THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

95-GT-197

RECESSED CASING TREATMENT EFFECTS ON FAN PERFORMANCE AND FLOW FIELD

C. S. Kang, A. B. McKenzie, R. L. Elder
School of Mechanical Engineering
Cranfield University
Bedford
England

ABSTRACT

An experimental investigation to examine the influence of the vaned recess casing treatment on stall margin, operating efficiency and the flow field of a low speed axial flow fan with aerospace type blade loading is presented. Different geometrical designs of the vaned passages were examined. The best configuration resulted in a stall margin improvement of 67%, a significantly higher pressure rise in the stall region and insignificant change in peak efficiency. Detailed 3-D flow measurements in the endwall region and in the casing recess were carried out with a slanted hot-wire, providing some insight to the operation of the device. The results revealed that the stall margin improvement was largely due to the removal of flow from the blade tip to the recess, and the elimination of the growth of the stall region at the tip, which occurs at stall in the solid casing build.

NOMENCLATURE

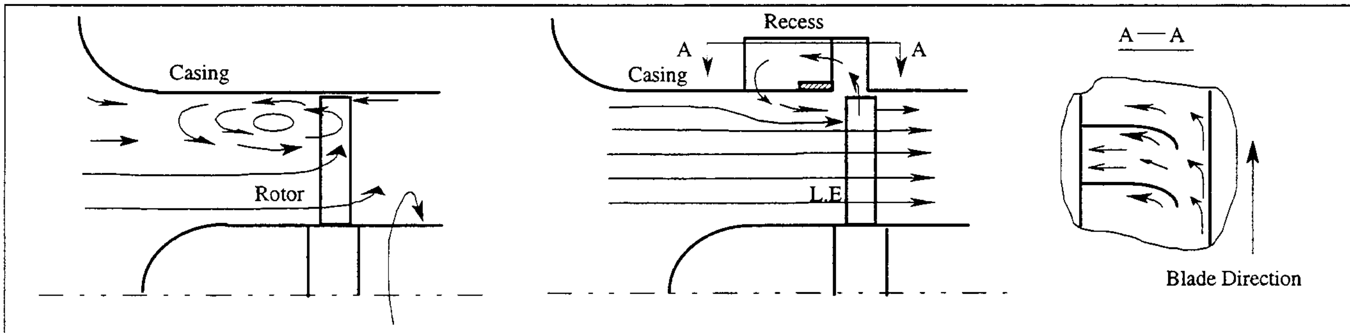
C_a rotor axial chord (mm)
Ce rotor axial chord exposure (mm)
P_{s2} outlet static pressure (N/m^2)
P_{t1} inlet total pressure (N/m^2)
U blade speed (m/s)
U_m mid span blade speed (m/s)
V_0, V_3 absolute inlet and outlet velocity (m/s)
V_1, V_2 relative inlet and outlet velocity (m/s)
V_a axial velocity (m/s)
V_t circumferential velocity (m/s)
V_r radial velocity (m/s)
ϕ flow coefficient, $\phi=V_a/U_m$
ΔH stage enthalpy rise
η total to total efficiency (%)
$\Delta\eta$ peak efficiency loss (%)
ρ air density (kg/m^3)
ψ total to static pressure rise coefficient $\psi=(P_{s2} - P_{t1})/(0.5\rho U_m^2)$
ϕ_{ss} flow coefficient at stall point for solid casing
ϕ_{ts} flow coefficient at stall point for treated casing
ψ_{sm} peak pressure rise coefficient for solid casing
ψ_{tm} peak pressure rise coefficient for treated casing
$\Delta\psi$ pressure rise improvement (%), $\Delta\psi=(\psi_{tm}/\psi_{sm})-1$
$\Delta\phi$ stall margin improvement (%), $\Delta\phi=1-(\phi_{ts}/\phi_{ss})$

INTRODUCTION

It is well known that the application of groove or slot type casing treatments over the tips of rotor blades (or to the rotating drum beneath the stator) can have a marked effect on stall margin improvement of axial compressors and fans. These techniques can lead to a limited but valuable stall margin improvement (around 20%), but unfortunately at the expense of 2-4% in stage efficiency.

An alternative kind of casing treatment called *vaned recess casing treatment* has shown a remarkable effect on the stall margin and pressure rise of industrial fans. The concept goes back to the patent work by **Ivanov** et al.(1984) and further research conducted by **Bard**(1984) and **Miyake** et al.(1987). More recently **Bard**(1993) reported a stabilising device integrated into the casing of axial fans, which completely eliminated stall without affecting the peak efficiency. Since aero-engine type fans generally have much higher blade loading than industrial fans, different behaviour and more difficulties are expected when this treatment is applied to an aero-engine fan or compressor. The effect of the vaned recess casing treatment on a fan with a typical aero-engine loading was first studied and reported by **Azimian** et al.(1989). The results were very encouraging with superior stall margin and less efficiency sacrifice when compared to conventional slot type treatments. In addition, **Ziabasharhagh** et al.(1992) demonstrated that these

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
Houston, Texas - June 5-8, 1995



a. Stall With Solid Casing

b. Flow Field With Casing Treatment

Fig. 1 Sketch of the Operation Of the Casing Treatment

casing treatments were capable of increasing stall margin for a range of machines and also able to increase tolerance to inlet flow distortion.

Research on the subject is in its early stages, and much work remains to be done before it can be widely used in practice and adopted as a standard item by designers. Indeed, at the treatment design has been mainly based on trial and error experiences so far. Clearly a better design depends very much on an understanding of the mechanisms involved. There have been a number of papers about the mechanism of slot type casing treatments, e.g. **Takata** et al.(1977), **Smith** et al.(1984), **Johnson** et al.(1987) and **Crook** et al.(1993), but few on vaned recess casing treatments. **Ziabasharhagh** et al.(1992), however, based on the 'wool tuft' flow study provided a hypothesis of the operation from which the following physical description of the flow may be deduced as sketched in **Fig.1**. The low momentum fluid, associated with stalling flow is centrifuged towards the blade tip and then recirculates axially as shown in **Fig.1-a** and also circumferentially in the untreated fan. When the treatment is applied, this tip flow passes radially into the treatment recess where the tangential flow component of the flow is removed by the stationary cambered vanes, then discharges out of the recess into the main flow as illustrated in **Fig.1-b**. The operation of these two types of casing treatments is not identical, though there may be many common points. This calls for more detailed studies on the flow phenomena associated with the treatment, as these will be essential to an understanding of the mechanism.

This work is therefore aimed at:

- further studying the effectiveness of the device to obtain additional information regarding the potential gain in stall margin available from optimum treatment configurations; and
- gaining a better understanding of the mechanism by studying the flow phenomena involved.

EXPERIMENTAL FACILITY & INSTRUMENTATION

The experimental facility used in this study was similar to that of **Azimian** et al.[1989], with the exception of a new computerised data acquisition system. The test fan was a single stage (rotor only) axial flow fan with a tip speed of 39m/s, tip diameter of 508mm, hub to tip ratio of 0.5 and near unity blade loading ($\Delta H/U^2$) at the hub. The rotor was equipped with 27 C_4 profile blades and driven by a variable speed, 3.7kw AC motor fitted with a dynamometer type torque meter. Previous studies had shown that the fan was susceptible to tip stall and was therefore considered a good vehicle for the present casing treatment studies.

A modified 3-D flow measurement technique with a 54.7° slanted hot-wire as proposed by **Whitfield** et al.(1972) was used. Due to the asymmetrical characteristics and limited calibration range of the slanted hot-wire, great care was taken to position the probe and interpret the measurements. A high speed (up to 100kHz) data acquisition system consisting of a 386 PC and a constant temperature hot-wire anemometer was developed and used. The static and total pressure for the performance were measured by a spirit filled manometer and a calibrated three-hole pneumatic probe respectively.

DESIGN OF CASING TREATMENTS

The casing treatment used by **Azimian** et al.(1989) was based on **Ivanov**'s design method and proved successful although rather arbitrary. The design is enhanced for further studies here. In all, three configurations were built and tested.

Configuration 1

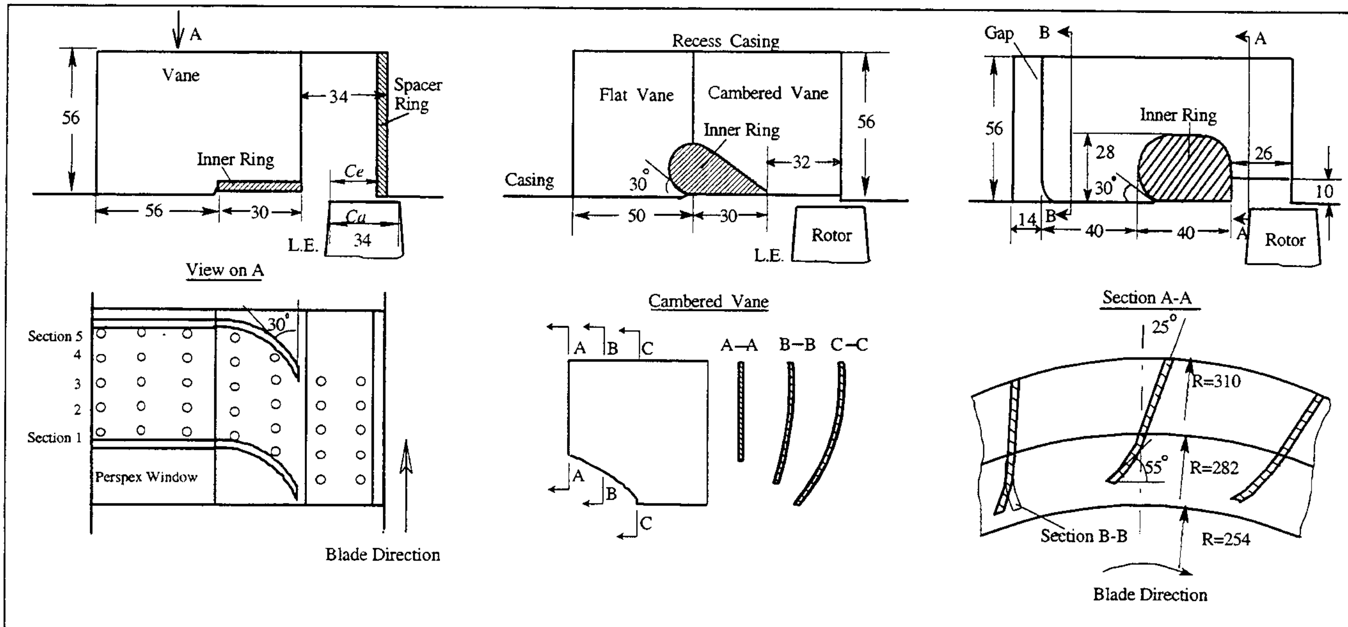
This configuration (**Fig.2-a**) comprised 24 vanes, half of that used by **Azimian et al.**(1989), giving an overall pitch/chord ratio of approximately unity. Several spacer rings (4mm in thick) were designed and placed inside the casing treatment such that the width of the inlet (unbladed) part of the recess could be varied. A perspex window with 35 probe mounts for wool tuft flow visualisation and flow traversing measurement was provided over a vane passage.

Configuration 2

This involved two major modifications as shown in **Fig.2-b**:

i) The "airfoil section" inner ring used by **Azimian** et al. (1989) was simplified to a 30° wedge crowned by a 10mm radius circular arc;

ii) New vanes covered the whole recess length. These vanes consisted of a radially placed uncambered (flat) part and a radially cambered part over the rotor (see **Fig.2-b**). The axial length of the recess was 8mm (approximately 7%) shorter



a. Configuration 1 b. Configuration 2 c. Configuration 3

Fig.2 Sketch of The Casing Treatment Design (*dimensions in mm*)

than that of Configuration 1.

Configuration 3

This treatment consists of an outer casing with 24 slightly curved vanes and an inner ring of substantial cross section (see **Fig.2-c**). The inlet part of the vane (section A-A) was cambered to 55° with respect to the circumferential direction in order to reduce the flow incidence to the vanes. The leading edge of the inlet portion was cut back by 10mm (18% of the recess height) to avoid additional noise. The outlet part of the vane had a 90° outlet angle (section B-B), re-introducing the flow into the main annulus in the radial direction. The vane was inclined at an angle of 25° with respect to the radial direction. Note that the radius of curvature of both inlet and outlet regions of the vanes has been increased from those used in Configuration 2.

OVERALL PERFORMANCE

The overall performance characteristics were measured at a constant (corrected) rotational speed of 1500rpm, to which all results are referred. Total to static pressure rise coefficient ψ and total to total efficiency η are presented as functions of the flow coefficient ϕ in **Fig.3**. The performance of the casing treatment is assessed below in terms of stall margin improvement $\Delta\phi$, pressure rise improvement $\Delta\psi$ and peak efficiency loss $\Delta\eta$. Each of the treatments was tested at various axial stations relative to the rotor leading edge and the following discussion relates to the optimum position (C_e/C_a was approxmately 2/3).

Solid Casing

The performance of the solid casing build which is used as a basis for comparison, and is shown in **Fig.3**. The pressure rise coefficient increases smoothly from 0.07 to 0.26 while the mass flow coefficient ϕ drops from 0.765 to 0.609. When ϕ is 0.609, the compressor flow breaks down with an abrupt fall in pressure rise and operation stabilises (with rotating stall cells) when ϕ has a value of 0.41. The compressor operation becomes more stable with pressure increasing as the flow rate is further reduced. As this occurs, the compressor enters a deep stall regime and nearly half of the annulus height ahead of the rotor is

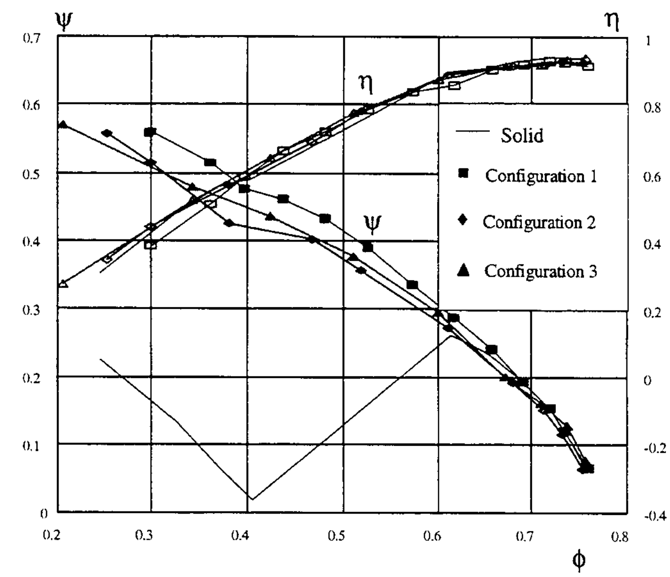


Fig.3 Overall Performance of the Isolated Rotor With Various Casing Treatments

occupied by axisymmetrical swirling flow with strong reverse flow in the tip region. In the stalled region, the work input is considerably reduced (**Fig.4**) until flow cut-off is approached.

Casing Treatments

It will be noted that Configuration 1 performed well ($\Delta\phi$ =51%, $\Delta\psi$=115% and $\Delta\eta$=0.5%) when compared against the datum configuration of the solid casing. This performance enhancement is very similar to that found by **Azimian** et al (1989) when a 48-vane build was used. This was not too surprising as the difference between these builds was the number of recess vanes and indicates that the flow is insensitive to blade number over this range.

Configuration 2 gave an improvement in stall margin over Configuration 1 ($\Delta\phi$=59% compared with 51%), but gave no advantage in pressure rise ($\Delta\psi$=114%) or efficiency ($\Delta\eta$ =0.75%). Additionally, a whistling noise was heard during the experiment, thought to be due to a vane/rotor blade interaction. Cutting back

the vane leading edges of the recess vanes by 20mm eliminated the noise but reduced the performance ($\Delta\phi$=17.3%, $\Delta\psi$=92.3% and $\Delta\eta$=1.03%). However, the results were not conclusive as after trimming, the vanes were not cambered sufficiently to ingest the reversed flow from the rotor tip as effectively.

The best results were obtained with Configuration 3 ($\Delta\phi$ =67%, $\Delta\psi$=120% and a slight increase in peak efficiency). These advantages are attributed to the new vane passage geometry. It is also noteworthy that the peak efficiency point was found to be moved to a higher flow rate (ϕ=0.75), compared to that for the other configurations at ϕ=0.72.

Generally, all the configurations improved stall margin with increased pressure rise toward zero flow and negligible change in peak efficiency. What is perhaps surprising is the similarity of the results considering the variation of the builds. In the extended unstalled range, all reverse flow and axial flow blockage found in the untreated build are removed, and no rotating stall was detected. However, at those reduced flow rates there was a noticeable increase in the noise level. There was also some signs of stall inception at flow coefficients around 0.4 (indicated by a dip in the pressure rise curve), which was found in all builds tested (see also **Azimian** et al. (1989) except for Configuration 3.

It will also be noted, **Fig.4**, that the treated builds involve considerably greater work input over the lower flow range than for the solid wall build.

Fig.5 shows the sensitivity of the Configuration 3 to different axial position (C_e/C_a is the amount of the rotor axial chord exposed to the treatment). Indicated are two distinct regions where performance is relatively insensitive to position. These are separated by a region (around the mid-chord) critically influenced by axial positioning and that the best results are achieved at C_e/C_a=67% where there is no apparent loss of efficiency.

FLOW MEASUREMENT RESULTS

Flow in the casing treatment recess and any flow change in the annulus flow due to the casing treatment are of great

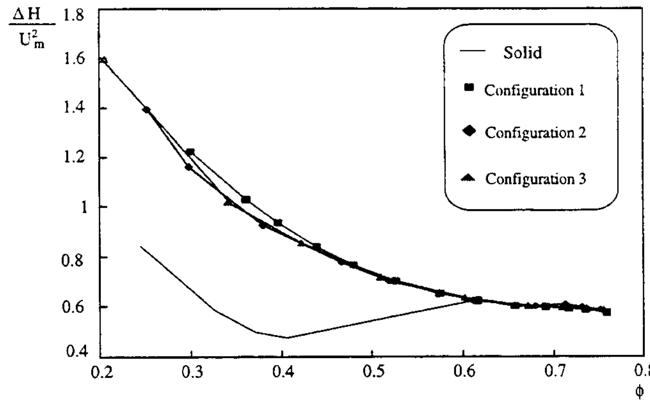


Fig.4 Work Input Characteristics

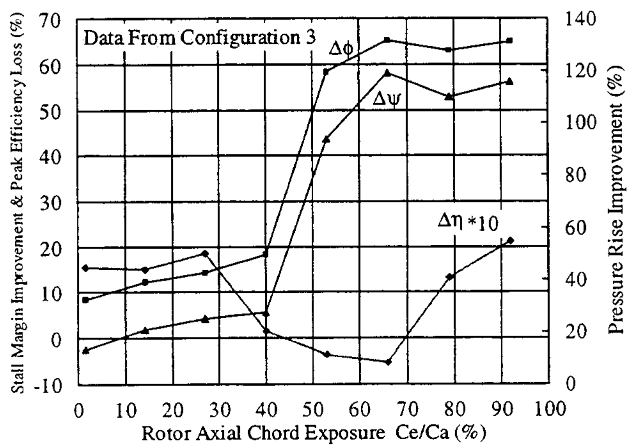


Fig.5 The Effect of Rotor Axial Chord Exposure

importance to an understanding the mechanism involved. The flows involved are very complex and cannot be treated as a two-dimensional flow, since the radial velocities can be very large in the treatment recess and even in the rotor region. Therefore, three-dimensional measurements using a slanted hot-wire (as previously described) were made to define the flow field. The supporting pressure data were taken by means of a calibrated three-hole pneumatic probe. For convenience and consistency, most measurements were taken at radially straight vaned configuration 1 but unfortunately the test data presented in Fig.5 and Fig.10 are not available from configuration1.

Velocity Traverse

The time-averaged radial profiles of the inlet (**Fig.6,** left column) velocity for the solid wall casing and treated casing configuration 1 (right column), were measured at a station 6mm upstream of the rotor. The outlet velocity (**Fig.7**), was measured at a station 56mm downstream of the rotor, because a casing flange made it difficult to arrange measurements close to the rotor.

Just before the natural stall point, **throttle=45%** (throttle fully open=85%), there is no significant difference to be noticed in the outlet velocity profile (between the builds), however,

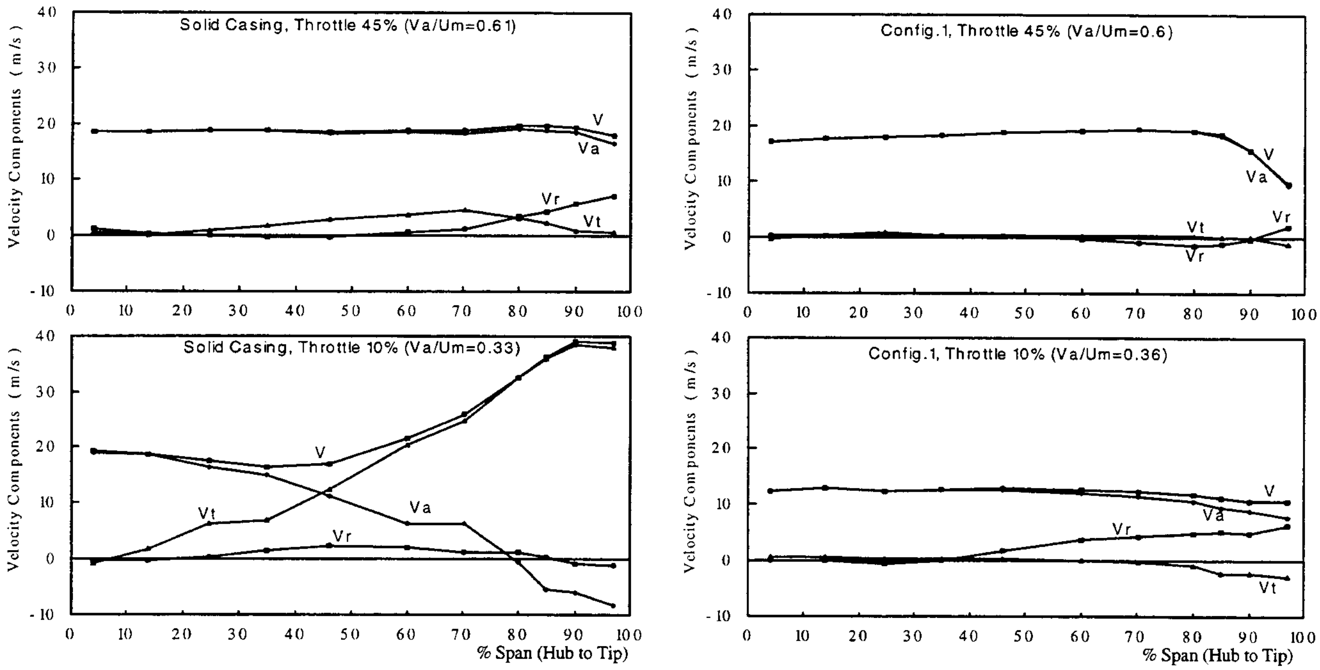


Fig. 6 Radial Distribution of Inlet Velocity for Solid and Treated Casing (6mm Upstream the rotor)

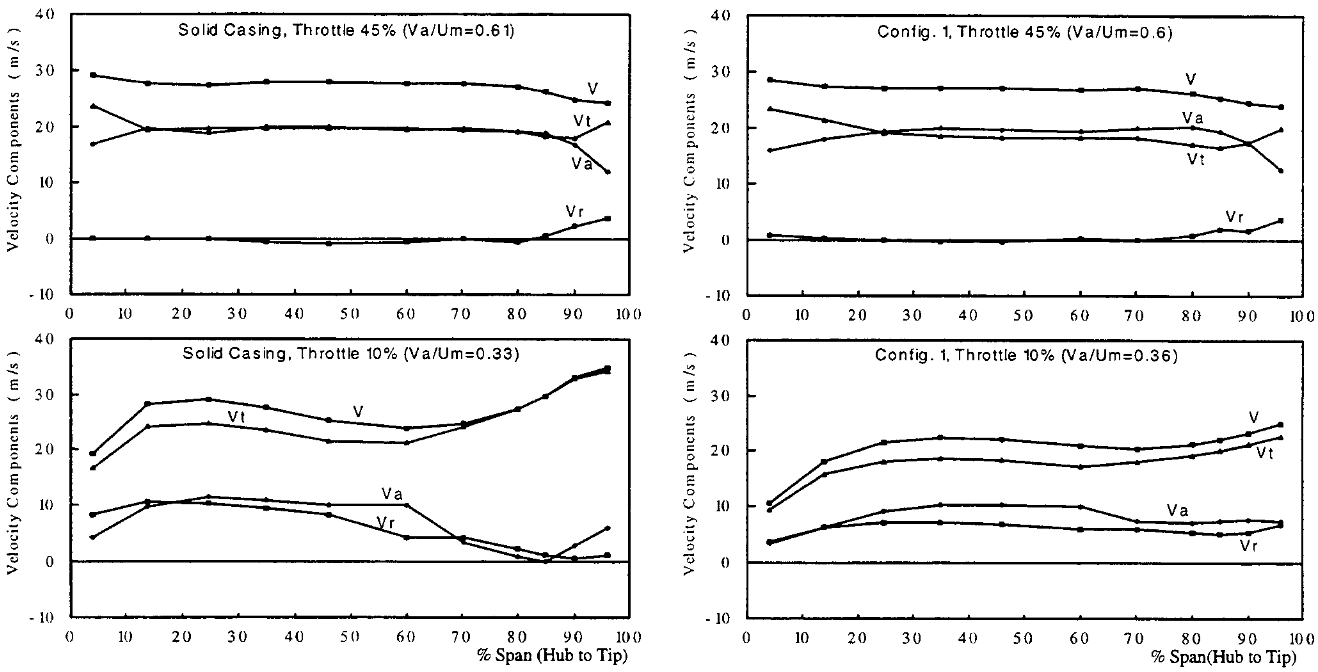


Fig. 7 Radial Distribution of Outlet Velocity for Solid and Treated Casing (56mm Downstream the Rotor)

there is a change of inlet velocity profile (due to the treatment) in the tip region (15% of span from the tip) where the velocity is decreased. This decrease is probably due to the inward radial flow from the outlet of the recess. This phenomenon was observed in all the builds at high flow rates (unstalled).

Near the stall point for the treated build (**throttle=10%**), the upstream radial traverse of the solid wall build shown in **Fig.6** demonstrates a dramatic deformation of flow pattern with a tangential velocity in the tip region approaching the rotor blade speed, and a high axial velocity in the hub region which decreases to a substantially negative value at the tip. With the treatment, the reverse flow in the tip region was completely

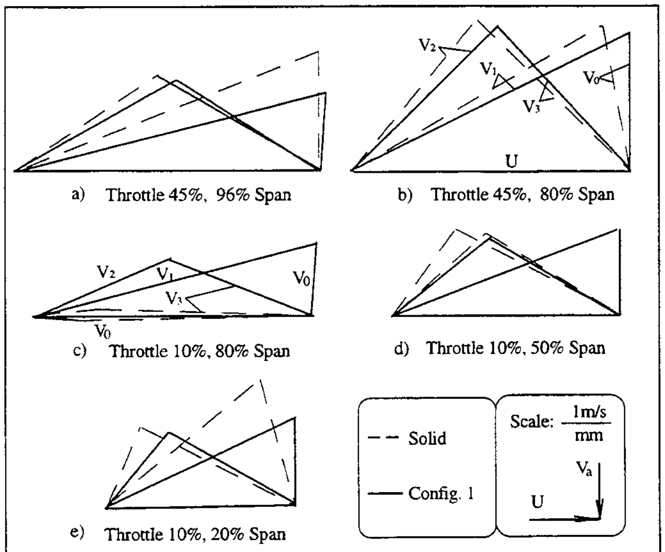


Fig.8 Comparison of the Velocity Triangle

removed and the axial velocity is much more uniform and the strong tangential component is eliminated. Downstream of the rotor, both the solid and treated cases have surprisingly similar profiles, although the treated profile tends to be more uniform.

It is interesting to note that the velocity triangles (see **Fig.8**) based on the flow measurements above reveal that the treatment does not have the function of reducing the inlet flow incidence onto the blade. On the contrary the reduced whirl in the tip region tends to increases the incidence in most cases. These observations suggest that the stall margin improvement due to the casing treatment was not achieved by reducing the incidence in the tip region.

Velocity Fluctuations

Fig.9 shows the instantaneous inlet velocity for the solid and treated casing measured in the plane 6mm upstream of the rotor at two radial positions, 50% and 75% span from the hub. The results (single rotor revolution traces) taken at the different spanwise positions substantially agree except for the solid casing result for the stalled flow case (throttle opening 10%) which differs due to the significant radial profile present (see also **Fig.6**). The interesting point to note in this set of results is the regular periodic nature of the treated cases and the comparative irregularity of the solid wall cases, indicating that the fluctuations other than blade to blade fluctuations which have been reduced (or removed) by the treatment. The implication here is that an unstable element in the flow (possibly the tip clearance vortex) has been significantly modified by the presence of the recess vanes.

Radial Pressure Profile

The total pressure and static pressure ratio (normalised by 0.5 ρU_m^2) before and behind the rotor are examined in **Fig.10** which presents the result measured at the two stations (48mm before the rotor and 56mm behind the rotor). In general, the casing treatment did not significantly alter the radial pressure profile

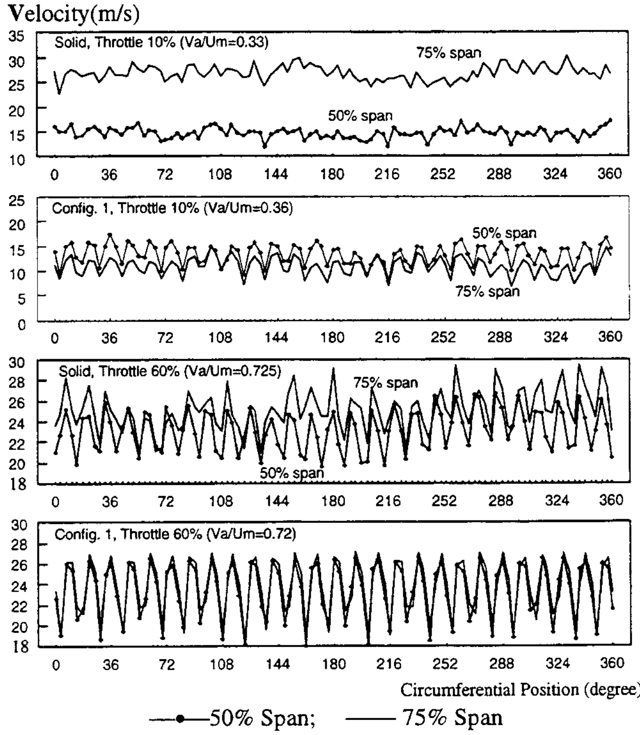


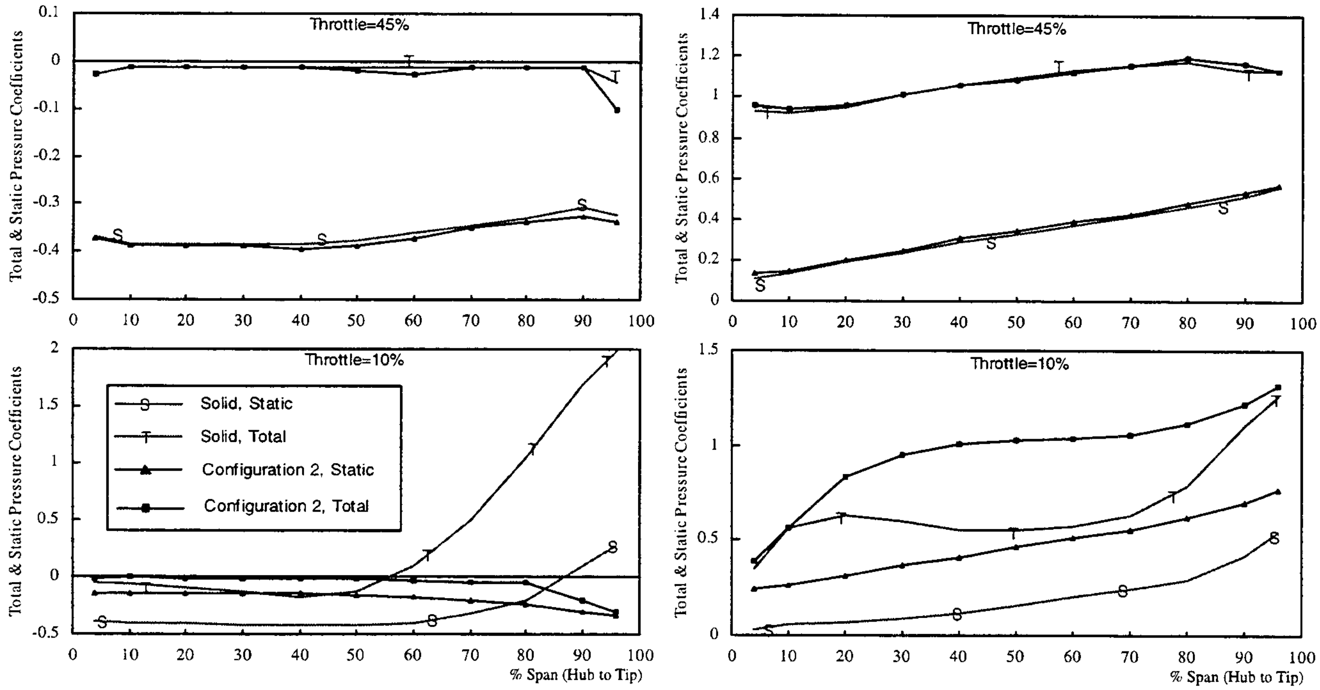
Fig.9 Comparison of Flow Uniformity

before the stall point (i.e. throttle=45%) but did change it very much in the extended unstalled flow range of the treated build (i.e. to throttle=10%). In this region, with solid wall casing, an excessively high total and static pressure occurred in the tip region upstream of the rotor and the pressure profiles under these conditions change markedly when the recess vanes are introduced. Perhaps the solid casing is of little interest (because the rotor is heavily stalled), but the regularity of the treated casing results are notable at this low flow rate. At both flow rates, for the treated build, the upstream tip region were found to have decreased total pressure, indicating that the treatment does not have the function of increasing the total pressure in the upstream tip region.

Flow in the Casing Treatment Recess

Flow measurements were taken in the casing treatment recess of configuration 1 by means of a slanted hot-wire in conjunction with wool tufts. Because of the intense fluctuation of the flow, it was almost impossible to obtain a repeatable result of instantaneous flow measurement and only time-averaged results are presented in **Fig.11** and **Fig.12.** for ϕ=0.36 and ϕ=0.6. The 3-D velocity components are shown by two sets of 2-D vector plots for various sections (as defined in **Fig.2-a)** and cylindrical surfaces

Comparing **Fig.11** with **Fig.12**, it is found that the magnitude of the vectors in the recess is increased with decreasing flow rate. This increase is most noticeable in the inlet of the recess over the rotor where the velocity measured at ϕ=0.36 is



a) 48mm Upstream the Rotor

b) 56mm Downstream the Rotor

Fig.10 Radial Profile of Total and Static Pressure Coefficients for Solid and Treated Casing

approximately equal to the blade tip speed of 39m/s, two times greater than that measured at ϕ=0.6. This radial velocity is presumably caused by an unbalanced radial pressure gradient in the exposed tip region, which increases with blade loading. As the loading increases with reduced flow so do the forces driving the recess flow. **Fig.11** and **Fig.12** also indicate the area of recirculation to become more confined within the vaned passage at high flow ϕ=0.6 compared to the conditions at ϕ=0.36.

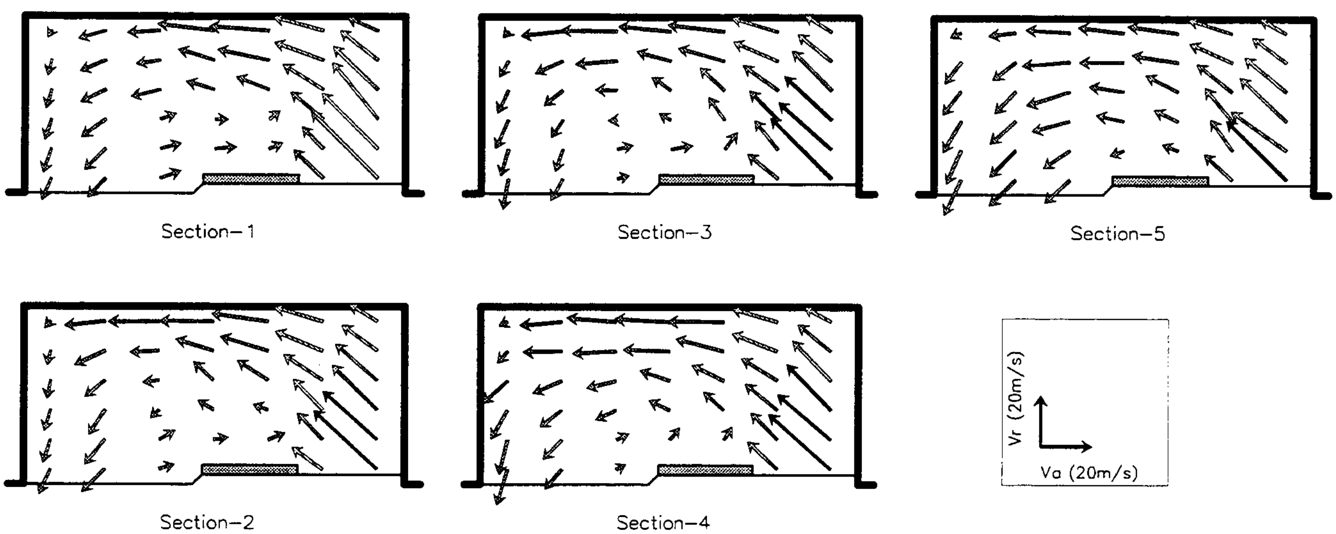
DISCUSSION OF THE MECHANISM OF THE TREATMENT

While these experiments have given some clues to the mechanism by which the treatment operates, some conjecture is still required and it is hoped that the following may provoke some further thought and suggestions from readers.

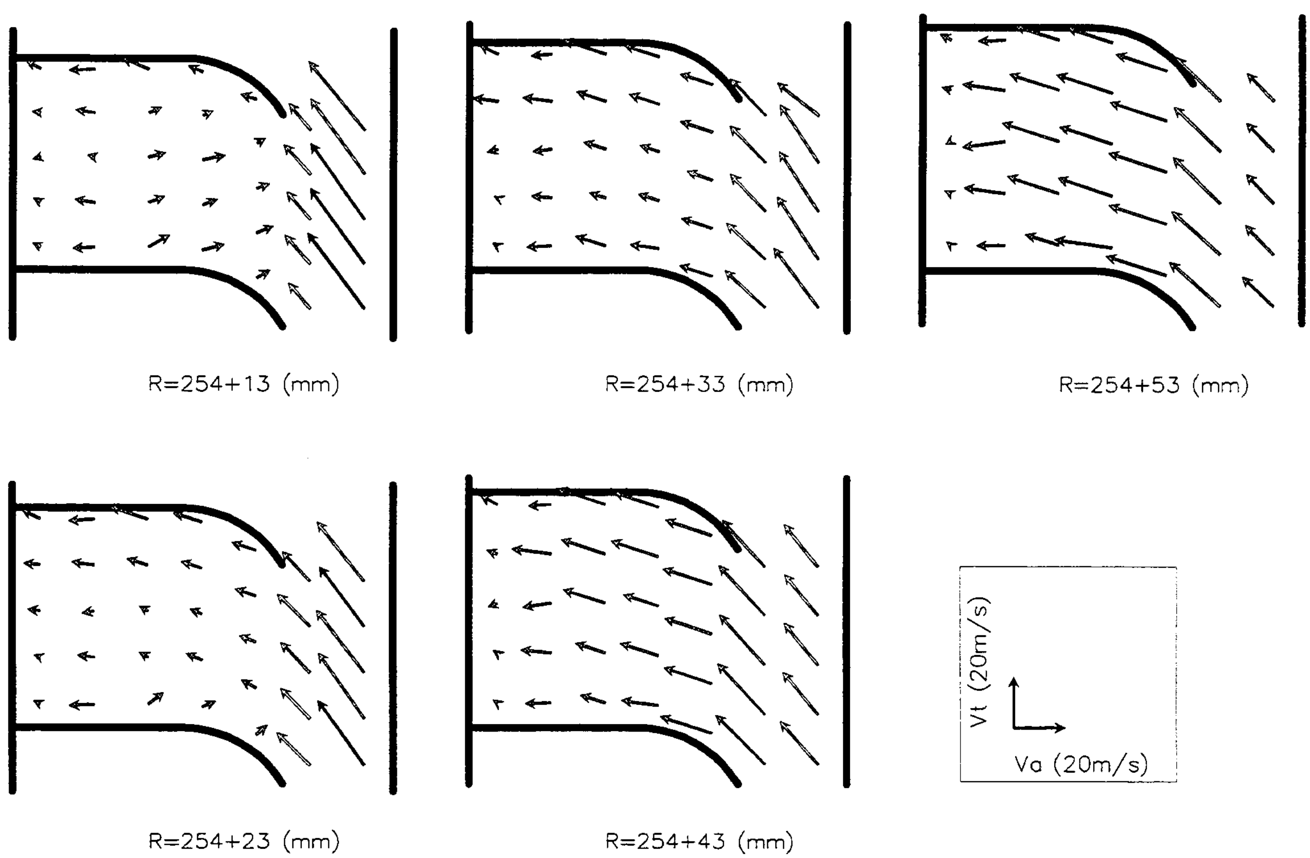
As already stated, the axial position of the treatment relative to the rotor chord is very important and the best results were obtained with the leading two thirds of the rotor chord portion exposed to the recess. **Lakshminarayana** et al. (1982) studied the three-dimensional flow field in the tip region of a low speed, moderately loaded axial compressor and found that the tip leakage flow originates near the quarter-chord and peaks around the mid-chord. It is interesting to relate this observation to the functional sensitivity of the treatment (**Fig.5**) where the change in flow regime changes when approximately 50% of the blade chord is exposed to the treatment. Furthermore, the measurements indicate a strong radial flow from the rotor tip into the treatment recess. Under these conditions, it is highly probable that the treatment significantly modifies the leakage flow, possibly eliminating the normal tip vortex flow with a radial flow into the recess instead. Such proposals are supported by the hot-wire measurements taken upstream of the rotor which were more repetitive with the treatment suggesting the removal of some non-periodic perturbation (possibly the tip vortex).

Having proposed this and observed the similarity between the rotor performance with and without treatment, it is possible to conjecture that, in the unstalled region, the losses associated with the tip clearance vortex (solid casing) and recess flow (casing treatment) are approximately equal but whereas the tip flow breaks down with the solid casing the recess vane permits a more progressive operation (where the recess flow and it losses increase with reduced flow rate). Additionally, it was noted (**Fig.4**) that the work input was higher for the treated builds. In the case of the solid wall build, rotating stall occurred and can be approximated to cells of zero axial velocity (zero mass flow), and regions of high, unstalled, axial velocity. This implies zero work in the stall cell and a low work coefficient in the unstalled zones consistent with the high flow coefficient leading to low averaged work input. With casing treatments, however, this rotating stall is not permitted and so the work coefficient goes on increasing as the flow rate reduces.

Furthermore, measurements taken suggest that whereas without the treatment there is a high whirl velocity ahead of the rotor in the stalled regime, the treatment separates the recirculating flow, removes the tangential velocity and re-introduces it into the rotor with a high relative velocity. This increases the work input and makes possible the higher pressure rises observed without development of rotating stall cells until a much lower flow rate. The action of the vanes in eliminating tangential flow in the rotor tip region is seen as essential to the

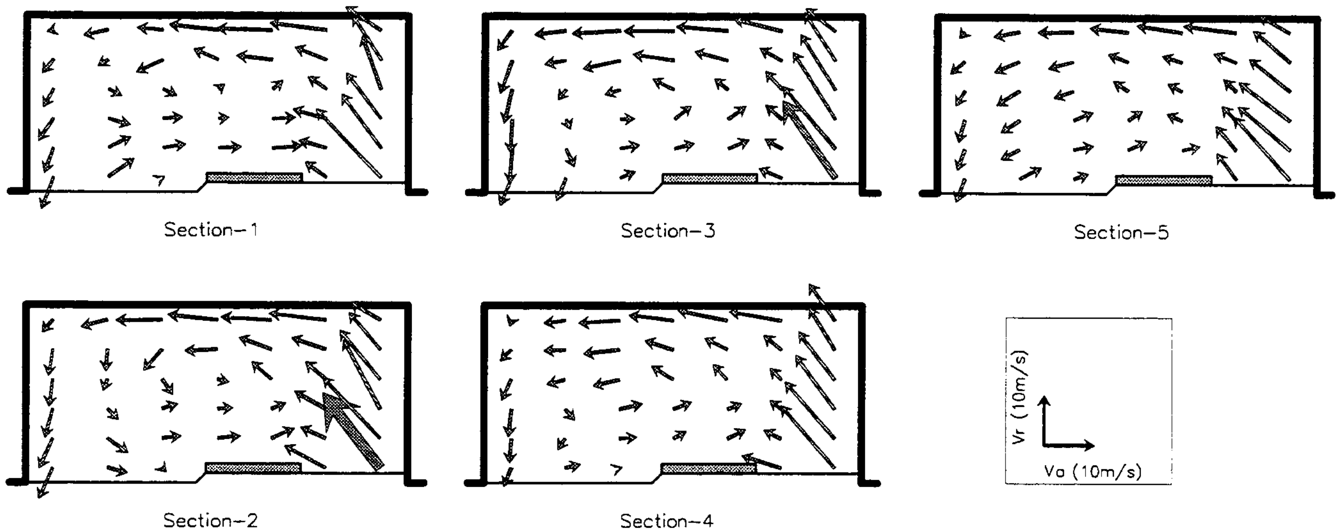


a) Vector Plot in Axial−Radial Plane

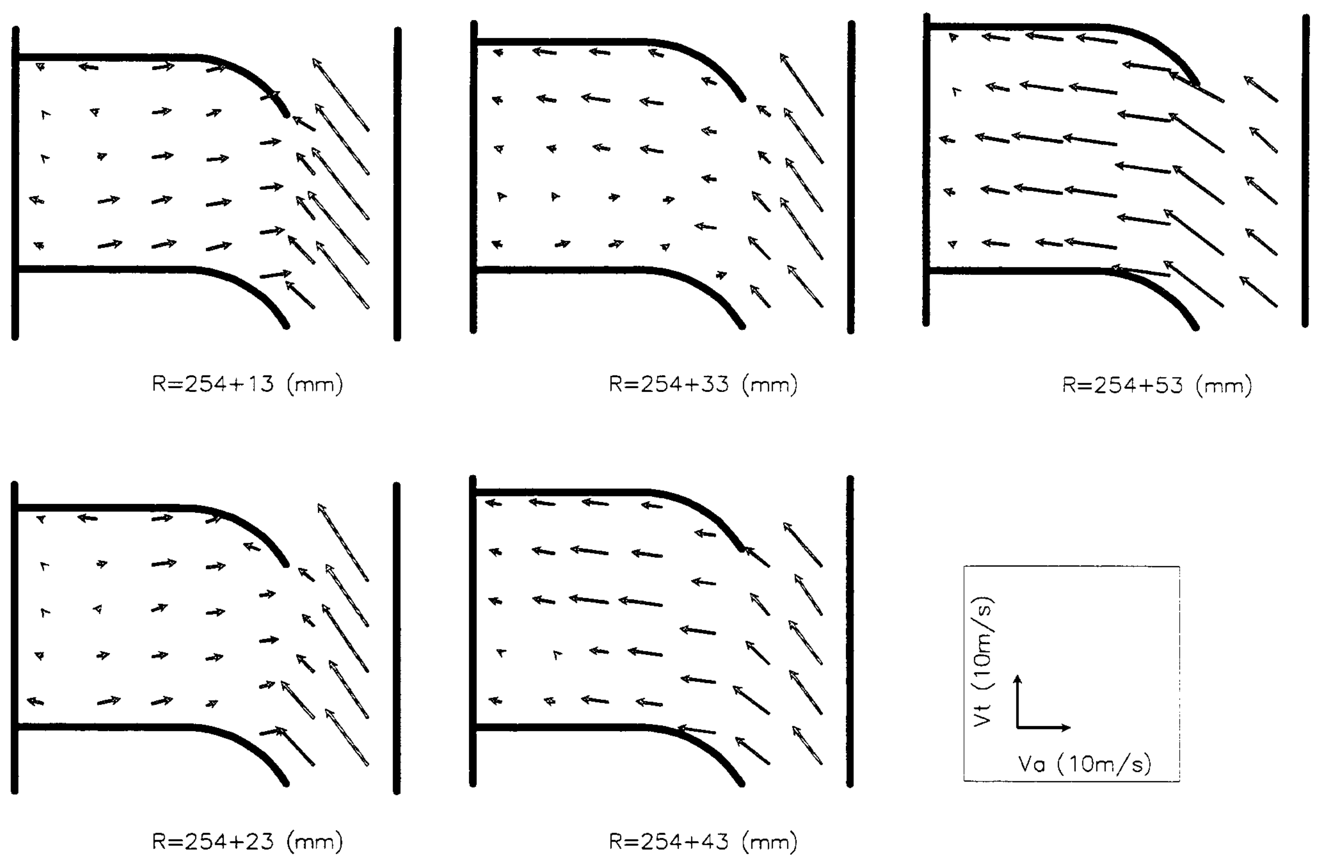


b) Vector Plot in Axial−Tangential Plane

Fig.11 Flow in Vane Passage of the Casing Treatment (Throttle 10%, Va/Um=0.36)



a) Vector Plot in Axial–Radial Plane



b) Vector Plot in Axial–Tangential Plane

Fig. 12 Flow in Vane Passage of the Casing Treatment (Throttle 45%, Va/Um=0.60) stall delay mechanism.

CONCLUSIONS

An experimental investigation to examine the influence of the vaned recess casing treatment on stall margin, operating efficiency and flow field of a low speed axial flow fan with aerospace type blade loading is presented. Detailed 3-D flow measurements in the endwall region and in the casing recess were carried out with a slanted hot-wire, providing some insight to the operation of the device. Based on this investigation, the following conclusions are drawn.

- Different geometrical designs of the vaned passages were examined and more than 65% of stall margin improvements and over twice its pressure rise (at stall) were obtained.
- The treatment did not significantly alter the efficiency curve. Neither the peak efficiency nor the efficiency in the extended unstalled range was significantly changed.
- The vane used in the recess has only a marginal effect on the overall performance of the compressor and different vane passages made much difference in stall margin but little difference in pressure rise and efficiency characteristics.
- It was found that the stall margin improvement due to the casing treatment was not achieved by reducing the incidence or by increasing the total pressure in the upstream tip region. It appeared that the flow leaving the exposed rotor tip (especially around mid-chord) and entering into the recess, and the elimination of the swirling flow by the recess vane are the most important features associated with the mechanism.

ACKNOWLEDGEMENT

The authors wish to express their gratitude to Mr. B. Charnley and other technicians for their assistance in the test arrangements.

REFERENCES

Azimian, A.R., Elder, R.L. and McKenzie, A.B.(1989) "Application of Recess Vaned Casing Treatment to Axial Flow Fans." ASME 89-GT-68

Bard, H.(1984) "Unstable Characteristics for Fans", AMCA Engineering Conference, Atlanta, Georgia, April 2-4, 1984

Bard, H.(1993) "Fan Stability by Anti-stall in Tunnels and Industrial Process Ventilation", Turbo Compressor and Fan Stability, I Mech E Seminar, April 8.

Crook, A. J., Greitzer, E. M., Tan, C. S., and Adamczyk, J. J. (1993) "Numerical Simulation of Compressor Endwall and Casing Treatment Flow Phenomena", Trans. ASME, *Journal of Turbomachinery*, Vol. 115/501

Ivanov, S. K. (1984) "Axial Flow Ventilation Fan", U. K. Patent Application (19) GB(11) 2 124, 303A.

Johnson, M. C. and Greitzer, E. M. (1987) " Effect of Slotted Hub and Casing Treatments on Compressor Endwall Flow Fields", Trans. ASME, *Journal of Turbomachinery*, Vol. 109/380

Lakshminarayana, B., Davino, R. and Pouagare, M. (1982) "Three-dimensional Flow Field in the Tip Region of a Compressor Rotor Passage-part I: Mean Velocity Profiles and Annulus Wall Boundary Layer; part II: Turbulence Properties", Trans. ASME, *Journal of Engineering for Power*, Vol. 104, Oct.

McKenzie, A. B. (1993) "Passive Stall Control for Fans and Compressors", Turbo Compressor and Fan Stability, I Mech E Seminar, April 8.

Miyake, Y., Inaba, T. and Kato, T.(1987) "Improvement of Unstable Characteristics of an Axial Flow Fan by Air-separator Equipment." Trans. ASME, *Journal of Fluid Engineering*, Vol. 109, P36-40, March.

Smith, G.D.J. and Cumpsty, N.A.(1984) "Flow Phenomena in Compressor Casing Treatment", Trans. ASME, *Journal of Engineering for Power*, Vol. 106, July 1984

Takata, H. and Tsukuda, Y. (1977) "Stall Margin Improvement by Casing Treatment--its Mechanism and Effectiveness." Trans. ASME, *Journal of Engineering for Power*, Vol. 99, Jan. 1977

Whitfield, C. E., Kelly, J. C. and Barry, B. (1972) "A Three-dimensional Analysis of Rotor Wakes" *Aeronautical Quarterly*, Nov. 1972

Ziabasharhagh, M., McKenzie, A.B. and Elder, R.L. (1992) "Recess Vane Passive Stall Control." ASME 92-GT-36