Engineering Journal of Qatar University, Vol. 1, 1988.

"THE POTENTIAL OF THE CO-ROTATING COMPRESSOR IN GAS TURBINE APPLICATIONS"

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ABSTRACT

The concept of co-rotating axial compressor was conceived primarily as a means of overcoming the inherent rotational speed mismatch between a fan and its driving turbine in moderate to high bypass ratio turbo-fan engines. In essence, the arrangement serves as a torque converter or "aerodynamic gear box" transferring a fraction of the energy supplied by the high pressure turbine to the compressor casing shaft via the co-rotating casing. Thus, considerable modifications in performance of gas turbine engines can be achieved. The present work examines the potential of exploiting the co-rotating compressor concept in major applications of gas turbines, among which are; turbo-fan, turbo-jet, turbo-propeller, turbo-shaft and basic industrial gas turbine engines.

The paper formulates torque and power constraints for engines which show feasibility of utilizing a co-rotating compressor. It also formulates speed relationships of components. Basic analyses are carried out, critical parameters are indicated, and characteristics of performance of engine components are discussed, with due consideration of the likelihood of load variation.

1. INTRODUCTION: HISTORY OF CO-ROTATING COMPRESSOR CONCEPT

Although the concept of co-rotating compressor casing was described by Howel (1), it does not seem to have been implemented until Rolls Royce proposed its use as means of driving a low-speed fan with a high-speed turbine, which device the company called a "Fanstat". In a collaborative program, the National Research Council of Canada modified a three-stage transonic research compressor provided by Rolls Royce to investigate the performance of both basic compressor and modified compressor with rotational casing. The test facility, procedure and results are described by Chappel (2) and by Chappel, Millar and Swiderski (3, 4).

The conventional version of the compressor had a design pressure ratio of about 3, at a mass flow rate of 11 kg/s and a rotational speed of 19,000 rpm. The modified compressor had design speed of 6800 rpm for the casing and 25,800 rpm for the rotor, maintaining the relative compressor speed of 19,000 rpm. Compressor performance was very similar to that of the conventional compressor, except that the low speed surge was so "soft" that it made hard to detect.

It may be concluded, from satisfactory aerodynamic and mechanical behaviour of an adaption of a conventional compressor, that a compressor designed from the start with a rotating casing should perform satisfactorily both aerodynamically and mechanically (5).

2. PRINCIPLE OF OPERATION

The objective of rotating the compressor casing in the direction in which air stream applies torque to it is to extract power from the air stream within the compressor, and this power can be utilized in different ways according to application. Simple conservation of angular momentum applied to the compressor of this arrangement shows that, with axial flow at inlet and outlet, the torque transmitted to the rotating casing must equal the torque provided to the compressor via the high speed rotor shaft. Consequently, the power transferred to the rotating casing (P_c) is related to the power supplied to compressor rotor (P_r) by the relation:

$$\frac{P_c}{N_c} = \frac{P_r}{N_r} \quad \text{or} \quad P_c = P_r \cdot \frac{N_c}{N_r} \qquad (1)$$

in which N_c and N_r are rotational speeds of compressor casing and rotor respectively. Equation (1) defines the fraction of the power supplied to the compressor which can be transferred through the rotating casing, or the "energy split".

Aerodynamically, the compressor performance is determined by the speed

difference between the rotor and casing, the "relatively speed" N_{rel}, i.e. N_{rel}
i.e.

$$N_{rel} = (N_r - N_c)$$

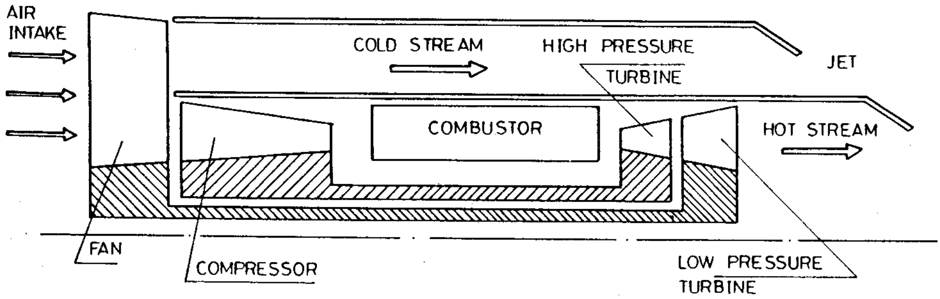
N_{rel} is equivalent to the compressor speed in a conventional compressor, and it is the quantity which governs compressor flow and pressure ratio. If the rotor speed N_r is held fixed by a governor, then any reduction of casing speed N_c will increase N_{rel}, and compressor flow rate and pressure ratio will increase, together with compressor power and torque, and consequently casing torque. These modifications in engine performance need to be examined in view of load characteristics and engine configurations of different gas turbine applications in order to determine the feasibility of the co-rotating compressor.

THE CO-TURBOFAN ENGINE

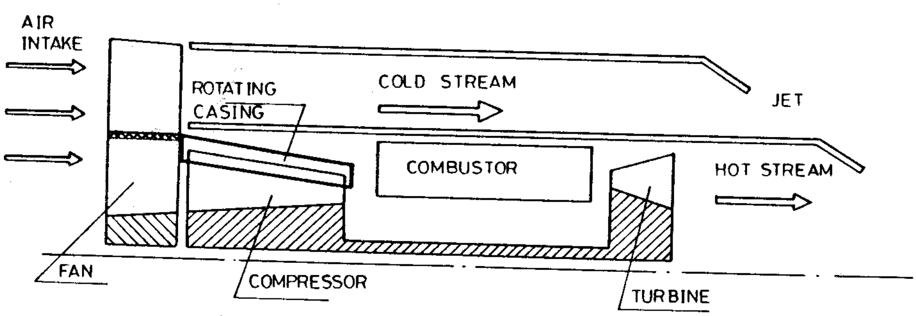
The concept of co-rotating blade rows within an axial compressor was conceived primarily as a means of overcoming the inherent rotational speed mismatch between a fan and its driving turbine in moderate to high bypass ratio turbofan engines. In essence, the arrangement serves as a torque converter or "aerodynamic gear box" which will permit both fan and turbine to operate closer to their individual optimum speeds. The co-turbofan engine is a turbofan engine which utilizes the co-rotating compressor, the fan being either mounted on, or driven by the rotating casing. A schematic representation of such an engine is presented in Figure (1).

A cooperative project was undertaken by the National Research Council of Canada (NRC) and Rolls-Royce (Canada) Ltd., to examine this concept of compressor spooling, called by Rolls-Royce, the "Fanstat". Analytical and experimental results of that project showed that the co-rotating compressor concept was both mechanically and aerodynamically feasible, (4). When the concept is applied however, to turbofan engines, the ratio of the power supplied to compressor casing P_c, which is used to drive the fan, to the power supplied to the core compressor to compress the hot stream P_r must be equal to the ratio of casing to rotor rotational speeds N_c/N_r. This is indicated by equation (1). At high bypass ratios, where the need to use a co-rotating compressor concept had originated in order to optimize fan and turbine speeds, typical value for power ratio implies that the ratio of fan to compressor speed should be relatively high, which is physically impractical.

Turbo-fan engines having bypass ratios of the order of 4 and above, fan pressure ratios of the order of 1.5 and above, may require casing-to-rotor speed ratios of the



a - Twin-Spool Turbofan Engine.



b - Proposed Co-Turbofan Engine.

Fig. 1: Schematic Representation of the Co-Turbofan Engine.

order of 40 to 80%. This in fact represents the major difficulty in applying the concept to turbo-fan engines. This difficulty diminishes however, as the bypass ratio and fan pressure ratio go below 3 and 1.5 respectively, and the co-turbofan engine retains then the advantage of dispensing with the fan shaft and its driving turbine by an extra turbine stage. A deeper insight of engine behaviour in this particular case can be obtained through discussion of co-turbojet engine presented in the next section.

4. THE CO-TURBOJET ENGINE

Among the gas turbine power plants proposed for aircraft applications is the Co-Turbojet Engine, which has the unique feature that the high pressure (H.P.) compressor casing co-rotates with its rotor at about one third speed, and is connected to the rotor of the low pressure (L.P.) compressor. The H.P. compressor casing derives power from the high pressure air stream passing through the H.P. compressor, which is used to drive the L.P. compressor, thus dispensing with the L.P. turbine and its shaft by adding one or two stages to the H.P. turbine.

A conventional two-spool turbojet engine is shown schematically in Figure (2). Figure (3) shows a schematic representation of the proposed co-turbojet engine. The additional complexity resulting from having a rotating casing of the H.P. compressor is more than offset by dispensing with L.P. turbine, avoiding having two long concentric high-speed rotating shafts, and saving in length and weight of engine.

The L.P. compressor rotor can be connected to the rotating casing in more than one way. The simplest way is a direct connection between L.P. rotor and rotating casing, Figure (3). This means that both of them would have to possess same speed. Other arrangements involving simple epicyclic gear trains may also be used. As they provide L.P. rotor speed different from casing speed, speeds remain proportional to each other. The additional complexity of design and the additional weight however, make such arrangement less attractive. For the present study, only direct connection between L.P. compressor rotor and H.P. compressor casing is considered.

As indicated by equation (1) the fraction of the H.P. turbine power reappearing at casing shaft is equal to the ratio of casing speed to H.P. turbine speed. This "torque constraint" or "energy split" equation can be written for a co-turbojet engine as follows:

$$P_c = P_{HPT} \cdot \frac{N_c}{N_{HPT}} \qquad (2)$$

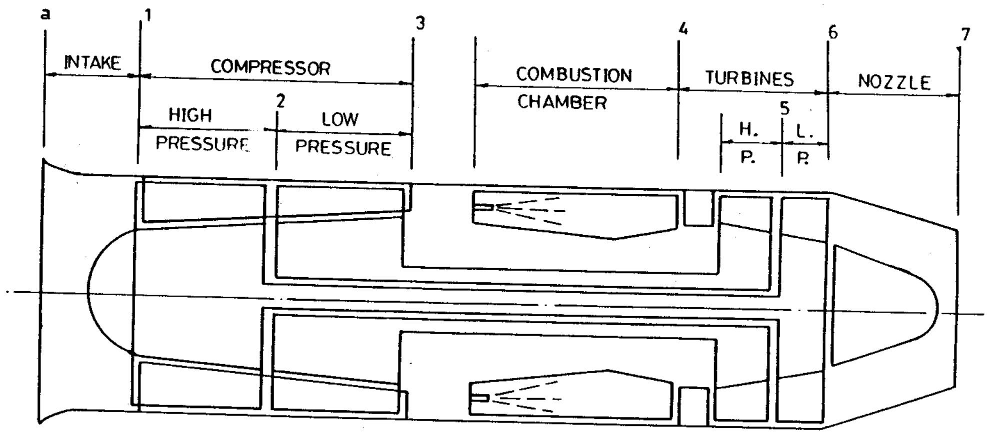


Fig. 2: Schematic Representation of a Conventional Two Spool Turbojet Engine

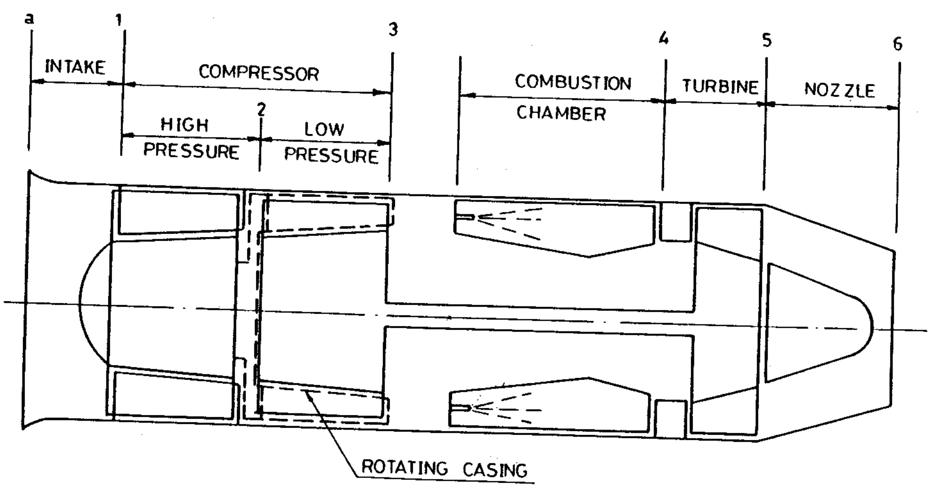


Fig. 3: Schematic Representation of a Co-Turbojet Engine.

the subscript HPT indicating high pressure turbine.

The H.P. turbine provides power to compress the air at H.P. compressor, plus the casing power, or:

$$P_{HPT} = P_{HP\ comp} + P_c \tag{3}$$

Thus:

$$P_c = \frac{N_c}{N_{HPT}} (P_{HP\ comp} + P_c) \quad , \quad \text{or}$$

$$P_c = \frac{N_c}{N_{HPT}} (P_{HP\ comp} + P_{LP\ comp}) \quad , \quad \text{since}$$

$$N_{rel} = N_{HPT} - N_c \quad , \quad \text{then}$$

$$P_c = \frac{N_c}{N_{rel}} \cdot P_{HP\ comp} \tag{4}$$

The quantity N_{rel}, the relative speed between H.P. and casing, $(N_{HPT}-N_c)$, is the effective H.P. compressor speed which governs its flow and pressure ratio. If the H.P. rotor speed is held fixed by a governor, then any reduction of casing speed N_c will increase N_{rel}, and H.P. compressor flow rate and pressure ratio will increase, together with H.P. compressor power and torque, and consequently casing power and torque. On the other hand, this reduction of casing speed reduces proportionally the L.P. compressor speed, which in turn decreases L.P. compressor flow rate and pressure ratio, together with L.P. compressor power. Two observations can be made referring to this discussion.

First, as far as stability of operation (stability of the assigned casing speed) is concerned, the proposed system ensures a great degree of stability. Any decrease of casing speed results in an increase in its driving torque while at the same time it results in a decrease in the casing output torque. (L.P. compressor torque). Both effects accelerate casing speed to its designed value, and vice versa.

Second as far as overall output (overall pressure ratio and mass flow) is concerned, it is controlled by two effects opposing each other. The H.P. compressor tends to increase engine output as N_c decreases, while L.P. compressor tends to do the reverse. The net result depends on some factors, among which are casing-to-rotor speed ratio at the design point, enthalpy rise of both compressors, location of design point on both L.P. compressor map and H.P. compressor map and ambient conditions. Specific selection of these factors by the designer can bring minimal effect on overall performance of compressors, and hence engine operation is almost non affected by variations in casing speed.

Another point of view considers casing speed control as a novel alternative for engine output control. The selection of the above factors would thus follow a different route.

Equation (4) can be rewritten in terms of component's enthalpy rise (Δh) and rate of mass flow (m) as follows:

$$m_1 \Delta h_{12} = \frac{N_c}{N_{rel}} \cdot m_2 \Delta h_{23} \qquad (5)$$

numbers referring to station numbers of Figure (3).

The ratio (N_c/N_{rel}), and the enthalpy rise Δh can be related to the variation in casing speed N^* and design speed ratio K as follows:

$$\frac{N_c}{N_{rel}} = \frac{K N^*_c}{1-K N^*_c} \qquad (6)$$

$$m_1 \Delta h_2 = \frac{K N^*_c}{1-K N^*_c} \cdot m_2 \Delta h_{23} \quad , \qquad (7)$$

$N^*_c = N_c/N^1_c$, N^1_c being the casing speed at design point conditions, and K being equal to N^1_c/N^1_{HPT}, with N^1_{HPT}, the speed of high pressure rotor at design point, assumed constant. Keeping in mind that $N^* = 1.0$ at design point, and assuming a value of $K = 1/3$, then:

$$\Delta h_{12} = \tfrac{1}{2} \Delta h_{23} \qquad (8)$$

Equation (8) shows that the compressors load sharing is practically acceptable.

Typical values of T_1 and overall pressure ratio (PR) result in a (PR_{12}/PR_{23}) ratio in the range of 0.7 to 0.8, or approximately 40 to 45% of total pressure ratio is provided by L.P. compressor, while the remaining 55% to 60% is provided by the H.P. compressor. This is not far from practical applications.

On the turbine side, the H.P. turbine, enhanced with an additional stage, becomes the only source of power for driving both compressors. It has to provide power according to equations (2) & (6), or

$$P_{HPT} = \left(\frac{1}{1 - KN^*_c}\right) \cdot P_{HP\ comp} \tag{9}$$

For $K = 1/3$ and $N^*_c = 1.0$, $P_{HPT} = 1.5\ P_{HPcomp}$. This means that the turbine in such a co-turbojet engine should be capable of supplying 50% more power than a conventional H.P. turbine of a two-spool engine of similar component characteristics. This extra power, which can be handled satisfactorily by an additional turbine stage, is used to drive the L.P. compressor, via the rotating casing.

Matching of two turbines in series, as in the case of conventional two-spool turbojet engine is now no longer required. Matching of a multi-stage turbine and a propelling nozzle should however, be considered carefully for determining engine performance. nequation (7) shows that load sharing between compressors is remarkably affected by the variations in casing speed N^*_c. An increase in N^*_c tends to bring the enthalpy rise of both compressors closer to each other; same can be said about compressor's pressure ratio, and vice versa. As far as the turbine is concerned, an increase in N^*_c means an increase in turbine power as indicated by equation (9). This, in turn, means an increase in overall engine power (thrust). This suggests the use of casing speed as an alternative for controlling turbine output while maintaining constant H.P. rotor speed.

The location of design point on compressor map determines to a great extent the variation in compressor efficiency at off-design operation. It should be always kept in mind that a variation of casing speed produces opposite effects on L.P. compressor and H.P. compressor speeds (i.e.) increases speed of one compressor and reduces speed of the other).

For predetermined values of compressors enthalpy rise (or total temperature rise), any change in ambient temperature is reflected on compressors pressure ratios PR_{12} and PR_{23}. Generally, a decrease of T_1 tends to increase the ratio (PR_{12}/PR_{23}) and increase engine overall pressure ratio for same amount of compressor power. On the other hand, as compressors power increase ($\Delta\ T_{12}$ and $\Delta\ T_{23}$ increase), at fixed ambient temperature, of course the engine overall pressure ratio increases, but does also the ratio (PT_{12}/PR_{23}), approaching 80% at relatively high overall pressure ratio.

Despite the attractive features displayed, the concept of co-turbojet engine needs however more investigation on both mechanical design and thermodynamic sides. Future work should involve a study of possible savings in cost, length and weight of such an arrangement. Control philosophies may be examined in view of the differing behaviour of the engine.

5. CO-TURBO-PROPELLER ENGINE

The co-turbo-propeller engine, with its propeller fully or partially driven by the compressor rotating casing, is schematically shown in Figure (4). It is typical of a propeller load, either for a ship or for an aeroplane, in which case the required driving torque increases with the square of speed. Consequently, the power absorbed varies with the cube of rotational speed of the propeller. When the transmission efficiency and gear ratio are known, the load characteristic in terms of the net power actually required from the turbine shaft, and the turbine shaft speed can be plotted, Figure (5).

Considering engine configuration (a) in Figure (4), as the propeller shaft speed increases, the compressor casing speed increases proportionally, while the propeller power requirement increases as the cube of speed ratio. As far as the co-rotating compressor is concerned, any increase in casing speed, however, decreases compressor effective speed, for fixed gas-generator speed, and consequently reduces engine power. This is exactly the reverse of load requirements. The same problem appears when considering the engine configuration (b), Figure (4). Consequently the co-turbopropeller engine is not a feasible gas turbine engine.

6. CO-TURBOSHAFT ENGINE

Among gas turbine power plants proposed for vehicular applications is the co-turboshaft engine. Unlike turbo-fan engines, when applying the co-rotating compressor to a turboshaft engine, the energy split need not be specified, so that the speed ratio can be selected through other criteria. Hence a co-turboshaft engine provides flexibility for the designer in selecting casing to rotor speed ratio, to optimize the performance to suit the application.

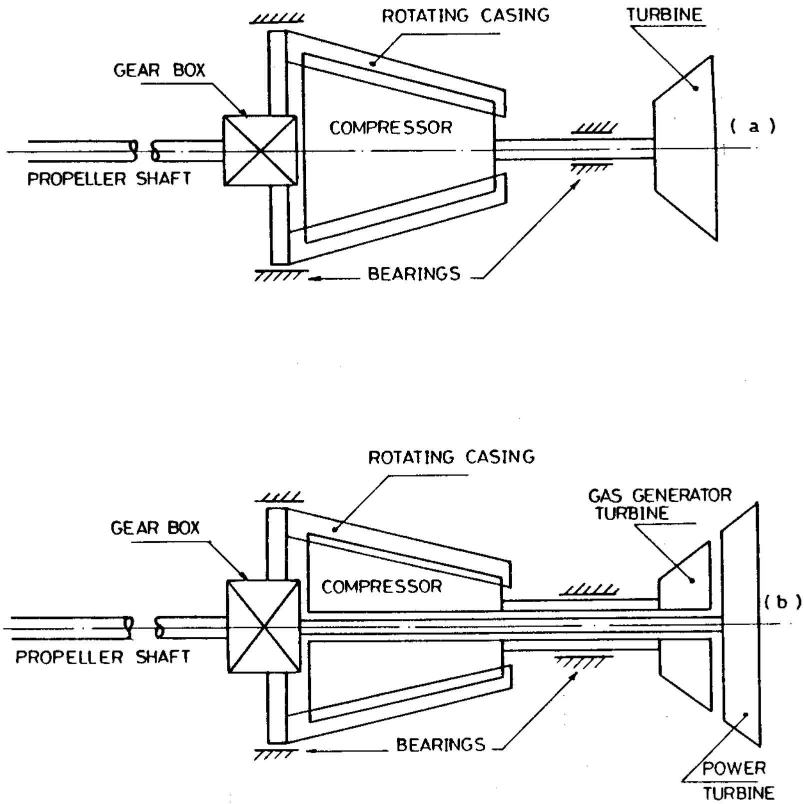


Fig. 4: Co-Turbopropeller Configurations.

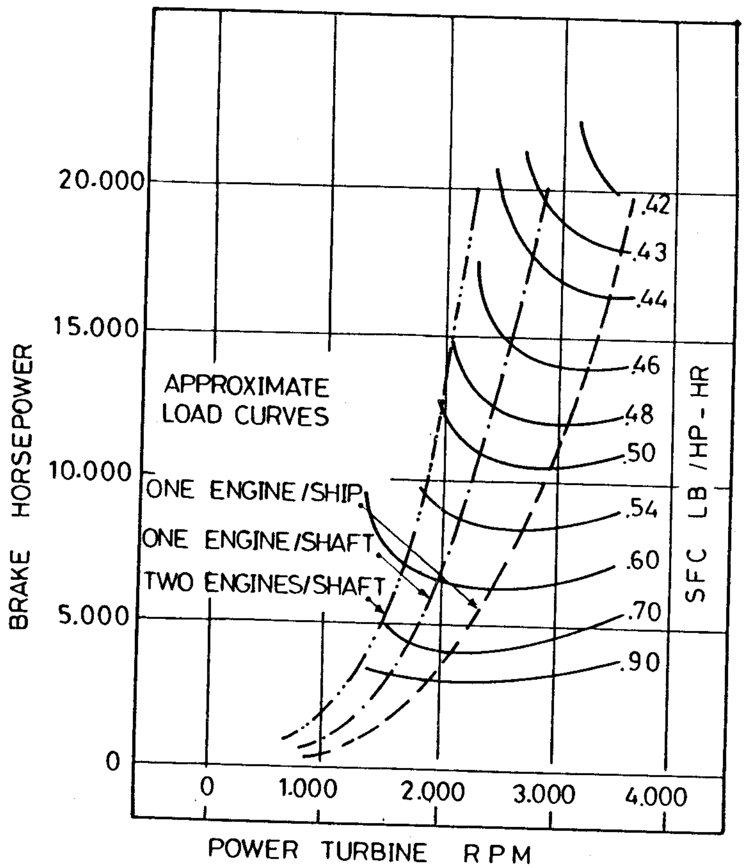


Fig. 5: Typical Load Characteristics of a Ship Propeller (adapted from reference (10)).

In a co-turboshaft engine arrangement, both compressor casing and power turbine contribute, via a gear box, to the output of the engine. The gear box maintains constant proportionality between output shaft speed, casing speed, and power turbine speed. With the gas generator rotor speed governed at 100%, decreasing output shaft speed will, therefore, simultaneously reduce both power turbine speed and compressor casing speed. The latter change will increase the relative speed of the core compressor and hence raise the air mass flow rate through the machine, thereby increasing both power turbine and casing output. The result will be evidenced by a significantly enhanced torque multiplication at full throttle part speed, which may offer considerable advantages in vehicular applications. A description of the co-turboshaft engine can be found in references (5) and (8) while a comprehensive discussion of its performance characteristics is presented in reference (7). Figure (6), adapted from reference (7) shows a schematic cross-section of a co-turboshaft engine.

The power turbine, being geared to the compressor casing and output shaft, will reflect any reduction in output shaft speed. In a conventional gas turbine, the power turbine will receive essentially constant gas power from a governed gas generator, and the torque-rise exhibited by such an engine is a result of the relation.

Output = Output Torque x Speed
= Power Turbine Efficiency x Gas Power

Because the power turbine efficiency falls off with decreasing shaft speed, this relationship results in a nearly linear increase of torque, rising to about twice the design torque at stall. In the co-turboshaft engine, this relationship still holds, but now the gas horsepower available to the turbine rises as turbine speed drops, so that output torque rises more rapidly with decreasing shaft speed than that of a conventional engine with same governed gas generator speed.

This steep torque rise made the co-turboshaft engine a promising candidate for such machinery as earthmovers, which have large load changes during their duty cycle. While this torque characteristic would be equally useful for lighter vehicles, it seems unlikely that this power plant would be economically attractive at power levels lower than about 400 KW. The stall torque is about 250% to 300% of the design torque, as compared to the factor of two which is typical for a conventional engine. At about 50% of the design speed, the ratio of the two torque curves is even higher if the power turbine design point is located at a position somewhat lower than the maximum efficiency point.

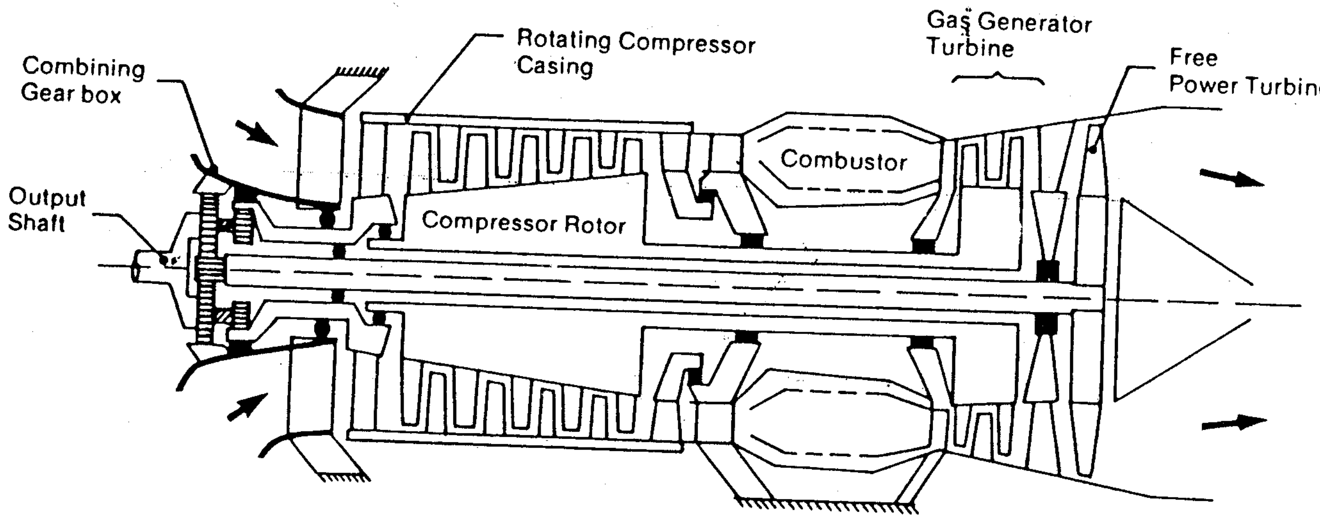


Fig. 6: Schematic Cross Section of a Co-Turboshaft Engine.

This increase in torque is reflected, of course, in an increase in power at part speed. The power is above the design point power down to about 25% of design shaft speed, whereas the power output of a conventional gas turbine engine falls as the speed drops below the design speed because the gas power is constant while the turbine efficiency falls off. For applications in heavy construction equipment, although the torque rise of a conventional engine is considerably better than that of a diesel engine, it is likely that a torque converter may be required in addition. The steeper torque curve of the co-turboshaft engine may obviate this necessity if that engine is used. Figures (7) and (8), adapted from reference (7) show the predicted part-speed performance of the co-turboshaft engine for two possible locations of design point on compressor map.

By judicious choice of compressor and power turbine design points, the increased torque and power at reduced shaft speeds can be achieved with little or no increase in turbine inlet temperature, and no change in gas generator speed and stress.

One incidental advantage which the co-turboshaft gives to its designer is the ability to load up the gas generator turbine somewhat, and to unload the power turbine. For engines of this pressure ratio, which is likely to be typical for vehicular engines with regenerators, the designer needs a total of three stages for the two turbines. This presents no problem on a single shaft engine, but on a free turbine engine about one and a half stages each would be desirable for the two turbines so that some compromise is necessary, with attendant sacrifice in efficiency. With the power transfer obtained naturally in the co-turboshaft, a two-stage gas generator turbine and a single stage power turbine match the demands well. This feature is similar in result to that achieved by such engines as the GM Power Transfer described about 25 years ago, or the more recent Kronogard KTT engine developed in Sweden. Neither of these machines, of course, was designed to increase shaft power or torque more than a conventional engine, with decreased output shaft speed.

At this early stage of development, little consideration has been given to the mechanical feasibility of the co-rotating compressor engine. While a 3-stage compressor has been operated with surprisingly little mechanical difficulty, it was a short, rugged compact design and might not be characteristic of longer, more flexible machine, which might suffer from vibrational problems which did not occur with the experimental compressor. Before tackling the formidable task of designing a full co-turboshaft engine, it was however, necessary to establish that the favourable performance characteristics which were predicted by intuitive arguments would be

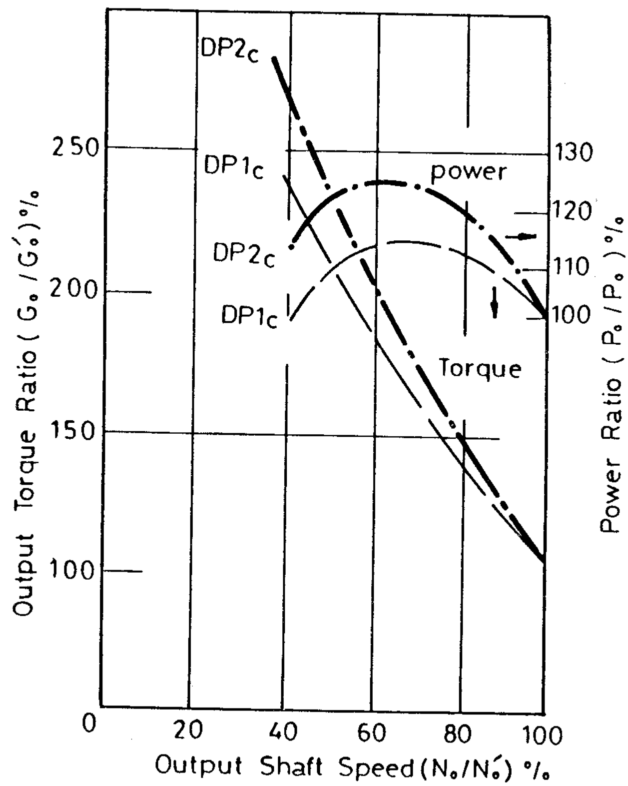


Fig. 7: Effect of Compressor Design Location on Co-Turboshaft Engine Torque and Power.

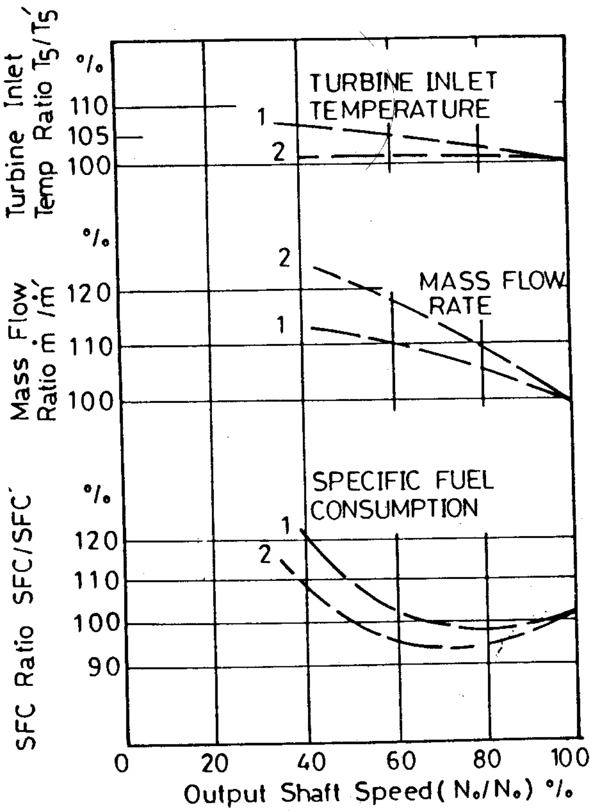


Fig. 8: Effect of Compressor Design Point Location on Co-Turboshaft Engine SFC and T_5.

confirmed by a more thorough investigation. Simulator studies have shown that the torque-speed behaviour is, in fact, as good as had been expected, and that this behaviour was obtained without the anticipated penalty in turbine inlet temperature. While further simulation studies are required, the results obtained so far would encourage the serious consideration of this type of power plant for future applications.

7. CO-ROTATING COMPRESSOR IN INDUSTRIAL GAS TURBINES

Utilization of the concept in industrial application depends on load characteristics (or power requirements) and suitability of methods of handling the power delivered by the compressor casing.

The most common type of load with a single-shaft industrial gas turbine is the electric generator which runs at constant rotational speed with load varied electrically. An engine having a co-rotating compressor is obviously not suitable for such application. This is mainly because it delivers power at two different speeds, (casing speed and rotor shaft speed). In addition, mechanical problems arising from massive-casing rotation, which may weigh tons in this case, represent another disadvantage.

In the fields of gas pipelines, liquid pipelines and repressurizing systems, where the number of gas turbines in-service has been increasing rapidly during the past decades, the main role of gas turbine engine is to compress the fluid, in compression stations, by means of a power turbine-gas generator arrangement. Performance of such gas turbines is similar to that of a free — turbine turboshaft engine. Therefore, it is expected that the application of the co-rotating compressor concept in pipeline pumping industry results in some advantages and disadvantages as that of the co-turboshaft engine herein discussed.

8. CONCLUSION

Remarkable modifications in performance can be achieved by introducing a compressor with co-rotating casing into gas turbine engines. Modifications of performance depend substantially on the fraction of energy transferred through compressor casing or "energy split", on the method of utilizing this fraction of energy and on load characteristics. This paper examines the potential of exploiting this concept in different gas turbine applications.

At high and moderate bypass turbofan engines, where the need to use a co-rotating compressor concept had originated, typical values for energy split implies that the ratio of fan to compressor speeds should be relatively high, which is physically impractical. When applied to multi-spool turbojet engines, the concept is shown to be attractive. With suitable arrangement, it is possible to dispense with the low pressure turbine and its rotor by adding one stage to the high pressure turbine. Possible savings in engine length, weight and complexity in addition to increased flexibility of thrust control are some of the engine merits.

Load characteristics represent the major difficulty in applying this concept to turbopropeller engines in marine and aircraft propulsion.

When applying the concept to turboshaft engines, energy split need not be specified and the designer can select the casing to rotor speed ratio value to optimize the performance to suit the application. Among the attractive performance modifications of free-turbine engines is the enhanced power and torque at part speed operation.

Application of the concept in utility power generation is shown to be unsuccesse. However, the concept retains its merits, as for the co-turboshaft when applied to gas and liquid pipeline compression stations.

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