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Recess Vane Passive Stall Control

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ABSTRACT

An experimental investigation has been carried out on the influence of a vaned recessed casing treatment on the stall margin improvement of axial flow fans with different hub to tip ratio, with and without inlet distortion.

The inlet distortion tests were conducted on a 0.5 hub to tip ratio fan and significant increases in the flow range with only small drops in operating efficiency were observed.

The clean flow tests were conducted on higher hub to tip ratio fans (0.7 and 0.9). In each case the stage characteristic was compared with the results obtained with a solid casing. Significant increases in the flow range, with only modest or no loss in operating efficiency, were observed for optimum configurations at both diameter ratios.

NOMENCLATURE

V_a	axial component of velocity
U_m	mid span blade speed
ϕ	flow coefficient, V_a/U_m
ϕ_{rs1}	stall flow for the solid casing
ϕ_{rs2}	stall flow for the treated build
ϕ_{p1}	peak pressure for the solid casing
ϕ_{p2}	peak pressure for the treated build
ρ	air density
Ψ_{TS}	(P(exit static) - P(inlet total)/ρU_m^2
η	efficiency
ΔH	work input
SM_{rs}	$1 - \phi_{rs1}/\phi_{rs2}$ - stall margin improvement based on RS inception
SM_p	$1 - \phi_{p1}/\phi_{p2}$ - stall margin improvement based on peak in pressure rise
h/s	height/span
θ	camber
ξ	stagger
s	space
c	chord

INTRODUCTION

During the last two decades many attempts have been made to achieve stall delay and suppression with passive devices (Takata and Tsukada (1977); Smith and Cumpsty (1984); Prince et al., (1975)). In these studies a wide variety of different techniques generally termed casing treatments (circumferential, axial and skew slots, straight or tapered holes, honeycombs, etc) have been employed. These each have useful applications but they also suffer some disadvantages. An alternative technique, which was initially developed by the low pressure axial fan industry, Ivanov (1965), and has shown encouraging results, is a recess vane or "large scale" casing treatment which is described below.

A large scale recess vane treatment, Fig. 1, consists of a ring cavity which has a vaned and vaneless region and is situated outside but open to the outer annulus wall of the fan. It is installed co-axially upstream of the rotor with the vaneless region partially over the rotor. The term "large scale" comes from the fact that it has dimensions of the same order as those of the blading, Ivanov et al., (1984), in contrast to small scale conventional treatments.

Azimian et al., (1987) & (1990), described recent studies in which such a device was applied to a low speed axial fan with aerospace type loading. The operating principle appears to be that, as stall is approached, the radial and reverse flows which tend to collect and stall the tip of such fans, Soundranayagam and Elder (1992), pass into the treatment and are thereby removed from the main flow annulus. The tip flow pattern then established, Fig. 2, appears to stabilise the flow over the rotor and significant increases in the flow range with insignificant reductions in operating efficiency for optimised builds can be achieved. Previous studies have already demonstrated very significant improvements for a 0.5 hub/tip ratio fan and this paper reports the effect of inlet flow distortion on the same fan and results from studies on higher hub/tip ratios (0.7 and 0.9) with uniform inlet flow.

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EXPERIMENTAL ARRANGEMENT

The compressor facility used in this study for the 0.5 hub/tip ratio tests was the same as that used in the previous investigation by Azimian et al., (1987) and (1990), consisting of a single stage low speed axial flow fan with a tip speed of 39.9 m/s, tip diameter of 508mm (20") and blade loading ($\Delta H/U^2_{hub}$) of near unity at the hub. A distortion screen (gauze) was placed upstream of the fan and configured to provide a "crescent" shape distortion representing an intake separation similar to that which can occur in a cross wind during ground operation. The fan with the bladed recess located partly over and mainly upstream of the rotor blade is shown in Fig. 1. The vanes used inside the recess region for this build are shown in Fig. 3. The dimensions of the gauze are shown in Fig. 4. An arrangement using 48 vanes in the recess was examined. To determine the total and static pressure downstream of the gauze (upstream of the rotor) a 4mm diameter 3-hole cylindrical yawmeter was used which had been previously calibrated for yaw angle, total and static pressure. Measurements were carried out by making a radial traverse of the flow at 30 degree (circumferential) intervals. The distortion gauze was situated 150mm upstream of the rotor providing a distortion coefficient, based on a 60 degree segment (DC_{60}), Reid (1969), of between 0.3 and 0.4 (depending on flow rate).

The higher hub/tip ratio studies were undertaken on another single stage low speed axial flow fan again with a tip speed of 39.9 m/s and tip diameter of 508mm. This fan rig had different hub configurations such that it could be tested with several hub/tip ratios including 0.7 and 0.9. The same casing treatments could be tested with either hub configuration. A casing treatment with overall dimensions similar to those of the distortion test was used for the high hub/tip ratio study except that the vane number was reduced to 24 (previous studies had indicated that the reduced blade number provided only modest changes, Azimian et al., (1990)). Two additional reduced sized casing treatments were designed and manufactured for these tests which, including the full scale treatment provided three in all. The different treatments had the same recess vane profile and general configuration but different overall dimensions. Accordingly they were termed "large", "medium" and "small" treatments which are defined in Fig. 5. This nomenclature is used throughout this paper. The "large" treatment design was based broadly on the ideas of Ivanov et al., (1984), for the 0.5 diameter ratio fan. The dimensions of the "medium" and "small" treatments were scaled in proportion to the blade heights of the 0.7 and 0.9 diameter ratio stages, i.e. the height of the "medium" treatment was scaled to 60% of the "large" treatment, and the "small" one to 20% of the "large" one. To maintain reasonable vane solidity the medium and small size treatments had the number of vanes increased to 40 and 90 respectively.

For both test rigs used the driving torque was measured by dead weights applied to an arm attached to the motor, which itself was mounted on trunnion bearings. The accuracy of the torque and other measurements appears to be such as to give a repeatability of $\pm 1\%$ on efficiency when no rotating stall is present.

The 0.7 and 0.9 diameter ratio blades were existing from previous unassociated work. Unfortunately the 0.9 diameter ratio rotor blades had suffered appreciable damage at the trailing edge during previous running, and this had to be removed by trimming away about 25% of the chord. This led to a lower pressure rise being available from these blades than from the 0.7 diameter ratio blading. Table 1 gives details of the blade geometries as tested.

EXPERIMENTAL RESULTS AND DISCUSSIONS

The stall margin improvements discussed below are used to quantify the performance change between the solid casing and the subsequent treated builds. The first parameter used was that defined by Takata and Tsukada (1977), and expressed as:

$$SM_{rs} = 1 - \frac{\phi_{rs2}}{\phi_{rs1}} \quad (1)$$

where SM_{rs} is the "stall margin" improvement and ϕ_{rs2} and ϕ_{rs1} are the mass flow coefficients at rotating stall inception for the treated build and solid casing respectively. A second parameter is introduced here which considers the change in flow rate at which the peak pressure rise occurs. In some circumstances it may be thought that the peak in pressure rise is a more practical limit to operation. This parameter is expressed as:

$$SM_p = 1 - \frac{\phi_{p2}}{\phi_{p1}} \quad (2)$$

where SM_p is the "flow margin" improvement and ϕ_{p2} and ϕ_{p1} are the mass flow coefficients at peak pressure rise for the treated casing and solid wall builds respectively.

INLET DISTORTION STUDIES ON 0.5 HUB/TIP STAGE

Solid Casing Without Distortion (Datum Build)

A characteristic for the stage is shown in Fig. 6. Beyond stall a region of reverse flow in front of the tip of the rotor was noted by means of wool tufts studies.

Solid Casing With Inlet Distortion

The fan characteristic measured is shown in Fig. 7 where it is compared with the datum build. As can be seen the change in mass flow coefficient is less significant than the change in pressure rise coefficient. Similar results were observed by Reid (1969).

Casing Treatment Without Distortion

Tests were undertaken where the optimum percentage of Axial chord of the rotor was exposed to the recess vane. Azimian et al., (1990), had previously demonstrated this to be 67%.

The results, Fig. 8, which compare the solid and treated casing configurations for undistorted flow, show significant improvement for the treated case with a stall margin improvement (SM_{rs}) of about 59% and a flow margin improvement (SM_p) of a more modest 8%. The pressure rise improvement was not very significant and there was also a small loss in peak efficiency with the casing treatment. Investigation into the form of the pressure rise characteristic for the casing treatment build indicated that the static pressure rise across the stator fell rapidly at flow coefficients below about 0.6 giving rise to the reduction in pressure rise coefficients for the stage at lower flows, Fig. 9. The rate at which static pressure rise fell reduced at a flow coefficient of about 0.4 and at lower flow rates the still increasing rotor pressure rise characteristic was able to provide a recovery in the stage

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characteristics which was followed by rotating stall inception as shown.

Casing Treatment With Inlet Distortion Results

The results of tests with the casing treatment, both with and without inlet flow distortion, are compared with the solid wall with inlet distortion in Fig. 10. Examination of the results show that imposition of the distortion on the casing treatment build reduces the flow range somewhat but has little influence on the peak pressure rise. An alternative way in which to consider these results is that the effect of the treatment on the fan operating with distortion was to significantly improve the peak pressure ratio and introduce a stall margin improvement (SM_{RS}) of 27%.

Table 2 provides a review of results for these experiments. In summary it appears that the casing treatment has performed well with significant increase in stall margin range and peak pressure rise with only a small reduction in efficiency.

HIGHER DIAMETER RATIO RESULTS

Solid Casings (Datum Builds)

The characteristics for the stages with 0.7 and 0.9 hub/tip ratios are shown in Fig. 11. Measurement with wool tufts have shown that in both cases the rotating stall was most severe at the rotor tip and therefore it appeared that the stall performance of the units was likely to be modified by changes to the tip geometry or outer casing.

Casing Treatments Results

As mentioned before, three sizes of treatments were tested with both diameter ratio fans.

The results, Table 3, show that significant improvements are possible with the optimum treated builds. It appeared that for these optimum builds, the flow process described in Fig. 2 could be established with little or no loss in efficiency. The optimum build results for the "large", "medium" and "small" treatments tested with the 0.7 hub/tip ratio configuration are shown in Figs. 12 to 14. A similar set of results for the 0.9 hub/tip ratio build are shown in Figs. 15 to 17. To achieve these results an extensive investigation was carried out to find the optimum position of the treatment for each diameter ratio and treatment tested. The results show that for 0.7 hub/tip ratio the large size treatment provided the largest stall gains for the smallest penalty whereas for the 0.9 hub/tip ratio build, the 'best' results were obtained with the medium size treatment. It therefore appears that the size of the recess treatment is another significant parameter influencing performance.

As previously reported, it appears that the flow exchange between the main flow and the recess area is of great importance. To understand better the mechanism involved a flow investigation inside the recess was carried out using wool tufts to indicate the local flow direction, Fig. 18, where multiple arrows indicate unsteadiness. Although the results presented are somewhat qualitative, they provide some considerable insight to the process. In Fig. 18 the radial height of 3% corresponds to a cylindrical plane within the treatment but close to the rotor tip and 97% represents another cylindrical plane near the outer casing of the recess vane cavity. The curved region (or leading edge) of the recess vanes shown in Fig. 18 are located near the rotor leading edge, the straight region of the vane being located upstream and farthest from the rotor leading edge. The results indicate that the flow is very unsteady in the vaneless region of the recess cavity until height/span > 40% where it is somewhat less unsteady. Flow in the recess vanes at height/span = 97% is fairly uniform except on the recess vane suction surface where some evidence of separation is present. At lower values of height/span (planes closer to the outer annulus of the fan) the flow gets increasingly more distorted with evidence of flow reversing into the direction of the compressor primary flow. At lower flow rates, the recirculating flow behaviour was stronger.

HYPOTHESIS OF OPERATION

It is suggested as a method of operation that low momentum fluid, associated with stalling flow on the blade, is centrifuged towards the blade tip. Without the casing treatment the centrifuged flow reaching the casing must be deflected forward or rearward or both by the casing as well as being entrained by the mainstream. The casing treatment recess provides an alternative path for the centrifuged air, and the vanes reduce the whirl velocity to near zero for air which is returned to the normal annulus at the forward end of the recess. Thus, when the mass flow falls below the value at which rotating stall cells would normally appear, the reversed flow from the rotor does not dramatically upset the incoming flow to the rotor. By this means the rotor is enabled to continue pumping to higher pressures and lower flows. Eventually, as the mass flow is further reduced, so much flow is centrifuged that the recess is too small to cope with all the centrifuged flow, and some of the reversed flow starts to occupy the outer region of the normal annulus immediately ahead of the rotor. This may cause rotating stall to be established, as when there is no casing treatment.

An intriguing feature of the test results is the minimal loss of peak efficiency with the treatments, where the leading 60% or so of the rotor chord has a massive tip clearance. No explanation is offered, but clearly further work to understand and exploit this phenomenon is called for.

CONCLUSION

A series of low speed axial flow fan tests (with aerospace type blade loadings) have been undertaken to investigate the influence of large scale casing treatments on their overall performance. Certain builds have also involved studies with inlet flow distortion. The general conclusions were as follows:

1. For the 0.5 diameter ratio stage it has been demonstrated that the casing treatment can restore the deficit of maximum pressure rise caused by a typical inlet flow distortion, such as arises where an aircraft turbofan is run statically in a cross wind.

2. With uniform inlet flow, large increments of stall margin are obtained with the treatments. When the size of the treatment and the axial position of the rotor are optimised the deficit of maximum efficiency is negligible, although this configuration may not give the maximum increase in stall margin.

3. The stall margin increments are substantially greater than reported in the literature for small scale treatments, which are most often contained within the axial chord of the rotor.

4. Three sizes of treatment were manufactured by scaling treatments in proportion to the blade height. The results indicate that this is not a correct means to obtain similarity of performance. Scaling in proportion to the blade

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chord may be more correct, but there is insufficient evidence to be certain of this.

5. It is not yet possible on the basis of these tests, and previous work on similar configurations, to offer more than a very general hypothesis for the means by which the treatment functions. Both this and the minimal loss of maximum efficiency, despite massive rotor tip clearance, remain to be explained by further, more detailed, investigations.

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Description	0.7 h/t			0.9 h/t		
Section	Tip	Mid	Hub	Tip	Mid	Hub
θ(deg)	13.5	33.0	45.0	20.0	27.0	34.0
ξ(deg)	42.5	34.5	29.0	40.0	37.0	29.5
s/c	0.903	0.779	0.682	1.314	1.174	0.863

Table 1 Geometry of blades for 0.7 & 0.9 h/t ratios.

Configuration	ϕ at peak	ϕ at Stall	ψ at peak	ψ at stall	SM_{rs}	SM_p
Solid Wall	.587	.587	.249	.249	—	—
Solid Wall with inlet distortion	.657	.657	.202	.202	-11.9%	-11.9%
Casing treatment	.541	.240	.260	.213	59%	8%
Casing treatment with inlet distortion	.597	.487	.259	.242	18%[1] / 27%[2]	-1.7% / 9%

Table 2 Summary of inlet distortion results.

SM_p = Flow margin improvement based on peak of pressure rise(see equ 2)

SM_{rs} = Stall margin improvement based on stall limit (see equ.1).

1 = Datum clean solid wall.

2 = Solid wall with inlet distrortion.

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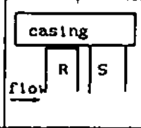
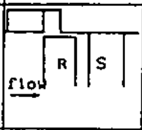
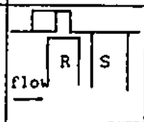
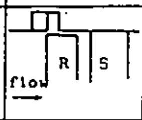
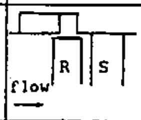
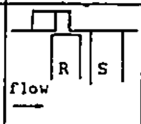
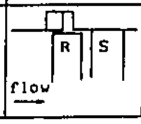
Build Number	Description	Sketch	percent of Axial chord of rotor exposed	Stall Margin Improvement	Loss in peak efficiency
0	Solid wall casing 0.7 & 0.9 h/t ratio builds		—	—	—
27	0.7 h/t ratio. recess with 24 vanes large size treatment		47%	58.5%	0%
6	0.7 h/t ratio recess with 40 vanes medium size treatment		63%	76%	2%
20	0.7 h/t ratio recess with 90 vanes small size treatment		32%	12.6%	1%
41	0.9 h/t ratio recess with 24 vanes large size treatment		89%	14%	2%
55	0.9 h/t ratio recess with 40 vanes medium size treatment		60.7%	34.1%	0%
54	0.9 h/t ratio recess with 90 vanes small size treatment		67.9%	50%	3%

Table 3 Results with different size treatments for 0.7 & 0.9 h/t ratios builds.

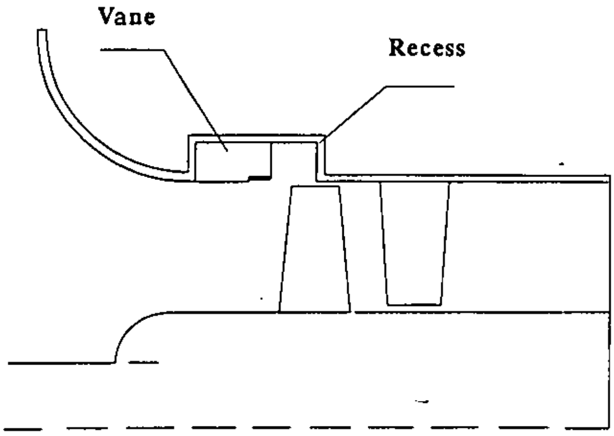


Fig. 1 General arrangement

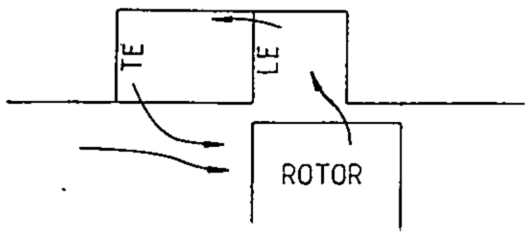


Fig 2. Flow pattern.

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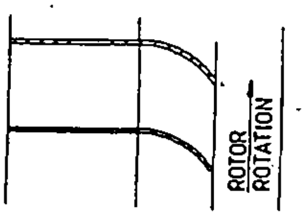
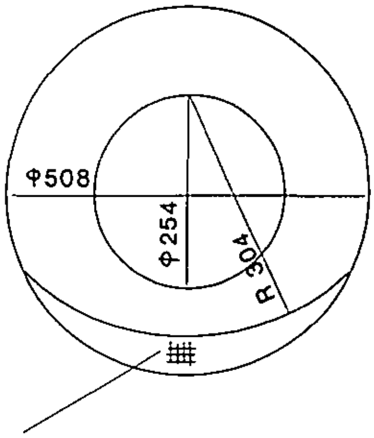


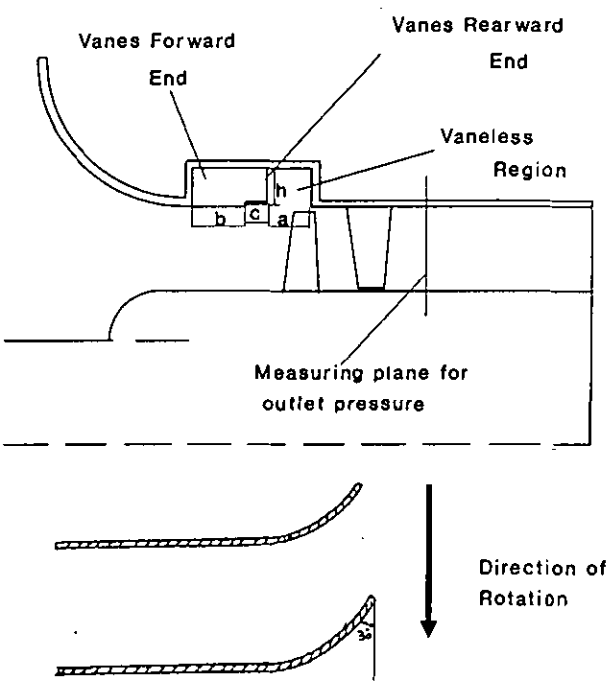
VANE NUMBER = 48 AND 24
SECTION OF RECESS VANE

Fig 3. Vane Arrangment



6 meshes per 25.4 mm

Fig. 4 The geometry for gauze (screen).



Large	Medium	Small
a = 33.8	a = 28.0	a = 12.5
b = 56.4	b = 34.0	b = 17.0
c = 30.0	c = 21.0	c = 10.0
h = 50.0	h = 31.0	h = 11.0
z1 = 48	z = 40	z = 90
z2 = 24		

z1 Vanes No. for distortion tests;

z2 Vanes No. for higher h/t ratio tests

Fig. 5 Recess vane casing treatment configurations.

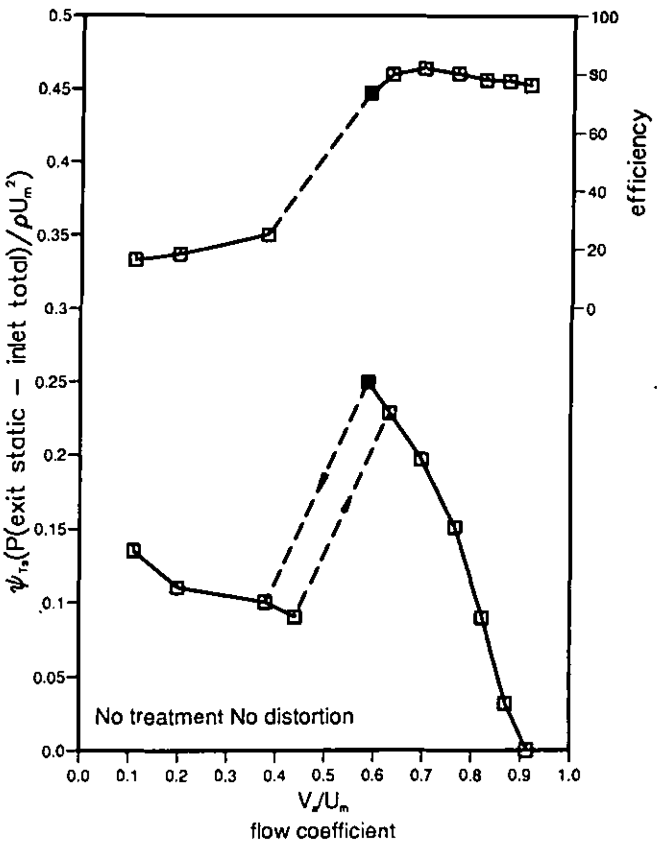


Fig. 6 Overall performance for stage (0.5 h/t)

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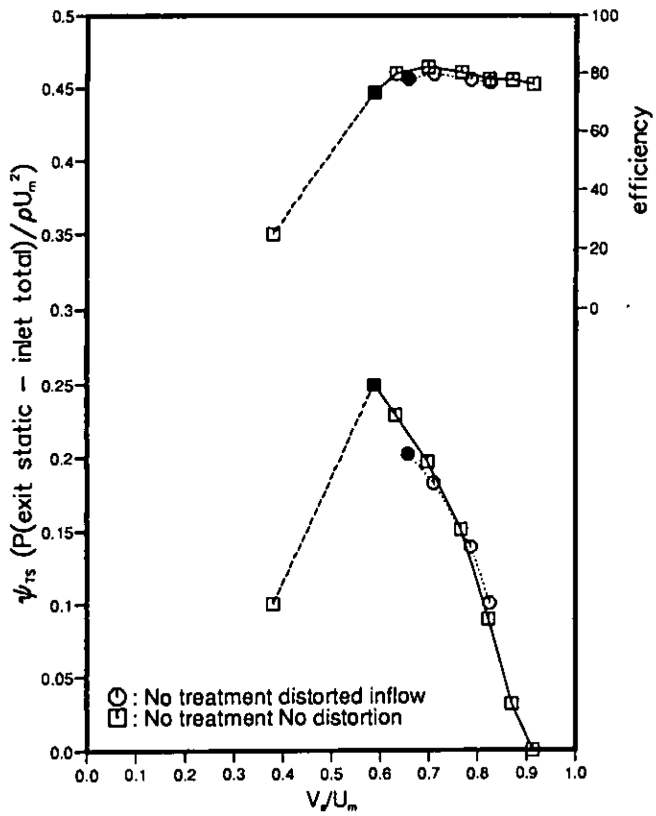

Fig. 7

Overall performance of solid wall with and without inlet distortion.

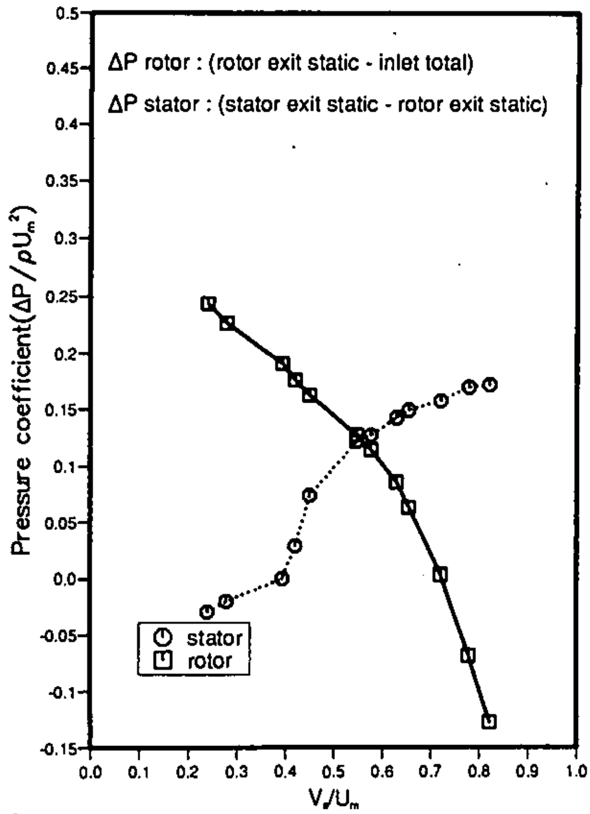

Fig. 9

Pressure rise characteristic for casing treatment (rotor and stator).

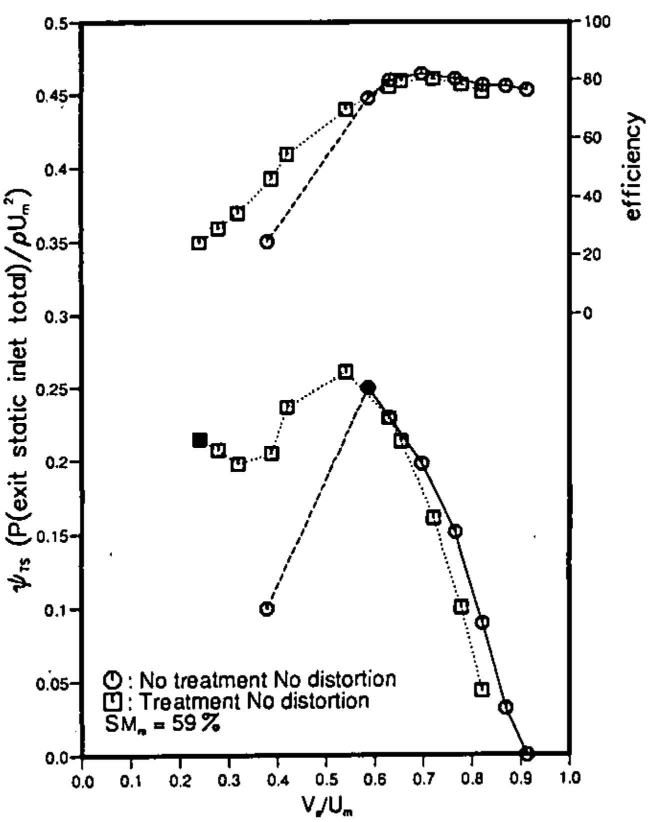

Fig. 8

Overall performance with and without casing treatment (0.5 h/t).

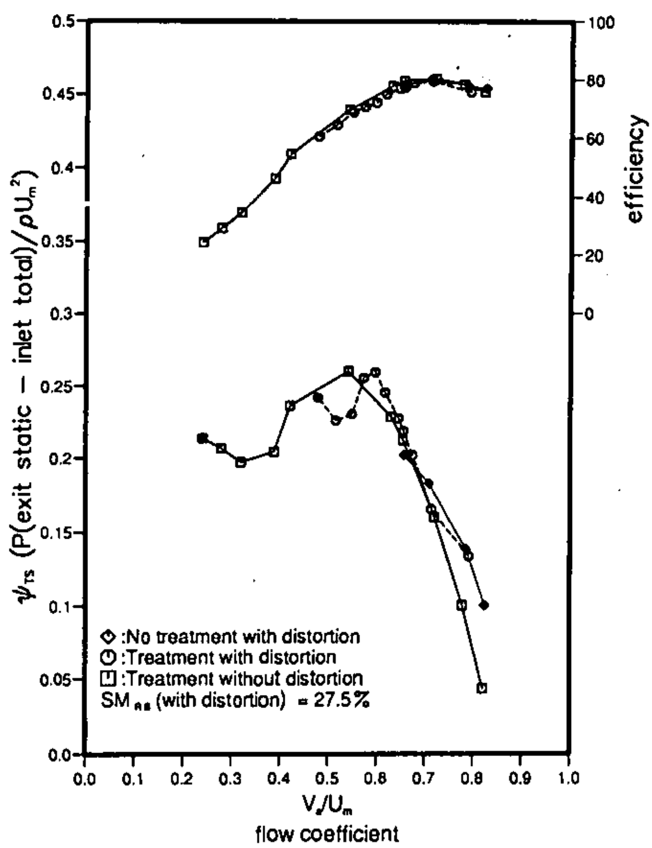

Fig. 10 Overall performance with distortion (0.5 h/t).

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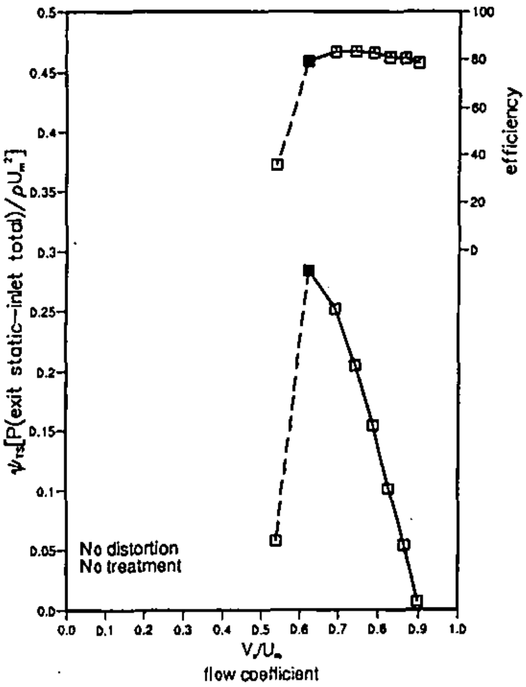

Fig. 11(a) Overall performance for 0.7 h/t build.

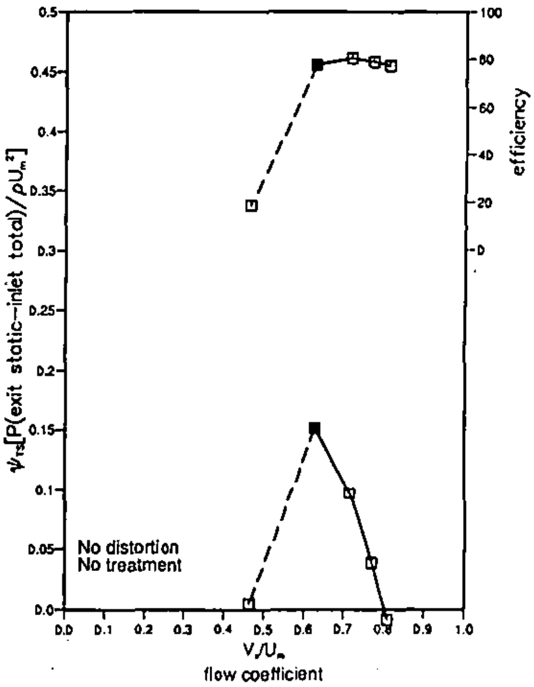

Fig. 11(b) Overall performance for 0.9 h/t build.

Fig. 11 Compressor Characteristics for Datum Clean Casing Builds.

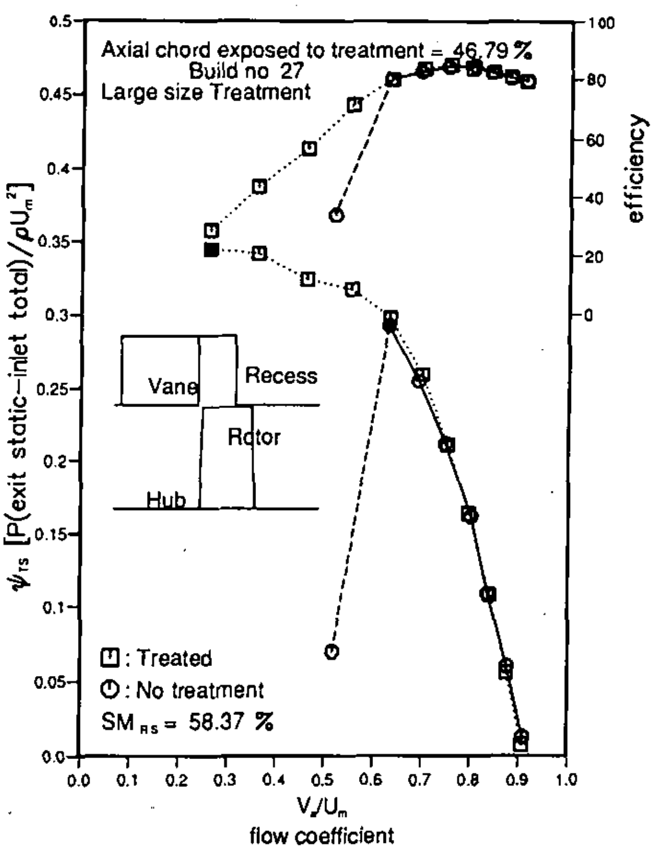

Fig 12. Overall performance for 0.7 h/t with large treatment.

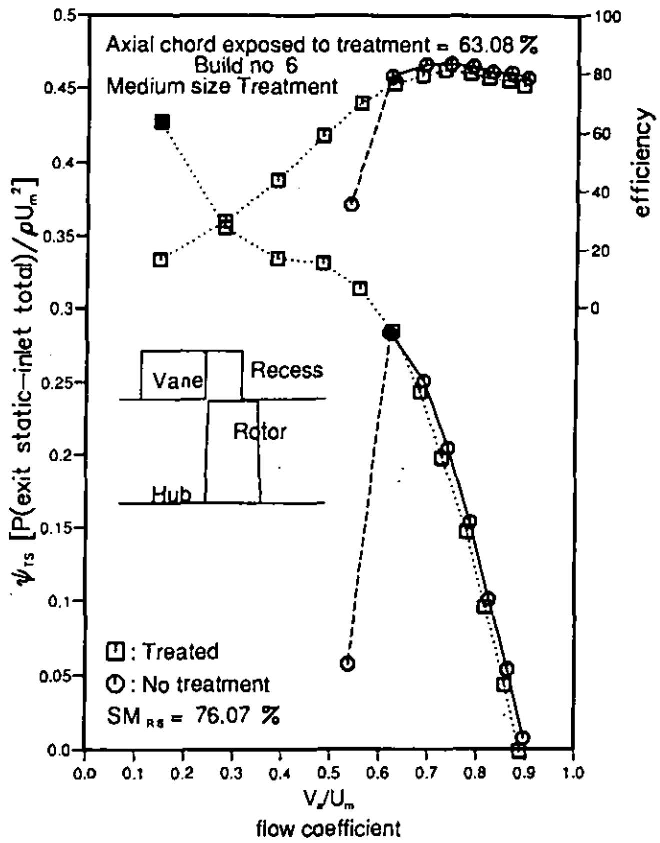

Fig 13.
Overall performance for 0.7 h/t with medium treatment.

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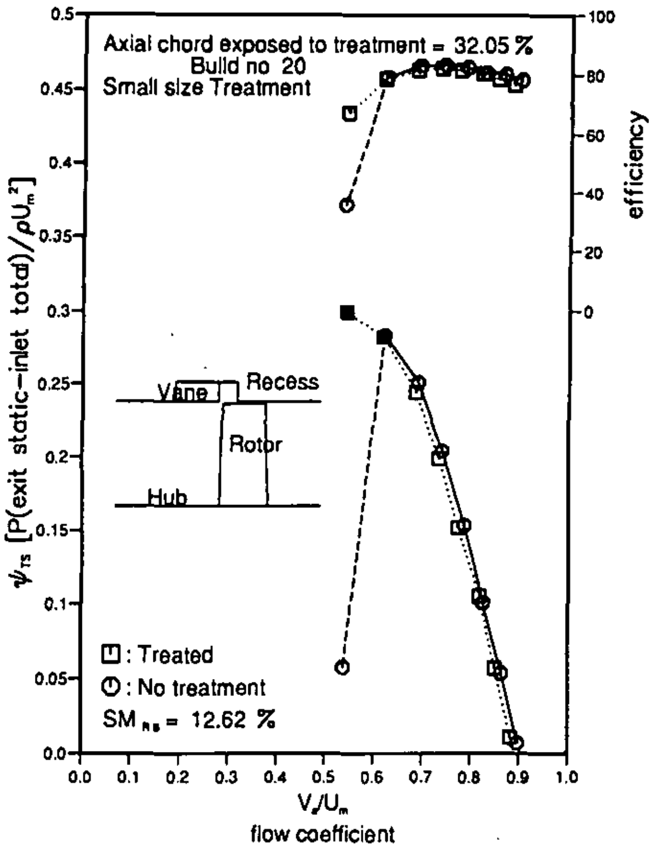

Fig 14. Overall performance for 0.7 h/t with small treatment.

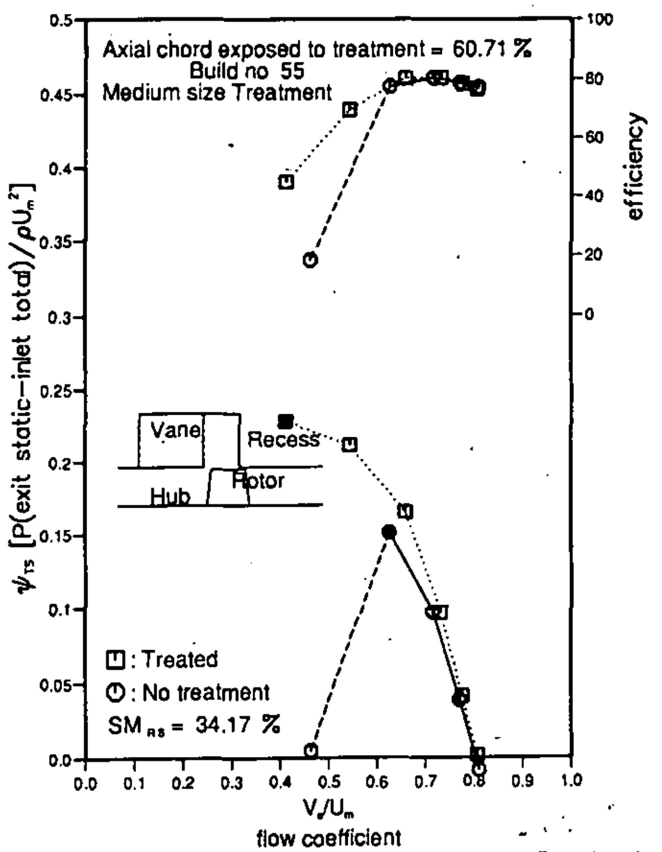

Fig 16. Overall performance for 0.9 h/t with medium treatment.

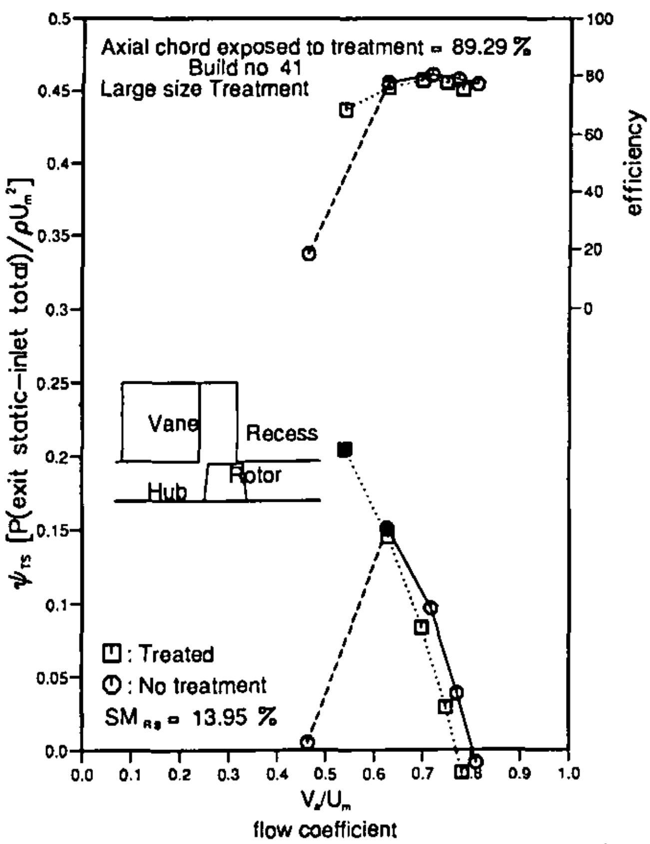

Fig 15. Overall performance for 0.9 h/t with large treatment.

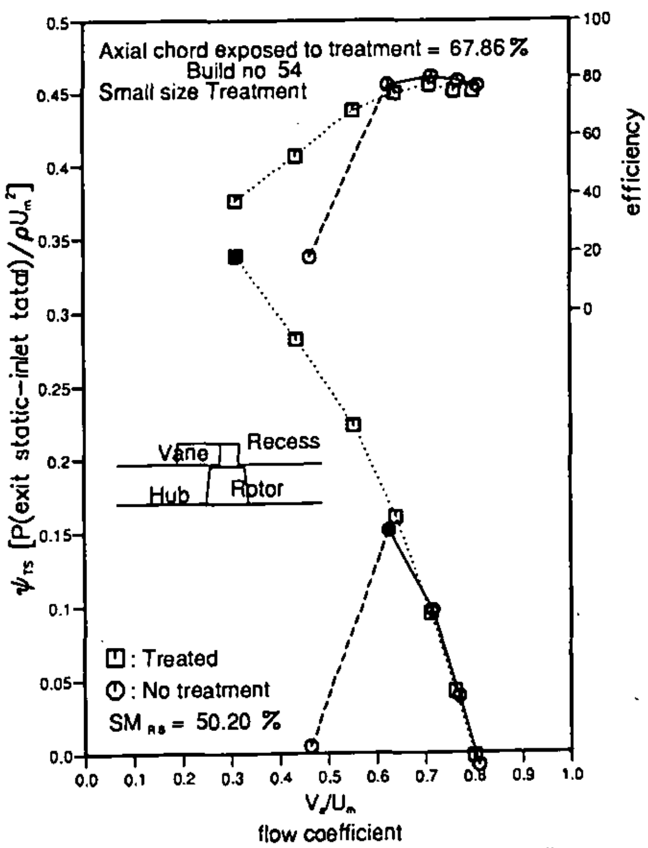

Fig 17. Overall performance for 0.9 h/t with small treatment.

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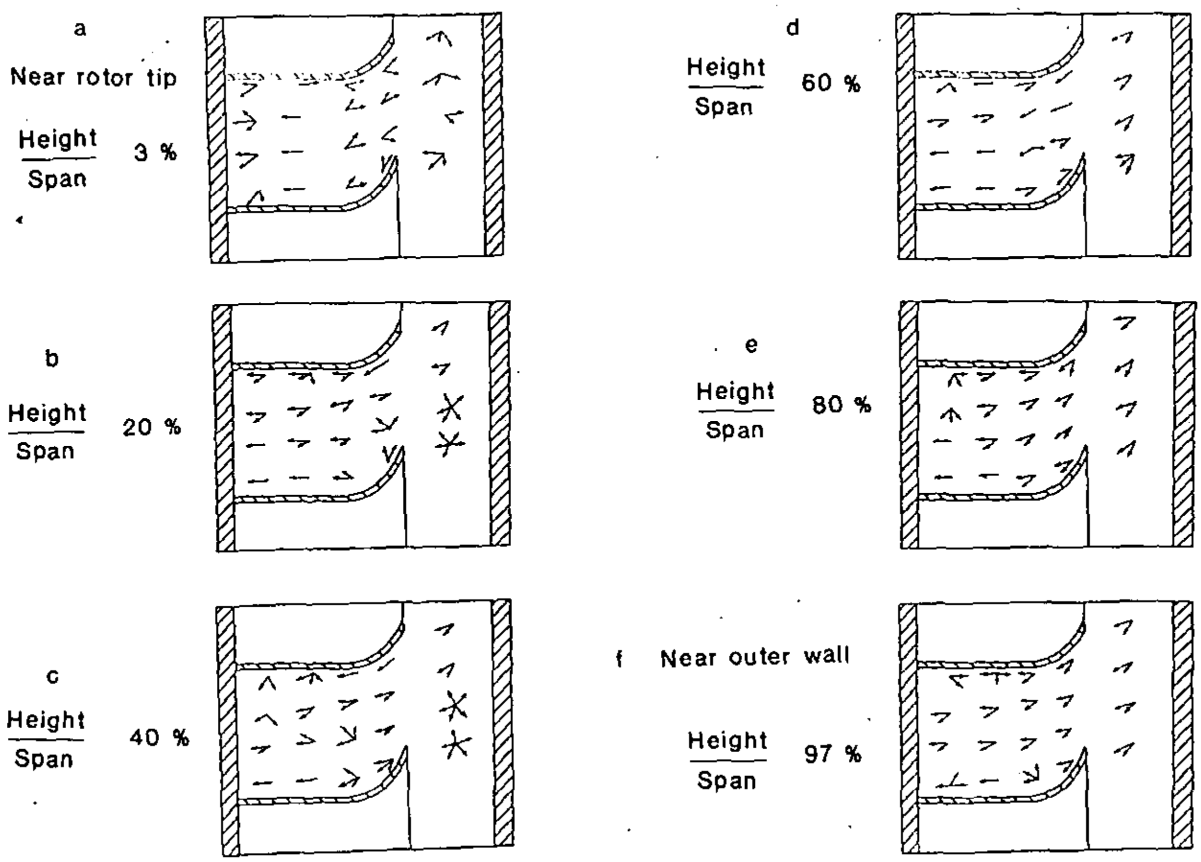

Fig. 18 Flow vectors inside the medium size treatment with 0.7 h/t ratio build.

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