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THE CO-TURBOJET ENGINE: A NOVEL CONCEPT FOR MULTI-SPOOL TURBOJET ENGINES

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ABSTRACT

The novel feature of the co-turbojet engine is the rotation of the casing of the high pressure compressor. This casing co-rotates, at lower speed, with the high pressure rotor of the turbojet engine, hence the name co-turbojet. The compressor casing derives power from the high pressure air stream passing through the compressor and this power is used to drive the low pressure compressor, which is mechanically coupled to it. This paper studies the consequencies of such basic modification in respect to both engine design and performance.

The paper presents what is thought to be a suitable mechanical arrangement of this engine. With this arrangement, it is possible to dispense with the low pressure turbine and its rotor by adding one stage to the high pressure turbine. Consequently a great saving in engine length and weight can be achieved. Avoiding problems and complexity, resulting from having two shafts rotating one inside the other at different varying speeds, are some of the attractive features of the co-turbojet engine.

Basic thermodynamic analysis of the engine are carried out to determine the critical parameters affecting its performance. The paper formulates torque and power constraints of the engine as well as components speed relationship. Performance of engine components in view of the likely different loading is disscussed. Among the attractive characteristcs of the engine is the control of thrust by means of speed control of the rotating compressor casing. The increased flexibility of engine control adds to its merits.

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INTRODUCTION

Among the gas turbine power plants proposed for aircraft applications is the Co-Turbojet Engine. This engine has the unique feature that the high pressure (H.P.) compressor casing co-rotates with its rotor at about one third the speed, and is connected to the rotor of the low pressure (L.P.) compressor. As will be seen in the next sections, this feature results in major changes in engine performance and design. The present paper is mainly concerned with introducing the concept, formulating components speed relationship, setting the basic performance constraints, indicating critical parameters and giving some guide lines for predicted performance of the Co-Turbojet engine.

EXPERIMENTAL VERIFICATION OF THE ROTATING CASING CONCEPT

While the concept of a rotating casing on a high speed compressor might appear to be mechanically forbidding, such a system has been tested extensively at the National Research Council of Canada, in a joint program with Rolls Royce. The concept of co-rotating blade rows within an axial compressor was conceived primarily as a mean 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 the fan and the turbine to operate closer to their individual optimum speeds. A co-operative project was undertaken by the National Research Council of Canada (NRC) and Rolls-Royce (Canada) Ltd. to examine this concept of compressor spooling, called the "Fanstat" by Rolls Royce.

Analytical and experimental results of that project showed that the concept was both mechanically and aerodynamically feasible [1]. The test unit was an experimental 3-stage compressor with shaft speed of 25,000 rpm and casing speed of 5,000 rpm. Compressor performance was very similar to that of the conventional compressor, except that the low speed surge was so "soft" as to be hard to detect.

POSSIBLE APPLICATIONS OF THE CONCEPT

The objective of rotating the compressor casing is to extract power from the air stream within the compressor, thereby extracting power from the the high pressure turbine. The high pressure turbine thus is assumed to provide the compressor air flow with power proportional to its enthalpy rise and the rotating casing with power proportional to its enthalpy drop. Simple conservation of angular momentum applied to the core 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 by the high pressure turbine via the high pressure rotor shaft. Thus the fraction of the H.P. turbine power which reappears at the casing shaft is equal to the ratio of casing speed to H.P. turbine speed. This is the basic expression for the energy split between the rotating casing and the core compressor. It represents the major difficulty in applying the co-rotating casing concept to a turbofan engine with the fan being mounted on, or driven by the rotating casing. 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 values for energy split implies that the ratio of fan to compressor speed should be relatively high, which is physically impractical.

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When applied to a turboshaft engine, the energy split need not be specified, so that the speed ratio can be selected by other criteria. Hence a co-turboshaft engine provides flexibility for the designer in selecting casing to rotor speed ratio, to optimize the performace to suit the application. While the torque characteristic is the most attractive feature of this engine, the power transfer from the gas generator turbine, through the compressor casing to the output gearbox, provides a degree of design flexibility which can also be an advantage. The performance of a co-turboshaft engine based on this concept was studied in a Ph.D. thesis by M.R.S. Okelah, and some of the results of that study are presented in references [2] and [3].

When applied to a two-spool turbojet engine, typical values for the energy split between the H.P. rotor and L.P. rotor implies that the casing-to-rotor speed ratio, necessary for transmitting sufficient power to L.P. compressor from H.P. rotor (via rotating casing), lies within a practically acceptable range.

CO-TURBOJET ENGINE CONFIGURATION

A conventional two-spool turbojet engine is shown schematically in Fig.1 . Fig.2 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 dipensing with L.P. turbine, avoiding having two long-concentric-high speed-rotating shafts, and savings in length and weight of engine.

The L.P. compressor rotor can be connected to the rotating casing in more than one way. The simplist way is a direct connection between L.P. rotor and rotating casing, as shown in Fig. 2, but it means that both of them have to have the same mechanical speed. Other arrangements involving simple epicyclic gear trains may also be used. They provide L.P. rotor speed different from casing speed, yet speeds remain proportional to each other. However, the additional complexity of design and additional weight make such arrangement less attractive. For the purpose of the present work, the direct connection between L.P. rotor and H.P. compressor casing is only considered.

PRINCIPLES OF OPERATION

The rotating casing torque is equal to the H.P. rotor torque, for axial flow into and out of the H.P. compressor, so that the rotating casing power is related to the H.P. turbine power by the "torque constraint", written as:

$$\frac{\dot{W}_c}{N_c} = \frac{\dot{W}_{HPT}}{N_{HPT}} \tag{1}$$

However, the H.P. turbine provides the power to compress the air, at H.P. compressor, plus the casing power, or $\dot{W}_{HPT} = \dot{W}_{HPcomp} + \dot{W}_c$, thus :

$$\dot{W}_c = \frac{N_c}{N_{HPT}} \cdot (\dot{W}_{HPcomp} + \dot{W}_c) \quad \text{or}$$

$$\dot{W}_c = \frac{N_c}{N_{rel}} \cdot \dot{W}_{HPcomp} \qquad (2)$$

The quantity N_{rel}, the relative speed between H.P. rotor 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 governer, 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 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), 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 to its designed speed, and vice versa.

Second, as far as overall output (overall pressure ratio and mass flow),is concerened it is controlled by two effects opposite to each other. The H.P. compressor tends to increase engine output as N_c decreases while L.P. compressor tends to do the opposite. 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. A specific selection of these factors by the designer can produce minimal effect on ovrall performance of compressors, and hence engine operation is almost not affected by variations in casing speed. Another philosophy of design may consider casing speed control a novel alternative for engine output control, the selection of the above factors accordingly has to follow a different course. A discussion of the effect of these factors is presented in the next sections.

COMPRESSORS SPEED RELATIONSHIP

As indicated by the torque constraint, equation (1),

$$\frac{\dot{W}_c}{N_c} = \frac{\dot{W}_{HPT}}{N_{HPT}} .$$

If we call the casing-to-rotor speed ratio at design point conditions K, then $K = N'_c / N'_{HPT}$ where the sign (') indicates dsign point values. By definition, $N_{rel} = N_{HPT} - N_c$ and $N'_{rel} = N'_{HPT} - N'_c$, thus:

$$\frac{N_{rel}}{N'_{rel}} = \frac{N_{HPT} - N_c}{N'_{HPT} - N'_c}$$

It we consider engine operation at fixed H.P. rotor speed, then N_{HPT} always equals N'_{HPT}, and the previous equation becomes:

$$\frac{N_{rel}}{N'_{rel}} = \frac{1 - N_c/N'_{HPT}}{1 - K} = \frac{1}{1-K} \left(1-K \cdot \frac{N_c}{N'_c}\right) \text{ , or}$$

$$N^*_{rel} = \frac{1}{1-k}\,(1-K\,N^*_c) \qquad (3)$$

Equation (3) relates variation in H.P. compressor effective speed ($N^*_{rel} = N_{rel}/N'_{rel}$) to variation in L.P. compressor speed ($N^*_c = N_c/N'_c$) for fixed H.P. rotor speed. Fig. 3 shows how this relationship is strongly affected by the factor K. Increasing K amplifies the effect of casing speed variations on H.P. compressor effective speed. For example a 20% reduction in casing speed produces 5% and 10% increases in H.P. compressor effective speed, for K = 1/5 and 1/3 respectively, as shown in Fig. 3. This in turn results in a remarkably different overall engine performance. A value of K ranging between 0.25 and 0.35 is thought to be a reasonable selection after considering the following factors:

a) The larger the value of K, the more effective the concept becomes

b) The larger the value of K, the higher the casing speed becomes, increasing the mechanical problems

c) The larger the value of K, the more over-sizing of the H.P. compressor in needed [2].

THERMODYNAMIC CONSIDERATIONS

The power split between the two compressors is determined from equation (2) as:

$$\dot{W}_{LPcomp} = \frac{N_c}{N_{rel}} \cdot \dot{W}_{HPcomp} \qquad (4)$$

or in terms of enthalpy rise:

$$\dot{m}_1 \Delta h_{12} = (N_c/N_{rel})\,\dot{m}_2 \Delta h_{23} \qquad (5)$$

where the numbers indicate station numbers of Fig. 2, $\dot{m}$ is the rate of mass flow and Δh is the enthalpy rise.

The ratio (N_c/N_{rel}) can be determined, using equation (3), thus:

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

Also Δh can be written in terms of the average specific heat C_p and total temperature rise ΔT, thus equation (5) becomes:

$$\dot{m}_1 C_{p_{12}} \Delta T_{12} = \left(\frac{KN^*_c}{1-KN^*_c}\right) \dot{m}_2 C_{p_{23}} \Delta T_{23} \qquad (7)$$

or in terms of compressor pressure ratio (PR) and efficiency n :

$$\dot{m}_1 C_{p_{12}} \cdot \frac{T_1}{n_{LPcomp}} \cdot [(PR_{12})^{r_1-1/r_1} - 1] = \frac{KN^*_c}{1-KN^*_c} \cdot \dot{m}_2 \cdot C_{p_{23}} \cdot \frac{T_2}{n_{HPcomp}} \cdot [(PR_{23})^{r_2-1/r_2} - 1] \qquad (8)$$

where r_1 and r_2 are specific heat ratios of L.P. and H.P. compressors respectively. Equations (7) and (8) show the interaction between L.P. and H.P. compressors with all variables involved. These equations represent the compatibility of work between both compressors and rotating casing. Also, the work compatibility of H.P. rotor can be formulated as follows:

$$\dot{W}_{HPT} = \dot{W}_{HPcomp} + \dot{W}_{LPcomp}$$

using equations (4) and (6)

$$\dot{W}_{HPT} = \left(\frac{1}{1-KN_c^*}\right) \cdot \dot{m}_2 C_{P_{23}} \cdot \Delta T_{23}/n_m \qquad (9)$$

where n_m is the mechanical efficiency , thus

$$\dot{m}_4 \cdot C_{P_{45}} \cdot \Delta T_{45} = \left(\frac{1}{1-KN_c^*}\right) \cdot \dot{m}_2 \cdot C_{P_{23}} \cdot \Delta T_{23}/n_m \qquad (10)$$

The compatibility of flow between the compressors yields

$$\left(\frac{\dot{m}\sqrt{T_2}}{P_2}\right) = \left(\frac{\dot{m}\sqrt{T_1}}{P_1}\right) \times \frac{P_1}{P_2} \times \sqrt{\frac{T_2}{T_1}} \qquad (11)$$

and between H.P. turbine and nozzle

$$\left(\frac{\dot{m}\sqrt{T_6}}{P_6}\right) = \left(\frac{\dot{m}\sqrt{T_5}}{P_5}\right) \times \frac{P_5}{P_6} \times \sqrt{\frac{T_6}{T_5}} \qquad (12)$$

where $\frac{\dot{m}\sqrt{T}}{P}$ is the non-dimensional flow [4]

DESIGN POINT PERFORMANCE

If we consider the simplified analysis where $\dot{m}_1 = \dot{m}_2 = \dot{m}$ throughout the engine, $Cp_{12} = Cp_{23}$ and $r_1 = r_2$, equation (7) then reduces to

$$\frac{\Delta T_{12}}{\Delta T_{23}} = \frac{KN_c^*}{1 - KN_c^*} \qquad (13)$$

Keeping in mind that $N_c^* = 1.0$ at design point, and assuming a value of $K = 1/3$,

$$\Delta T_{12} = 1/2\ \Delta T_{23} \qquad (14)$$

In terms of compressor pressure ratio

$$[(PR_{12})^{r-1/r} - 1] / [(PR_{23})^{r-1/r} - 1] = \frac{KN_c^*}{1-KN_c^*} \cdot \frac{T_2}{T_1} \cdot \frac{n_{LPcomp}}{n_{HPcomp}} \qquad (15)$$

For same compressors design point efficiency,

$$[(PR_{12})^{r-1/r} - 1] / [PR_{23})^{r-1/r} - 1] = \frac{1}{2}\ \frac{T_2}{T_1} \qquad (16)$$

Equations (14) and (15) show that the compressors load sharing is practically acceptable. Typical values of T_1 and overall pressure ratio 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 ramining 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 compressor. It has to provide power according to equation (9), or

$$\dot{W}_{HPT} = \left(\frac{1}{1 - KN_c^*}\right) \cdot \dot{W}_{HPcomp} \tag{17}$$

for $K = 1/3$ and $N_c^* = 1.0$, $\dot{W}_{HPT} = 1.5 \ \dot{W}_{HPcomp}$

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 characteistics. 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 engines is not required any more. However, matching of a multi-stage turbine and a propelling nozzle should be considered carefully for determining engine performance.

PERFORMANCE AT OFF-DESIGN CONDITIONS

Equations (13) and (15) show that the load sharing between compressors is remarkably affected by variations in casing speed N_c^*. An increase in N_c^* tends to bring temperature rise of both compressors closer to each other. the same can be said about compressor's pressure ratio, and vice versa. As far as turbine is concerned, an increase in N_c^* means an increase in turbine power as indicated by equation (17). 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 compresor map determines to a great extent the variation in compressor efficiency at off-design operation. Variations of η_{LPcomp} and η_{HPcomp} affects compressors pressure ratio and power as indicated by equation (8). 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 ambinet temperature reflects on compressors pressure ratios PR_{12} and PR_{23} as indicated by equation (15). 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 (ΔT_{12} and ΔT_{23} increase), at fixed ambient temperature, of course engine overall pressure ratio increases, but also the ratio of (PR_{12}/PR_{23}) increases approaching 80% at relatively very high overall pressure ratio.

However, the concept of co-turbjet engine needs more investigations on both mechanical design side and thermodynamic performance side. Future work should involve estimations of savings in cost, length and weight of engine. Engine control philosophies have to be examined in view of the expected behaviour of engine. Detailed engine peformance at both design point and off-design conditions based on typical component characteristics and selected design parameters is another area proposed for future work.

CONCLUSION

A novel concept for turbojet engines, featuring a rotating casing of the high pressure compressor is introduced. With the proposed mechanical arrangement, it is possible to dispense with the low pressure turbine and its shaft by adding one stage to the high pressure turbine. Consequently, savings in engine weight, length, complexity and cost can be achieved. Engine performance is shown to be remarkably affected by casing speed, which suggests using it as an alternative for engine control. Basic engine characteristics and compressors speed and load relationships are formulated. Effects of casing speed variations, location of design points on compressor maps, and ambient conditions on compressors relative loading are discussed. Proposed areas for future work are indicated.

REFERENCES

1. Chappell, M.S., and Millar, D.A.J., "Analytical and Experimental Studies of an Axial Compressor with Co-Rotating Stators". Paper presented at the 3rd International Symposium on Air Breathing Engines, Munich, Germany, Paper No. 76014, March 1976.

2. Okelah, M.R.S., and Millar, D.A.J., "Some Performance Characteristics of a Gas Turbine Engine with Enhanced Torque Amplification at Part Speed", Paper presented at the 1985 International Congress and Exposition of the Society of Automotive Engineers, Detroit, Feb. 25 - March 1, 1985, SAE Paper No. 850031.

3. Millar, D.A.J., Chappell, M.S. and Okelah, M.R., "The Co-Turboshaft - A Novel Gas Turbine Power Plant for Heavy Equipment". Paper presented at the Gas Turbine Conference and Exhibit, San Dieago, Calfornia, March 1979, ASME paper No. 79-GT-132.

4. Cohen, H., Rogers, G.F.C. and Saravanamuttoo, H.I.H., "Gas Turbine Theory", Second Edition, Longman, 1975.

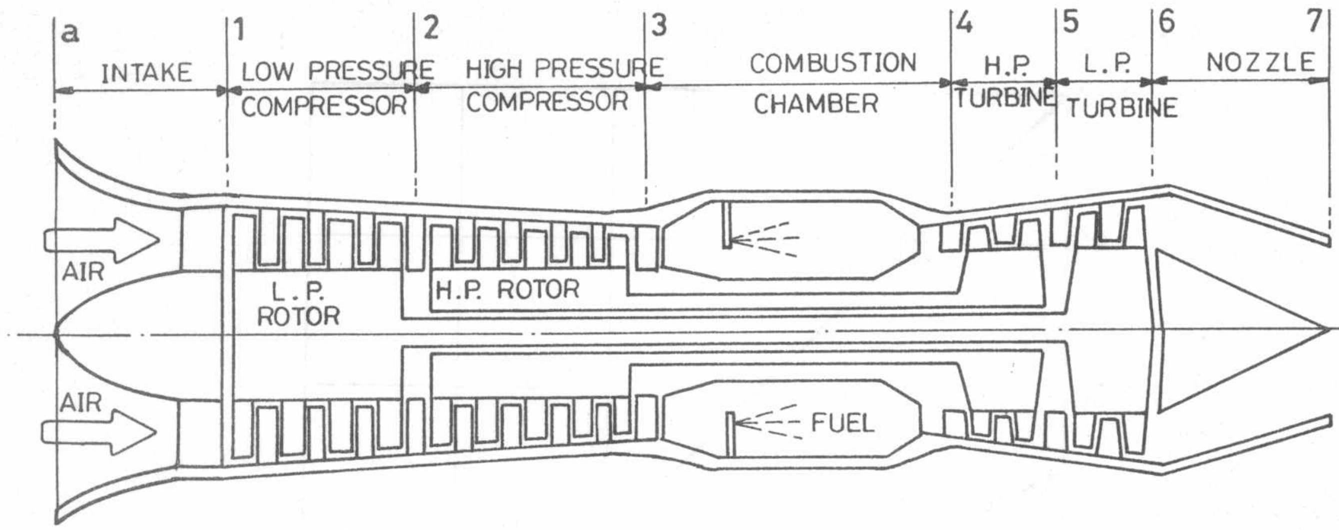

Fig. 1 SCHEMATIC REPRESENTATION OF A CONVENTIONAL TWO SPOOL TURBOJET ENGINE

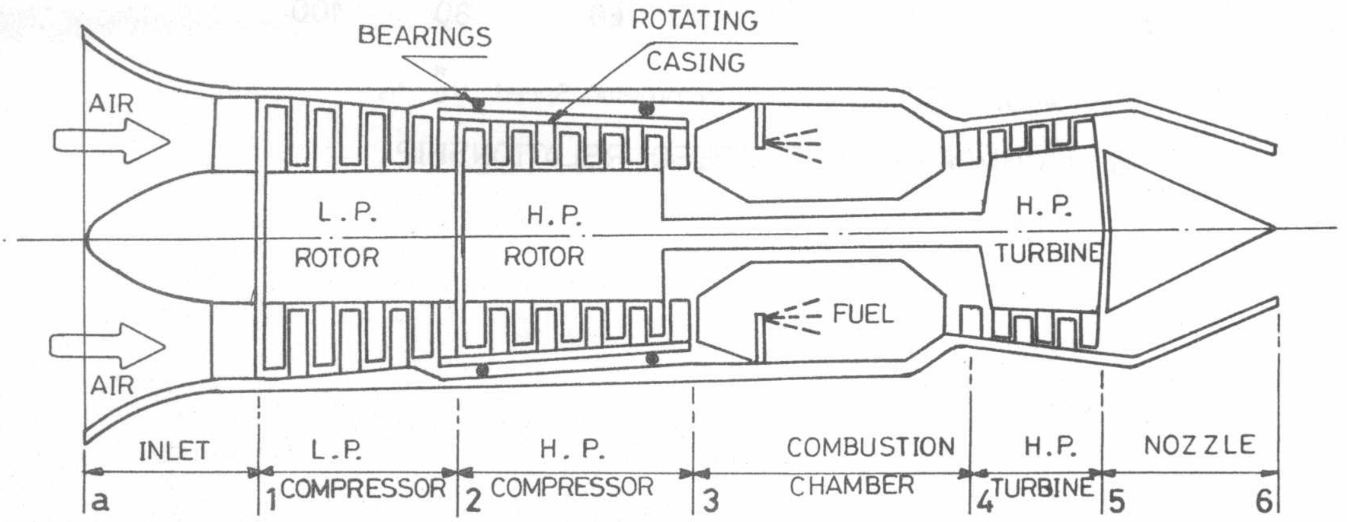

Fig. 2 SCHEMATIC REPRESENTATION OF A CO TURBOJET ENGINE

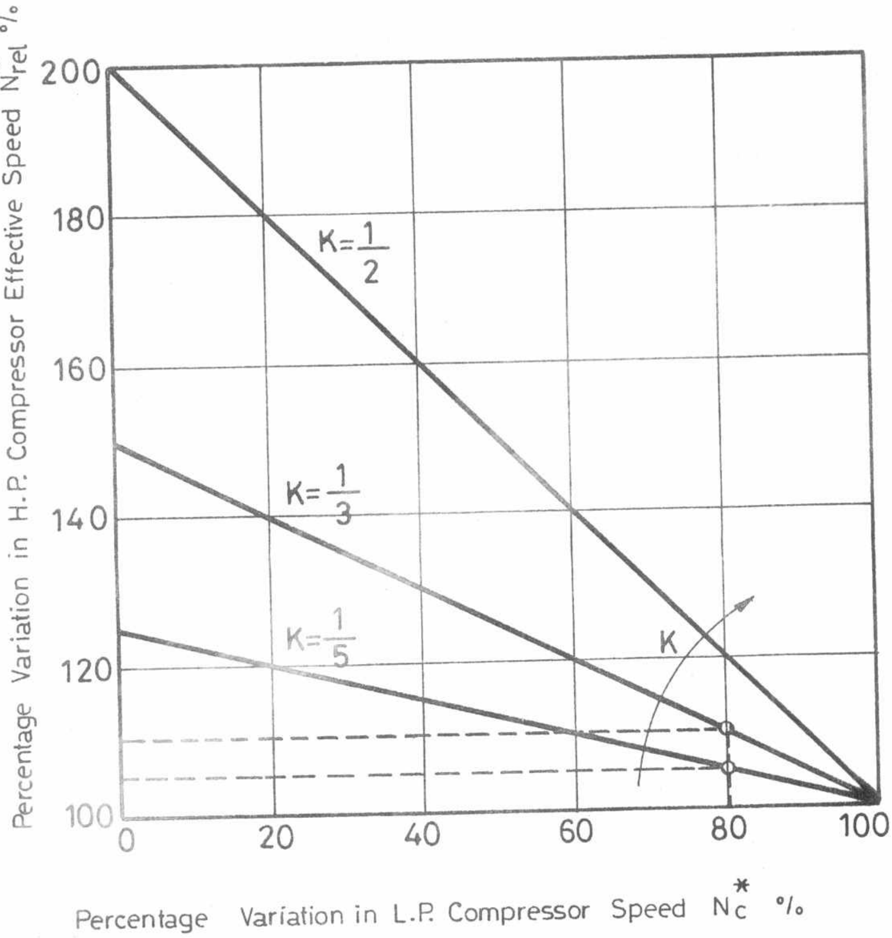

Fig. 3 COMPRESSOR'S SPEED RELATIONSHIP