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The effect of very low axial clearances on the performance of an axial-flow compressor stage

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The results of tests on an axial-flow compressor stage designed to allow operation at unusually low axial clearances are presented. The axial clearance was varied between 1 and 33 % of the blade chord whilst the compressor was running, and the stage pressure rise showed a peak at a clearance around 4 % of blade chord. The pressure rise was 8 % higher at this clearance than at 33 % of chord, with no significant change in the stage efficiency. In addition, reducing the clearance was found under some conditions to bring the stage from a stalled to a fully unstalled state. The sound output of the stage was measured at all clearances, and the results show a significant increase in the high harmonics of the blade-passing frequency at low axial clearances.

1. Introduction

The apparatus used to obtain the results described in this paper was constructed specifically to allow the investigation of the effect of very low axial clearances on stage performance. This variation on conventional compressor geometry had previously been largely ignored because of the obvious additional mechanical difficulties the clearance constraint would present to the designer, but recent work by the author (Furber & Ffowcs Williams 1979; Furber 1980) has suggested that significant performance improvements might arise in a stage designed to exploit the rotor–stator interaction which results from such a low clearance.

The compressor was designed with a variable axial clearance in order that direct comparisons between the performances at various clearances might be possible, and also because of the possibility that reducing the gap while running might result in the stage coming out of a stalled state. This is because the potential-flow interaction between the rotor and stator reinforces the blade circulations at each blade passing when the clearance is very low, discouraging the flow from separating, and possibly encouraging it to reattach when stalled. That this has been found to happen in the experimental stage is one of the most interesting results of the tests.

An unexpected but no less interesting discovery is that, although the stage-pressure rise increases as the clearance is reduced from 'normal' values, the increase does not continue indefinitely. There is a peak in the pressure rise at a clearance of around 4 % of the blade chord, and thereafter the pressure rise reduces as the clearance is lowered. There has been no suggestion of this effect in previous work.

The sound output of the rig was measured and 'ensemble averaged' to give only the component related to the blade-passing frequency. As expected, the noise increases with reduced clearance, and the high harmonics dominate the sound at the lowest clearances.

In recent years much work has been done to investigate the relative importance

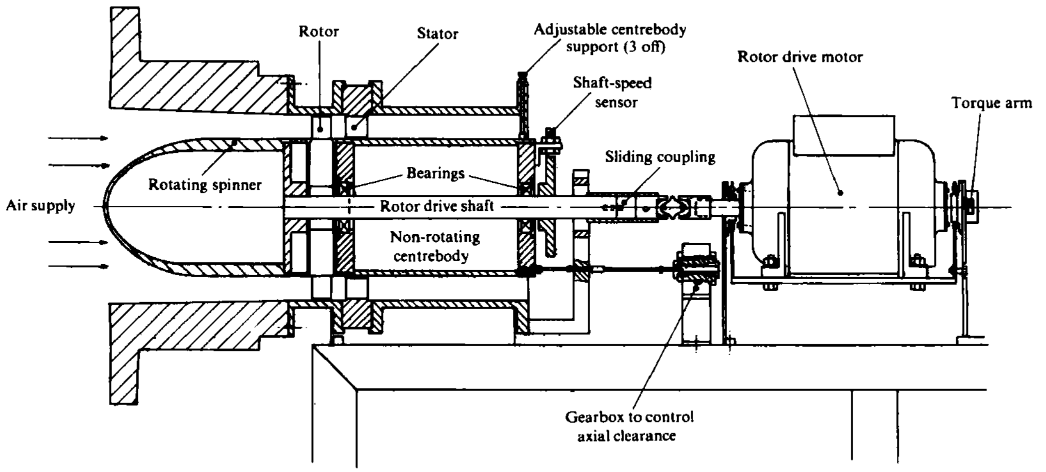


FIGURE 1. The working section of the experimental apparatus.

of the many geometric factors that determine a complete compressor design, and some attention has been paid to the effect of axial clearance. Aschenbrenner (1966), Smith (1970) and Jenny (1976) all tested various builds of single- or multi-stage axial compressors, and found an increased pressure rise at slightly increased efficiency as the gaps were reduced. The one exception was a test by Jenny where the reduction of the gap between an upstream stator and a downstream rotor led to a reduced efficiency. None of these tests allowed for changing the axial clearance while running, and therefore they were incapable of demonstrating the unstalling phenomenon, but taken together they show that the increased work output is obtainable from different blade sets under different conditions, and is not peculiar to the exact configuration used in the tests described here.

2. Apparatus

The general layout of the apparatus is shown in figure 1. The working section is supplied with air from a centrifugal pump, through a throttle, settling chamber and convergent duct. The stator blades are machined from aluminium integrally with the outer blade support ring, which is bolted between the entry and exit outer casings. All the flow instrumentation is mounted in this outer unit, which is itself connected to the air supply and supported rigidly from the laboratory floor.

The inner unit is made up of a non-rotating centrebody downstream of the rotor, through which the rotor drive shaft passes, and the rotor and rotating spinner. The centrebody is supported by phosphor-bronze inserts in the inner faces of the stator blades, and by adjustable radial supports at the downstream end of the exit outer casing. All these supports bear directly on the surface of the centrebody, leaving the entire inner unit unconstrained in the axial direction.

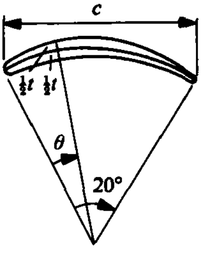
The rotor drive motor is mounted in a torque cradle and connected to the rotor drive shaft by a slotted coupling to allow for the axial movement of the inner unit, and finally the axial position of the inner unit is accurately controlled by a lead screw driven manually through a reduction gearbox.

The instrumentation consists of static pressure tappings in the supply settling chamber and either side of the stage in the flow direction, and a carrier for a hot-wire

rotor profile	10C5 on 20° circular arc
chord	38.1 mm
stagger	34°
spacing	39 mm at mid span
number of blades	23
mounting	dovetail into hub
stator profile	10C5 on 20° circular arc
chord	38.1 mm
stagger	13°
spacing	39 mm at mid span
number of blades	23
mounting	machined integral with outer support ring, 3 mm radius fillets
tip diameter	305 mm
hub/tip ratio	0.75
rotor speed	2600 r.p.m.
tip speed	42 m/s
Mach number	0.14
Reynolds number based on chord	100000
maximum stage pressure rise	750 Pa

TABLE 1. Details of the tested stage

$\theta/20°$ (%)	$t/2c$ (%)
0	0
1.25	1.65
2.5	2.27
5	3.08
7.5	3.62
10	4.02
15	4.55
20	4.83
30	5
40	4.89
50	4.57
60	4.05
70	3.37
80	2.54
90	1.6
95	1.06
100	0



leading-edge radius of curvature is 1.5% of chord
trailing-edge radius of curvature is 0.5% of chord

TABLE 2. Details of the 10C5 profile

probe, which was inserted in the outer casing just upstream of the rotor. The stalling of the stage was clearly audible, but the hot wire was used to confirm the condition of the stage as being stalled, and also to identify the mode as a single-cell rotating stall. The rotor speed is measured by counting pulses from a magnetic sensor mounted near a suitable disk on the rotor shaft, and the motor torque is obtained by counterbalancing the cradle to a null position.

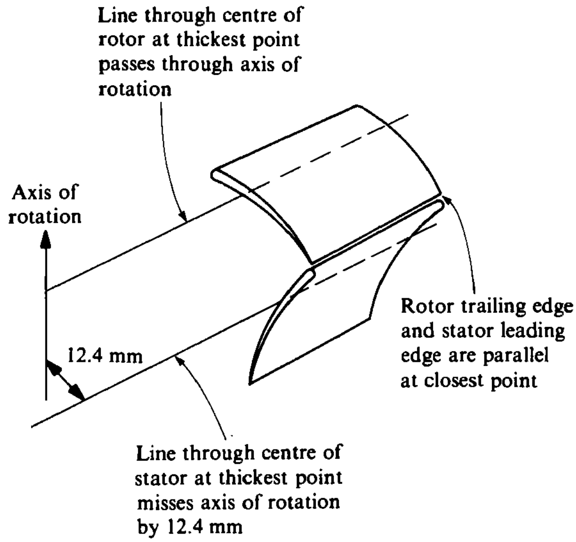


FIGURE 2. The radial alignment of the blading, which was designed to minimize leakage between the rotor trailing edge and the stator leading edge.

A half-inch Bruel and Kjaer condenser microphone was placed just outside the airstream emerging from the downstream end of the working section to measure the sound output.

The blading details are given in table 1, and the coordinates of the 10C5 profile used for both blade sets are given in table 2. Both the rotor and the stator used two-dimensional blading; the only unusual feature was that the stator-blade centreline was skewed from the radial direction so that the stator leading edge was parallel to the rotor trailing edge at the closest point (see figure 2). Thus the blade passing was not a 'scissor' action, but a simultaneous approximate closure of the gap along the whole span. At the lowest axial clearance the flow between the blade pair was therefore cut off altogether rather than redirected in a radial direction, this being the feature required to maximize the potential-flow interaction between the blades.

3. Results

The results of the tests on stage performance are shown in figures 3, 4 and 5. In figure 3(*a*) the stage loading ($\psi = \Delta P/\rho u^2$, where ΔP is the measured rise in static pressure across the stage, ρ is the air density and u is the circumferential speed of the mid-span point of the rotor blades) is shown against axial clearance for various settings of the air-supply blower and throttle. A peak in the pressure rise is observable at all the unstalled-flow conditions, and occurs at a clearance in the region of 4–6.5 % of blade chord. The height of the peak increases from around 5 % of the mean level at high flow rates to around 8 % near stall. The extreme case is unstalled at low clearance, and then the peak is 25 % higher than the mean level.

Figure 3(*b*) shows the corresponding efficiency ($\eta = \Delta P\, Av/G\Omega$, where A is the total cross-sectional area of the flow through the stage, v is the mean axial velocity, G is the measured torque and Ω the rotational speed of the motor) against axial clearance. The bearing losses in the centrebody affect this efficiency, though the losses in the motor bearings are cancelled out by the method of measuring the torque. The

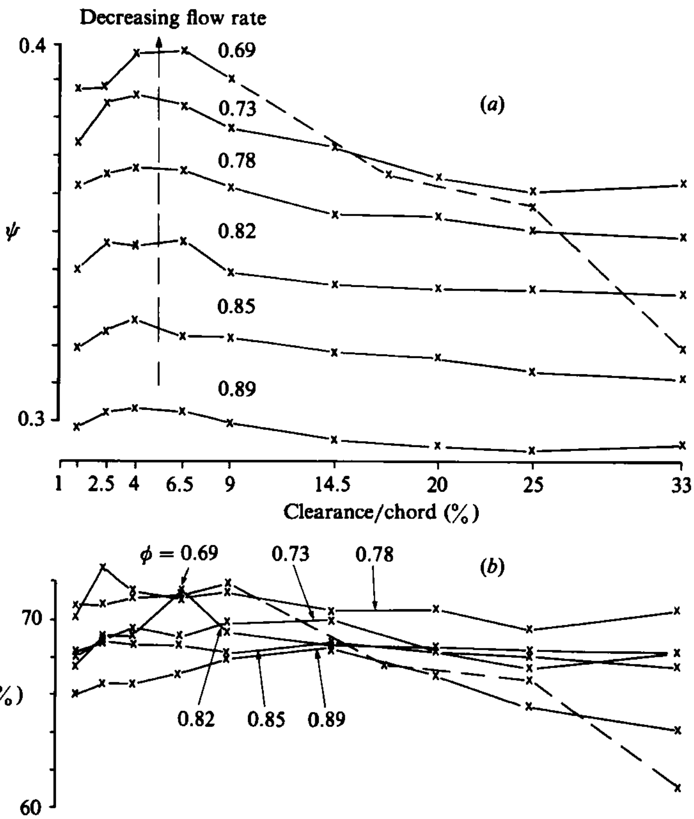


FIGURE 3. The stage loading (*a*) and efficiency (*b*) as a function of axial clearance for various settings of the air supply. The indicated values for ϕ were constant along the curves to within 1 %, except along the dashed section, where the stage was stalled.

dependence of the efficiency on the clearance is less clear than that of the stage loading. There is no apparent peak at 4 % clearance, but nor is there any tendency for the efficiency to drop at low clearances. It seems fair to conclude from these results that the increased work output from the stage is not obtained at the cost of reduced efficiency.

Some of the results have been replotted in figure 4 in the form of characteristics and efficiency curves. The stage loading ψ and efficiency η are defined above, and the flow coefficient ϕ is the ratio of the mean axial velocity of the air v to the circumferential speed of the mid-span point of the rotor blades u. The results shown are for clearances of 1, 4 and 33 % of blade chord, representing the minimum, optimum and maximum tested clearances, respectively. The increased stage work output at no efficiency loss is clear from these graphs, and is obtainable over the entire unstalled working range of the stage.

The dependence of the stalling behaviour of the stage on the axial clearance is of special interest, and the characteristics of figure 4 show that this dependence is marked. The stalling discontinuities in the characteristics were investigated in more detail, and the results are shown in figure 5. At the three lowest clearances a hysteresis cycle was discernible in that at certain flow coefficients the stage could operate either stalled or unstalled, depending on whether the throttle was being opened from the stalled part of the characteristic or closed from the unstalled part. The cycles are

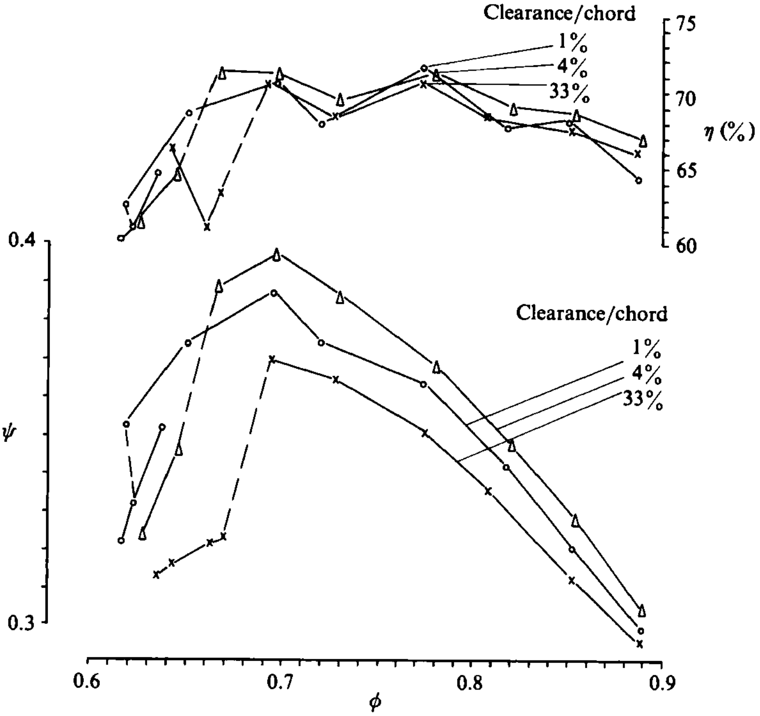


FIGURE 4. Stage characteristics and efficiencies at minimum, optimum and maximum clearances.

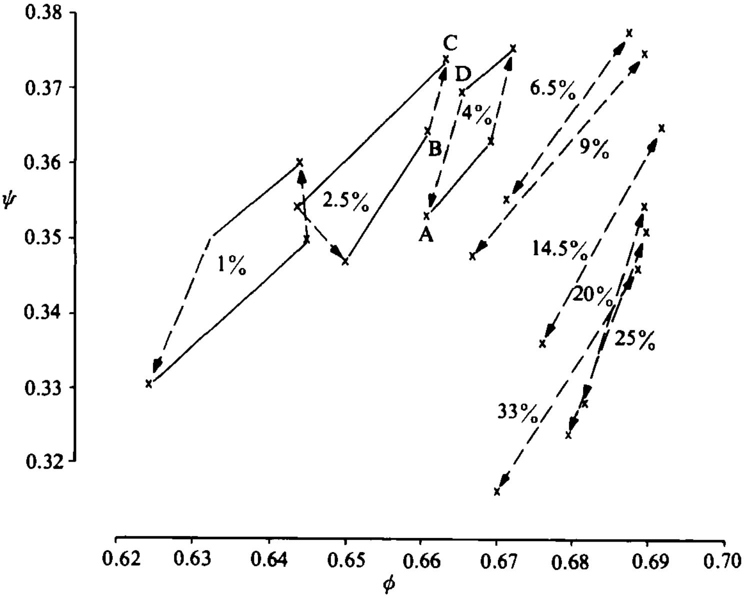


FIGURE 5. The discontinuities in the stage characteristics at various axial clearances, marked as a percentage of the blade chord. The hysteresis cycles were found to be too small for accurate measurement at clearances of over 4% of chord. (The points marked A, B, C and D are referred to in the text.)

shown in figure 5, and the direction of the transitions between stalled and unstalled states are indicated. At the higher clearances the discontinuities are reversible to experimental accuracy.

As the curves in figure 5 suggest, it is possible to start the apparatus in a fully stalled state and to bring it out of stall simply by reducing the axial clearance. Consider, for example, a stage with clearance equal to 4 % of chord which has been stalled by closing the throttle. It will be operating at point A in figure 5. If the clearance is reduced to 2.5 % of chord the operating point moves to B (approximately, and if we ignore the small change in flow coefficient caused by the reduced clearance). Point B is marginally unstable, and a small perturbation in the flow will cause a switch to point C, which is on the unstalled section of the characteristic. Conversely, increasing the clearance will move the operating point near to D, which is again unstable and will collapse back to A. It was possible to take the experimental stage in and out of stall in this way, by varying only the axial clearance.

The stalling-flow coefficient decreases with reduced axial clearance, and, unlike the increased pressure rise, this effect has not been found to peak at a particular clearance. The stage which stalls at the lowest flow coefficient is the one with the lowest clearance. This result, though not predictable in detail, was expected because of the tendency of the rotor–stator interaction to restore the blade circulations (Furber & Ffowcs Williams 1979). The increase in the stage-pressure rise with reduced clearance was in line with earlier results (Aschenbrenner 1966, Jenny 1976, Smith 1970), though the peak had not previously been observed.

4. Sound output

The sound output of the rig was measured at various clearances using a half-inch Bruel and Kjaer microphone. The decibel level was taken from the meter on the microphone amplifier, and the microphone output was digitized and 'ensemble averaged' using the magnetic sensor on the rotor shaft as a trigger. This procedure averages out the random noise caused by the jet, and leaves only the harmonics of the blade-passing frequency. This component of the sound is caused by the unsteady loading on the blades, and may be used as an estimate of the relative strengths of those loadings at the different clearances.

The power spectra of the sound at clearances of 1, 4 and 33 % of chord are shown in figure 6. At 33 % clearance the sound is dominated by the fundamental, with a few low harmonics present. At 4 % clearance harmonics up to the twelfth are quite strong, though the fundamental is still predominant. At 1 %, however, the seventh and ninth harmonics have higher amplitude than the fundamental, and the subjective quality of the sound is quite different. The total sound level at 4 % clearance was 6 dB higher than at 33 % clearance, and that at 1 % clearance was 17 dB higher, showing a strong dependence on clearance at the lower clearances.

5. Conclusions

The results of the tests which have been presented in this report have confirmed the conclusions of previous work listed in the references, principally that the performance of a conventional axial turbomachinery stage may improve as the axial clearance between the rotor and the stator is reduced. Stage-blade profiles are often selected on the basis of cascade data, where a row of blades is tested in completely steady flow, and it is therefore perhaps surprising that blading that is optimized in

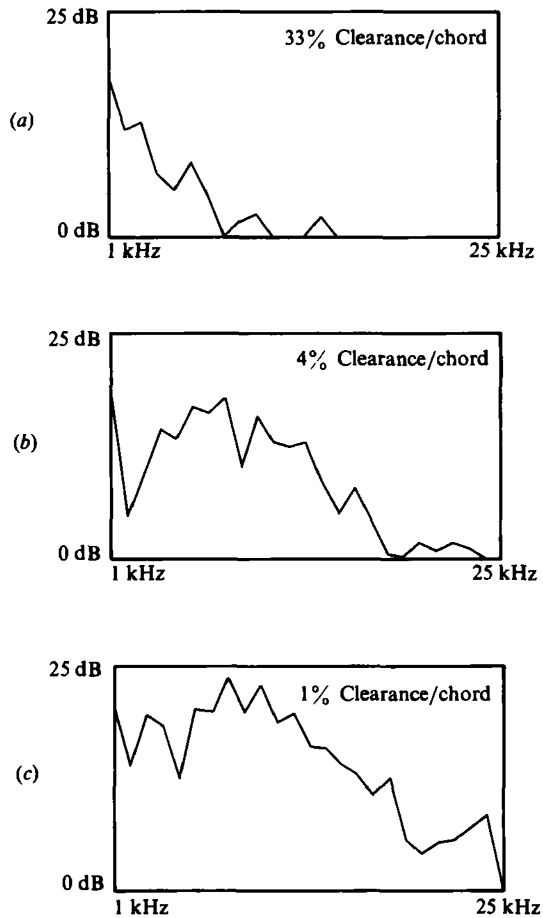


FIGURE 6. Log power spectra of the blade-passing frequency harmonic components of the sound output of the stage at a clearance of (*a*) 33 %, (*b*) 4 % and (*c*) 1 % of blade chord. The blade-passing frequency was 1 kHz. The total sound level in (*b*) is 6 dB higher than in (*a*), and that in (*c*) is 17 dB higher than that in (*a*).

steady conditions should demonstrate an improved performance in the very unsteady flows of the low-clearance tests reported here. If new blade profiles could be optimized to exploit the strong interactions of low-clearance configurations then further improvements might be possible, though a design method for such blades is not yet available.

A possible theoretical explanation for this improved performance at low clearance has been suggested by the author (Furber & Ffowcs Williams 1979), where a mechanism for determining blade circulations in zero-clearance stages is described. This mechanism is independent of the Kutta condition, and yields higher circulations (and stage loadings) under unstalled operating conditions. The performance improvements observed here are consistent with the hypothesis that such a mechanism comes into play at low clearances, though the performance peak at a clearance in the region of 5 % of chord is unexplained.

It should be noted here that the two alternative methods for predicting the stage performance, as detailed in the author's earlier work, cover only two axial-clearance conditions. The 'Weis-Fogh' model assumes zero axial clearance, whereas the normal

'Kutta condition' model applies only when the clearance is much larger than the chord. Therefore only one point on each of the curves of figure 3 is described by each theory. It might be natural to assume that the stage behaviour should move monotonically between these points as the 'Weis-Fogh' mechanism comes into effect, but the results in figure 3 show this assumption to be false. The explanation of the maxima in the curves of figure 3 must await a more detailed theory of rotor–stator interaction at low (but non-zero) axial clearance.

A remarkable demonstration of the performance change at very low clearance is the unstalling of a stage which had been in steady rotating stall at a higher clearance. The increased pressure rise and reduced flow coefficient at stall could enable a compressor to operate at a higher pressure ratio against a fixed flow resistance without requiring special devices such as bleeds to ensure unstalled operation at start-up.

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