bras issus du moyeu. La partie cylindrique a été calculée et dimensionnée mais la partie compresseur et la partie turbine sont inaccessibles au calcul. De plus la partie turbine est extrêmement critique pour la résistance du rotor.

2) <u>Rotor à chambres de combustions radiales</u> :
Le dessin est moins élégant que le précédent et conduit à une machine beaucoup plus encombrante. Néanmoins il respecte mieux les qualités spécifiques du générateur de NERNST. De plus la structure est beaucoup plus simple à calculer et à réaliser, la combustion est plus facile à organiser et le refroidissement des parties chaudes plus aisé.

On est parti de l'idée que les rendements de compression et de détente seront d'autant meilleurs que les canaux seront allongés. On s'est alors imposé de donner à la roue d'entrée un diamètre égal au 1/10è du diamètre du rotor. Le débit de la machine est alors limité par la roue d'entrée et peut être traité par un petit nombre de chambres de combustion, situées à l'extrémité du bras, dans lesquelles se font la compression et la détente (planche 15). La description qui suit concerne un générateur de 500 ch à trois bras vissés dans un moyeu central. Chaque bras est constitué d'une enveloppe extérieure froide travaillant à la traction et d'un tube télescopique interne chaud dont les éléments travaillent au

flambage en appui sur la structure froide. Le tube intérieur débouche à l'intérieur de la coupole.
La compression s'effectue dans l'espace annulaire entre les deux tubes, la combustion se fait dans la coupole périphérique et la détente dans le tube interne.

Résistance du tube extérieur

Pour s'assurer de la faisabilité d'une telle structure un programme a été mis au point pour calculer les contraintes dans un corps de révolution à surface interne cylindrique et à épaisseur évolutive.(figure 12)
On s'est donné les valeurs numériques suivantes :

Rayon de giration du sommet du bras	= 0,50 m
Rayon intérieur du bras	= 0,05 m
Densité du métal	= 8
Vitesse périphérique au sommet	= 800 m/s

La figure (12) donne une évolution d'épaisseur et l'évolution correspondante des contraintes suivant les courbures principales : n_φ dans le plan méridien et sa perpendiculaire n_θ .
On voit que la coupole qui est la partie la plus chaude (environ 350°C) est la moins chargée mécaniquement. Le bras travaille à la traction pure et pourrait être allégé par emploi de fibres à haute résistance. Quoiqu'il en soit la faisabilité d'une telle structure avec les aciers actuels est démontrée. Le bras réel supportera un tube télescopique interne faisant fonction de turbine. Celui-ci modifie peu le résultat précédent compte tenu que ce tube peut être mince et que le bras est renforcé par les nervures de centrage du tube interne (figure 13).

Niveau thermique

La température du tube extérieur sera sensiblement celle de l'air en cours de compression, soit l'ambiante au moyeu et 350°C dans la coupole. Le tube interne aura des contacts linéaires avec le tube externe pour limiter la conduction vers ce dernier. En aval de la zone de combustion une face de ce tube sera à 900°C et l'autre sera refroidie par l'air en compression. Le fractionnement de ce tube sera choisi en fonction de la contrainte admissible aux appuis de chaque élément travaillant au flambage. La coupole interne sera soumise au rayonnement de la flamme. C'est une chance que cet endroit du générateur soit le seul où la pression statique côté air est supérieure à celle côté gaz. On pourra donc organiser simplement un refroidissement par transpiration ou par film.

Combustion

La combustion constitue le point le plus aléatoire du projet. On a cependant quelques raisons d'être optimiste.
Le problème est rendu difficile par le faible volume disponible pour effectuer la combustion. Il s'agit donc de réaliser un foyer à haute intensité, c'est-à-dire d'accélérer l'échange énergétique entre produits qui ont réagi et produits qui vont réagir chimiquement. Dans une chambre classique l'échange est assuré par mélange turbulent entre gaz chauds et air primaire. Dans une chambre tournante le champ de gravitation intense (plus de 100.000 g dans le cas présent) permet de forcer une pénétration des couches froides dans les couches chaudes pourvu que l'air soit injecté dans la zone de combustion par nappes perpendiculaires au vecteur accélération et dans le sens des potentiels décroissants.

C'est sur ce principe que la chambre a été imaginée (figure 14). On cherchera à localiser la combustion dans une sphère entourée d'une couche sphérique d'air froid. Cette enveloppe d'air est alimentée par une couronne de tuyères, également inclinées sur la surface de la coupole interne, qui induisent un moment cinétique autour de l'axe de bras. La stabilité gyroscopique acquise par les particules fluides entraîne un basculement du moment cinétique dans le référentiel du rotor autour d'un axe parallèle à l'axe de rotation. Ce brassage vigoureux devrait réaliser la dilution sur une très courte distance. Le carburant peut être introduit dans la zone de combustion en phase liquide fractionnée pour éviter les trop fortes pressions d'injection, ou en phase gazeuse.

Conlusion

La figure 15 montre une organisation possible d'un générateur à 3 bras. Cette première étude de faisabilité montre que rien n'interdit à priori de réaliser une chambre de combustion tournant à 800 m/s. L'architecture proposée n'est sûrement pas optimale. Rappelons encore que pour une vitesse périphérique donnée, plus le diamètre du rotor est grand plus faible est l'accélération dans la chambre de combustion et meilleurs sont les rendements de compression et de détente.

L'organisation générale de la machine n'a pas été abordée et en particulier la façon d'entretenir la rotation du générateur et de créer le vide dans le carter. A ma connaissance aucune expérimentation n'a été tentée sur ce principe exceptés des travaux de PRATT et WHITNEY sur la combustion dans un champ de gravitation intense qui mettent en évidence un accroissement très important des vitesses de combustion.

TRANSMISSION DE LA PUISSANCE D'UNE TURBINE A GAZ

L'avenir de la turbine à gaz en propulsion terrestre dépend beaucoup des solutions qu'on apportera au problème de la transmission de puissance. La machine de propulsion idéale se caractérise par une relation hyperbolique entre le couple et la vitesse qui permet l'utilisation de toute la puissance installée à chaque régime. De ce point de vue le moteur Diesel suralimenté est une mauvaise machine dont le couple croît avec la vitesse de rotation. La turbine à deux arbres se comporte comme un convertisseur de couple de rapport 2. Ceci suggère d'associer à la turbine une boîte automatique classique dont on aura retiré le convertisseur de couple hydrocinétique. Des problèmes surgissent lors de changement de rapport : en effet, la fonction de transfert d'une turbine à gaz est très différente de celle d'un moteur Diesel. On notera par exemple que le danger de survitesse de la turbine libre impose pratiquement un passage de vitesse sous charge. Ceci étant, la grande énergie cinétique restituée par la turbine libre quand elle passe d'un régime à un régime inférieur provoque une fatigue importante des embrayages de la boîte. Il n'est pas douteux que la transmission d'une turbine à gaz relève d'une solution spécifique.

Fonctionnellement la transmission hydrostatique résout assez bien le problème, mais l'ensemble est très hétérogène : la turbine est légère et se passe d'un circuit de refroidissement; la formule hydrostatique est lourde et dégage beaucoup de calories. Je pense qu'une solution d'avenir doit respecter les impératifs suivants :

- la transmission doit être automatique
- la puissance calorifique dissipée dans l'huile de la transmission doit être suffisamment faible pour être évacuée par le radiateur d'huile de la turbine.
  Ceci exclut pratiquement tous les convertisseurs hydrocinétiques et les éléments hydrostatiques qui nécessiteraient un encombrant dispositif de refroidissement.
- le flux issu du générateur de gaz doit être utilisé comme fluide de travail du convertisseur de couple afin d'évacuer dans l'échappement les pertes de transmission.
  La solution décrite ci-après respecte ces impératifs :

Convertisseur de couple aérodynamique

L'idée de combiner l'effet convertisseur de couple de deux turbines de travail est due à Monsieur KRONOGARD (SAE Transaction 1960).

La transmission de Monsieur KRONOGARD est schématisée sur la figure (16). Le générateur alimente successivement un aubage fixe à calage variable, et deux aubages mobiles contrarotatifs. Ainsi au lieu de transformer une énergie mécanique reçue (couple moteur x vitesse moteur) en une énergie désirée (couple récepteur x vitesse récepteur), on produit directement l'énergie désirée (couple récepteur x vitesse récepteur). La turbine I et la turbine II attaquent respectivement le pignon solaire et la couronne d'un train différentiel dont le porte-satellites est relié au bâti par une roue libre. La prise de mouvement s'effectue sur la couronne.

Nous analyserons le fonctionnement de cet ensemble en imaginant que l'arbre de sortie est calé sur un frein que l'on desserre progressivement. Le diagramme (17) donne l'évolution du couple et de la vitesse de sortie. Quand le couple résistant décroît de C_0 à C_1, la vitesse de sortie croît de 0 à N_1. Pendant cette phase la turbine II est alimentée par le tourbillon résiduel de la turbine I. Les couples aérodynamiques de sens contraire tendent à entraîner le porte-satellites dans le sens interdit par la roue libre. Les vitesses des turbines I et II sont donc proportionnelles pendant cette phase.
Pour C_1 et N_1 le couple appliqué au porte-satellites s'annule, la roue libre décolle et le couple résistant de la turbine II s'annule. La turbine I effectue alors tout le travail avec le handicap de la perte de charge créée par le moulinage de la turbine II. Pour éviter cet inconvénient, nous expérimentons avec la division Hispano Suiza de la SNECMA une nouvelle association de deux turbines contrarotatives. Quatre idées directrices ont conduit à cette configuration :

- pendant la phase où une seule turbine travaille, le fonctionnement de celle-ci ne doit pas être gêné par la turbine qui ne travaille pas, qui sera donc placée en amont.
- la phase, où les deux turbines se partagent la chute d'enthalpie, est présumée donner le meilleur rendement global de détente. Cette phase assurera donc la demi plage des vitesses supérieures. En effet, en propulsion terrestre la puissance appelée est statistiquement fonction croissante de la vitesse d'avancement. Pour la plupart des profils de parcours la meilleure consommation moyenne est obtenue en plaçant le meilleur rendement à une vitesse d'avancement supérieure à la demi vitesse maximale.
- dans tous les cas de fonctionnement les trois rendements de grille devront rester acceptables. Les angles d'incidence sur les aubages ne devront pas sortir d'une plage évaluée par

la méthode de AINLEY et MATHIESEN. Pour satisfaire à cette condition, le calage de l'aubage fixe devra être réglable.

- dans la phase contrarotative la liaison cinématique entre les deux turbines doit permettre un bon rendement de détente sur toute la demi plage de vitesse. Pour ce faire la chute d'enthalpie doit être transférée progressivement d'une turbine à l'autre. Il est donc évident que les vitesses de rotation des deux turbines devront varier en sens inverse, contrairement à la solution KRONOGARD où les deux vitesses sont proportionnelles dans la phase contrarotative.

La figure (18) représente schématiquement un convertisseur de couple respectant ces quatre principes. Il est constitué par un aubage distributeur à calage réglable suivi de deux aubages mobiles contrarotatifs I et II fonctionnant en série. Les deux turbines contrarotatives sont reliées par un train épicycloïdal de raison k.

La turbine I attaque la couronne extérieure par l'intermédiaire d'un réducteur de rapport k_1. La turbine II attaque le planétaire central par l'intermédiaire d'un réducteur inverseur de rapport k_2. Une roue libre RL interdit la contre rotation de la turbine I. Le mouvement est pris sur l'arbre du porte-satellites.

Soient C_1 et C_2 les couples développés par les turbines I et II. En partant de l'arrêt et en augmentant la vitesse de l'arbre de sortie on voit apparaître deux phases de fonctionnement (figure 19).

Phase I. Si $k\,k_2C_2 > k_1\,C_1$ la turbine I reste bloquée et sert de distributeur à la turbine II. Pendant la montée en vitesse C_1 est sensiblement constant et C_2 diminue jusqu'à

$C_2 = \dfrac{k_1\,C_1}{kk_2}$, couple de débloquage de la roue libre et fin de la phase I.

Phase II. A partir de cette vitesse les deux turbines tournent ensemble en se répartissant le travail pour assurer l'équilibre du train épicycloïdal.

$$k = \frac{k_1C_1}{k_2C_2}$$

La turbine I accélère et la turbine II ralentit conformément au quatrième impératif.

Détermination des profils

Pour définir les profils on s'est imposé trois points de fonctionnement :

- fonctionnement A

  Les deux turbines tournent ensemble et la vitesse absolue de sortie de la turbine II est axiale.

- fonctionnement B

  C'est le point de transition entre la phase I et la phase II, caractérisé par l'arrêt de la turbine I et $k = \dfrac{k_1C_1}{k_2C_2}$

- fonctionnement C

  La roue libre bloque la turbine I et la vitesse absolue du fluide à la sortie de la turbine II est axiale.

Il serait trop long d'écrire l'ensemble des équations de compatibilité. Il suffit de savoir qu'on peut définir par itération les triangles de vitesse en A,B,C et les profils des trois grilles d'aubes quand on se donne :

- les caractéristiques de l'écoulement issu du générateur de gaz
- la valeur du paramètre $k\,\dfrac{k_2}{k_1}\cdot\dfrac{U_{1a}}{U_{2a}}$, où U_{1a} et U_{2a} sont respectivement les vitesses périphériques des turbines I et II au point A.
- la valeur de la variation du calage du distributeur
- certaines caractéristiques dimensionnelles de la veine fluide.

Résultats des calculs

L'angle d'incidence i a été calculé pour les points A, B, C :

Distributeur :

A	i	= + 8°
B	i	= + 1°
C	i	= - 4°

Turbine I :

A	i	=	- 25°
B	i	=	+ 20°
C	i	=	+ 14°

Turbine II :

A	i	=	- 37°
B	i	=	- 21°
C	i	=	+ 28°

Le calcul des pertes fait à partir de ces incidences a donné des résultats suffisamment encourageants pour lancer la fabrication d'une machine d'essai dimensionnée pour un débit de 6 kg/s, et prévue pour des essais aérodynamiques en écoulement froid. L'installation, prévue pour permettre un réglage du calage des aubages fixes et mobiles, doit donner l'influence respective de ces trois paramètres.
Les deux turbines peuvent être freinées indépendamment jusqu'au calage.
A l'issue des essais aérodynamiques, la liaison cinématique pourra être établie entre les turbines afin d'analyser les régimes transitoires.

Conclusion :

La transmission aérodynamique semble donc être la plus simple et la plus compacte pour une turbine à gaz.
L'automatisme est acquis au prix d'une roue libre qui se bloque et se débloque sans choc et d'un asservissement du calage du distributeur à la vitesse de sortie.
Un rapport de conversion de six, peut être obtenu en choisissant une vitesse nominale un peu supérieure à N_A.
Le rendement énergétique ne sera connu qu'après les essais. Il sera à comparer au produit du rendement d'une turbine libre classique par le rendement de la transmission associée.

LA TURBINE A GAZ FACE AU MOTEUR DIESEL

Pour fixer les idées nous placerons la comparaison sur le terrain de la propulsion des trains routiers. La puissance installée sur ces véhicules est actuellement de 300 chevaux; elle devrait atteindre 800 chevaux dans les années qui viennent.
Pour les turbines de camion l'unanimité s'est faite autour du cycle peu comprimé à fort taux de récupération (90%). Nous admettrons donc que la turbine est du type à régénérateurs tournants (Ford 707).
Pour les moteurs Diesel nous envisagerons deux cas :

a - les moteurs à quatre temps actuellement sur le marché fonctionnant à une pression moyenne de 11 Kg/cm2 et à 2500 t/mn.

b - les moteurs à quatre temps à bas rapports volumétriques en cours de développement fonctionnant à une pression moyenne de 25 kg/cm2 à 2500 t/mn.

Ces hypothèses conduisent aux caractéristiques suivantes :

Pour 300 chevaux :

hypothèse a - cylindrée 10 litres (6 ou 8 cylindres)
poids = 800 kg

hypothèse b - cylindrée 4,5 litres (3 ou 4 cylindres)
poids = 400 kg

Pour 800 chevaux :

hypothèse a - cylindrée 27 litres (au moins 16 cylindres)
poids = 2500 kg

hypothèse b - cylindrée 12 litres (6 ou 8 cylindres)
poids = 1 000 kg

Il existe des camions expérimentaux qui permettent de comparer la turbine et le Diesel type a pour une puissance de 300 chevaux. Si l'on tient compte du filtre à air et du silencieux la turbine est plus encombrante mais moins lourde que le moteur Diesel. Elle consomme plus de carburant à toutes les charges et spécialement aux faibles charges et au ralenti. Le bilan est donc plutôt favorable au moteur Diesel.
Au contraire pour 800 chevaux la turbine est beaucoup plus avantageuse que le moteur Diesel type a.
Avec l'hypothèse b le Diesel sera sans doute plus avantageux pour toute la plage de puissance.

MACHINES HYBRIDES

Les machines volumétriques sont limitées par leur incapacité à aspirer de forts débits d'air sous un faible encombrement. Le débit volumique est proportionnel à la cylindrée et à la vitesse de rotation.
Les lois de similitude montrent qu'une machine k fois plus lourde qu'une machine homothétique de débit volumique Q_v, ne peut traiter qu'un débit volumique $k^2/3 \times Q_v$. Les petites machines seront donc relativement plus puissantes que les grosses.
Pour accroître le débit d'air on agira de préférence sur la densité de l'air d'admission. Il s'agit donc de trouver un dispositif qui élève la densité de l'air au meilleur prix. La turbine à gaz est bien conçue pour jouer ce rôle.
Le succès de la turbo suralimentation des moteurs Diesel est un exemple de cette évolution. A ce sujet il faut noter que la turbosoufflante est encore considérée comme un accessoire du moteur Diesel, qui peut être utilisé sans elle.
Il y aura intérêt à concevoir desmachines hybrides où la partie basse pression du cycle est traitée par une turbine à gaz, et la partie haute pression par une machine volumétrique. Il faudra exploiter les aptitudes respectives de ces deux conceptions, et partager judicieusement les fonctions.

Fonctions à attribuer à la partie turbine d'une machine hybride

Pour respecter la simplicité de la turbine en assurant une grande durée de vie et un faible prix de revient il faudrait respecter les impératifs ci-après :

- réduire au maximum la température des gaz admis aux turbines
- éliminer tous les dispositifs de géométrie variable
- éliminer toute liaison cinématique avec l'arbre de la turbine
- en conséquence, on ne prélèvera aucune puissance mécanique sur l'arbre de la turbine.

La seule fonction de la turbine est alors de comprimer l'air d'admission de l'élément volumétrique. Il faudra donc être très exigeant sur le rendement de compression et de détente. L'aérodynamique de ces machines sera aussi élaborée que celle des turbines destinées à fournir de la puissance. Le problème de l'adaptation sera d'ailleurs beaucoup plus simple puisque la machine sera optimisée pour une puissance nulle. On obtient ainsi une température d'autonomie extrêmement basse comme le montre l'exemple suivant :

température ambiante	$T_o = 288°\ K$
rendement polytropique de compression	$\eta_c = 0,85$
rendement polytropique de détente	$\eta_t = 0,85$
pertes de charge totales du circuit (admission, chambre de combustion, échappement)	$\sum \frac{\Delta p}{p} = 10\ \%$

La puissance dissipée dans les roulements a été négligée, les pertes de charges ayant été surévaluées.

rapport de pression du cycle	$\frac{P_2}{P_1} = 6$
température sortie compresseur	$T_2 = 523°\ K = 250°\ C$
température devant turbine	$T_3 = 700°\ K = 427°\ C$
augmentation de température dans la chambre de combustion	$\Delta T_c = 177°\ C$

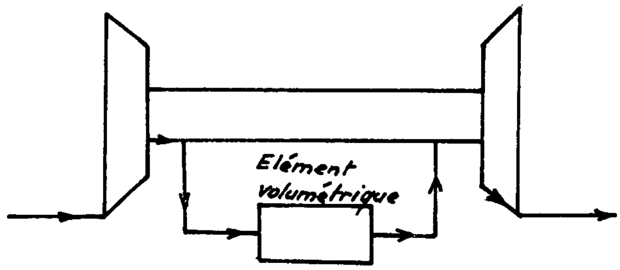
Ces basses températures permettent de très longues durées de vie avec des matériaux courants.

Fonctions attribuées à la partie volumétrique

C'est à elle qu'il incombe de fournir la totalité de la puissance mécanique de la machine hybride. La technologie nous impose de respecter les limites ci-après :

- une pression maximale de cycle imposée par la tenue mécanique
- une température moyenne maximale imposée par la tenue thermique
- une température maximale d'échappement
- pour les machines Diesel, une température minimale en fin de compression d'autant plus basse que la pression est élevée.

Notons que l'élément volumétrique fonctionne en dérivation sur le circuit haute pression de l'élément turbine comme le montre le schéma ci-après :



On abandonne ainsi la possibilité de récupérer toute l'énergie disponible dans le cycle mixte :

- énergie de bouffées, libérée à l'ouverture des clapets d'échappement
- amélioration du rendement volumétrique obtenu par augmentation du rapport de balayage
- énergie excédentaire récupérable dans les gaz d'échappement de l'élément volumétrique après prélèvement de l'énergie de compression

La faible valeur relative de ces énergies ne justifie pas les complications constructives que nécessiterait leur récupération. Ces conditions aux limites étant fixées on cherchera la valeur optimale du taux de suralimentation de l'élément volumétrique.
Une étude complète a été faite pour les moteurs Diesel à bas rapport volumétrique dont l'exposé sortirait du cadre de ce séminaire.
On se contentera de dire que pour une technologie de moteur donnée, la pression moyenne indiquée croît proportionnellement à la pression de suralimentation jusqu'à 12 bars environ.
L'évolution du fluide dans le cyclindre se fait à plus forte densité et à plus basse température que dans un cycle classique. Ceci explique que les charges thermiques n'augmentent pas et que l'énergie transmise à l'eau de refroidissement soit beaucoup plus faible en valeur relative. Ainsi on récupère dans l'eau ce qu'on perd en abaissant le rapport de compression, et donc le rendement du cycle Diesel.

CONCLUSION

Il semble acquis qu'un marché important va s'ouvrir à des moteurs légers de puissances comprises entre 300 et 1 500 chevaux. Ces moteurs seront essentiellement destinés à propulser des engins terrestres, c'est-à-dire à des usagés industriels. Un poids de 1,5 kg/ch devrait satisfaire la plupart des utilisateurs. Un poids inférieur sera certes apprécié, mais pas à n'importe quel prix. En particulier la consommation de carburant restera un poste important du coût d'exploitation.

D'autre part, les dispositifs d'admission d'air, de filtration, d'insonorisation et d'échappement sont des équipements onéreux et encombrants. Proportionnels au débit d'air, ils seront trois fois plus importants pour une turbine que pour un moteur Diesel.

Le prix au cheval des moteurs Diesel actuels reste un objectif pour la turbine à gaz. Or ce prix devrait encore décroître considérablement avec la haute suralimentation en même temps que le poids au cheval, qui devrait atteindre 1 kg/ch dans un futur proche.

Pour toutes ces raisons je pense qu'à moyen terme, l'atout maître de la turbine restera son extrême légèreté, atout que lui fait perdre le récupérateur de chaleur. Le champ des utilisations où la légèreté est impérative, restera donc la chasse gardée des turbines à cycle simple.
Ce champ qui contient la propulsion aérienne n'interfère que très peu avec le domaine terrestre qui restera sans doute le fief du moteur Diesel très suralimenté.
La turbine à gaz à récupérateur qui soutient difficilement la comparaison avec le Diesel d'aujourd'hui, doit se préparer à l'offensive du Diesel de demain.

Il n'est cependant pas douteux qu'à plus ou moins long terme, on trouvera la manière de décrire élégamment le cycle d'une turbo-machine, si satisfaisant pour l'esprit.

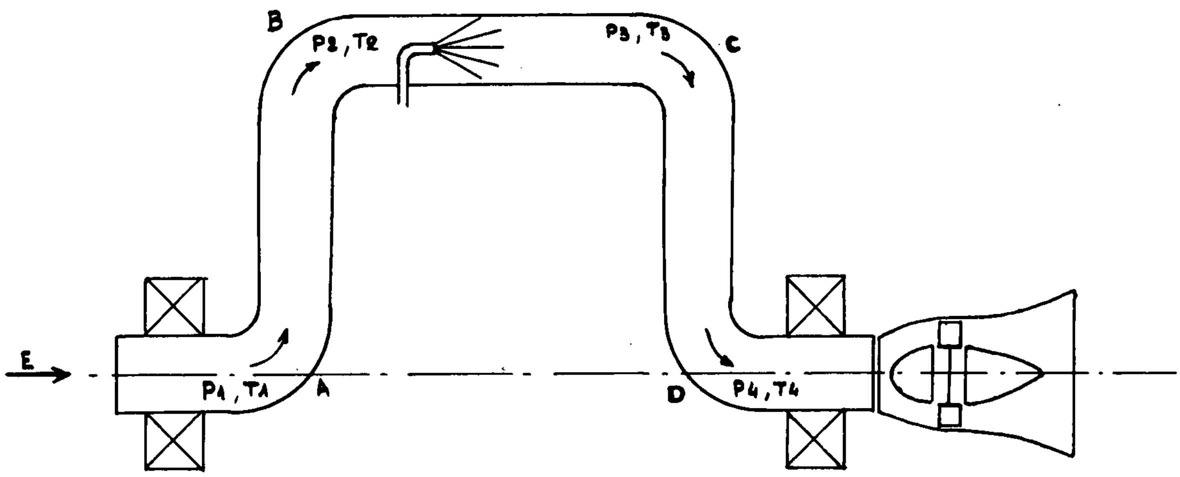


Fig.1 Schéma de principe d'un générateur de NERNST
Principle diagram of a NERNST generator

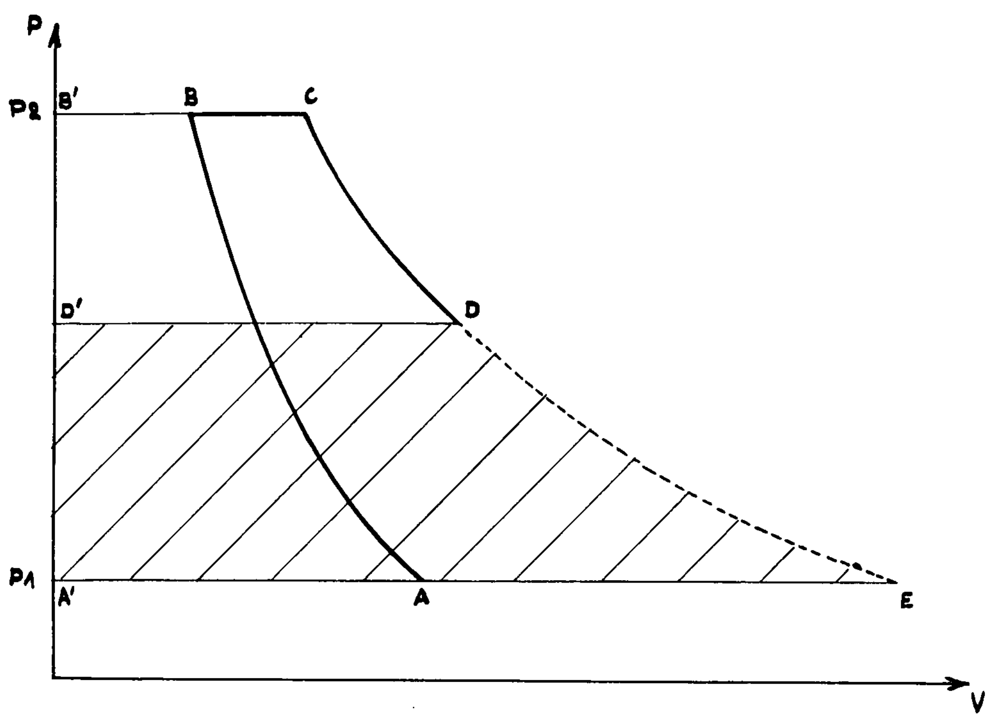


Fig.2 Diagramme de l'évolution du fluide
Fluid proceeding diagram

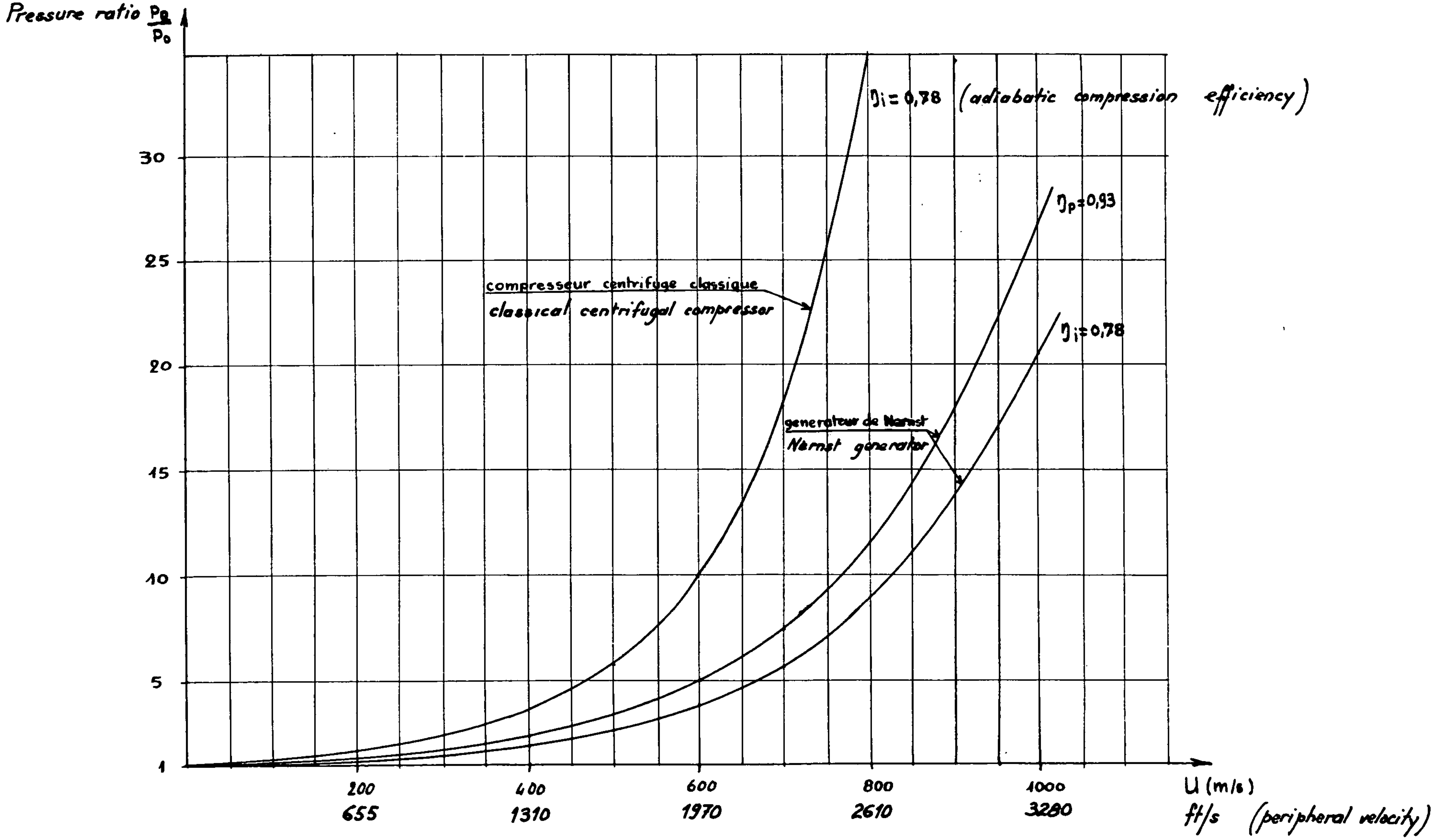


Fig.3 Taux de compression fonction de la vitesse périphérique et du rendement de compression pour un compresseur classique et pour un générateur de NERNST
Pressure ratio in terms of peripheral velocity and compression efficiency for a classical compressor and a NERNST generator

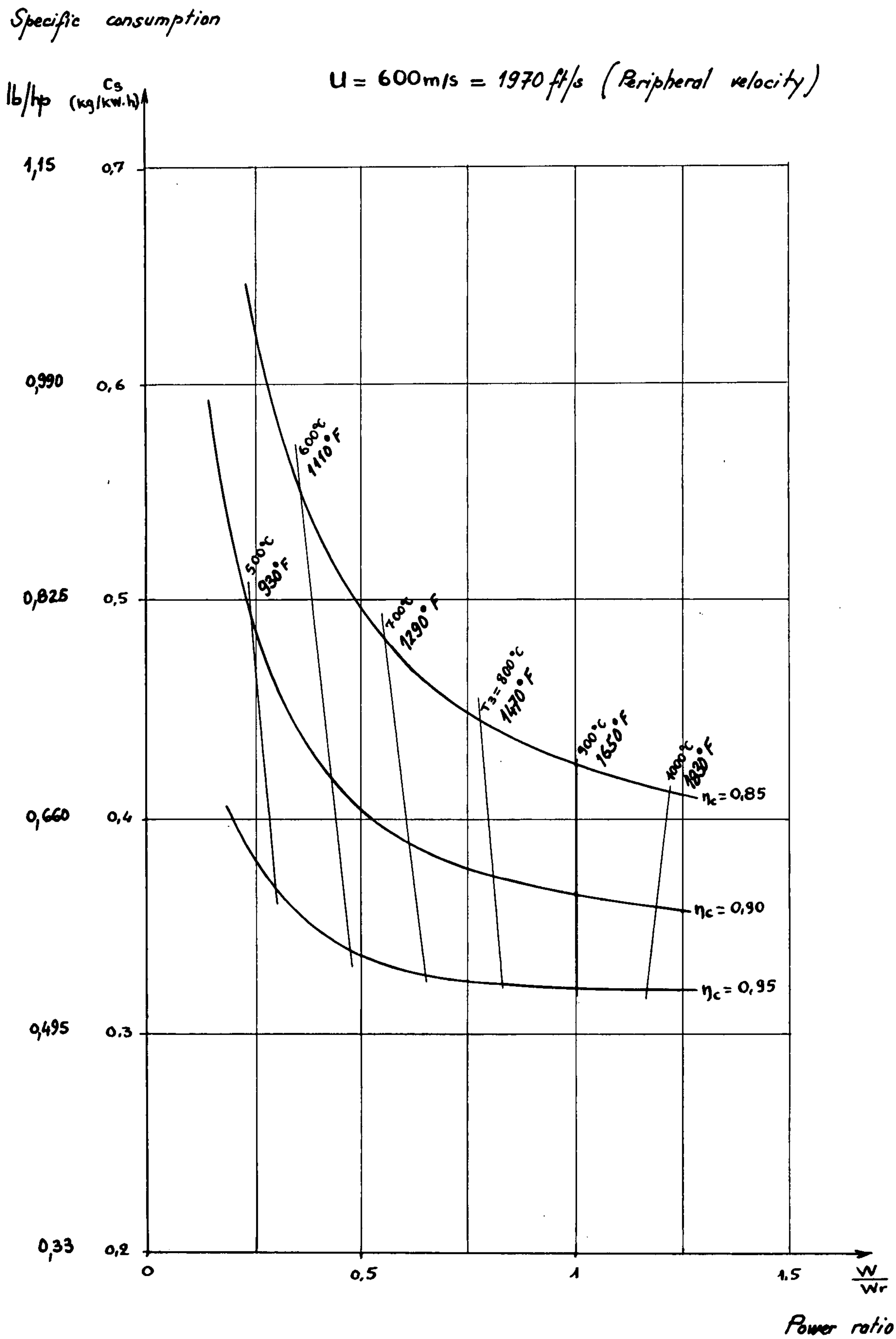


Fig.4 Consommation spécifique d'une machine constituée d'un générateur de NERNST et d'une turbine de travail de rendement polytropique 0,86 en fonction de la température

Specific consumption in terms of temperature, of a NERNST generator breeding a free turbine of 0.86 polytropic efficiency

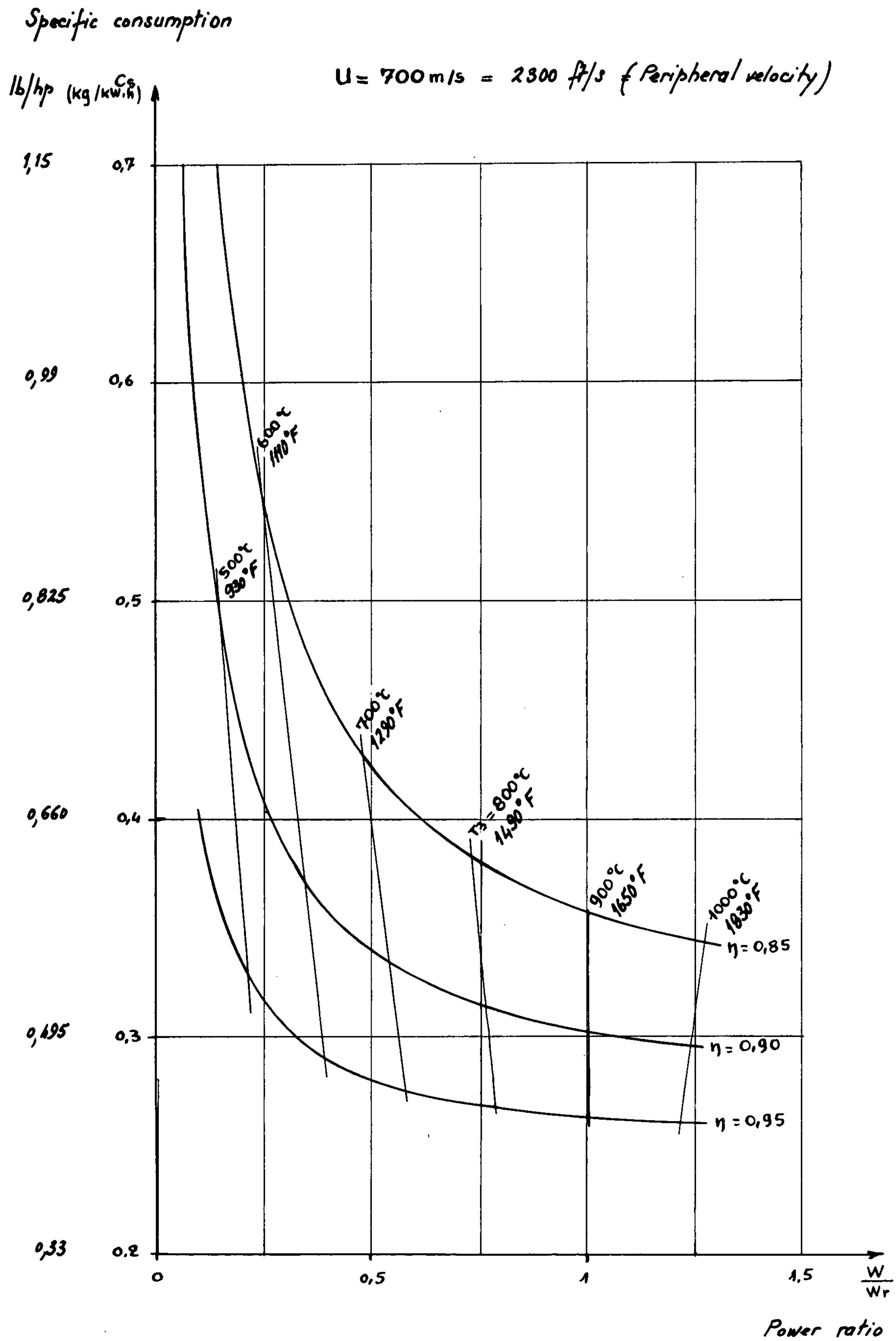


Fig.5 Consommation spécifique d'une machine constituée d'un générateur de NERNST et d'une turbine de travail de rendement polytropique 0,86 en fonction de la température

Specific consumption in terms of temperature, of a NERNST generator breeding a free turbine of 0.86 polytropic efficiency

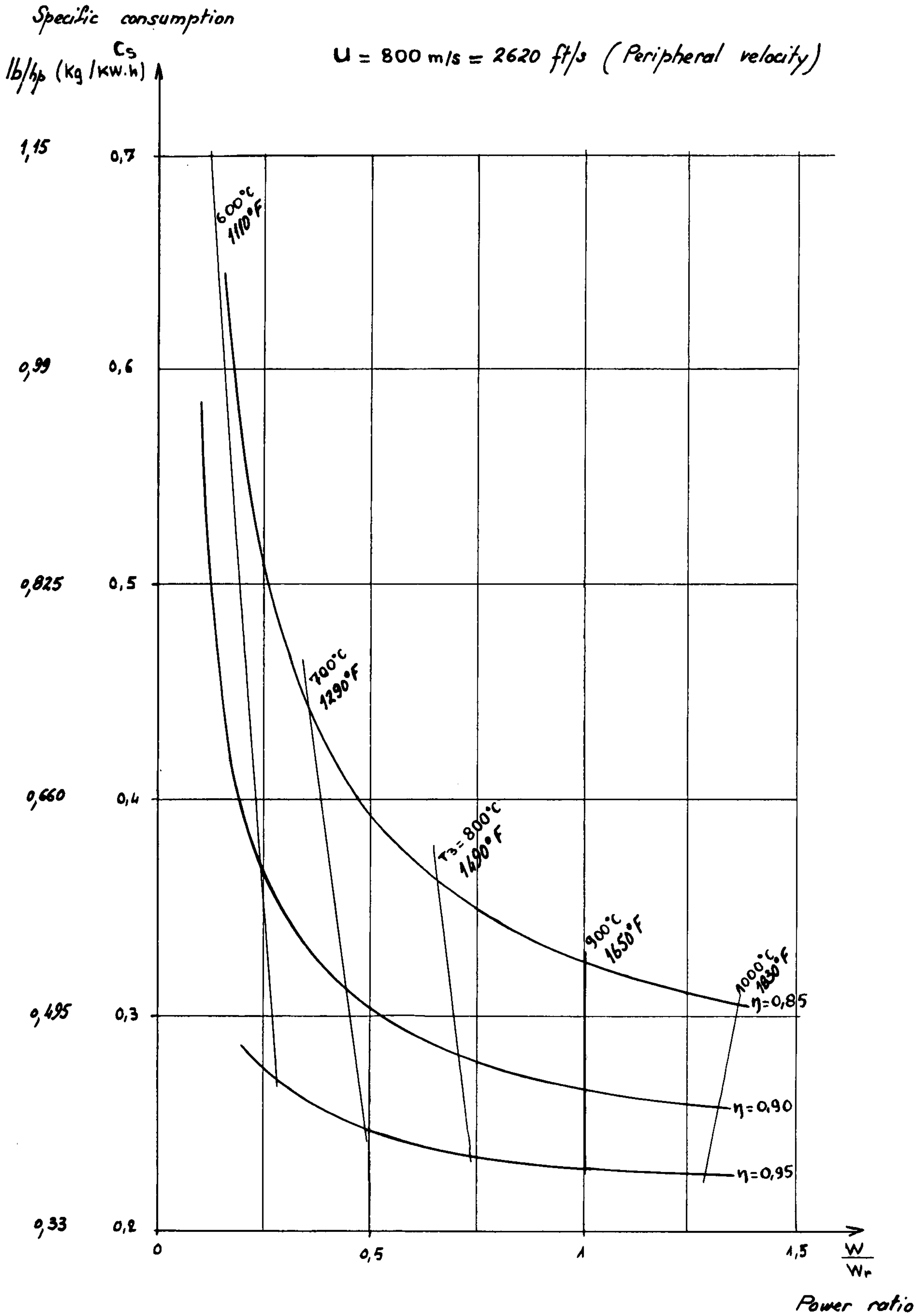


Fig.6 Consommation spécifique d'une machine constituée d'un générateur de NERNST et d'une turbine de travail de rendement polytropique 0,86 en fonction de la température

Specific consumption in terms of temperature, of a NERNST generator breeding a free turbine of 0.86 polytropic efficiency

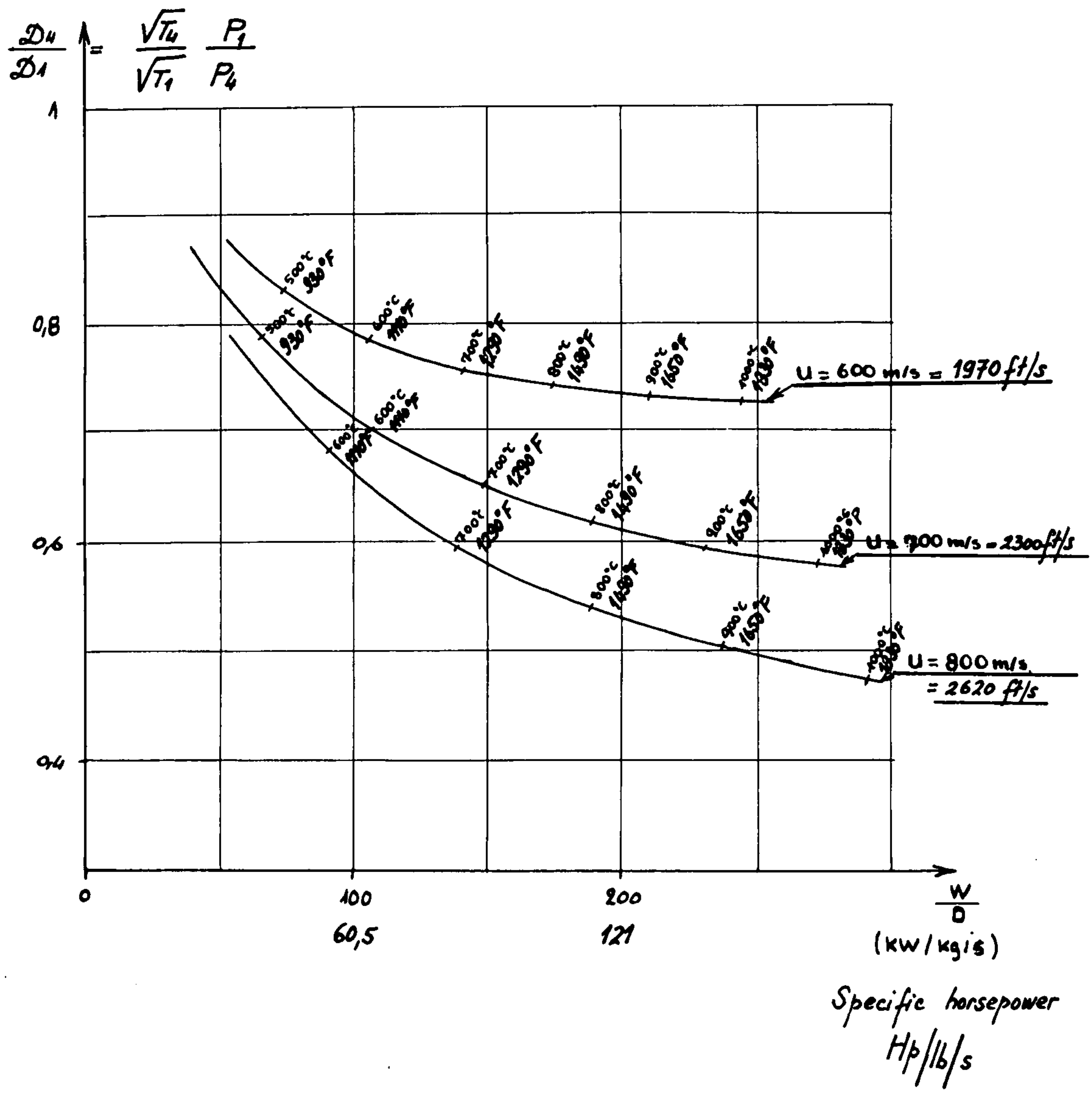


Fig.7 Rapport du débit réduit refoulé D_4, au débit réduit aspiré D_1, en fonction de la température maximum de cycle et de la vitesse périphérique pour un rendement de compression et de détente de 0,95

Ratio of the inlet value of the quantity $D = \frac{Q\sqrt{T}}{P}$, to its outlet value, in terms of maximal gas temperature and peripheral velocity for a NERNST generator. (Q massflow)

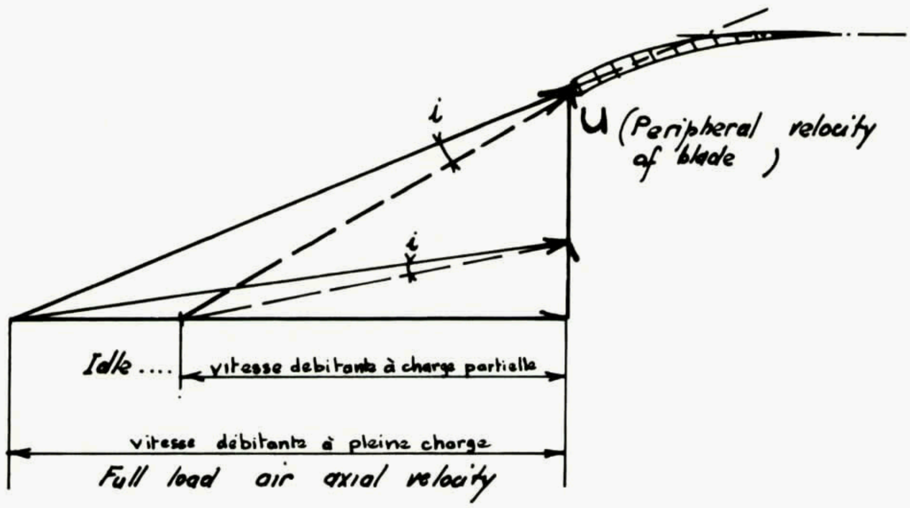


Fig.8 Valeur maximale de l'incidence i sur l'aubage d'entrée du générateur de NERNST
Maximal value of incidence angle i of the airflow at rotor inlet of the NERNST generator, breeding a constant cross-section distributor

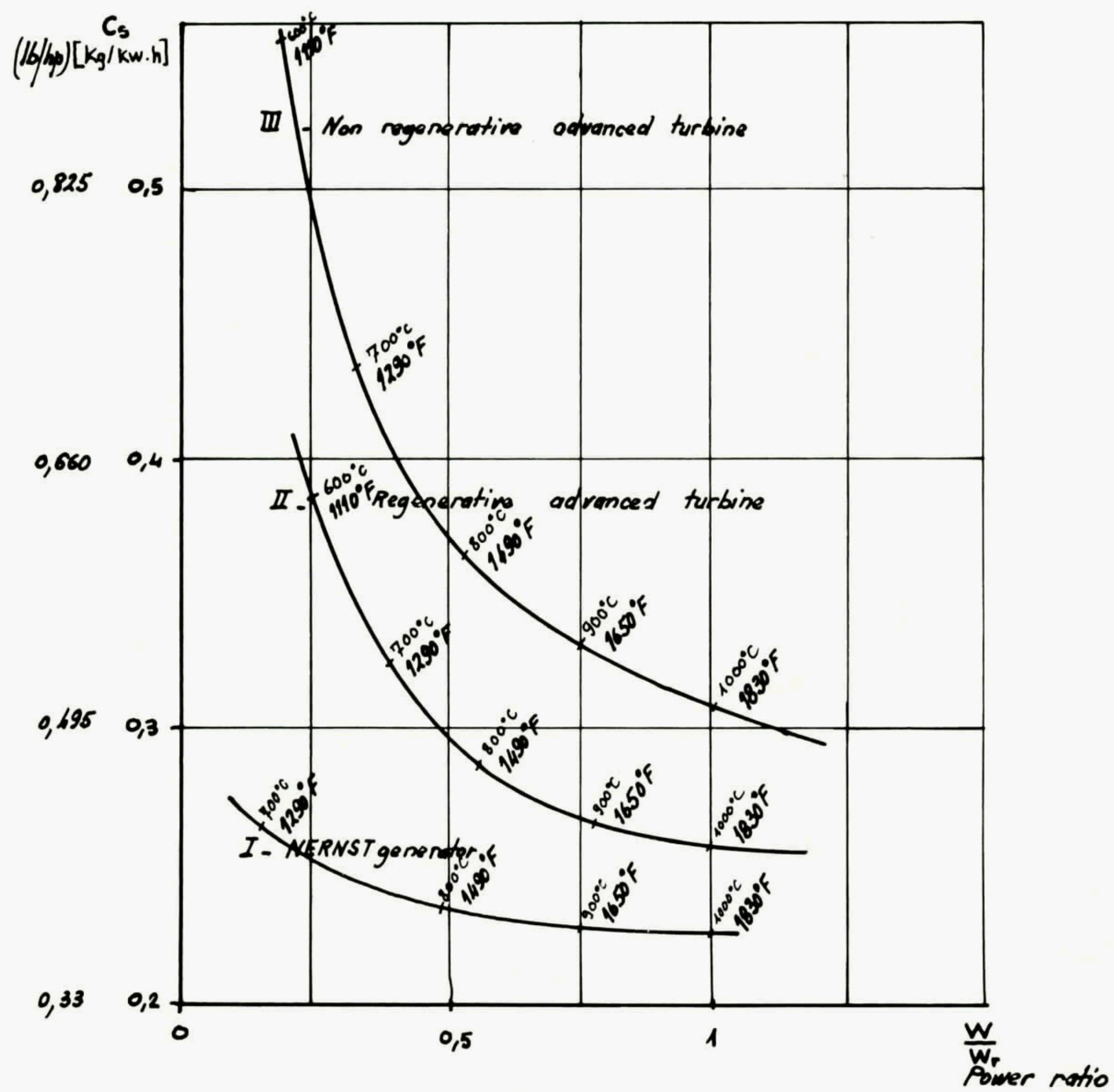


Fig.9 Compared part load specific fuel consumptions for the same nominal temperature of 1830°F

I – NERNST generator: U = 2600 ft/s, $\eta_t = \eta_c = 0.95$

II – regenerative advanced turbine: Pressure ratio = 4, regenerator efficiency = 90%, $\eta_c = 0.825$, $\eta_t = 0.87$

III – non regenerative advanced turbine: Pressure ratio = 10, $\eta_c = 0.825$, $\eta_t = 0.87$

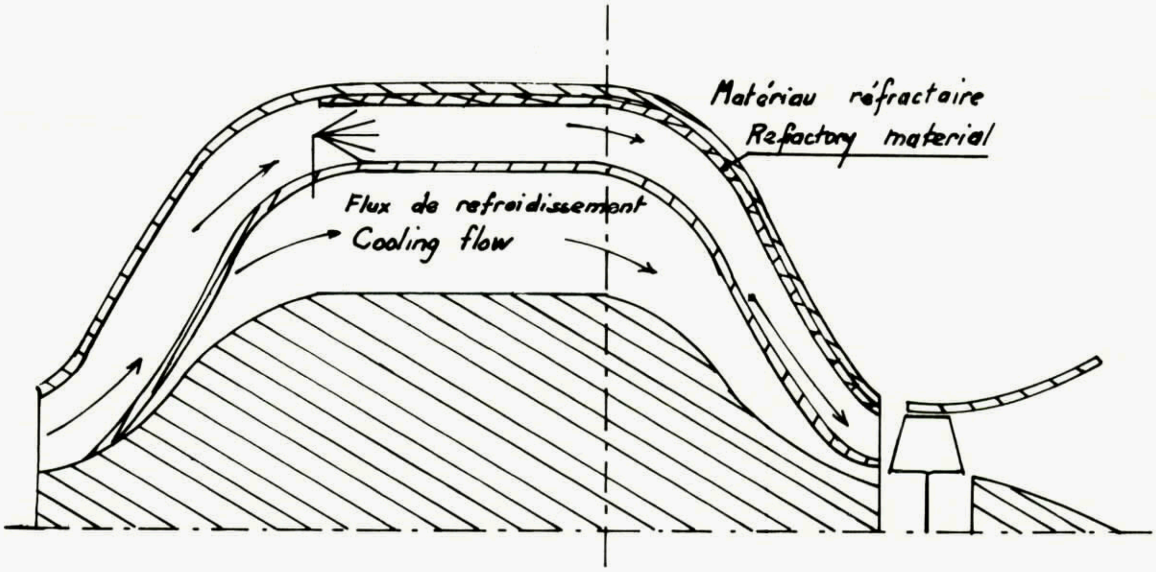


Fig.10 Coupe méridienne d'une structure voûte
Axial section of a dome-structure

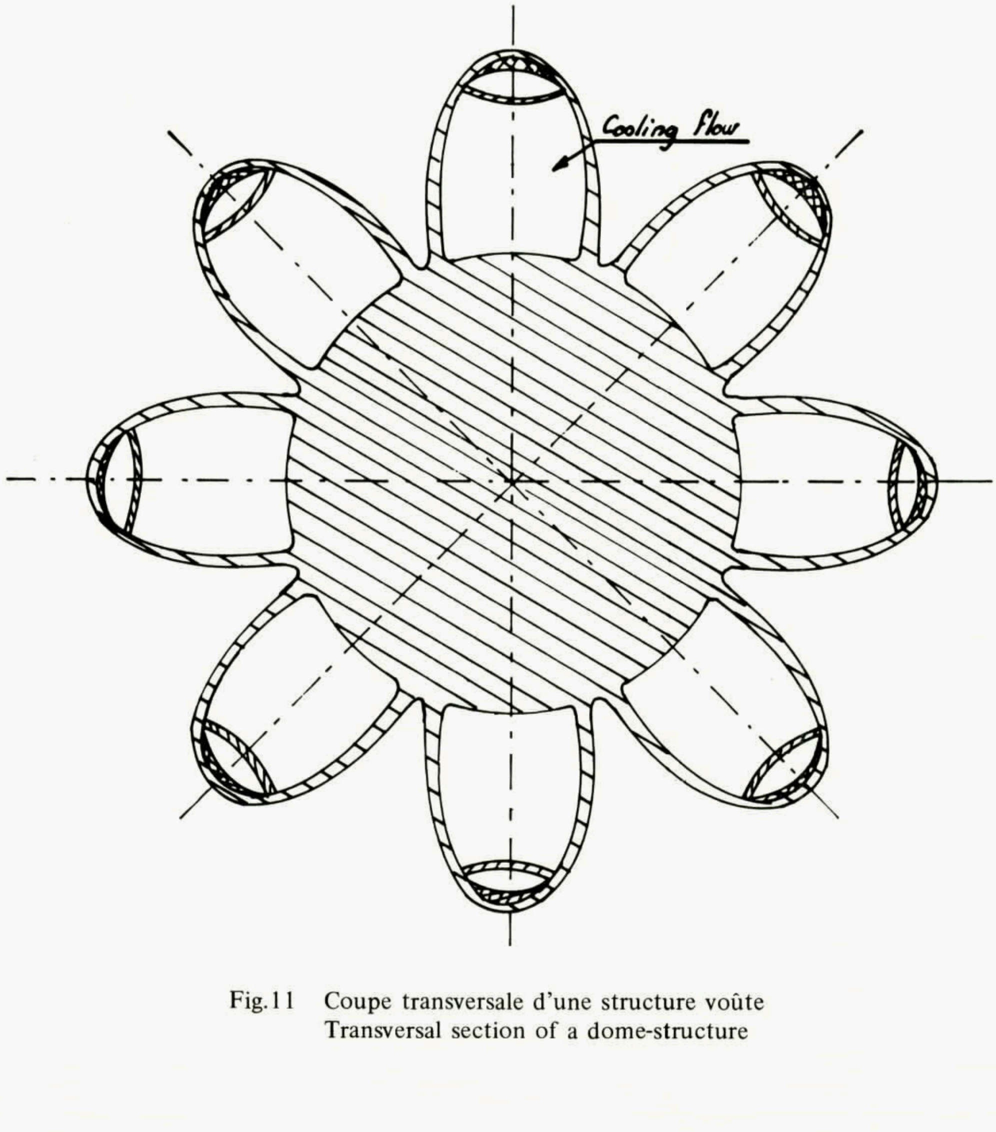


Fig.11 Coupe transversale d'une structure voûte
Transversal section of a dome-structure

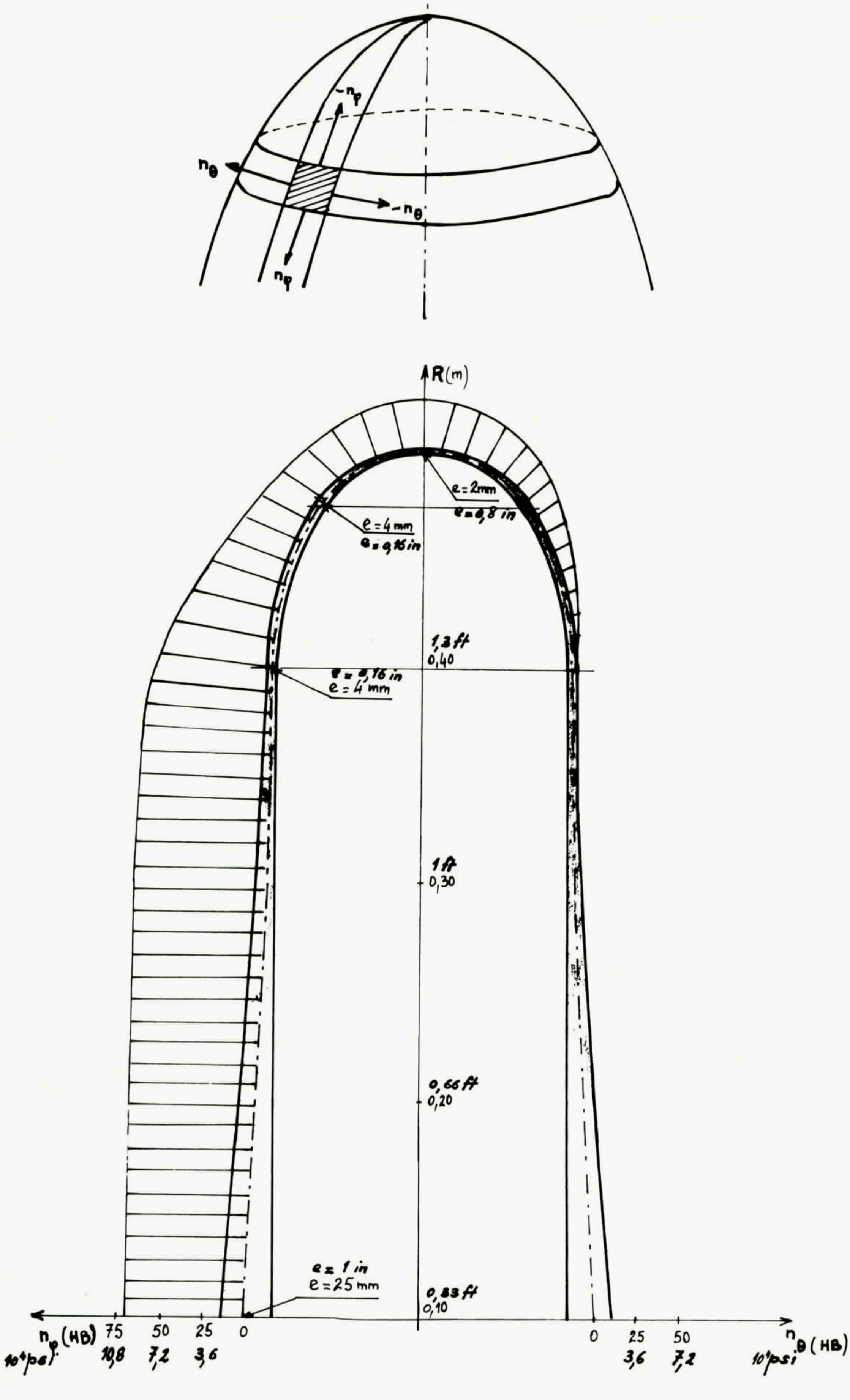


Fig.12 Evolution des contraintes n et n en fonction de l'épaisseur d'un bras en acier, tournant à une vitesse périphérique de 800 m/s

Stresses n and n diagram in terms of the thickness of a steel element, for a peripheral velocity of 2620 ft/s

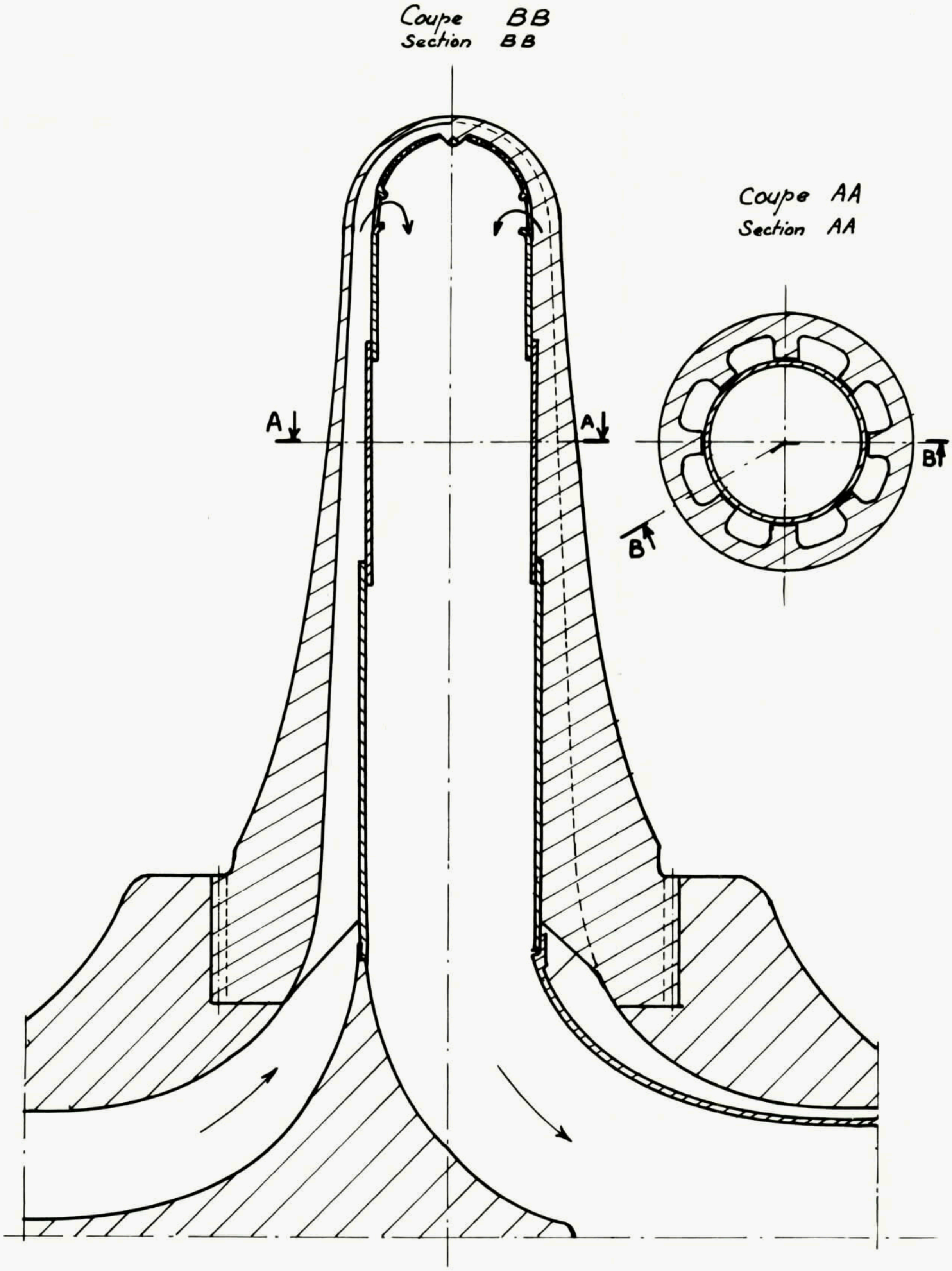
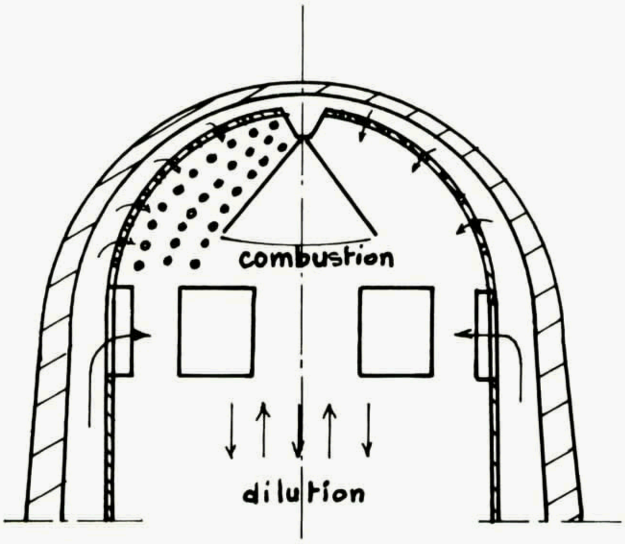


Fig.13 Exemple d'architecture d'un bras d'un générateur de NERNST
Architecture example of an element of a NERNST generator



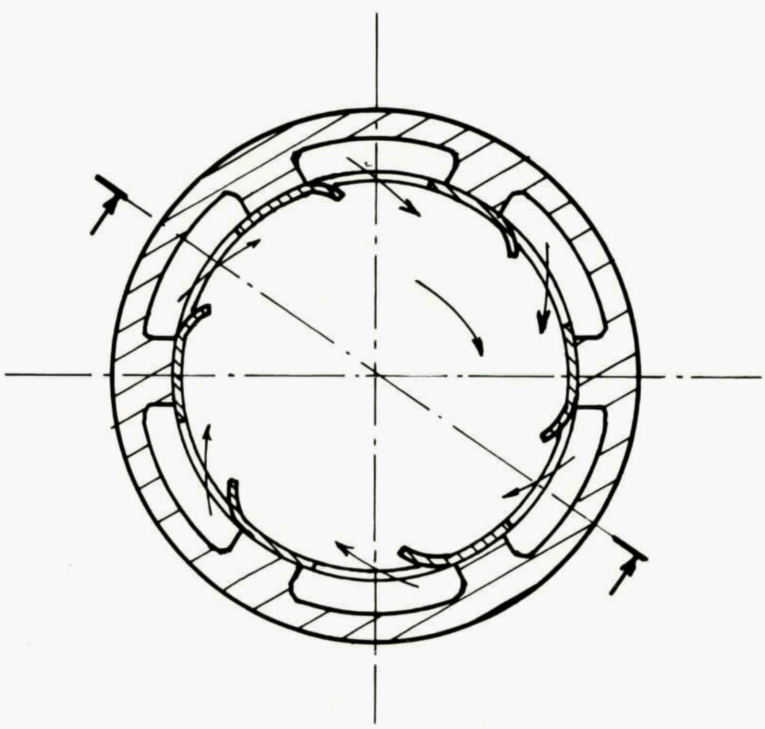


Fig.14 Projet de chambre de combustion d'un générateur de NERNST
Project of a combustion chamber of a NERNST generator

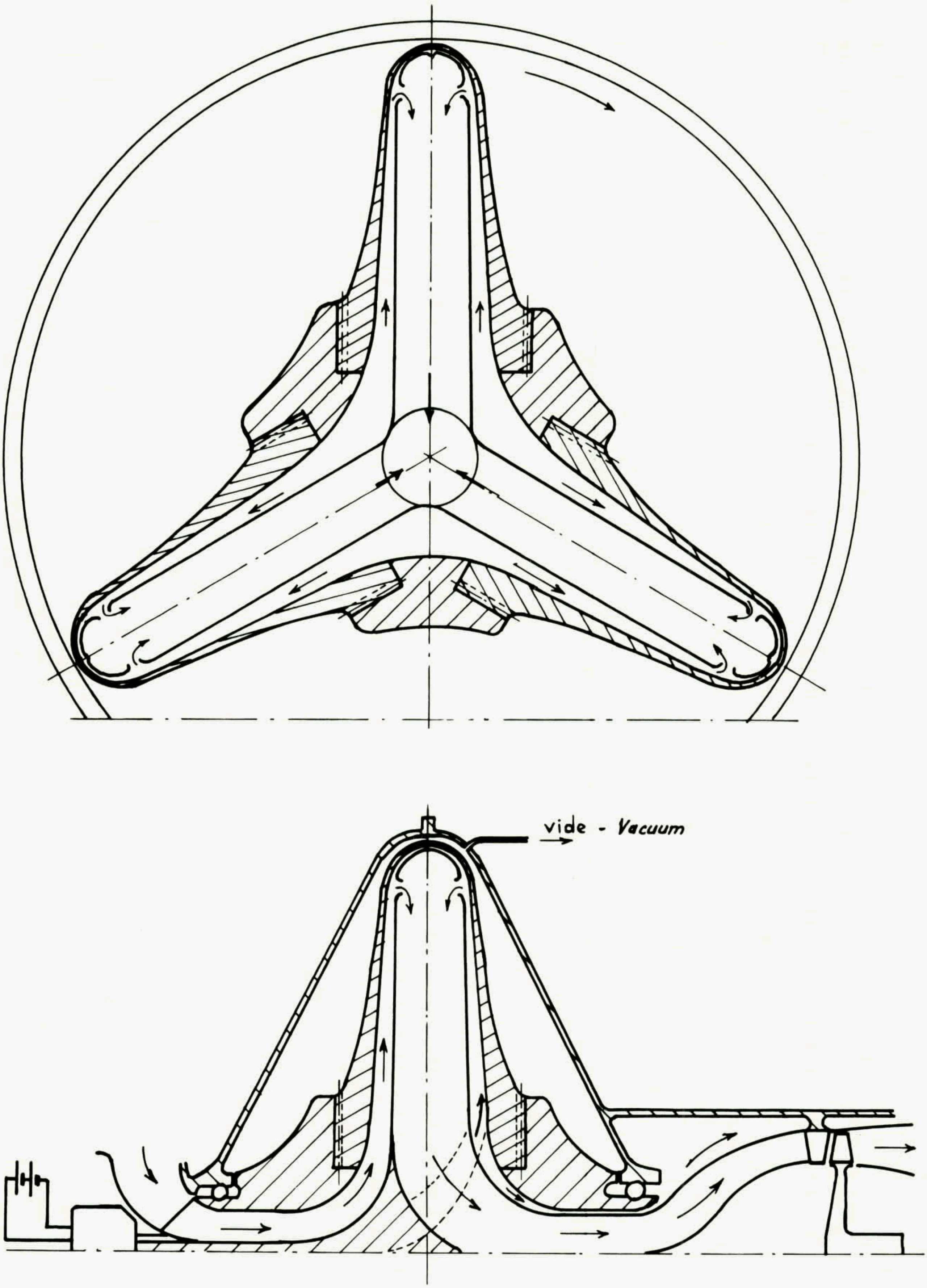


Fig.15 Exemple de conception d'une machine à générateur de NERNST
Example of conception of an engine running on NERNST cycle

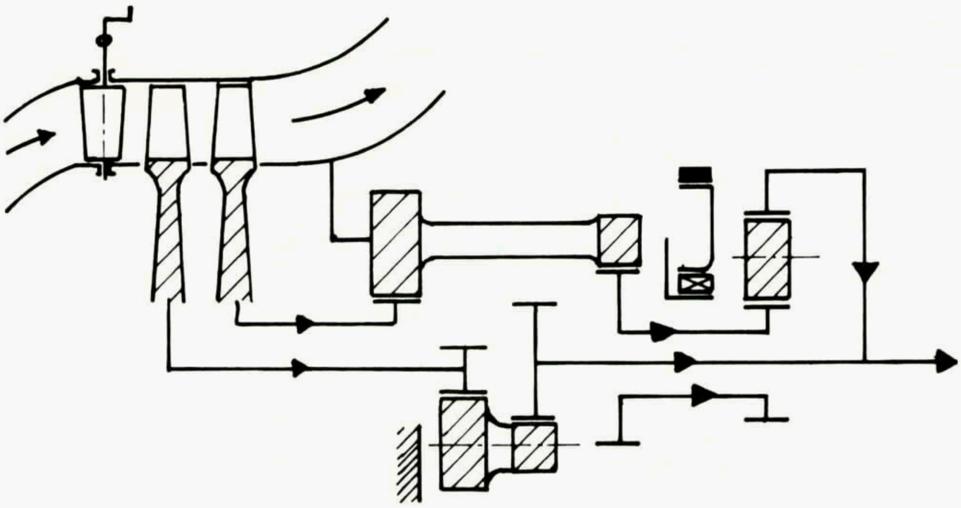


Fig.16 Convertisseur de couple proposé par M. Kronogard
Kronogard's torque convertor

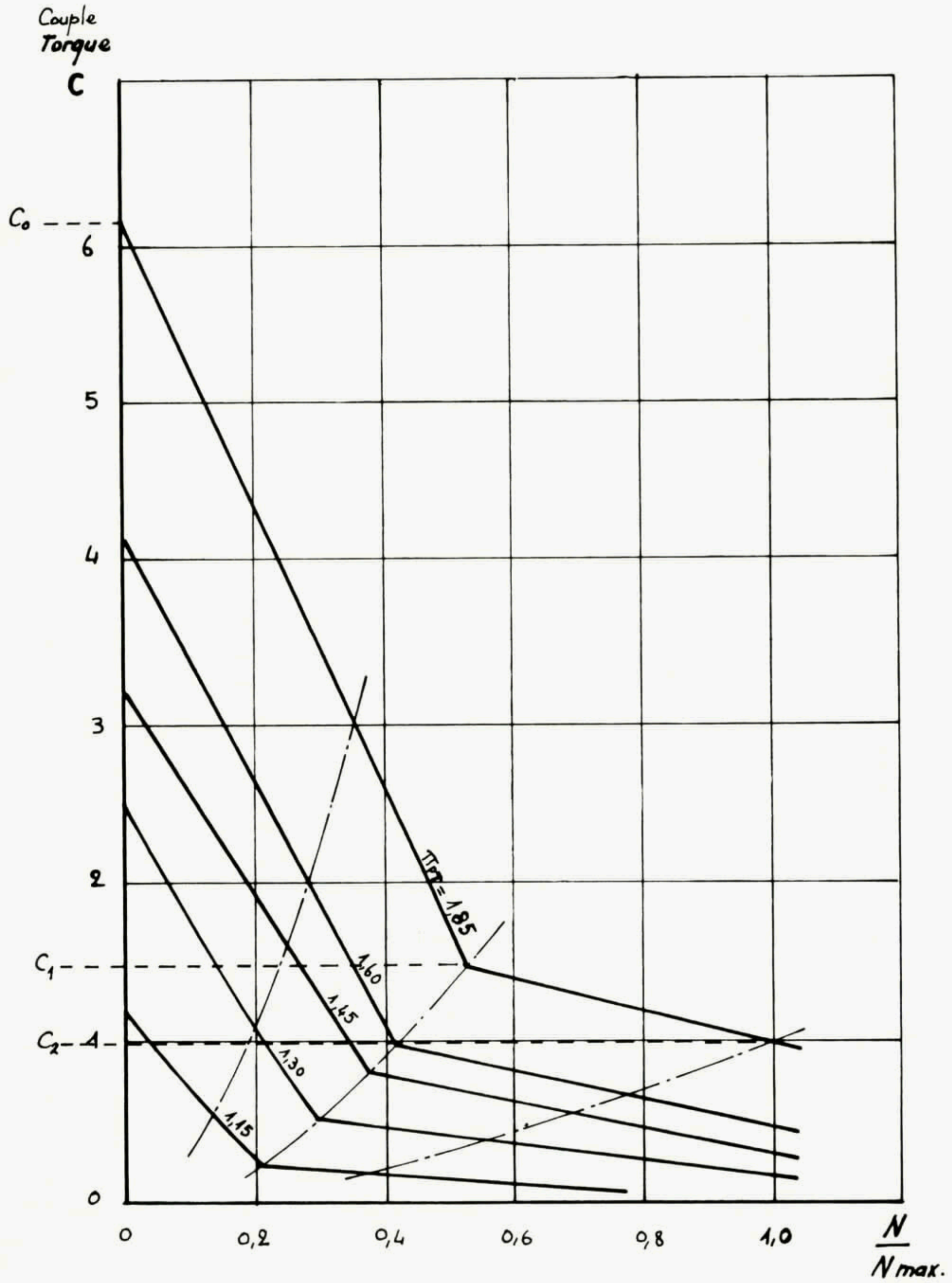


Fig.17 Courbe caractéristique du convertisseur Kronogard
Characteristic diagram of the Kronogard's torque convertor

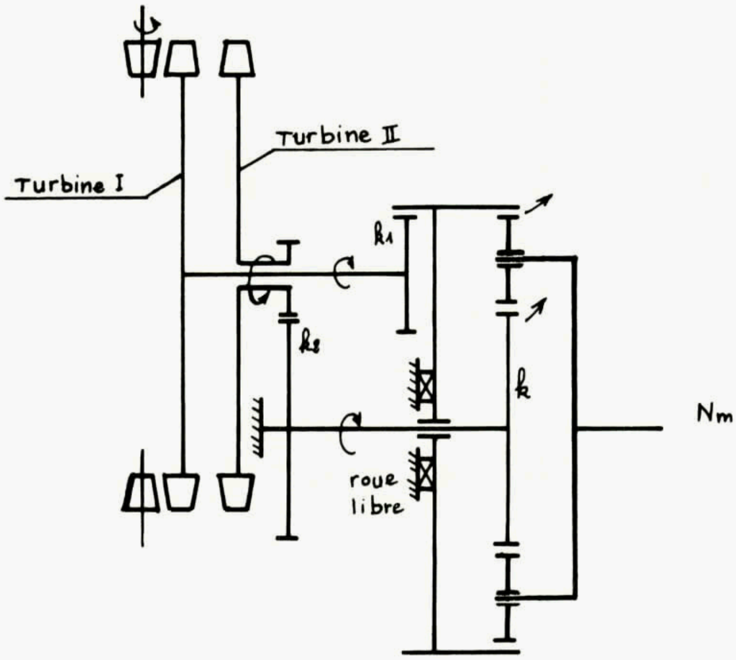


Fig.18 Convertisseur de couple Hispano-Suiza
Hispano-Suiza's torque convertor

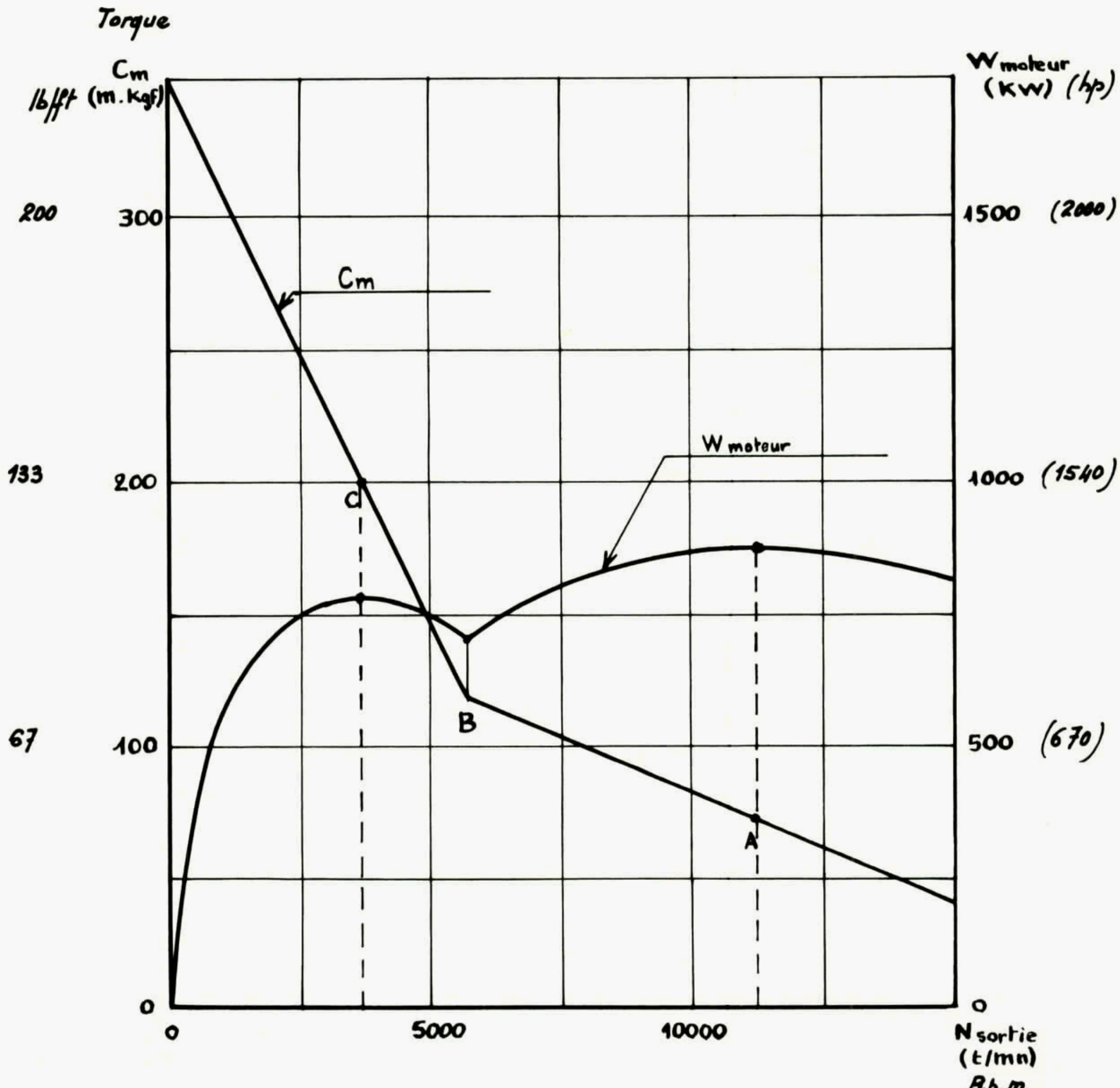


Fig.19 Courbe caractéristique du convertisseur de couple Hispano-Suiza
Characteristic diagram of the Hispano-Suiza's torque convertor

FUTURE DEVELOPMENTS OF SMALL GAS TURBINES

by

Jean MELCHIOR (France)

Ingénieur Principal de l'Armement

INTRODUCTION

Being instructed to proceed to prospective research on an optimum propulsive device for combat vehicles, I have acquired a broad knowledge of the various mobile sources of energy. I have analysed their respective operational qualities and investigated the specific limits of each architecture. In the light of this analysis, I went more deeply into the study of solutions likely to extend these limits.

I was concerned with diesel engines as well as with gas turbines, of course, and therefore feel authorized to venture making a prediction as regards the upshot of the competition which has begun between these two sources of energy. I do not feel qualified to add anything to the previous papers on the evolution of present techniques. Therefore, I shall be satisfied with stating what the limits of these techniques are and go more fully into the details of a few original concepts.

Prior to contemplating the future evolution of gas turbines, I would like to outline briefly the history of their development up to their present status. Initially, the attractiveness of the gas turbine lay in its great mechanical simplicity. Unfortunately, the drawback of this simplicity was bad performance which made it difficult to use. However, the necessity to give up the propeller to increase the speed of aircraft gave it spectacular impetus after World War II.

It should be pointed out, incidentally, that the turbine took the lead, at that time, owing to its jet effect, that is to say its ability to speed up a considerably air flow within a small frontal area. The effort to achieve high thermo-propulsive efficiency came much later on, with high by-pass ratio ducted fan engines. It was found out, then, that for cruise speeds of approximately 1000 km/hr. it is advantageous to separate the propulsive function from the energy generating function associated together in the conventional jet engine. This evolution gave impetus to the thermal efficiency of gas generators, at the cost of ever increasing complexity. One can say nowadays that the propeller has been replaced by a fan, and the reciprocating engine by a gas turbine.

Had it been known in 1945 that the 1970 commercial aircraft would be propelled by large fans, the development of reciprocating engines to drive these fans would have been continued. As a matter of fact, the immense progress achieved since then to bring down the specific fuel consumption of gas turbines to that of diesel engines could not be forecast. In view of such performances, the turbine has now almost completely superseded the reciprocating engine in the aeronautical field, and is attempting to impose itself for ground and sea vehicles.

Success will likely be more difficult to achieve in this field. For aeronautical propulsion, the turbine operates under particularly favourable conditions :

- High average power per unit
- Considerably appreciated light weight
- Total upstream pressure higher than total downstream pressure
- Low temperature environment
- No filtering nor insulation
- Hardly variable power and rate in cruise
- Very rare operation under part load conditions

Operating conditions for ground vehicles are usually very different :

- Low power per unit
- Low bulk more appreciated than light weight
- Total upstream pressure lower than total downstream pressure
- Great variations in environmental temperature
- Filtering and insulation almost always required
- Abrupt variations in power supply requirements
- Frequent part load operation

It should also be pointed out that while it acquired these performance capabilities for the sake of aeronautics, the gas turbine lost its original advantage - simplicity.

EVOLUTION OF SMALL, CONVENTIONALLY DESIGNED TURBINES

Increasing the cycle efficiency implies simultaneous variations in several parameters. Reducing the size of the engine brings out incompatibilities between these variations. For instance, the higher the compression efficiency, the higher also is the optimum compression rate. Now, an increase of the compression rate induces a reduction of the fluid stream section and a degradation of the compression efficiency. Likewise, an increase of the pressure ratio should be associated with an increase of the turbine inlet temperature. Now, reducing the stream section makes vane cooling more difficult.

These scale effects, many more examples of which could be quoted, hamper the low power applications of gas turbines unless considerably complicated devices are provided: heat exchangers, connection devices between the free power turbine and the gas generator turbine, variable setting guide vanes, etc. All these devices involve the same adjustment difficulties and failure sources as those inherent in reciprocating engines, and which were expected to disappear when the use of the latter was discontinued.

Engineers will unquestionably overcome the present difficulties, but one should bear in mind the fact that industrial achievements are generally based on the simplest technical solution to reach an objective. For this reason, I believe that the industrial turbine will be simple or will not exist. For the power range with which we are concerned here, the gas generator might consist of a supersonic centrifugal compressor and a cooled radial turbine.

The pressure, temperature and efficiency levels which it will be possible to reach with these simple components will determine the scope of possible uses. However, I doubt it will ever be possible to achieve the performances required for ground vehicle propulsion without adding heat exchangers. As a matter of fact, in view of their very design, small turbomachines have a poor efficiency under design conditions and especially under part-load conditions.

REASONS FOR EFFICIENCY LIMITATIONS OF SMALL GAS TURBINES

One hopes to be able to reach compression rates, in the future, ranging from 12 to 15 for a single stage. At supersonic velocities, an adiabatic efficiency above 0.75 appears hardly realistic. The only means of reducing the specific fuel consumption consists therefore in increasing the turbine inlet temperature. Now, advances in the field of high temperature behaviour will become increasingly slow and difficult, and will always be associated with costly cooling techniques, which are undesirable on small size engines. The latter will be particularly handicapped by the deterioration of compression and expansion efficiencies, due to the increase of the compression rate and rotation speed.

To associate simplicity and performances under design conditions, the limitation is imposed by the admissible turbine inlet temperature.

Part-load efficiency is always lower than rated power efficiency. Owing to the turbo-compressor surge, the pressure ratio must be reduced simultaneously with the air flow rate.

Therefore, for the power to be reduced, either the maximum cycle temperature or the pressure ratio, or both must be decreased. In any case, the cycle efficiency is diminished. Finally, idling a turbine consumes at least 25% of its maximum consumption, whereas the diesel engine consumes 2%. A highly efficient heat exchanger provides a solution which proves very costly in terms of complexity and bulk.

The feasibility of new designs which do not imply thermal exchanges between the air and gases will be discussed hereunder.

FEASIBILITY OF NEW DESIGNS OF ARCHITECTURE

- Gas generators with rotating combustion chambers

A turbine with rotating combustion chambers is particularly well adapted to the propulsion of ground vehicles. This cycle was proposed by the physicist WALTER NERNST (1) in the early part of the century, but it was deemed impracticable with the materials then available. Progress made since that time in the field of composite and refractory materials, as well as in that of cooling techniques, has induced me to reconsider the question.

- Operating Principle

A NERNST generator is diagrammatized by a U-shaped tube ABCD rotating around the axis AD. (See Fig. 1).

The air gets in at E, is compressed by centrifugation in AB, burns at a constant pressure in BC and is partially expanded in CD; then, it gets out through S with the usable cycle energy which will be collected in a propulsive turbine, for instance.

To maintain the rotation of the tube, it is enough to overcome the frictions of the bearings and of the ambient fluid on the external surface of the rotor; as a matter of fact, the kinetic moment imparted to the rotor by the fluid in branch AB is recovered by the rotor in branch CD. If, besides, the air intake and the gas exhaust are axial, the aerodynamic torque applied to the rotor is non-existent.

Figure 2 represents the cycle on a pressure-volume diagram between the isobaric curves P_1 and P_2, and the polytropic curves AB and EC.

Supposing U is the peripheral velocity of the rotor part CD, the compression ratio is given on Figure 3 for two values of the polytropic compression efficiency. From U = 600 m/s onwards, pressure ratios are of consequence. Let us note that velocity U is only limited by the rotor resistance, since the relative Mach number within the tube is independent of U.

Thus schematized, the rotor is symmetrical, and the flow will have to be induced in a given direction by means of air intake guide vanes. Once it has been induced, the flow is maintained by the difference in specific mass between columns AB and CD located in the centrifugal field of gravitation.

If the rotation speed of the system is constant (the maximum speed allowed by the materials used will be chosen), the pressure ratio is set at its peak value.

The fuel flow rate will determine the maximum temperature of cycle T_3 and, consequently, the generator exhaust temperature T_4. We shall see that, for a given exhaust section, the engine air flow depends on T_3.

In order to avoid frictions on the outer surface, the rotor will rotate within a casing, in partial vacuum. The radial acceleration in the combustion area is given by the formula $\frac{U^2}{R}$, R being the radius of the rotor.

The stresses to which hot parts are subjected will be reduced by increasing the radius of the rotor.

PERFORMANCE EVALUATION

- Compression Efficiency

This efficiency has been evaluated by applying the laws formulated for conventional centrifugal compressors. The main causes of loss in a centrifugal compressor are the frictions exerted on the casing of the compressor, flow distortion due to flow "slipping" and poor diffuser efficiency. Such loss sources are all removed in a NERNST generator.

The duct is closed and a low relative velocity may be selected to reduce internal frictions. The slipping cancels out in the peripheral area, when the radial velocity cancels out (suppression of Coriolis acceleration); owing to the high radial aspect ratio, it remains low all along the duct. The diffuser is removed since the whole compression is achieved by increasing the static enthalpy.

In view of all the above remarks, the polytropic compression efficiency is assessed to be 0.93.

- Expansion efficiency in the generator

For the same reasons, the expansion efficiency is estimated to be 0.95.

- Cycle efficiency

We are approximating to the perfect cycle where the thermal efficiency is independent of the temperature, and depends only on the pressure ratio. The peripheral velocity and the efficiency being equal, the pressure ratio of a conventional centrifugal compressor is much higher than that of a NERNST generator. Figure 3 shows the pressure ratio as a function of the peripheral velocity for a conventional compressor (η_c = 0.78) and for a NERNST generator.

It should be pointed out that beyond 500 m/s the efficiency of conventional compressors is degraded by the shock-waves created in the diffuser. The peripheral velocity of the NERNST generator is only limited by the rotor resistance. It should obviously be chosen as high as possible since the compression efficiency does not depend on it. It should be possible to reach a speed of 800 m/s with steel if the structure is adequately designed. Figures 4,5 and 6 show the specific fuel consuption as a function of the peripheral velocity and of efficiencies ($\eta_c = \eta$) for a conventional free power turbine, with a polytropic efficiency 0.86. Let us note the following point:

$U = 800$ m/s $P_2/P_1 = 11.5$ $\eta_c = \eta_t = 0.95$ $T_3 = 800°C$ $C_s = 170$ g/ch/h & $W_s = 240$ ch/Kg/s

These performances are higher than those of the best engines currently available.

OPERATING CONDITIONS

In order to obtain the best cycle efficiency for any load, the generator operation at a constant rotation speed will be chosen. Furthermore, this solution offers the advantage of being the simplest and of suppressing the response delay time to a power supply demand. The pressure ratio is then invariable.

Two solutions are possible to modulate the power :

1. To keep the maximum temperature T_3 constant and vary the air flow which is proportional to the power. This solution allows the best consumption at reduced rates. It implies necessarily :

   - that no surge of the generator occurs at very low flow rates (of the order of 1/10th of the nominal flow rate). This requirement is certainly met, owing to the absence of stator capacity under pressure.

   - that the free turbine accepts a variation in the gas flow rate - from 1 to 10 approximately - which cannot be contemplated. The second solution will, therefore, be chosen.

2. Modulate the maximum cycle temperature T_3. Figure 7 shows the variation of the ratio of the reduced outgoing flow rate to the reduced flow rate getting into the generator, as a function of T_3. We see that this ratio hardly varies between 900°C and 450°C; it rises from 0.6 to 0.85, while the power changes from W to 0.17 W. This indicates that, while the nozzle ring of the free power turbine is critical, the flow rate through the generator decreases from 1 to 0.7, which corresponds to a very slightly off-design operation of the inlet wheel. As a matter of fact, the tangential velocity of the latter, at blade tips, will be under 100 m/s. Along the whole height of the blade, the axial velocity will be considerably higher than the tangential velocity. When this velocity varies from W to 0.7 W, the velocity triangle is very little distorted and the incidence angle remains small (See Fig. 8).

   This solution, which is very simple since it only requires fixed geometry, gives excellent part-load performance. (See Fig. 9).

Optimum Cycle

In brief, the objective to reach will be the development of a turbine composed of a NERNST generator operating at a constant peripheral velocity 800 m/s $\frac{P2}{P1} = 11.5$ at a maximum temperature of 900°C, and supplying a conventional free power fixed geometry turbine. One will attempt to obtain :

$\eta_c = 0.93$
$\eta_t = 0.95$ for the generator
$\eta_t = 0.86$ for the free power turbine

the performances will then be :

under design conditions:

$W_s = 240$ KW/Kg/s $= 325$ ch/Kg/s
$C_s = 230$ g/KW/h $= 170$ g/ch/h

at a fifth of the power:

$C_s = 290$ g/KW/h $= 215$ g/ch/h

Examples of Configurations

We have shown that the performances of a NERNST generator are associated with the achievement of high peripheral velocities. The problem consists therefore, in developing a closed structure rotating at 800 m/s, within which combustion will take place.

Two structures have been considered so far :

1. Rotor with axial combustion chambers :

   Figures 10 and 11 show a cross-section perpendicular to the axis and a meridian section. The air intake and exhaust guide vanes are connected by 6 or 8 ducts, each equipped with a combustion chamber. These ducts are delimited by a tensile stressed outer vault, and by a compression stressed inner vault. These vaults are hanging from arms protruding from the hub. The cylindrical part is the result of calculations and dimensioning; however, the compressor part and the turbine part are not amenable to computation. Besides, the turbine part is extremely critical. as far as the rotor strength is concerned.

2. Rotor with radial combustion chambers :

   The design is less attractive than the previous one and leads to a much bulkier machine. However, it complies better with the specific qualities of the NERNST generator. Furthermore, the structure is much easier to design and develop, combustion is easier to arrange, and the cooling of hot parts is more easily achieved.

   This design is based on the following principle : compression and expansion efficiencies will be all the higher as ducts will be longer. This is why the diameter of the intake wheel has been set at 1/10th of the rotor diameter. The engine mass-flow is then limited by the intake wheel and can be burnt by a small number of combustion chambers located at the arm-tips, and in which compression and expansion take place. (Fig.15). The following description is that of a 500hp generator, with three arms screwed into a central hub. Each arm is composed of an outer, cold, tension stressed shroud and of an inner, hot, telescopic tube, the components of which rest on the cold structure and undergo buckling stresses. The inner tube emerges inside the dome.

   Compression takes place within the ring-shaped space between both tubes; combustion takes place in the peripheral dome, and expansion in the inner tube.

Outer Tube Strength

To establish the feasibility of such a structure, a computer program has been developed to determine the stresses in a body of revolution with a cylindrical internal surface and an evolutive thickness (Fig. 12).

The following numerical values have been selected :

Gyration radius of the arm top :	0.50 m
Inner radius of the arm :	0.05 m
Metal density :	8
Peripheral velocity at the top :	800 m/s

Figure 12 shows a thickness evolution and the corresponding stress evolution according to the main curves: η_φ in the meridian plane and its perpendicular η_θ.

It is evident, from all this, that the dome, which is the hottest part (approximately 350°) is the least mechanically loaded. The arm undergoes pure traction stresses and could be lightened by the use of high strength fibres. Anyway, the feasibility of such a structure with current steels has been demonstrated. The real arm will support an internal telescopic tube operating as a turbine. This does not alter the previous data, as this tube may be thin, and the arm is reinforced by the ribs keeping the internal tube co-axial (Fig. 13).

Thermal Level

The temperature of the external tube will be roughly that of the air during the compression process, that is, room temperature at the hub and 350°in the dome. The internal tube will have linear contacts with the external tube to limit conduction towards the latter. Downstream of the combustion area, one side of this tube will be at a temperature of 900°C, and the other will be cooled by the air in the compression process. The splitting of this tube into sections will be based on the allowable stress at the supporting points of each buckling stressed element. The internal dome will be subjected to the flame radiation. It is fortunate that this spot of the generator is the only one where the static pressure is higher on the air side than on the gas side. It will therefore be possible to arrange cooling by a simple method: transpiration or film cooling.

Combustion

Combustion is the most critical point of the project. However, we have reasons for being optimistic.

The small volume available for combustion renders the problem difficult. It is therefore a question of developing a high intensity combustion chamber, that is to say, to speed up the energy transfer between products which have chemically reacted, and products which will. In a conventional chamber, the transfer is insured by a turbulent mixture between hot gases and primary air. In a rotating chamber, it is possible, owing to the intense field of gravitation (above 100,000 g in the present case), to induce forcibly a penetration of the cold layers into the hot layers, provided the air is injected into the combustion area in the form of sheets perpendicular to the acceleration vector, along the direction of decreasing potentials.

The design of the chamber has been based on this principle (Fig. 14). One will attempt to have the combustion take place within a sphere surrounded by a spherical layer of cold air. This air envelope is supplied by a ring of nozzles, also slanting towards the surface of the internal dome, which create a kinetic moment around the arm axis. The gyroscopic stability acquired by the fluid particles brings about a rocking motion of the kinetic moment around an axis parallel to the rotation axis in the reference axes of the rotor. This powerful mixing (stirring) should lead to dilution on a very short distance. The fuel may be introduced into the combustion area either in a broken up liquid phase to avoid too high injection pressures, or in a gaseous phase.

Conclusion

Figure 15 shows a possible layout for a three-arm generator. This preliminary feasibility study shows that, a priori, there is no obstacle to the development of a combustion chamber rotating at 800 m/s. The proposed architecture is certainly not optimum. Let us state again that, for a given peripheral velocity, the larger is the rotor diameter, the lower is the acceleration in the combustion chamber and the higher the compression and expansion efficiencies.

The general operation of the machine has not been dealt with, in particular how to maintain the generator rotation and to create a vacuum in the casing. As far as I know, no experimenting has been carried out based on this principle, apart from PRATT WHITNEY's research on combustion in an intense field of gravitation, which reveals a considerable increase in combustion speeds.

TRANSMISSION OF GAS TURBINE POWER

The future prospects of the gas turbine in the field of ground propulsion depend to a large extent on the solutions which will be found for the power transmission problem. The ideal propulsion machine is characterized by a hyperbolic relation between the torque and the rotation speed which makes it possible to use the total rated power under any working condition. From this viewpoint, the turbo charged diesel engine is a bad machine, the torque of which increases with the rotation speed. The two-shaft turbine behaves like a ratio 2 torque converter. This leads to the suggestion of associating with the turbine a conventional gear box, from which the hydrokinetic torque converter has been removed. Problems arise for changing gears; as a matter of fact, a gas turbine transfer function is quite different from that of a diesel engine. It should be pointed out, for instance, that the over-speed hazard of the free power turbine practically imposes changing gear underload. In view of this, the considerable kinetic energy returned by the free power turbine when its rotation speed decreases induces considerable fatigue in the gear box clutches. The power transmission of a gas turbine unquestionably requires a specific solution.

Hydrostatic transmission provides a rather good functional solution to the problem, but the overall structure is very heterogeneous: the turbine is light and can do without a cooling system; the hydrostatic solution results in a heavy system which produces a considerable amount of calories. I believe that any future solution should comply with the following requirements :

- transmission must be automatic
- the thermal power dissipated in the transmission oil must be low enough to be rejected by the oil radiator of the turbine. This practically precludes the use of any hydrokinetic converter and hydrostatic element which would require a bulky cooling device.
- the flow issuing from the gas generator must be used as a working fluid for the torque converter, in order to clear away the transmission losses in the exhaust.

The solution described below meets these requirements :

Aerodynamic Torque Converter

We are indebted to Mr. KRONOGARD (SAE Transaction 1960) for the idea of combining the torque converter effect of two turbines.

Mr. KRONOGARD's transmission system is schematized on Figure 16. The generator supplies successively a fixed, variable setting angle vane and two counter-rotating mobile vanes. This way, instead of transforming the mechanical energy received (engine torque x engine speed) into the desired energy (driven shaft torque x driven shaft speed), the energy desired is directly produced (driven shaft torque x driven shaft speed). Turbine I and Turbine II act respectively on the sun gear and the ring gear of a differential gear train, the planet wheel carrier of which is connected with the stand by a free wheel. The power outlet is derived from the ring gear.

We shall analyse the operation of this set up, assuming that the outlet shaft is wedged on a brake which is progressively released. Diagram 17 shows the evolution of the torque and the outlet rotation speed. When the resistance torque decreases from C_o to C_1, the velocity increases from 0 to N_1. The aerodynamic torques in the opposite direction tend to drive the planet wheel carrier along the direction of rotation blocked by the free wheel. During this phase, the velocities of turbines I and II are therefore proportional.

For C_1 and N_1, the torque exerted on the planet wheel carrier cancels out, the free wheel is released and the resistance torque of the turbine II cancels out. Then, turbine I produces all the work, though handicapped by the pressure loss induced by the windmilling of turbine II. To remedy this drawback, we are testing in cooperation with the Hispano-Suiza section of SNECMA a new combination of counter-rotating turbines. Four leading ideas have resulted in this configuration :

- In the course of the phase during which a single turbine operates, its operation must not be hindered by the turbine which does not operate, and which, therefore will be located upstream.

- The phase during which both turbines share the enthalpy drop is assumed to yield the best overall expansion efficiency. This phase will then insure the half-range of higher speeds. As a matter of fact, in ground propulsion, the power required is statistically an increasing function of the forward velocity. For most terrain profiles, the best average consumption is obtained by setting the highest efficiency at a forward velocity higher than the half maximum velocity.

- Under all operation conditions, the three cascade efficiencies will have to remain acceptable. The angles of incidence on the vanes should not exceed a range evaluated according to AINLEY's and MATHIESEN's method. To meet this requirement, the setting angle of the fixed vane should be adjustable.

- In the counter-rotating phase, the kinematic link between the two turbines should allow a good expansion efficiency over the whole half-range of speeds. For this purpose, the enthalpy drop must be progressively transferred from one turbine to the other. It is therefore obvious that the rotation speeds of the two turbines should vary in opposite directions, contrary to KRONOGARD's solution where both speeds are proportional in the counter-rotating phase.

Figure 18 shows a diagram of a torque converter complying with these four principles. It is composed of an adjustable setting angle nozzle guide vane, followed by two mobile, counter-rotating vanes I and II, series operating. The two counter-rotating turbines are connected by an epicyclic train of ratio k.

Turbine I acts on the outer ring gear through a k_1 ratio speed gear. Turbine II acts on the sun gear through a k_2 ratio reducing reversing gear. A free wheel RL prevents the counter-rotation of Turbine I. Power output is derived from the shaft of the planet wheel carrier.

Let C_1 and C_2 be the torques produced by Turbines I and II. If we start from the resting point and increase the speed of the output shaft, we can observe two phases of operation (Figure 19).

Phase I If $k\,k_2C_2 > k_1C_1$, Turbine I remains locked and plays the part of a distributor for Turbine II. While the speed rises, C_1 remains fairly constant, and C_2 decreases down to

$$C_2 = \frac{k_1C_1}{kk_2}$$

which is the release torque of the free wheel and the end of Phase I.

Phase II From this speed onwards, both turbines rotate together and share the work between each other to insure the balance of the epicyclic train.

$$k = \frac{k_1C_1}{k_2C_2}$$

Turbine I accelerates and Turbine II slows down in conformity with the fourth requirement.

Profile Determination

To define the blade profiles, three operating points have been imposed :

Operation A - Both turbines rotate together and the absolute outlet velocity of Turbine II is axial.

Operation B - This is the transition point between phase I and phase II, characterized by the stopping of Turbine I and $k = \frac{k_1 C_1}{k_2 C_2}$

Operation C - The free wheel locks Turbine I and the absolute velocity of the fluid at the outlet of Turbine II is axial.

Writing the overall compatibility equations would be too long. It is enough to know that the speed triangles can be defined in A, B and C, as well as the profiles of the three cascades, when the following information is provided :

- characteristics of the flow issuing from the gas generator
- value of the parameter k $\frac{k_2}{k_1} \cdot \frac{U_{1a}}{U_{2a}}$, where U_{1a} and U_{2a} are respectively the peripheral velocities of Turbines I and II in point A.
- value of the distributor setting angle variation
- some dimensional characteristics of the fluid flow.

Results of Calculations

The angle of incidence i was calculated for points A, B and C:

Distributor :

A	i	= + 8°
B	i	= + 1°
C	i	= + 4°

Turbine I :

A	i	= - 25°
B	i	= + 20°
C	i	= + 14°

Turbine II :

A	i	= - 37°
B	i	= - 21°
C	i	= + 28°

The calculation of losses, based on these incidences, led to sufficiently encouraging results to initiate the manufacturing of a test machine, dimensioned for a flow-rate of 6 kg/s, and designed for aerodynamic tests in a cold flow. This set up, designed so as to allow the setting angle of fixed and mobile vanes to be adjusted, should provide the respective influence of these three parameters.

Both turbines can be slowed down separately until stoppage. After the aerodynamic tests are over, it will be possible to establish the kinematic link between the turbines, in order to analyse transitory power rates.

Conclusion :

Aerodynamic transmission seems therefore to be both the most simple and the most compact for a gas turbine.

Automatism is achieved owing to a free wheel which gets locked or released without shock, and to a control of the distributor setting by the outlet rotation speed.

It is possible to obtain a conversion ratio $\simeq$ 6 by choosing a rating speed slightly higher than N_A.

The power output will not be know until after the tests. It will have to be compared with the product of the efficiency of a conventional free power turbine by the efficiency of the type of transmission associated with it.

THE GAS TURBINE AS COMPARED WITH THE DIESEL ENGINE

For the sake of practicality, this comparison will apply to the propulsion of heavy trucks. These vehicles are at present fitted up with a 300 hp power unit, which should become an 800 hp power unit within the forthcoming years.

As far as truck turbines are concerned, a low compression, high recovery rate (90%) cycle has been unanimously selected. We shall therefore assume that the turbine is of the rotating generator type (Ford 707). For diesel engines, we shall consider two cases :

a. Four-stroke engines in current use, operating with a brake-mean effective pressure of 11 kg/cm^2, and at 2500 rpm.

b. Low compression ratio, four-stroke engines, in the development process, operating under b.m.c.p. of 25 kg/cm^2 and at 2500 rpm.

These assumptions lead to the following characteristics :

- for 300 cv : Assumption A : displacement 10 litres (6 or 8 cylinders)
  weight 800 kg.
  Assumption B : displacement 4.5 litres (3 or 4 cylinders)
  weight 400 kg.

- for 800 cv : Assumption A : displacement 27 litres (at least 16 cylinders)
  weight 2500 kg.
  Assumption B : displacement 12 litres (6 or 8 cylinders)
  weight 1000 kg.

There are research trucks on which the turbines and the type (A) diesel engine can be compared for a 300 hp power. If we take into consideration the air filter and the silencer, the turbine is bulkier but less heavy than the diesel engine. For the whole range of loads, and especially at low loads and when idling, it consumes more fuel. The overall comparison is therefore in favour of the diesel engine.

On the contrary, for 800 hp, the turbine is much more advantageous than the type (A) diesel engine. In the case of assumption B, the diesel engine will probably be more advantageous for the whole range of powers.

HYBRID MACHINES

Volumetric engines are limited by their inability to breathe in large air flows for low bulk configurations. The volume flow is proportional to the displacement and the rotation speed.

Similarity laws show that a machine which is k times heavier than a homothetic machine with a volume flow Q_v can only handle a volume flow $k^{2/3} \times Q_v$. Small machines will therefore be relatively more powerful than large ones.

To increase the air flow, it is preferable to modify the density of the inlet air. The problem lies, therefore, in developing a device capable of increasing the air density in the most inexpensive way. The gas turbine is well designed to play this part.

The success of the turbo-charging of diesel engines is an example of this trend. It should be pointed out, in this respect, that the turbo-charger is still regarded as accessory to the diesel engine which can operate without it.

It will be interesting to design hybrid machines in which the low pressure part of the cycle is handled by a gas turbine, and the high pressure part by a volumetric machine. The respective possibilities of both designs will have to be exploited, and the tasks judiciously shared.

Tasks to assign to the turbine part of a hybrid machine

To maintain the simplicity of the turbine while achieving long life and low cost, the following requirements should be complied with :

- Reduce as much as possible the turbine inlet gas temperature;
- Rule out all variable geometry devices:
- Remove any kinematic link with the turbine shaft
- Consequently, no mechanical power whatsoever will be derived from the turbine shaft.

The only function of the turbine then consists in compressing the inlet air of the volumetric component. Therefore, one should be very exacting as regards the compression and expansion efficiency. The aerodynamic design of these machines will be as elaborate as that of the turbines designed to supply power. Besides, the operating conditions will be better defined and the engine will be optimized for a zero power. Thus, an extremely low autonomy temperature is obtained, as it is shown by the following example :

Room temperature	To	= 288° K
Polytropic compression efficiency	ηc	= 0,85
Polytropic expansion efficiency	ηt	= 0,85
Total pressure losses of the system (intake, combustion chamber, exhaust)	$\sum \frac{\Delta p}{p}$	- 10%

The power wasted in the bearings has been overlooked, as the pressure losses have been overrated.

Pressure ratio of the cycle :	$\frac{P_2}{P_1}$	= 6
Compressor outlet temperature	T_2	= 523° K = 250° C
Turbine inlet temperature	T_3	= 700° K = 427° C
Temperature increase in combustion chamber	ΔT_c	= 177° C

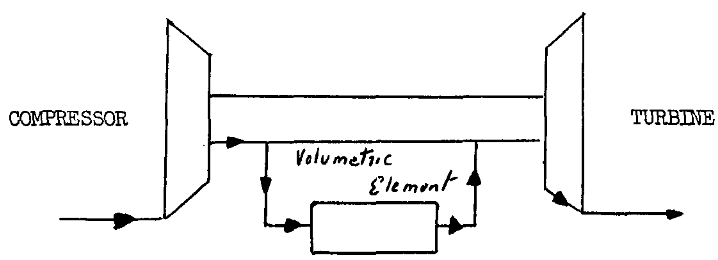
These low temperatures allow long life duration with usual materials.

Tasks to assign to the volumetric part

To this part falls the task of supplying the overall mechanical power of the hybrid machine. For technological reasons, we have to abide by the following limits :

- a maximum cycle pressure imposed by the mechanical behaviour;
- a maximum average temperature imposed by the thermal behaviour;
- a maximum exhaust temperature;
- in the case of diesel engines, a minimum temperature at the end of the compression phase; this temperature is all the lower as the pressure is higher.

Let us note that the volumetric component derives its airflow from the high pressure circuit of the turbine, as shown in the following diagram :



Thus we give up the possibility of recovering the whole of the energy available in the composite cycle :

- pulse energy, released when the exhaust valves open;
- improvement of the volumetric efficiency obtained by increasing the scavenging ratio.
- excess energy recoverable in the exhaust gases of the volumetric element after the compression energy has been derived.

The low relative value of these energies does not justify the engineering complications which their recovery would involve. Once these boundary conditions have been set, one will try to determine the optimum value of the super-charging rate of the volumetric element.

A comprehensive survey has been made for low compression ratio diesel engines; reporting the results would exceed the scope of this seminar.

We will merely state that, for a given engine technology, the indicated mean pressure increases in direct ratio to the boost pressure, up to approximately 12 bars.

The evolution of the fluid in the cylinder takes place at a higher density and a lower temperature than in a conventional cycle. This accounts for the fact that thermal loads do not increase and that the energy transferred to the cooling water is much lower in relative value. Thus, we recover in the water what we lose by decreasing the compression ratio, and therefore the diesel cycle efficiency.

CONCLUSION

It now seems well established that light engines with powers ranging between 300 and 1500 hp will find considerable avenues of trade. These engines will be essentially used for the propulsion of ground vehicles, that is for industrial applications. 1.5 kg of weight per hp. should satisfy most users. A lower weight will be appreciated, of course, but not at any cost. Fuel consumption, in particular, will remain an important item in the cost of operation.

Besides, the air intake, filtering, sound-proofing and exhaust devices are costly and bulky. Since they are proportional to the air flow rate, they will be three times larger for a turbine than for a diesel engine.

The cost per hp. of present diesel engines remains an objective for the gas turbine to reach Now, this cost should still decrease considerably with high super-charging simultaneously with the weight per hp., which should reach 1 kg/hp. in the near future.

For all these reasons, I believe that, as a medium range prospect, the essential asset of the turbine will remain its extremely light weight , which is however counterbalanced by the heat recovery device. The field of applications in which light weight is imperative will then remain the monopoly of simple cycle turbines.

This field of applications, which includes aircraft propulsion, overlaps only very slightly the field of ground propulsion, which will probably remain the domain of the highly super-charged diesel engine.

The gas turbine, fitted with a recovery device, which hardly stands comparison with today's diesel engine, should get ready to carry out an offensive against tomorrow's diesel engine.

It is, however, unquestionable that, in the more or less distant future, we will succeed in describing elegantly a turbine engine cycle, which is so satisfactory to the mind.

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