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Flow Studies in Ducted Twin-Rotor Contra-Rotating Axial Flow Fans

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ABSTRACT

The design and testing of a 400 mm diameter contra-rotating fan unit was undertaken to study the flow behavior through the contra-rotating fans, and to find ways and means of improving their design and performance. The performance characteristics of the two-fan unit have shown that large overall stall margins can be achieved. Also, the effect of axial gaps showed that at the design speed combination best performance was observed at an axial gap of 50% of the 1st Fan chord.

Studies on the 2nd fan exit flow field, performance characteristics of individual fans and casing boundary layer development have been made. Significant performance enhancement is observed with serration on 2nd fan rotor blade surface. When casing boundary layer suction is employed in between the two blades, the 2nd fan exit flow shows better uniformity and increased total pressure at all radii. However, to obtain a large operating range, careful optimization of the 2nd rotor blade design would be required, taking into account peculiarities shown by the present study in variation of *deviation and exit flow angles of the individual fans, and casing boundary layer development with increased axial pressure gradient* .

NOMENCLATURE

c - blade chord	α Absolute air angle, Degrees
C - Absolute Velocity, M/S	β Relative air angle, Degrees
C_a - Axial Velocity, M/S	δ Deviation angle, Degrees
C_w - Whirl Velocity Component, M/S	i Incidence angle, Degrees
C_l - Lift Coefficient	σ Blade Solidity, (c/S)
D* Diffusion Factor	Θ Blade camber angle, Degrees
P_{static} - Static Pressure, N/M^2	Blade Stagger angle, Degree
P_{total} - Total Pressure, N/M	STATIONS
U - Tip Vel. of the Fans, M/S	1 First Fan Inlet
V - Relative Velocity, M/S	2 First Fan Exit
S - Blade spacing	3 Second Fan inlet
SPR,TPR - Static, Total Pressure Rise	4 Second Fan Exit

INTRODUCTION

The potential of contra-rotating fan has been explored earlier by many (Sharma et al, 1985 ; Celestina et al, 1986; and Sullivan et al, 1990). The stator-less configuration which emerged out of these studies offers a large saving in weight and size for any aircraft or any other vehicular gas turbine application . However, to-date very little detailed study has been reported in open literature. Sharma et al [1985, 1988] dealt with two axial gaps, 30% and 200% of the first fan chord. Our results on axial gap [32% to 110%] effect have been reported earlier [Roy and Rao, (1989)] and indicated that axial gap optimization is required. A summary of those results are reproduced here. A flow study has also been conducted to look into various flow parameters at design & some off-design operating points. Our results indicated that the contra-rotating fans in series behave somewhat differently than co-rotating fans. Two contra-rotating fans seem to combine the benefits of two fans in series and of those in parallel, at the speed combinations they were tested. The latter part of the study was initiated when it was observed that, the 1st and the 2nd fans showed differing characteristics : the second fan goes into partial stall at higher flow rate than the first fan and is more vulnerable to variation in speed and flow rate. Thus some flow studies of the wake of the 2nd fan (deviation and exit flow angles) and casing boundary layer development across the rotors was initiated. Flow near the casing was studied to get a measure of the flow uniformity, and specifically the boundary layers. It seems that the flow develops thick boundary layer across the rear fan. Two methods to improve the performance were attempted : i) Serrations on 2nd fan suction surface, & ii) Casing boundary layer removal by suction.

EXPERIMENTAL DETAILS

The CONTRA-ROTATING fan unit was designed (Table 1), built and installed (Ravibabu, 1987) in a test bed for testing axial fans [Fig.1]. The fans were to run at 2400 rpm and deliver 3.8 Kg/S at 2000 N/M^2. The fans (tip dia. of ≈ 400 mm) are housed in a 406 mm dia. casing.

TABLE-1 : BLADE DESIGN PARAMETERS

PARAMETERS	SECTION I F1, F2	SECTION II F1, F2	SECTION III F1, F2	SECTION IV F1, F2	SECTION V F1, F2
Dia.(M)	0.20, 0.20	0.25, 0.25	0.30, 0.30	0.35, 0.30	0.40, 0.40
β_1, β_3	32.1, 53.1	38.1 , 52.4	43.3 , 52.8	47.7 , 53.7	51.4 , 54.9
β_2, β_4	-12.9 23.7	5.6 , 30.3	20.3 , 35.8	31.3 , 40.5	39.6 , 44.3
σ = c/S	1.593, 1.93	1.27 , 1.54	1.06 , 1.28	0.90 , 1.10	0.79 , 0.96
$C_l \times \sigma$	1.62, 1.0	1.25, 0.80	0.95, 0.60	0.72, 0.45	0.60, 0.30
D*	0.36, 0.485	0.42 , 0.43	0.42 , 0.39	0.39 , 0.36	0.36, 0.33

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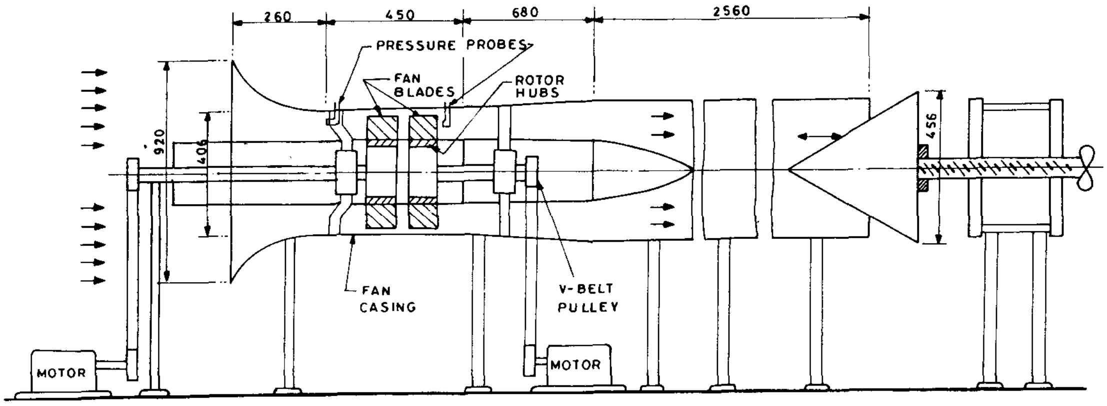


FIG. 1 . TEST BED FOR THE FAN UNIT

The fan blades were designed by Blade Element Theory, (NASA SP-36, 1965) and the flow was assigned a near-free-vortex operating characteristics, with varying incidences assigned to each blade section. A total of 5 blade sections were designed for each fan. No through flow prediction technique was available to the investigators and hence, was not attempted. A Blade Element based performance prediction method was carried out. The final performance was measured from the test bed. The 1st fan has 10 blades and second fan has 11 blades with varying solidity, stagger and camber from root to tip. The two fan blades were designed with modified C4 type blade profiles (F-series) bladings from root to tip [Wallis, 1977 & 1983], and the resulting twisted blades had entry velocity vectors of the 2nd fan matched to 1st fan outlet flow field at all the 5 radial blade sections, and thus, the fans have completely different shapes. The first accommodates axial entry of the flow and the 2nd fan exits the flow at low exit angle (10°), . Both are low aspect ratio blades, aspect ratio being 1 (Wennerstrom, 1986), with constant chord. Twist of the 1st fan was 33 ° and that of the 2nd fan was 23 °. However, second fan had higher stagger angles all the way from root to tip [Table 1].

The test bed (Fig.1) has the capability to vary fan speeds and mass flows independently. The two fans are separately powered by 2 D.C. motors through V-belt drives. Power input to each fan was measured directly from motor control panels separately for calculation of efficiency of each fan. Inlet static and total pressures were measured by standard pitot-static probes and the fan exit pressures were measured by shielded total pressure probe. The mass flow was varied by moving a cone axially, at the duct exit. The upstream measurement for inlet velocity and static pressure was made by standard pitot-static probe at 1-chord distance upstream of the first fan leading edge. The casing static pressure taps were made from 0.5 upstream of 1st fan to 1 chord distance downstream of the 2nd fan trailing edge. A total of 20 casing taps have been arranged for continuous monitoring of static pressure variation in the axial direction. At three stations, upstream of the first fan, in between the two fans & downstream of the 2nd fan, 8 circumferential casing static readings are taken to monitor axisymmetry of the flow at these stations. The downstream total pressure probe is traversed in radial direction, and readings are area averaged and plotted in the characteristics plots. Static pressure rise characteristics plots are based on casing static pressure readings. Mass flow has been calculated on the basis of inlet readings. The shielded total pressure, pitot-static and 4 hole yawmeter probes used have been calibrated in a low speed calibration tunnel (200 mm x 200 mm test section, Max. Velocity = 25 M/S).

The wake flow survey downstream of the 2nd fan was carried out with the help of a 0.5 mm outer dia. stainless steel probe at 0° yaw angle setting, which essentially measures the local total pressure. This probe was traversed every 1 mm to record the local flow field pressure. This was not strictly a boundary layer study, but a study of deterioration of pressure profile in the near-wake (x/c = 0.50) of 2nd fan resulting from the stator less contra-rotating fan configuration. At low mass flows the flow near casing at the 2nd fan rear studied with a TSI velocimeter (Hot-wire based, ≤30 M/S) with and without suction of casing boundary layer.The casing boundary layer removal was attempted through suction of boundary layer by suction blowers. The axial station for casing boundary layer suction was found from the axial variation of static pressure. All noise measurements were made 2 meter outside, axially upstream, of the inlet bellmouth.

OVERALL PERFORMANCE RESULTS

The results presented showed [Fig.2] the fan speeds varying from 2000 to 2400 rpm with 32% C_1 axial gap. At a particular speed combination, measurements are made with varying mass flow settings. Flow measurements were made with the digital micromanometers, digital non-contact rpm meters and in a IBM PC-AT microcomputer through a multichannel data logger.

The important first observation was the excess mass flow obtained from the contra-rotating unit. The first fan, when tested alone, on its own (Fig.2a), did produce, at individual design pressure rise of 1000 Pa , flow ≈3.65 Kg/S (≈ 3.85 Kg/S with 2nd fan removed). When the fans were run at speed combinations of 2000/2800 and 2800/2000 the mass flows (5.905 & 6.191 Kg/S respectively) came out to be nearly same as that of 2400/2400 (6.054 Kg/S); 1st fan speed seems to have greater effect on mass flow and the second fan speed seems to have greater effect on pressure rises and operating ranges. The total mass flow is almost always nearly equivalent to the sum total of individual mass flows of each fan. Thus, two fans in series and contra-rotating in close proximity produce the pressure rise as in two fans in series, and mass flow almost as if the two fans are in parallel. The contra-rotating fan unit which produced 60% more mass flow and 10% more pressure rise than designed for, was found to be partially stalled at the design mass flow (3.8 Kg/S), with full stall occurring immediately after 3.6 Kg/S. Actually, the pressure rise achieved at the design mass flow(≈3.8 Kg/S) is same as that, obtained at about 5 Kg/sec, where efficiency is much higher. The design pressure rise was obtained at the about 5.75 Kg/S mass flow . The characteristics of the contra-rotating fans (rpm≈ 2000 to 2400) shown in Fig.2 are plotted from full throttle to full stall conditions. The efficiency calculations were based on motor input power, calibrated motor losses, and constant belt-drive efficiency of 90%, the accuracy being ±3% .

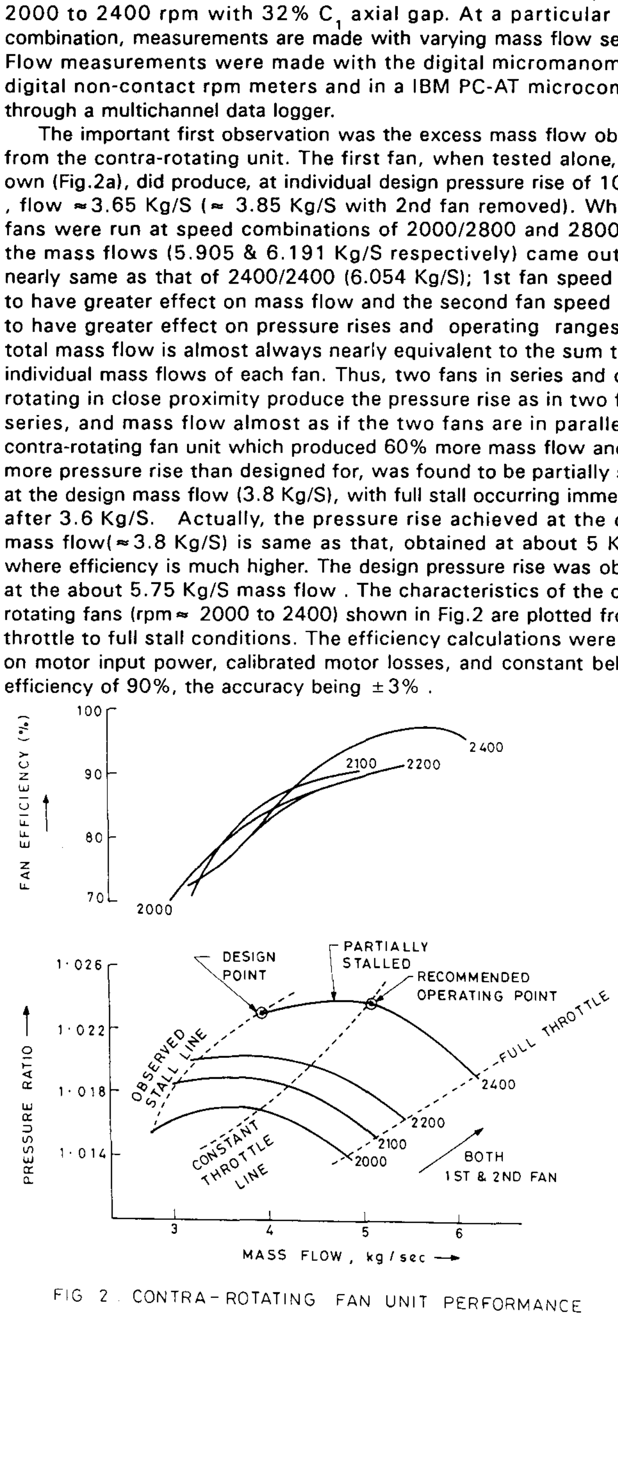


FIG 2 . CONTRA-ROTATING FAN UNIT PERFORMANCE

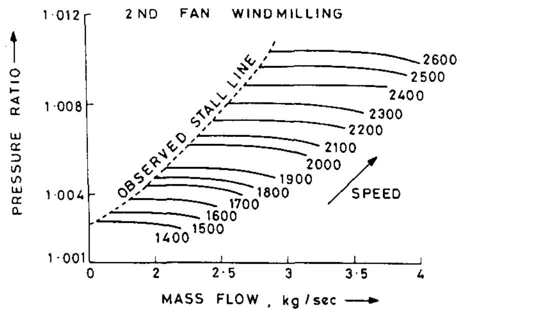
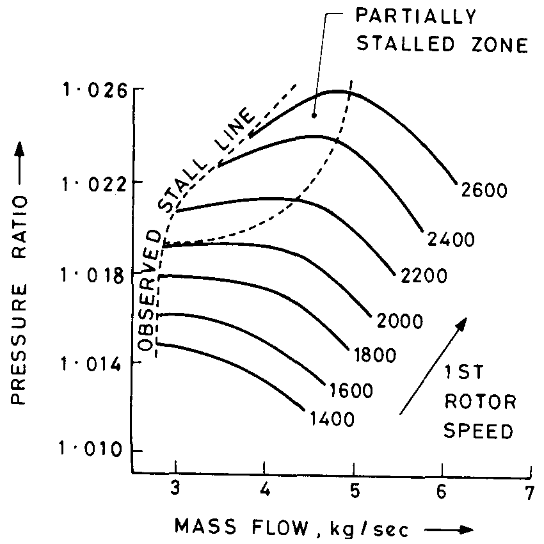


FIG 2a. FIRST FAN PERFORMANCE CHARACTERISTICS



SECOND ROTOR SPEED : 2400 R.P.M.

FIG.3

Stalling Behavior of the Contra-Rotating Fan Unit

The test results showed the effects of the contra-rotation of the 2nd fan on the overall performance of the unit [Fig.2,3,4]. It was observed (Rao, 1988, Basu 1989) that when the 2nd fan runs at higher speeds than the first fan, the maximum pressure operating zones are extended to lower flow rates. That means onset of even partial stall is delayed or avoided by running the first fan at lower and lower speeds, while the second fan speed is held at a higher rpm [Fig.3]. Similar nearly stall free flow has been reported (Sharma et al, 1985), when the 2nd fan speed was twice the 1st fan speed. On the other hand, when the second fan speed is lower than the first fan speed the pressure rise curves show a narrower operating range, i.e. early stall [Fig.4]. It was also observed that measured efficiency values continue to rise at higher mass flows when the second fan speed is higher. When the first fan speed is constant, with the 2nd fan speed varied the same trend is observed, [Fig.4] i.e. when the second fan speed is higher large high pressure operating zone is extendible to lower mass flows without stall. The partial stall zone in both the graphs vanish as the second fan attains a higher speed. Wherever the pressure falls sharply, indicating stall, with decrease in mass flow, the graphs have been discontinued. When the pressure fall is gradual, due to partial stall, graphs have been plotted to show the progressive deterioration. The behavior exhibited in Fig.s 3 & 4 may be somewhat explained from a typical 2-D velocity diagram [Fig.5].

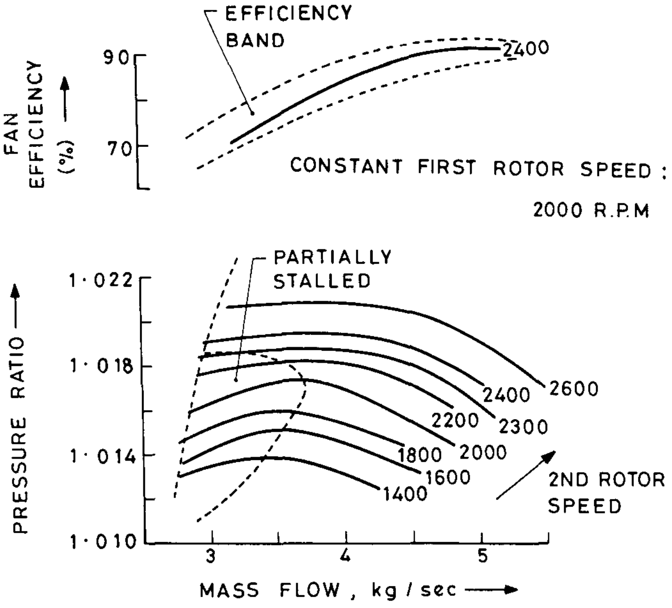


FIG. 4

When the 2nd fan speed in contra-rotation is increased, keeping the 1st fan constant, the flow would be incident on the 2nd fan at high positive incidence angle [Fig.5a]. The effect would be similar if the mass flow is throttled to lower values. As both the blades are made with rounded leading edges and with additional 3% droop to the circular arc camber, this high positive angles do not lead to separation or stall. On the 1st fan the stall tendency (increasing deviation) [Fig.5b] is suppressed by the contra-rotating 2nd fan. The explanation may be as follows: 2nd fan blade suction surface moves in opposite direction to the 1st fan suction surface, and thus the suction effect of the 2nd fan pulls the 1st fan wake along with it in a direction opposite to that of 1st fan motion and thus tends to suppress separation and lower the deviation angle.Thus at when 2nd fan speeds are higher, even with flow throttling, unit total pressure rise characteristics remained flat and stall is delayed to far lower mass flows [Fig.4].

This characteristic demonstrates that the contra-rotating fan does not follow a parabolic stall or surge line, when the speed ratio varies. At higher second fan speeds [Fig.3,4] the stalling tends to occur at lower mass flows. Thus, there would be two parts of the stall lines: one; when the first fan speed is higher; second, when the 2nd fan speed is raised above the 1st fan speed.

The behavior of overall large stall margin, however, when studied with individual fan rotor performance characteristics showed that the 2nd fan stalls earlier and has a considerably lower mass flow operating range than the 1st fan. For example [Fig.6] when the first fan running at 2400 rpm first shows flat characteristics at high mass flows and increasing pressure rise, & as the mass flow is throttled, the 2nd fan, also running at 2400 rpm, partially stalls when the 1st fan static pressure rise is in ascending mode. At lower rpm the operating ranges of the 2nd fan characteristics shift to the lower mass flows, shows steep ascending modes, steeper at lower speeds, before stalling. 2nd fan wake flow study is presented below.

Fig.7 shows the difference between the predicted (using Blade Element Theory with Carter's deviation rule) performance at design speeds and the measured ones at 2400/2400 rpm. While matching at the design point is close, the wide differences at off-design mass flows needed further studies. This was done, first, by studying the effect of axial gap between the two fans, and then, by studying the deviation angles and flow qualities at 2nd fan exit at 50% C axial gap. The first fan wake was left out of the study as, (i) it was observed that its performance was already better than expected and any further improvement in the two fan unit would be connected to the 2nd fan performance improvement and (ii) first fan wake was not easily accessible for study of deviation angle variation.

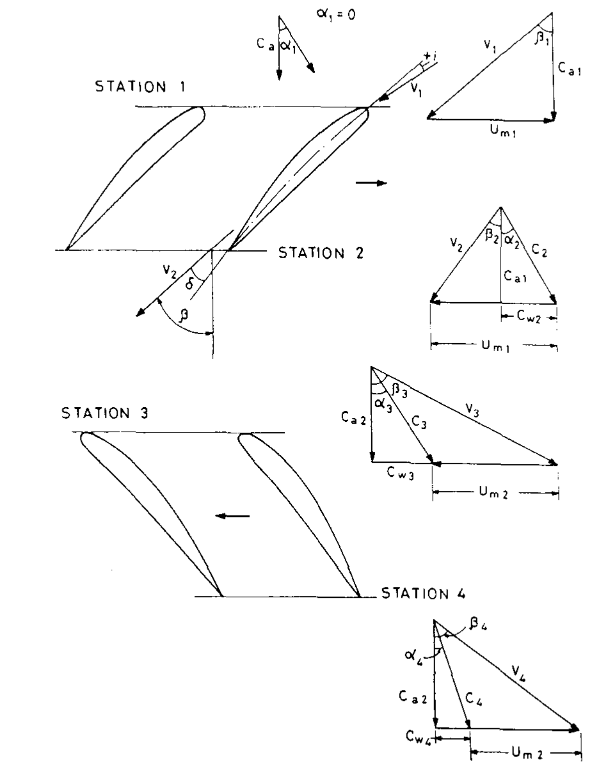


FIG.5 TYPICAL VELOCITY TRIANGLES AT MEAN DIAMETER OF THE CONTRA-ROTATING FANS

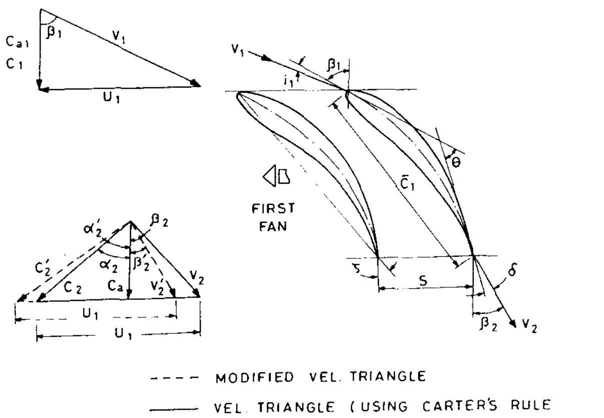


FIG. 5a. MODIFICATION VELOCITY VECTORS FROM FIRST FAN EXIT OUT TO CONTRA ROTATING 2ND FAN SUCTION

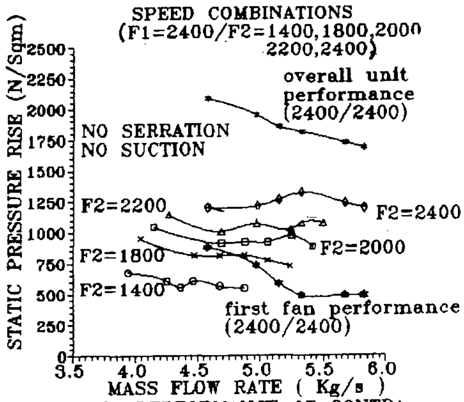


FIG. 6: PERFORMANCE OF CONTRA-ROTATING FANS

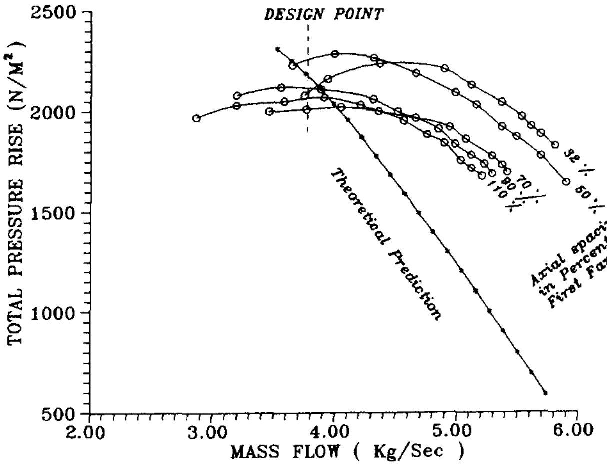


FIG : 7 : Comparison of Test Results with Prediction

Effect of Axial Gap on Performance & Noise

The axial gap between the two fans was varied from 32% to 110% of the length of chord of the first fan blading. The noise measurement was made at 2 meter axially upstream of the inlet bellmouth, with a RION digital noise meter, (with 1/3 octave filter and frequency analyzer). The noise was measured in the F-scale. Further noise readings are not presented here.

The axial gap change first confirmed that the standard inverse-design computations for predicting the off-design performance characteristics were inadequate for predicting with acceptable accuracy. With increase in axial gap at design speed combination the performance curves showed reduced pressure ratio. However, they still did not come down to the predicted characteristics, in which axial gap had not been considered as a parameter [Fig.7]. Predictions were based on Carter's deviation angle formulation, when incidence varies with operating point (blade stagger angle being fixed). However, the test results have later indicated that deviations have probably been over-predicted, and better matching with higher axial gaps indicate that deviations, especially of 1st fan probably increase with axial gap towards the predicted values. Fig.8 shows the overall performance parameters with variation of axial gap between the two fans. This variation shows better performance when axial gap was at 50% of c_1, than when at 32% of c_1. Thereafter, the performance continuously decreased with further increase of axial gap. Thus, on one hand performance needs to be sacrificed for noise reduction, on the other hand, axial gap optimization is needed for maximization of performance. Thus, increasing the gap between the two fans permit a significant reduction in the source noise of about 8 to 10 dB for an aerodynamic penalty of ≈10% in fan efficiency. This indicates the possibility of variable-axial-gap contra-rotating prop-fans, for optimizing between noise and performance in flight. All studies reported hereafter were conducted at axial gap of 50% c_1.

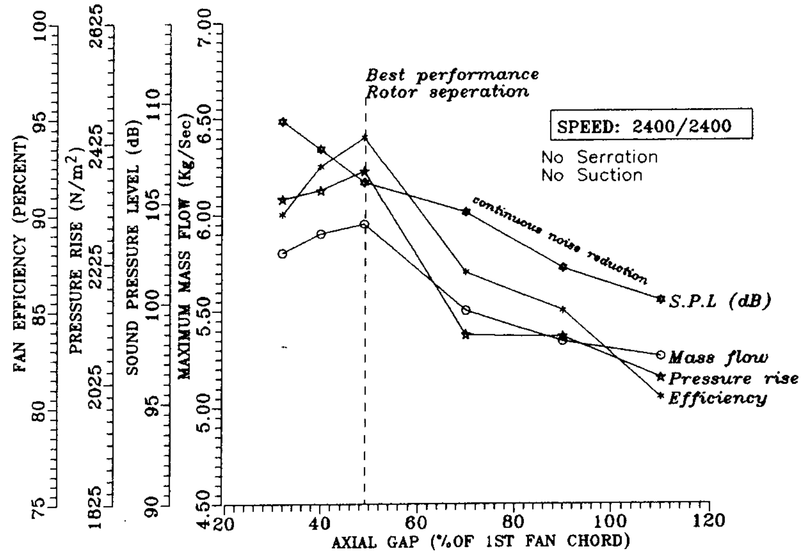


FIGURE : 8 : EFFECT OF AXIAL GAP BETWEEN TWO ROTORS

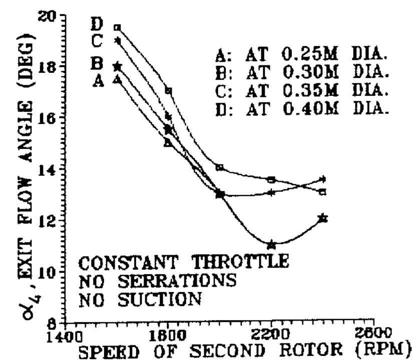


FIG.9a: VARIATION OF EXIT FLOW ANGLE AT 1xAXIAL GAP

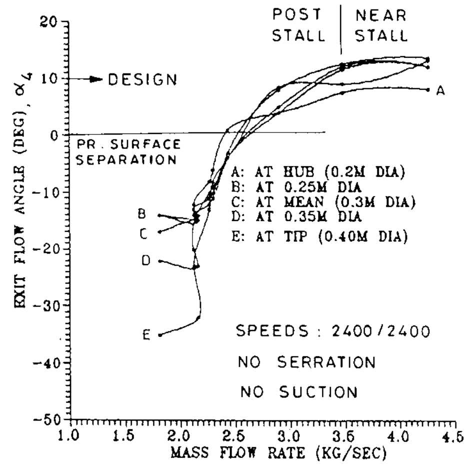


FIG. 9b. VARIATION OF EXIT FLOW ANGLES AT DESIGN SPEED WITH MASS FLOW.

2nd Fan Exit Flow Field Studies

The 2nd fan exit flow angle measurements indicate that absolute flow angle matches very well with the design value i.e. 10° (Fig.9) at speeds (2400/2400). However, at off-design near-stall and post-stall operating points (when 2nd fan speed is varied) the exit flow angles increase away from the design value (Fig.9) . Interesting results were obtained when exit flow angle values were translated to deviation angles (Fig.10A).

The deviation angles at design point conform closely to design prediction. With decrease of mass flow deviation angles becomes negative . When the speed of the 2nd fan is reduced significantly the deviation goes over to negative values. Considering the contra rotating velocity vectors at the 2nd fan inlet (Fig.5) it is possible to explain this: with reduction of the 2nd fan speed the entry relative velocity, V_3, over most of the flow field, shift over to high negative incidence angles; this, then gave rise to *separation on the pressure side of the 2nd fan blades* (Fig.10b). The problem must have been aggravated by the modification of C4 blade profiles with an extra nose droop, intended to accommodate high positive incidence angle. The discrepancy between design and measured values are insignificant at the 2nd fan wake over the entire blade span. The droop is a low speed fan feature, and may be absent in high speed fans, and from this study should not be used on 2nd fan. The blade stagger angles may also be assigned in such a way during design, as to avoid this problem of pressure surface separation on the 2nd fan during operation.

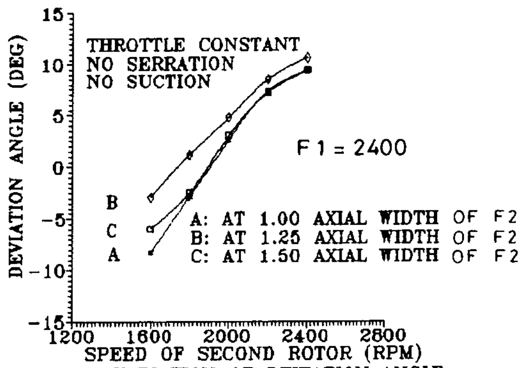


FIG 10 : VARIATION OF DEVIATION ANGLE AT MEAN RADIUS OF F2

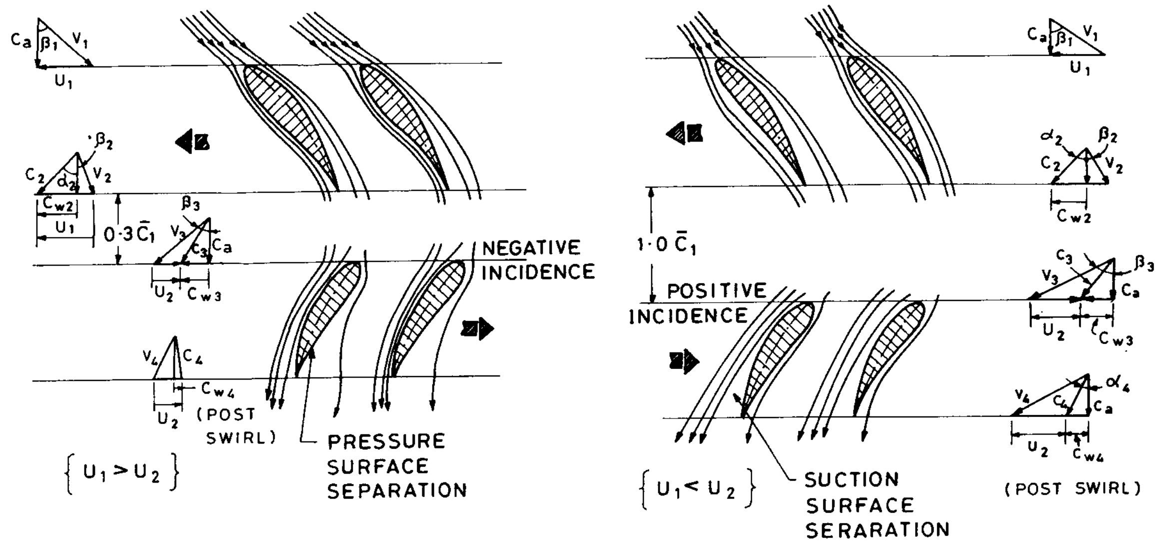


FIG.10b. EFFECT OF SPEED AND AXIAL GAP ON DEVIATION

<u>Near-Casing Flow Field Study at 2nd Fan Wake</u>

Studies done indicate that the casing boundary layer thickness increases from 4/5 mm upstream [Fig.11a](the pressure profile in the upstream flow showed very little variation as entry axial flow velocity varied from 25 M/S to 53 M/S, low speed stall to high speed full throttle) to 25/30 mm downstream of the fan unit [Fig.11b]. With increase of 2nd fan speed to 2400 rpm the pressure field showed almost parabolic radial distribution, signifying thick boundary layer and mixing zone (Murthy, 1991).

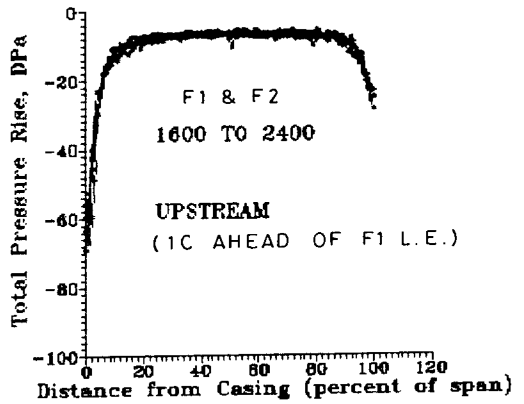


FIG 11a : ENTRY FLOW FIELD

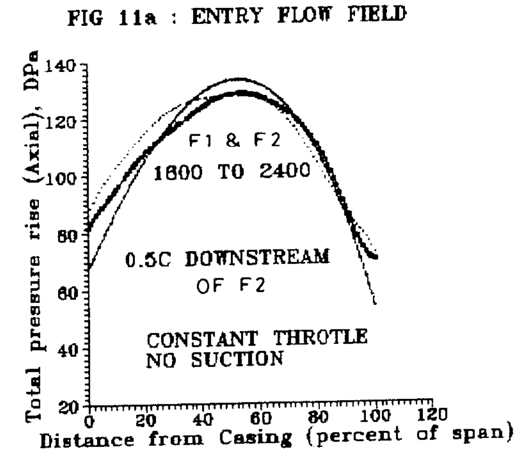


FIG 11b : EXIT FLOW FIELD

The wake flow field was studied with a 100 mm long, 0.5 mm o.d., thin probe with measurements made at 0.5 chord behind the 2nd fan. This is by no means a proper boundary layer study. However, the casing boundary layer was so thick, almost like a fully developed flow, that in absence of hot-wire anemometer, ordinary probe traverse was considered adequate for a qualitative study into the effect of increased axial pressure gradient with the fan speed. Improvement, if any, by casing boundary layer removal was also studied the same way. Near transition zones (from Boundary layer to free stream flow) a hot-wire based velocimeter was used which, however, can't be used within 4 mm of the casing wall.

<u>Performance Improvement Studies</u>

Serration on 2nd fan blade : Attempts to improve the 2nd fan performance was initiated with serrations employed on the suction surface. Fig.12 shows the serrations employed on the surface. The serrations were given zig-zagshapes to stagger the tripping of boundary layer chordwise from root to tip. This was expected to result in staggered release of trailing edge vortices from root to tip. The thickness of serrations was determined after a number of trials. 10% of maximum blade thickness serration placed at 45% chord position produced worse performance than unserrated blades; the tripping was either too early or too violent or both. 2.5% of maximum thickness serration placed at 55% chordwise position didn't have any effect on the performance; tripping was either too late or serration was well buried within the boundary layer. Next, 5% of maximum thickness was employed at 50% chordwise position. Significant improvement was observed with this serration. All the results presented here are of 5% thickness serration 2nd fan blades placed at 50% of chordwise position.

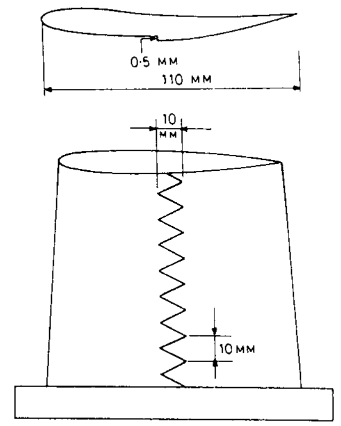


FIG. 12 DESIGN OF SERRATIONS

Speed RPM	Total Pressure Rise N/Sq.M.		Percentage of Improvement
	With serrations	Without Serrations	
2400/1800	2063.4	1750.00	17.9
2400/2000	2243.7	1850.00	21.2
2400/2200	2457.2	1950.00	26.0
2400/2400	2567.4	2100.00	22.2
2400/2600	2864.7	2100.00	36.4

TABLE 3: TOTAL PRESSURE RISE OF THE UNIT : SERRATED & UNSERRATED FAN 2

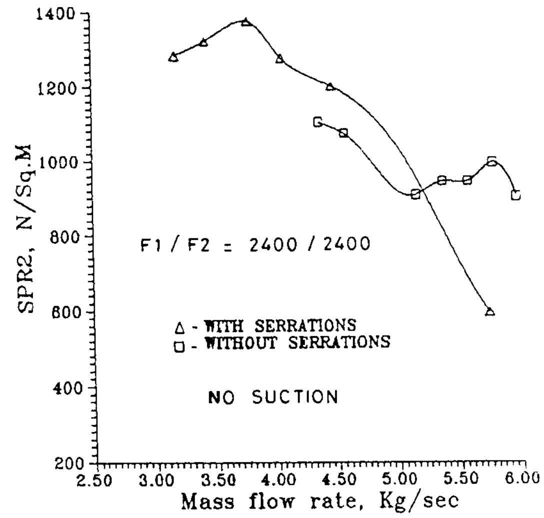


FIG. 13 . STATIC PRESSURE CHARACTERISTIC COMPARISON OF SECOND FAN AT 2400/2000 RPM

It may be seen [Fig.13] that performance shown in terms of static pressure rises showed significant improvement. The stall characteristics of the 2nd fan shifted more to the left accommodating operation at lower mass flows, especially at design speeds. The overall unit performance also showed significant improvement in both pressure rise and shift of stall mass flow to lower values [Fig.14]. A Summary of improvement achieved is shown in Table.2. Serration was not tried on 1st fan at all.

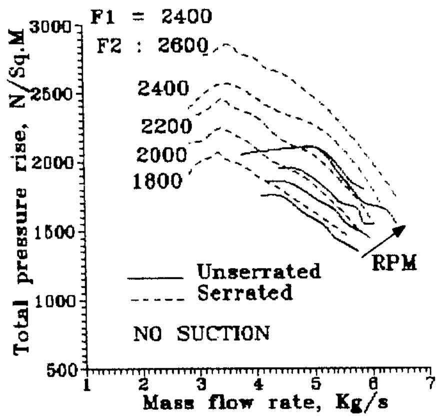


FIG 14: IMPROVEMENT WITH SERRATIONS

Suction of Casing Boundary Layer

Suction of casing boundary layer (as per Lee et al,1990) was employed at the axial stations between 16 - 60% of axial gap, (chosen from the Fig.15a axial static pressure distribution) downstream of 1st fan (Murthy, 1991). With the suction blowers kept open at full throttle and power, RPM of both the fans were varied to observe the effect on 2nd fan wake. The flow profile studied with a 0.5 mm dia. probe traversed at the 2nd fan rear show an improved pressure profile with the entire total pressure lines shifting upwards (Fig.15b). Overall suction flow rate was measured to be varying from 1.7% to 4% of main flow rate. The improvement in flow profile over the entire blade span, and not only at near-tip section (Fig.16) probably indicates arrest of tip induced secondary flow development over the 2nd fan. At low flow rates the second fan wake was studied with a hot wire based velocimeter at a station 25%-of-2nd-fan-chord rear of 2nd fan trailing edge; the effect of suction (≈4%) near the casing is shown in the Fig.16. The present study has not gone beyond 4% of suction for casing boundary control.

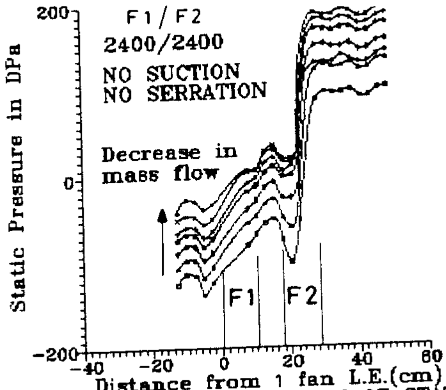


FIG 15a: AXIAL DISTRIBUTION OF STATIC PRESSURE ALONG THE UNIT

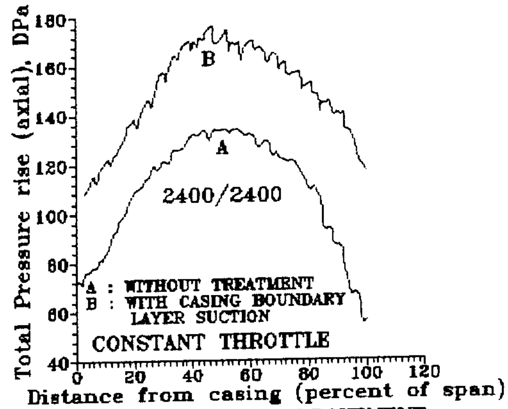


FIG.15b: FLOW IMPROVEMENT OF II FAN WAKE

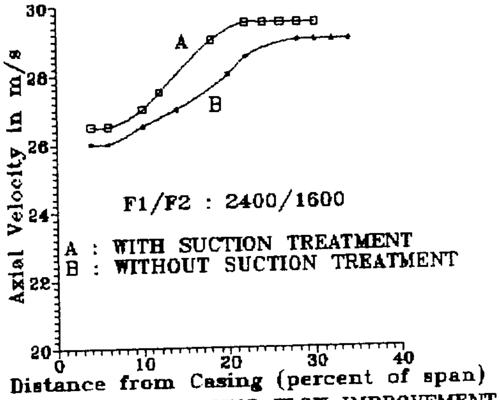


FIG 16: NEAR CASING FLOW IMPROVEMENT

CONCLUSIONS

The blade element inverse-design performance prediction method adapted from conventional co-rotating axial flow compressors could not be utilized to predict : a) Increased mass flow processing capacity, b) Large stall margin for the 1st fan, c) Early stall of the 2nd fan, d) effect of axial gap on flow rate and pressure rise, d) discrepancy of deviation and exit flow angles with design predictions. It is felt after this study that contra-rotation fans/compressors require : i) axial gap optimization, ii) separate deviation and loss correlations, specifically for 1st fan, for accurate design and prediction, and iii) An iterative method of arriving at design value of mass flow processing capacity, which is observed to be an inherent feature of the contra-rotation. The requirements and flow conditions of unducted and ducted units are separate, as the unducted fans are expected to allow flow entry through the uncovered tips of the blades [Celestina et al, 1986], and the present study is of greater relevance to ducted contra-rotating fan or compressor units only.

MEASUREMENT ACCURACIES

PARAMETER	INSTRUMENT	ACCURACIES
1) Total & Static Pressures	Digital Micromanometer , Standard & Shielded probes	0.1 mm H_2O
2) Noise	RION Noisemeter with 1/3 octave filter	0.1 dB
3) Boundary layer	0.5mm Boundary layer probe	1 mm
4) Flow angle	4-hole spherical yawmeter	1 degree
5) Flow velocity	Pitot-static probe with Digital Micromanometer, &	0.1 M/S
	Hot-wire Anemometry based TSI Velocimeter	0.1 M/S

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