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EFFECT OF AXIALLY NON-UNIFORM ROTOR TIP CLEARANCE ON THE PERFORMANCE OF A HIGH SPEED AXIAL FLOW COMPRESSOR STAGE

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BREAK

ABSTRACT

An axially non-uniform type of rotor tip clearance was conceived and tried on a single stage compressor. This concept is based on the advantages of a smaller tip clearance in the front portion of the blade and a larger clearance in the rear portion which allows a higher tip leakage flow to interact with the passage secondary flow, casing wall boundary layer, separated flow on the blade suction surface and the scraping vortex, which are more prominent at the rear portion of the blade. Experimental results indicated that an axially non-uniform clearance can provide improved performance of a compressor stage. Providing the tip clearance in the compressor casing instead of at the blade tip indicated certain advantages. An 'optimum' value of rotor tip clearance was noticed for this compressor stage, both for axially uniform and axially non-uniform clearance.

NOMENCLATURE:

- t = Clearance between rotor tip and casing liner
- Po = Total pressure
- To = Total temperature
- η = Adiabatic efficiency
- CN = Corrected speed of rotation, rpm.
- γ = Ratio of specific heats
- m = Corrected mass flow rate
- Cp = Specific heat at constant pressure
- PR = Total Pressure Ratio

Subscripts:

- 1 = Compressor inlet (Total pressure measuring station)
- 2 = Rotor inlet
- 3 = Rotor outlet
- 4 = Compressor outlet (Total pressure measuring station)
- s = With reference to smaller tip clearance
- l = With reference to larger tip clearance

INTRODUCTION:

The rotor tip clearance in an axial flow compressor has been an important topic of investigation as it considerably influences the pressure rise, efficiency, stall margin and radial shift of streamlines in the flow passages of the compressor. Many investigators have conducted experiments on compressors with widely varying geometric and aerodynamic parameters to find out the influence of rotor tip gap on compressor performance [1,2,3,4,5,11,15].* And all the experiments that have been conducted so far have axially uniform rotor tip clearance i.e., the tip clearance has the same value from the leading edge to the trailing edge. These investigations have revealed that as the tip clearance is reduced the efficiency of the compressor increases. But too small a clearance may not be the most desirable clearance, as the tip clearance flow is known to beneficially interact with the passage secondary flow, annulus wall boundary layer and blade suction surface boundary layer [4,5,12,15]. Therefore, at very low levels of tip clearance, the efficiency actually decreases with the decrease in tip gap thereby reversing the trend that prevailed at higher levels of tip clearance. This gives rise to the concept of 'optimum' tip clearance, which gives the maximum efficiency of the compressor stage [4,5,15]. But this phenomenon (i.e., optimum clearance) was not noticeable in all the compressors, as in some compressors the efficiency continuedtto increase with the decrease of tip clearance all the way [4,5].

Further, it was observed that with the alteration of the tip gap the stall margin limit was altered; and tip gap affected both the inception and geometry of rotating stall.

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* Numbers in the square brackets are the references listed at the end of the paper.

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Non-uniform Clearance:

It is clear that tip clearance flow is influenced by several factors and also it will, in turn, influence other flows in the blade passage like the main flow, passage secondary flow, annulus wall boundary layer, blade suction surface boundary layer, etc. Since the tip clearance flow is in the opposite direction of passage secondary flow near the casing (Fig.3), it will reduce the magnitude of the secondary flow and consequently the losses due to secondary flows. The leakage flow is also found to be beneficial in decreasing the annulus wall boundary layer growth [11] and also in reducing the boundary layer separation on the blade suction surface [12]. Further, the 'Scraping effect' (due to the relative motion between the casing wall and blade tip) influences the pressure distribution on the blade surfaces near the tip and its effect is opposite to that caused by the tip clearance [10, 15].

Further, as the air flows from the leading edge to the trailing edge of the blade the thickness of the boundary layer also continuously increases, which results in a thicker boundary layer in the rear half of the blade and also on the rear half of the end walls. The passage secondary flow is due to the flow of air through the boundary layers on the two blade surfaces as well as on the hub and casing. Since the thickness of the boundary layer is more in the rear portion of the blade surfaces and end walls, the passage secondary flow and the annulus wall boundary layer in the casing, will be more pronounced at the rear portion of the blade. Further, the blade suction surface boundary layer gets separated close to the trailing edge; and the scraping vortex often appears in the rear half of the blade tip [14].

Since the tip clearance flow interacts beneficially with all the four phenomenon indicated above (i.e., secondary flow, casing boundary layer, blade boundary layer and the scraping vortex) and these are more pronounced in the rear half of the blade, it becomes obvious that more tip leakage flow would be desirable at the rear half of the blade. At a particular value of the larger tip gap (optimum gap) it is quite likely that the tip clearance flow will substantially neutralise the adverse effects caused by the above four phenomenon. Therefore, a larger tip clearance at the rear half would be beneficial. Based on this analysis the new concept of axially non-uniform clearance has been formulated. In this new concept a smaller clearance is provided from the leading edge to midchord (which provides the well established advantages of a smaller clearance), and a larger clearance from midchord to the trailing edge (Fig.4).

In view of the above considerations, experimental investigations were carried out at the National Aerospace Laboratories, Bangalore, to determine the influence of axially non-uniform rotor tip clearance on the performance of a high speed compressor stage. Further, experiments were also conducted on axially uniform tip clearance, so that the compressor performance with both axially non-uniform and uniform tip clearances could be compared.

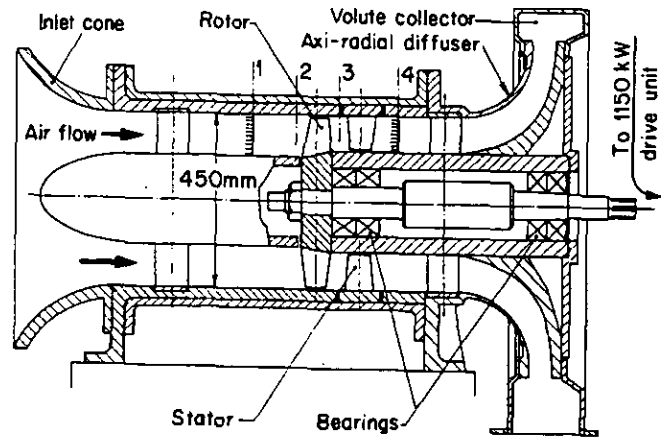


Fig. 1 Schematic of the research compressor

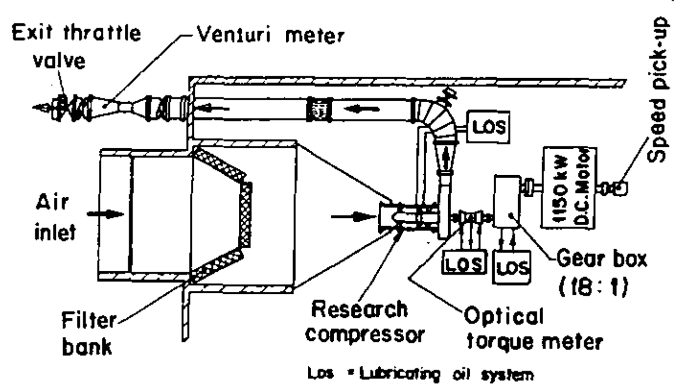


Fig. 2 Schematic layout of axial flow compressor research facility

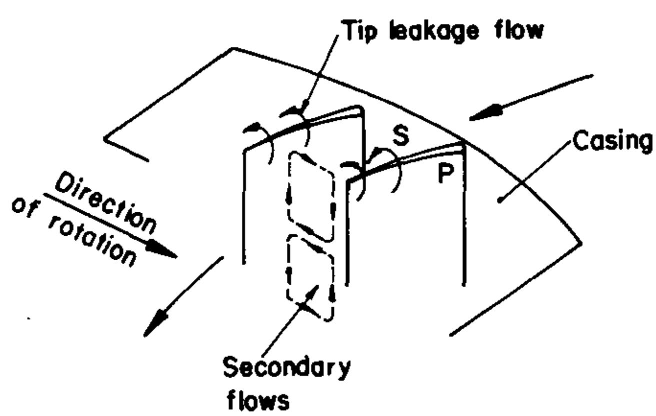
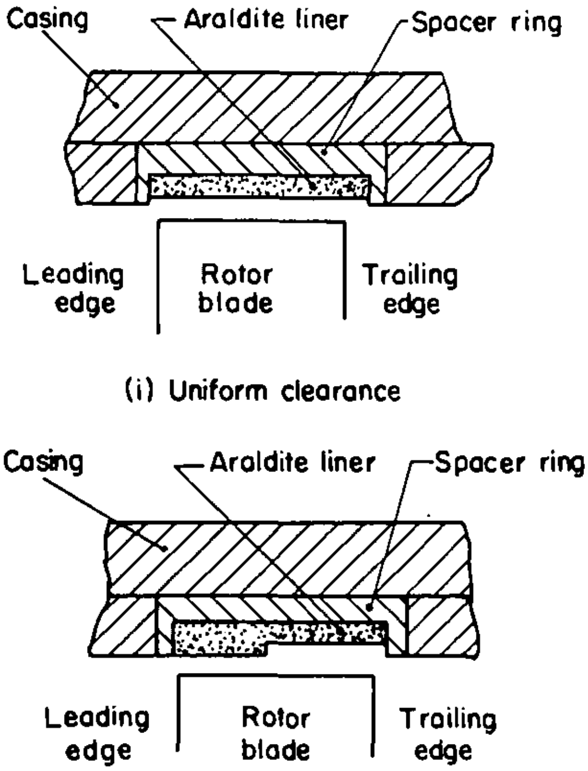


Fig. 3 Interaction of tip leakage flow with secondary flows



(ii) Non uniform clearance

Fig. 4 Method adopted for providing rotor tip clearance

Method of Providing Rotor Tip Clearance:

In the present investigation the rotor tip clearance was provided by machining the casing ring surrounding the rotor blade (Fig.4). In this method, a softer material like araldite was embedded in the mild steel spacer ring enveloping the rotor blade, and this araldite ring was machined to the required specification. Initially, the liner was machined to provide a uniform rotor tip clearance of 0.25 mm and subsequently to higher values of uniform & non-uniform clearance (clearance refers to non-rotating tip clearance). This method was adopted as it would not only be economical and easier, but also it may have several aerodynamic advantages, which have been discussed elsewhere in the paper.

Description of the Experimental Facility:

Experimental investigations were conducted using the single stage axial flow compressor (Fig.1) in the Propulsion Division of the National Aerospace Laboratories. The details of the compressor are as follows:

Rotor tip diameter, D	=	0.450 m
Hub-tip ratio at rotor inlet	=	0.525
Hub-tip ratio at rotor outlet	=	0.60
Number of blades in the rotor	=	21
Type of blading = NACA design (6% thick, symmetrical circular arc profile).		
Blade chord at tip, C	=	0.0842 m
Stagger angle at tip	=	47°
Solidity at tip	=	1.25
Solidity at hub	=	2.05
Blade mean height, h	=	0.1008 m
Aspect ratio	=	1.04
Number of blades in the stator	=	18
Solidity at tip for the stator	=	1.05

Mach triangles at tip corresponding to 9054 rpm are as follows:

Rotor inlet relative mach no	=	0.79
Rotor inlet axial mach no	=	0.43
Rotor blade tip mach no	=	0.65
Rotor outlet relative mach no	=	0.59
Stator inlet mach no	=	0.51
Stator outlet mach no	=	0.46

The compressor was driven by a variable speed 1150 KW D.C. Motor through a step up gear box and an optical torquemeter. The mass flow through the compressor and the pressure ratio were controlled by operating the exhaust throttle valve located downstream of the compressor. This is an open loop facility with the air flowing through the bell mouth, experimental compressor, axi-radial diffuser, volute, exhaust ducting, venturimeter and exhaust throttle valve (Fig.2). Four total pressure rakes were installed at measuring stations 1 and 4 (Fig.1) to measure the total pressure at compressor inlet and outlet. These total pressure rakes consisted of 15 pitot tubes stacked radially. Details of the experimental facility and instrumentation are given in refs.6 and 7.

Experimental Procedure:

Tests were conducted first with five uniform tip clearances of 0.25, 0.5, 1.0, 1.5 and 2.0 mm and the compressor stage performance was evaluated. Subsequently, test runs were made with four axially non-uniform tip clearance of 0.5 - 1.0 mm, 0.5 - 1.5 mm, 0.5 - 2.0 mm and 0.5 - 2.5 mm. In this case the tip clearance from leading edge to mid-chord was maintained constant at 0.5 mm. For each of the above tip clearances, test runs were made at three different speeds of the compressor (corrected speeds of 5561 rpm, 7760 rpm and 9054 rpm) and at each speed the air mass flow was varied from maximum (exhaust throttle valve fully open) to the mass flow corresponding to the stall point.

Evaluation of Test Parametes:

The average total pressure at both the inlet and outlet of the compressor stage was found by the area averaging method. The total pressures, mass flow rates, and input torque, were measured using total pressure rakes, venturimeter and optical torquemeter respectively. The mass flow rates and the rotational speeds were corrected to standard sea level conditions. The adiabatic efficiency of the compressor stage was calculated based on torque measurement and applying appropriate corrections for bearing and disc friction losses [8,13]. The efficiencies were also calculated based on air temperatures at inlet and outlet of the compressor stage. The stall point was determined by observing the sudden change in compressor noise level and tone. The observations on stall were also cross checked using hot wire anemometer probes. All the experimental data was fed to a computer based data acquisition system (MACSYM 350) for on-line processing and obtaining compressor performance.

Results and Discussions:

The results obtained on the influence of uniform and non-uniform rotor tip clearance on the performance of the compressor stage are shown in Figs.5,6,7,8 and 9.

The results for axially uniform and axially non-uniform tip clearance are discussed separately in order to indicate quantitatively the differences in compressor behaviour with the alteration in tip gap. In the case of uniform tip clearance the clearance was altered over the entire chord length, whereas in the case of non-uniform clearance the clearance from the leading edge to midchord was NOT altered (Kept constant at 0.5 mm), but only the clearance from midchord to trailing edge was altered.

Axially uniform rotor tip clearance:

An analysis of the results indicated that the adiabatic efficiency of the compressor stage continuously increased as the tip clearance was reduced from 2.00 mm to 0.50 mm at each of the three corrected speeds i.e., 5561 rpm (131 m/s), 7760 rpm (183 m/s) and 9054 rpm (207 m/s). The increase in efficiency for each of the speeds were 6.0% 4.34% and 3.45% respectively. The trends are shown in Fig.5 and 6.

But when the tip clearance was reduced lower than 0.5 mm (to a value of 0.25 mm), the efficiency actually decreased by 2.88%, 3.37% and 3.28% respectively at each of the three speeds. This indicated that this compressor stage reached its maximum stage efficiency when the tip clearances was 0.5 mm (this clearance represented 0.5% of blade height and 0.6% of blade chord). This, perhaps, is the optimum tip clearance for this particular compressor stage (Fig.6). However, there was only a marginal change in the pressure ratio with the decrease of tip clearance. The efficiencies shown in Fig.6 (as well as in Fig.8) are the maximum values obtained for any tip gap and for a constant speed.

Regarding the stall margin, with the increase of tip clearance upto 1.5 mm the stall margin also increased at all the three speeds, (Fig.5). But there was no further improvement in stall margin when the tip clearance was increased from 1.5mm to 2.00mm.

Axially non-uniform tip clearance:

When the clearance from mid-chord to trailing edge was increased in steps of 0.5 mm, the efficiency decreased for each of the three speeds (Figs.7 & 8).

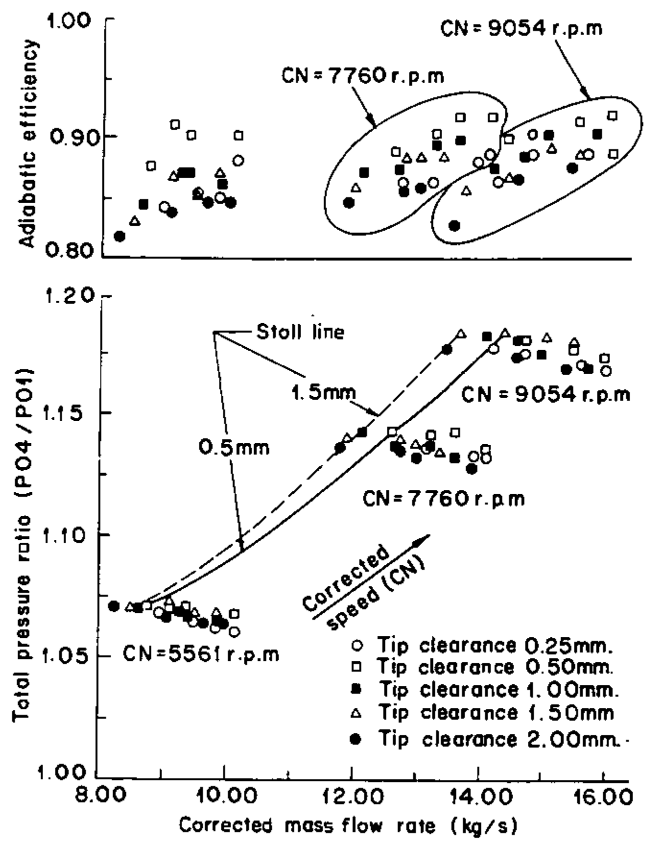


Fig. 5 Compressor performance characteristics with different axially uniform tip clearance

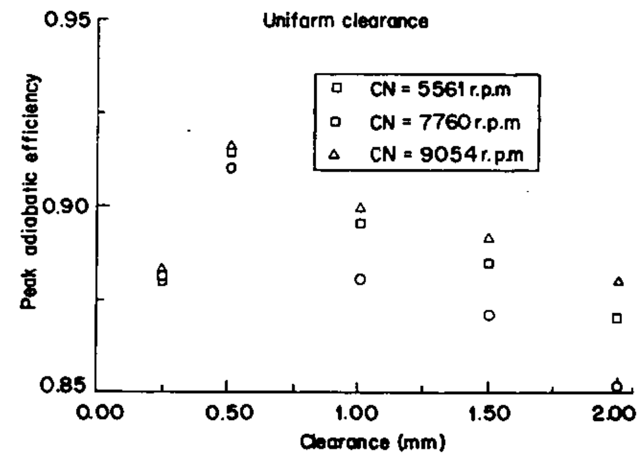


Fig. 6 Variation of efficiency with tip clearance (axially uniform)

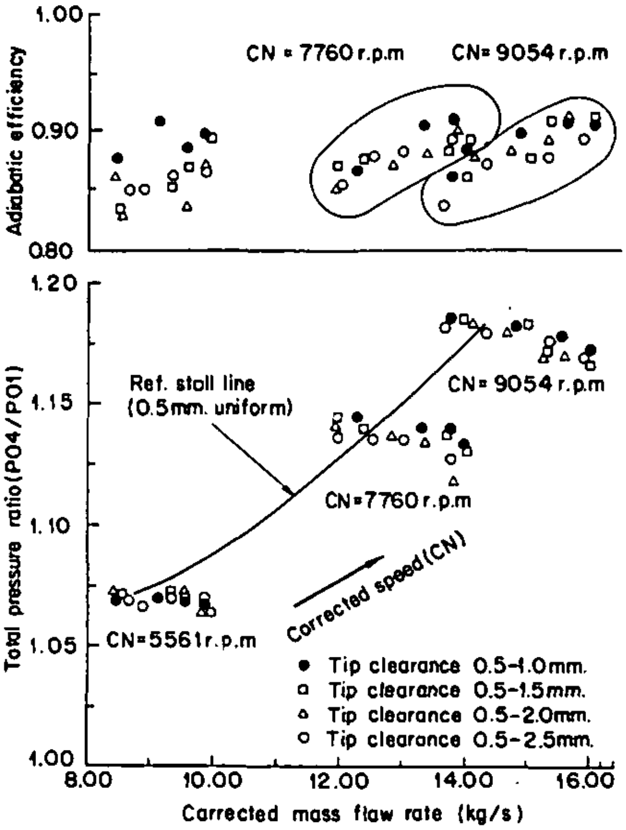


Fig. 7 Compressor performance characteristics with different axially non-uniform tip clearance

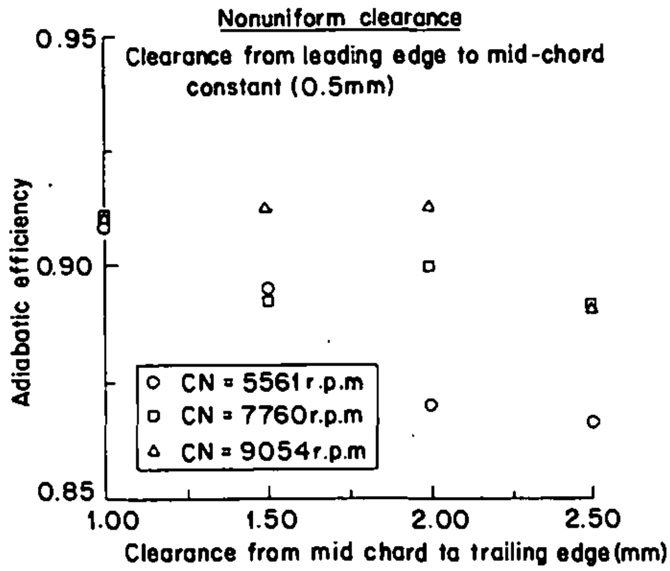


Fig. 8 Variation of efficiency with tip clearance (axially non-uniform)

The decrease in efficiency when the clearance was increased from 1.0 mm to 2.5 mm was 4.19%, 1.92% and 1.86% respectively for each of the three speeds. Similarly, when the clearance was 2.0 mm, the decrease in efficiency was 3.80%, 1.92% and -0.61% (actually an increase). The trends clearly indicate that the increase of tip clearance in the rear half of the blade, does not reduce the efficiency significantly, (whereas in the case of uniform clearance the decrease in efficiency with the increase of clearance was quite significant, as indicated earlier).

The efficiencies of 0.5 - 1.5 mm combination non-uniform clearance appear to be as good as (or even slightly better than) 1.0 mm uniform clearance and much better than 1.5 mm uniform clearance. Similarly, 0.5 - 1.0 mm combination of non uniform clearance appears to have almost the same efficiency as 0.5 mm uniform clearance and much better than 1.0 mm uniform clearance. Further, the stall margin for the combination of 0.5 - 1.0 mm was higher than with uniform clearances of either 0.5 or 1.0 mm (Fig.9). Thus, the combination of 0.5 - 1.0 mm was not only better than all other non-uniform configurations but also better than even 0.5 mm uniform clearance.

This indicates that although the tip clearance is higher in the rear portion of the blade, the efficiency has not decreased. It is possible that the higher tip clearance flow at the rear portion of the blade has effectively neutralized the adverse effects due to the passage secondary flow, casing wall boundary layer, separated blade boundary layer and the scraping vortex, resulting in the improved performance of the compressor.

Regarding the changes in stall margin, it did not follow any particular order when the tip clearance (from midchord to trailing edge) was altered (Fig.7). However, for all the four combinations of non uniform tip clearance the stall margin was positive (i.e. batter) as compared to the reference line. (For determining the stall margin for any value of tip clearance - both uniform and nonuniform - the stall line at a uniform clearance of 0.5 mm was taken as the 'reference line').

Effects of Providing the Clearances in the Casing:

It is necessary to examine the effects of providing the tip clearance by machining the casing liner instead of the blade tip (for both uniform and non-uniform cases). The increase of stall margin with the increase of tip clearance may be also due to the fact that the higher tip clearance was obtained by increasing the depth of the groove in the casing liner. Although this configuration is not similar to the casing treatment, where the depth of machining (of the treated casing) is about 20% to 50% of the blade height [16], but still the machined groove in the casing liner appears to have provided the required benefit for the prevention of stall.

Further, machining the casing (instead of the blade tip) may have certain advantages - the radial

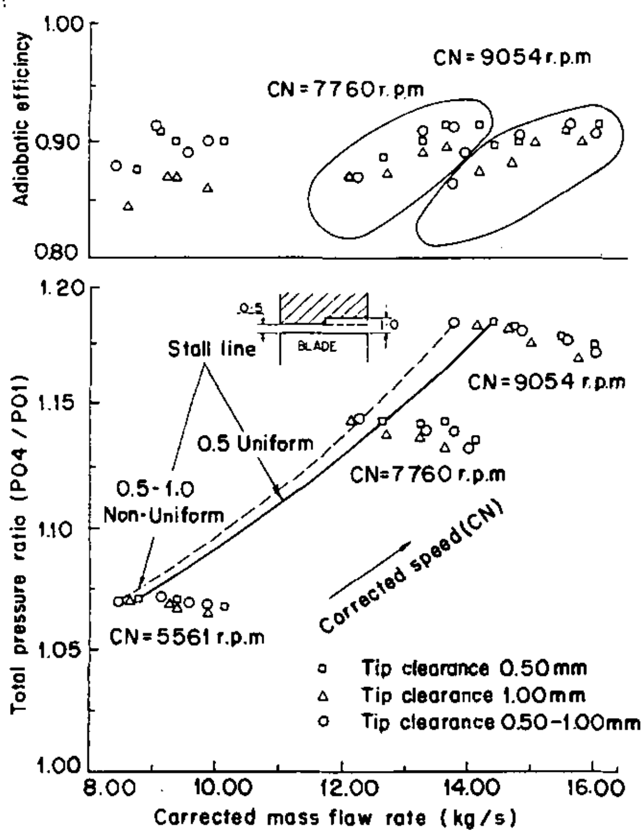


Fig. 9 Compressor performance characteristics with uniform and non-uniform tip clearances (comparison)

shift of the streamlines would be reduced, and the main flow is much less disturbed. It is quite likely that the leakage flow through the tip would also be reduced because the leakage has to pass through the boundary layer material entrenched in the groove. The step in the groove could reduce the direct flow (undeflected) through the clearance space. Further, any configuration of the tip clearance can be obtained without disturbing the blade tip and the profile. The clearance can also be effectively combined with the casing treatment, and the blade tip and the casing wall can be at the same level.

Among the possible disadvantages of providing the clearance in the casing are the effects of the steps in the casing associated with the formation of the clearance groove in the casing. But since the steps are in the annulus wall boundary layer, where the velocities are comparitively smaller, the problems, if any, may not be serious. Even in the casing treatment many configurations are introduced in the casing, like skewed slot, honey comb, circumferential groove, etc. Compared to these configurations, the clearance groove in the casing will cause far less disturbance in the airflow. Further, if the adverse impact of the steps is considerable, the edges can easily be rounded off or champhered.

In this context, the implications of machining the blade tip to provide the increase in clearance was also examined. This method also has certain disadvantages. In this case there will be a direct flow through the clearance space which is undeflected and is, therefore, not available for doing work. Further, by machining the blade tip, the height and geometry of the blade would be altered; consequently the tip speed and the work output would be different. Therefore, the change in compressor performance cannot be attributed solely to the change in the tip gap. Other factors mentioned above could also affect the compressor performance. And in the case of non-uniform clearance, the changed geometry of the blade tip may have more significant impact on compressor performance. And in the case of non-uniform clearance, the changed geometry of the blade tip may have more significant impact on compressor performance.

Comparison of Efficiencies:

When experiments are conducted with different rotor tip clearances or configurations, accurate determination of the changes in efficiencies (with the changes in tip clearance or configurations) is important.

When ever the compressor efficiency with respect to a particular tip clearance is mentioned, it is necessary to clearly indicate whether the efficiency is at the stall point, or it is the peak efficiency for that speed, or it is with reference to any other operating point on the constant speed line. For comparing the efficiency with respect to any other value of clearance, both the efficiencies must have the same reference.

Here again it could be further pointed out, that even if the efficiencies are evaluated, say, at their respective stall points for the two clearances, the operating points in the compressor map will be different; with the result the mass flow rates, pressure ratios and power input will also be different. Therefore, from an analysis of the experimental results although it may be possible to infer qualitatively that with the decrease of tip clearance the efficiency increases, quantitatively the values obtained could be misleading. This is because the quantum of change in the efficiency is not solely due to change in the value of tip clearance, but it may be partly due to alterations in other parameters like pressure ratio, mass flow, etc., since the locations of the operating points in the compressor map are different in the two cases. For, it is well known that for any compressor (even with a fixed tip clearance and fixed speed) the efficiency undergoes a change when the operating point in the compressor performance map is shifted.

In order to overcome the above difficulties and anamolies, a different procedure could be used which is proposed below:

The adiabatic efficiency of a compressor stage is given by the expression :

$$\eta = \frac{\text{Isentropic work of compression for a given pressure ratio and a given mass flow rate}}{\text{Actual work of compression for the same pressure ratio and same mass flow rate.}} \qquad (1)$$

The above expression can be rewritten [8] as

$$\eta = \frac{m\,C_p\,T_{01}\left[\left(P_{04}/P_{01}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\text{(Net Power Input to the Compressor stage)}} \qquad (2)$$

When we try to compare the efficiencies of the compressor stage with two different clearances (or configurations), say 'ts' (smaller tip clearance) and 'tl' (larger tip clearance), first the efficiency 'η' with 'ts' is experimentally evaluated corresponding to a mass flow rate of m(s), pressure ratio of (PR)s and a speed of (CN)s. Next, when the compressor stage is operating with the larger tip clearance 'tl', the exhaust throttle valve and also the <u>speed</u> are slightly readjusted so that we get the same mass flow rate [(m)l = (m)s] and the same pressure ratio [(PR)l = (PR)s]. This manipulation of exhaust throttle valve and speed will have to be done iteratively till we get the same operating point for 'tl' (i.e., the mass flow rate and the pressure ratio are the same), but the speed (CN)l will be different. Since the losses are apparently more when the tip clearance is large, additional work input is required to account for these higher losses. To obtain this additional work the compressor blade speed has to be increased. Therefore, (CN)l will be higher than (CN)s.

Referring to equation (2) the numerator will be the same for both the cases ('ts and 'tl'). But the denominator will be different for each case - higher in the case of 'tl' and lower for the case 'ts'. Therefore, the efficiency for 'tl' (larger clearance) will be lower than for 'ts' (smaller clearance). The values of efficiencies obtained by the above procedure are accurate and are truly indicative of the change in efficiency due only to the change in tip clearance. No other factor, other than the change in tip clearance, is involved in effecting this change of efficiency. It must be pointed out here, that this procedure should be used <u>only for comparing</u> the efficiencies. However, for obtaining the complete performance map of the compressor (with 'tl'), the usual standard procedure (of keeping the speed constant) should be followed.

<u>Optimisation of the Non-Uniform Rotor Tip Clearance</u>

In the experiments described in this paper since the combination of 0.5 mm - 1.0 mm clearance has given encouraging results we have to examine whether it is possible to improve the tip configuration still further. The following factors are to be considered for obtaining further improvements. A larger clearance should be provided only at those locations where it can be beneficial and at other locations along the blade tip only smaller clearance should be provided. Therefore, from the leading edge up to about 50% of the chord, very small clearance (purely from mechanical considerations) may be maintained. From about 50% of the chord the tip gap could be increased gradually (not necessarily in a single step as in the present experiment) in such a way, the interaction of the tip leakage flow with the secondary flows, casing wall boundary layer, etc., can give beneficial results.

In order to estimate the magnitude of the tip clearance flow that would be required in the rear portion of the blade (and the corresponding tip gap) it is necessary to know approximately the magnitude of the secondary flows, thickness of the casing wall boundary layer, location of the separation of blade boundary layer, etc. (It should be mentioned here that the passage secondary flow in a compressor rotor is somewhat weaker as compared to that in a corresponding cascade. This is because in a compressor rotor, there will be a streamwise component of vorticity at the inlet. Therefore, the secondary flow, which depends upon the difference between the streamwise components of vorticity at exit and inlet, get reduced. However, if the flow turning is high, the rotor blades have low aspect ratio and the inlet boundary layer has a steep velocity gradient, then the secondary flow in a compressor rotor could still become quite significant. This appears to be the case in the present investigation with the NAL compressor).

After making an analysis of the four parameters mentioned above the tip clearance flow and the tip gap could established. Because of the complexity of tip clearance flow and its interaction, this has to be done only by a trial and error process (as it has been for casing treatment). Further, in the case of non-uniform clearance, the tip gap flow will be much more complex than the uniform clearance. Flow visualisation techniques like laser anemometry and holography could be valuable aids for determining this complex type of improved tip configuration.

<u>Optimisation of the Uniform Tip Clearance</u>:

In the present investigation, it was found that for this compressor stage an optimum tip clearance (i.e., the tip clearance at which the adiabatic efficiency has the maximum value) exists. It is also known that all compressors do not have optimum tip clearance. It is not clear why this anomaly exists. One reason could be that the tip clearance flow is extremely complex and is influenced by so many factors. Therefore, the pattern of air flow through the tip gap does not necessarily depend only on the value of tip clearance but also on several other factors, with the result the existence or otherwise of an optimum clearance (uniform) depends upon the pattern of air flow through the tip and its interaction with secondary flows, casing boundary layer etc. It may also be pointed out that expressing the tip clearance as a percentage of blade height or blade chord could be misleading; because any two compressors having the same percentage value of tip clearance (expressed as a fraction of the blade height or chord) may have entirely two different types of tip clearance flows if other influencing factors are different for the two compressors. With the result each type of compressor may have its own value of tip clearance, which will give the best efficiency.

Concluding Remarks:

Experimental investigations were carried out on a new type of rotor tip clearance to find out the possible improvements in the performance of an axial flow compressor stage.

Different types of axially uniform and axially non-uniform rotor tip configurations were used in the investigations. For each configuration the adiabatic efficiency and stall margin were evaluated. The effects of providing the tip clearance in the compressor casing instead of the usual method of providing the clearance at the blade tip were examined. Analysis was made of a method of accurately determining the change in compressor efficiency due to the change in tip clearance or configuration.

The results are summarised as follows:

1. An axially non-uniform rotor tip clearance will have definite advantages over axially uniform clearance. The clearance from the leading edge upto midchord should be smaller (purely from mechanical considerations); and the clearance from mid chord to trailing edge should be larger.

2. An 'Optimum' value of tip clearance was noticed for this compressor stage, for the cases of both axially uniform and non-uniform clearances

3. With the increase of tip clearance there was improvement in stall margin for both the cases.

4. Clearance formed in the compressor casing appear to provide more benefits than the clearance obtained by machining the blade tip.

5. Improvements in adiabatic efficiencies obtained by changing the tip clearance or configuration, should be determined by very accurate methods or procedures. This is very essential in order to compare and suggest better tip clearance configurations.

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