RESUMÉ OF PAPERS

41[ST] TECHNOLOGICAL CONFERENCE

February 12 & 13, 2000

A Joint Publication of
ATIRA, SITRA, NITRA and BTRA

Bombay Textile Research Association
L.B.S. Marg, Ghatkopar (West)
Mumbai 400 086.

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DESIGN ASPECTS CONTRIBUTING TO ENERGY EFFICIENCY OF AXIAL FLOW FANS

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INTRODUCTION

Conversion of fibres into fabric requires specific level of R.H. depending on the process, for which humidification plants are installed. These plants mostly use axial flow fans to pump the required quantity of air. It is estimated that more than 40,000 axial flow fans are in operation in the Indian textile industry. By and large, these fans require about 10 kW power/fan, thus total requirement by these fans works out to be around 400 MW power needing Rs.2000 crore investment. Considering the scopes of improvement and energy saving potential from each of the specific engineering aspects of fan system, a joint R&D project between M/s. Parag Fans & Cooling Systems Ltd. and ATIRA is persued.

Amongst the several specific engineering aspects, two aspects namely (i) nose and tail fairings and (ii) tip clearance and ovality were identified as a first phase of R&D work. The set up on these were designed, fabricated and evaluated on AMCA Standard Test Rig. The fairings have yielded energy saving of about 10%. The impact of tip clearance and ovality correction on actual energy saving depends on how bad the original fans are. Correction of tip clearance/ovality in a few shopfloor cases confirms theoretical savings. Also collection of actual data on existing tip clearance of 374 fans from the textile mills indicates about 6 to 8% saving potential. This on a national level works out to be around 50 MW saving, i.e. a saving of Rs. 250 crores on capital expenditure on power plants and about Rs.138 crores on annual saving by the mills. The scopes of improvement through other areas are also looked into.

COMPONENTS OF AN AXIAL FLOW FAN SYSTEM

As shown in Fig.1, the axial flow fan system essentially consists of :

1. Fan Blade
2. Hub
3. Prerotators
4. Straighteners
5. Ring
6. Nose Fairing
7. Tail Fairing
8. Inlet Cone
9. Outlet Cone

Each component is having its own specific effect on capacity, pressure and energy consumption. Also, the surface finish of the materials and overall weight of the rotating mass has some effect on energy efficiency but of marginal contribution.

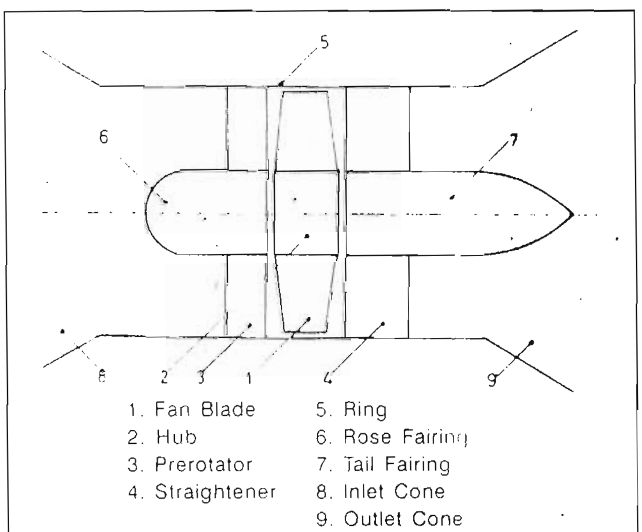


Fig. 1 : Axial Flow Fan System

ENERGY EFFICIENCY OF A FAN IN A SYSTEM

The total energy efficiency of the utility system can be defined as :

$$EE_s = \frac{\text{Energy Supplied to the Air}}{\text{Energy Supplied to the Motor}}$$

$$= \frac{\text{Air Energy Delivered}}{\text{Motor Energy Consumed}} \quad \ldots\ldots\ldots\ldots (1)$$

The motor efficiency and the transmission efficiency for an indirect drive system contribute to EE_s. In the present study, the load on the electric motor was 50% and above where motor efficiency is quite high and does not change much. Also, in the test rig, fans are mounted directly on the motor shaft and hence there is no transmission loss.

Energy of the air is the product of pressure and mass flow. The pressure depends on the system characteristics. The axial flow fan system is installed in a system comprising dampers, filters, heating or cooling coils, air washers, eliminators, etc., each offering a specific resistance to the air flow through it. The resistance offered by these components can be assessed by the formula.

$$\backslash P = \frac{4f\, Lv^2}{2\, g} \quad \ldots\ldots\ldots\ldots (2)$$

and

$$v = \frac{Q}{A} \quad \ldots\ldots\ldots\ldots (3)$$

where,

- f is the coefficient of friction between the air and the surface over which it flows - dimensionless
- L is the length of travel, ft
- v is the velocity of air, ft/sec
- A is the area of cross section of the air path, ft^2
- Q is the volumetric flow of air, ft^3/sec
- g is the gravitational constant ft/sec^2

$$\therefore EE_s = \frac{K \cdot Q^2}{P} \quad \ldots\ldots\ldots\ldots (4)$$

where K is a constant

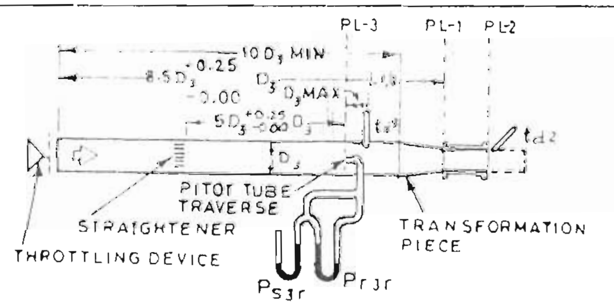
and P is energy consumed by the motor, kW

The energy efficiency of a fan system (Fig.1) in an air conditioning system can be assessed by measuring air flow and energy consumed by the motor. A test rig as per AMCA standard (Fig.2) had been designed and installed on site. The test rig comprises :

1. A long round duct
2. Throttling device to control flow/pressure
3. Straightener to streamline the flow
4. Pitot tube traverse to measure dynamic or velocity pressure
5. Arrangement to measure static pressure
6. Transformation piece to connect fan of required size.

The present study reported here confines to the effect of following parameters on the fan efficiency :

- Tip Clearance and Ovality
- Nose and Tail Fairings



FLOW AND PRESSURE FORMULAE

$$P_{v3} = \left(\frac{\Sigma\sqrt{P_{v3r}}}{n}\right)^2 \qquad P_v = P_{v3}\left(\frac{A_3}{A_2}\right)^2\left(\frac{\rho_3}{\rho_2}\right)$$

$$V_3 = 1096\sqrt{\frac{P_{v3}}{\rho_3}} \qquad P_{t1} = P_{s3} + P_{v3} - f\left(\frac{L_{1,3}}{D_{h3}}\right)P_{v3}$$

$$Q_3 = V_3 A_3 \qquad P_{t2} = P_v$$

$$Q = Q_3\left(\frac{P_3}{\rho}\right) \qquad P_t = P_{t2} - P_{t1}$$

$$P_{s3} = \frac{\Sigma P_{s3r}}{n} \qquad P_s = P_t - P_v$$

Fig. 2 : Test Rig As Per Ansi/AMCA 210-85 Fig. 16

SET-UP AS PER AMCA TESTING METHOD

The Test Rig (Fig.2) used for evaluating the fan performance was designed as per ANSI (American National Standards Institute)/AMCA (Air Movement and Control Association) standard 210-85, Fig.16 by IIT, Mumbai. The manufacturing and installation were done by M/s. Parag Fans & Cooling Systems Ltd., finally inspected and approved by IIT, Mumbai.

The parameters monitored are :

(i) Static pressure, mm of water column

(ii) Total pressure, mm of water column

The velocity pressure and total flow are derived from these two.

(iii) Power consumption : this consists of all electrical parameters like voltage, current, power and power factor

(iv) Speed through VFD

From the various data collected, as per the AMCA method and subsequent calculation therefrom', following performance curves are generated.

(i) Flow Vs. (a) Static Pressure

& (b) Total Pressure

(ii) Flow Vs. Fan BHP

(iii) Flow Vs Percentage Efficiency

TIP CLEARANCE AND EFFICIENCY LOSSES - THEORETICAL ANALYSIS

The tip clearance value for an efficient fan operation is recommended as one percent of the blade length, beyond which the rotor efficiency falls. This is estimated as :

$$\text{additional loss in percentage rotor efficiency}^{(2)} = 2 \times \frac{\text{additional clearance}}{\text{blade height}}$$

The losses occur because of : (i) reduction in net air flow, and (ii) reduction in pressure rise. The losses in net pressure and flow through the fan system occur because there is positive pressure on the delivery side and negative pressure on the suction side and this causes recirculation of air through the annular space between the ring and the rotor referred as tip clearance, thus reducing net air power as tip clearance increases.

The effect of tip clearance on rotor efficiency is shown in Figure - 3.

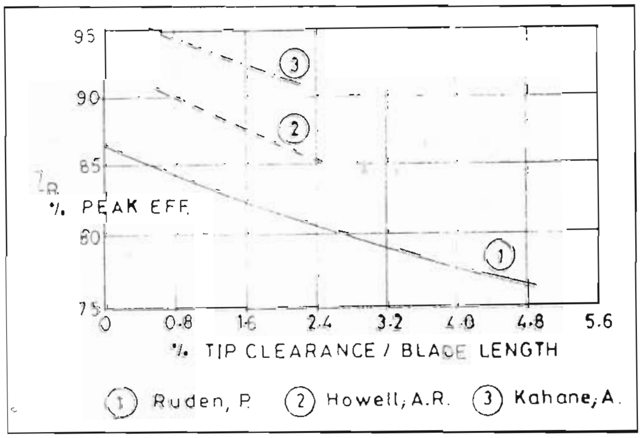


Fig. 3 : Effect of Tip Clearance on Rotor Efficiency

TIP CLEARANCE AND OVALITY IN THE INDUSTRY AND POTENTIAL SAVINGS

The data related to tip clearance and ovality as

Table-1
Frequency Distribution of Tip Clearance and Ovality

Figures show number of fans in each

Grade	Range (mm)	Min. T.C. (mm)	Max. T.C. (mm)	Ave. T.C. (mm)	Ovality (mm)
A	0 < A ≤ 3	52	1	4	163
B	3 < B ≤ 6	209	74	99	131
C	6 < C ≤ 9	83	87	163	52
D	9 < D ≤ 12	21	132	82	19
E	12 < E ≤ 15	6	49	17	8

Table-1 (Contd.)
Frequency Distribution of Tip Clearance and Ovality

Figures show number of fans in each

Grade	Range (mm)	Min. T.C. (mm)	Max. T.C. (mm)	Are. T.C. (mm)	Ovality (mm)
F	15 < F ≤ 18	2	20	5	1
G	18 < G ≤ 21	1	8	4	
H	21 < H ≤ 24	-	2	-	-
I	24 < I ≤ 27	-	1	-	-
TOTAL		374	374	374	374

prevailing in the textile industry were gathered through a specially designed questionnaire which was sent to 55 mills all over India, as well as through actual measurements in a few local mills. These data as received from 8 mills were analysed and are given in Table-1.

Power Saving Potential

The data of 374 fans were further analysed to arrive at potential power saving if all these fans are corrected for removing ovality and maintaining tip clearance to a maximum of 3 mm. Considering the techno-economics it is assumed that only 80% of this saving is aimed at i.e., only those fans are attended which have higher tip clearance and ovality values. This has reduced down the number of fans from 374 to 256 i.e., to about 2/3rd fans. It is worth noting from Table-2 that the saving potential per fan, depending on original tip clearance and ovality, ranges from as low as 0.24 kW/fan to as high as 0.78 kW/fan when it is aimed for 80% saving. For a typical case of Mill No. 2 having 28 fans, this means a saving of 0.78 x 28 = 21.84 say 22 kW after correcting 21 fans for tip clearance and ovality. This will mean an annual energy saving of 1.45 lakh units on the basis of 22 hr / day and 300 days / year working.

Table-2
Power Saving Potential by Correcting Tip Clearance and Ovality

(1) Mill No.	(2) Total Fans	(3) Pareto Fans 80% or 3 mm	(4) Total Saving Potential (kW)	(5) Pareto Saving Potential (kW)	(6) kW Saving Potential Per Fan All Fans	(7) kW Saving Potential Per Fan Pareto Fan	(8) % Fans As Per Pareto	(9) Gain Due to Pareto Fans i.e., 7 ÷ 6
1.	102	74	48.60	38.88	0.48	0.53	72.55	1.10
2.	28	21	20.52	16.42	0.73	0.78	75.00	1.07
3.	42	25	13.59	10.87	0.26	0.44	48.08	1.67
4.	6	5	1.62	1.41	0.27	0.28	83.33	1.04
5.	51	38	21.79	17.43	0.43	0.46	74.51	1.07
6.	22	16	4.70	3.77	0.22	0.24	72.73	1.10
7.	31	24	7.83	6.26	0.25	0.26	77.42	1.03
8.	36	20	11.45	9.16	0.32	0.46	55.55	1.44
9.	38	23	12.0	9.60	0.32	0.42	61.00	1.32
10.	18	10	6.15	4.92	0.342	0.492	55.55	1.44

REDUCING TIP CLEARANCE - AN EXPERIMENT - A CASE STUDY

The specifications of the test fan were :

Fan Dia (D)	:	1200 mm nominal
Hub Dia (d)	:	405 mm
Ratio, d/D	:	33.75%
Fan Speed	:	1440 rpm
No. of Blades	:	Six
Blade Material	:	FRP
Blade Profile	:	Aerodynamic

Measurements were carried out for static pressure, total pressure, kW consumed and fan speed. These were analysed to plot fan performance curves and calculate fan efficiency.

The measurements were carried out for following five conditions :

(1) Original - without incorporating any correction, and having ave-tip clearance of 6 mm.

(2) After correcting for ovality and reducing the tip clearance to an average of 2 mm.

(3) Fan diameter cut by 5 mm to increase the tip clearance to 4.5 mm.

(4) Fan diameter cut further by 5 mm to increase the tip clearance to 7 mm.

(5) Fan diameter cut still further by 10 mm to increase the tip clearance to 12 mm.

The performance data of Total Efficiency are shown in Figure - 4.

The data of 374 fans collected from various mills indicated an overall average tip clearance of about 8 mm, and with an assumed blade length of 400 mm as overall average of 374 fans, the ratio of Tip Clearance / Blade Length is 2% which corresponds to about 82% rotor efficiency. Similarly, there were fans having ave. tip clearance as high as 19.5 mm with a ratio of Tip Clearance / Blade Length as about 5%. The rotor efficiency corresponding to this value is about 76% from the literature which indicates a drop of 8% since the maximum efficiency that can be achieved is 84% from the graph, with minimum tip clearance that is recommended.

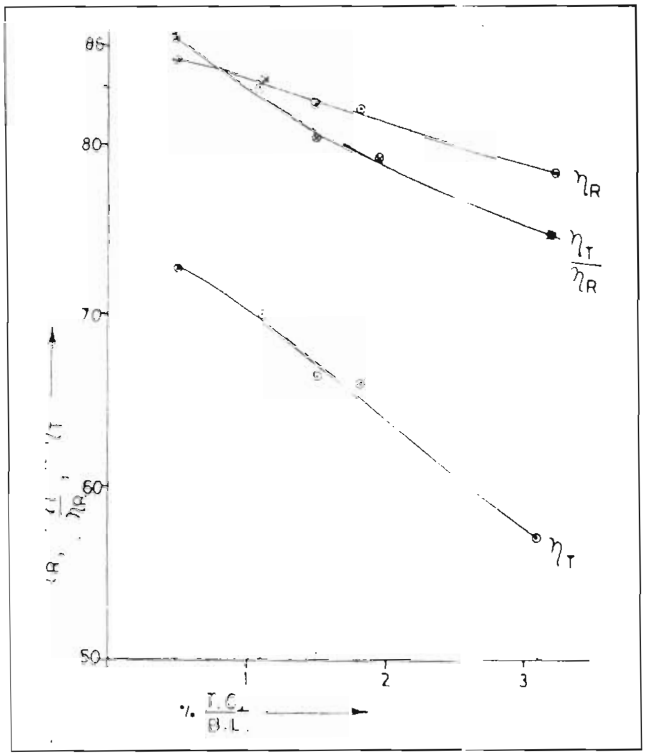


Fig. 4 : Total Efficency at Varying Tip Clearance

ATIRA and M/s. Parag Fans are at present in the process of developing and evaluating various methods of reducing tip clearance and ovality. Considering that this job has to be done on site and fans are working at speeds as high as 1440 rpm, the most reliable, and cost effective methods as well as corresponding tools, jigs and fixtures, etc., are being developed.

NOSE AND TAIL FAIRINGS

The hub diameter in an axial flow fans is about 30 to 40 percent of the fan diameter i.e., about 9 to 16 percent of area which obstructs the preceding streamlined flow of air and is one of the reasons for reduction in energy efficiency of the fans. This happens on both suction and delivery sides of the rotor. To avoid these losses and thereby increase the fan efficiency, the air should be streamlined (devoid of turbulence) by proper guide known as NOSE FAIRING on suction side and TAIL FAIRING on delivery side, as shown in Figure - 1.

The design of the fairings for 1200 mm nominal dia fan having 6 aerodynamically designed FRP blades and rotating at 1440 rpm was made by

ATIRA, and fabricated and installed by M/s. Parag Fans in the AMCA TEST RIG. The effect of fairings was assessed from various data as per AMCA method which consisted of Fan BHP, Total Pressure, Velocity Pressure and Efficiency at one tip angle position. The results of efficiency levels under each of the 4 cases i.e., (i) without fairings, (ii) with only suction fairing, (iii) with only delivery fairing and (iv) with both the fairings, are as shown in Figure - 5.

Curve 1	No Fairing
Curve 2	Only Suction Fairing
Curve 3	Only Delivery Fairing
Curve 4	Both Side Fairings

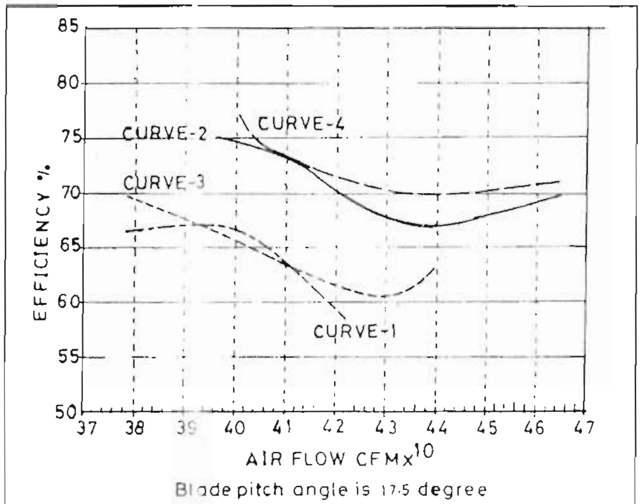


Fig. 5 : Effect of Fairings on Fan Efficiency

The following observations are made :

(i) Without any fairing, the maximum efficiency is about 67-68% at a flow of about 40,000 CFM.

(ii) Increase in efficiency by nose fairing is substantially larger than that of tail fairing.

(iii) The combined effect of both the fairings on improving efficiency is approximately cumulative total of two individual effects.

(iv) Nose fairing seems very economical, easy to implement and payback period will be hardly few months.

(v) Installation of tail fairing may be expensive and difficult also, with only marginal improvement in efficiency.

However, it will help by reducing resistance and swirl and hence minimise the separation losses. This needs further confirmation by experiments which is planned, particularly for fans used in textile humidification plants.

INLET AND OUTLET CONES

In the axial flow fans, the air first enters a total cross sectional area of the plant room, but as it approaches the fan blade i.e., rotor, it occupies only the blade portion of the rotor; thus there is no air in front of the area than ring created

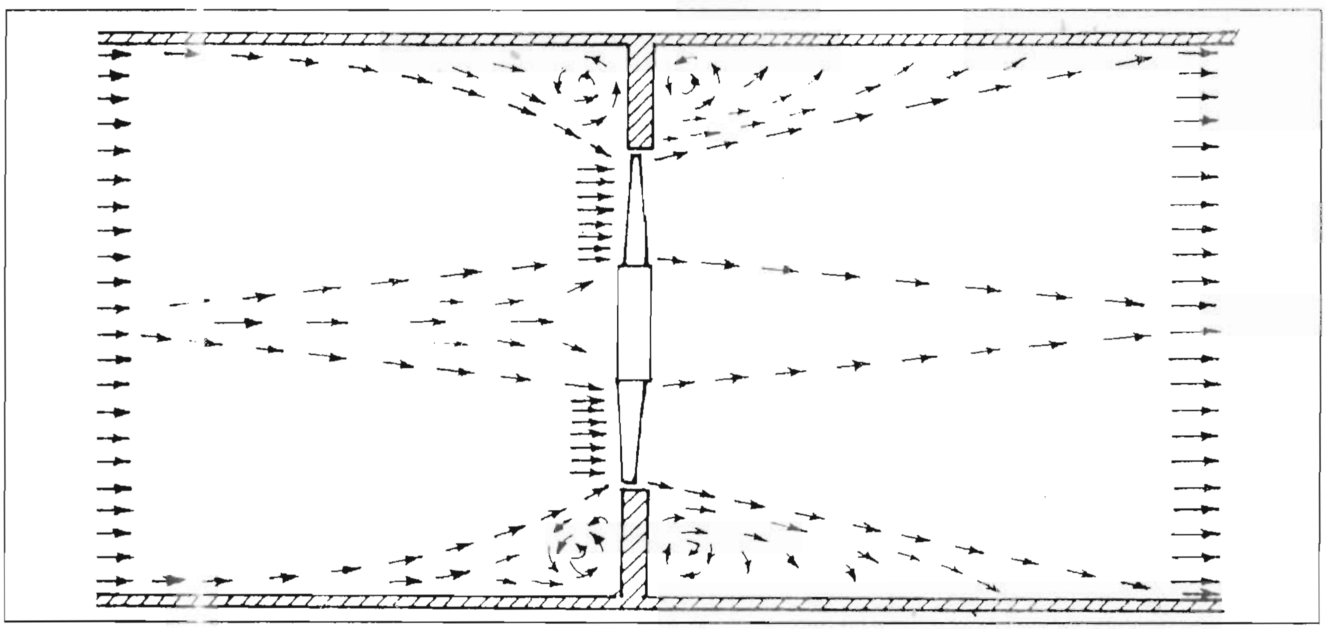


Fig. 6 : Vena Contracta at Fan Inlet

by the blades. This results in an acceleration of air flow near the rotor which increases with increased hub diameter. In absence of any guide at the inlet, this will result in a vena contracta as schematically shown in Figure - 6.

Attaching a bell shaped inlet guide to the fan ring will help to avoid this to occur, particularly if the fan rotor is located near the fan ring inlet. The use of inlet bell is reported to : (i) boost the flow rate by 10-15% as reported in the literature, (ii) increase the fan efficiency and (iii) reduce noise level.

Similarly, a suitably designed cone at the fan outlet just after fan ring will help in guiding the air by minimising eddying flow which otherwise leads to loss of conversion of velocity head to static head. This cone is also called static regain cone.

Various design alternatives of inlet and outlet cones are being explored to finally select one which gives best performance in each, based on the performance testing on AMCA Test Rig.

STATIONARY GUIDE VANES

These are used before as well as after the fan rotor called prerotators and straighteners respectively, to subside swirl momentum caused by change of angular momemtum in the air stream, and convert it to static pressure.

Further details of these, their merits in the context of textile industry's fans, expected gains in the efficiency, techno-economics, etc., are planned to be looked into in the next phase of the project.

OTHER MOUNTINGS

The fan assembly consists of all important components discussed herewith as above. It also consists of other components like mounting brackets, pedestal bearings, etc. All these obstruct the air flow and hence providing a smooth aerodynamic shapes to all is likely to improve the performance and thus efficiency although the impact may not be significant. The role and contribution of all these is planned to be studied and if necessary, suitable design modifications will be considered.

CONCLUSIONS

Of the total energy consumed by the humidification plants, 60 to 70% is by the fans only. Various aspects of the fan system such as ovality, tip clearance, fairings, cones, etc., if incorporated in the fan of the humidification plants, contribute substantially to the total energy efficiency of the fans. Two aspects namely: (i) tip clearance and ovality and (ii) Nose and Tail Fairings were studied in detail, optimum design evolved, fabricated and tested, using the AMCA/ASHRAE Standard Test Rig. They showed substantial energy savings in comparison with the cost of conversion.

In Indian textile industry, about 40,000 axial flow fans are in operation. Industry's data from a few modern mills indicate energy saving potential of about 50 MW if these two parameters investigated so far are implemented. The payback period of implementing these two parameters is likely to be a few months only.

ACKNOWLEDGEMENTS

The contents of the paper focus on the actual work done jointly by M/s. Parag Fans & Cooling Systems Ltd., Dewas, and ATIRA. Messers K.R. Mishra and A.S. Sachania of ATIRA helped in data collection in mills and also Shri A.I. Malwat in preparing various gauges to measure tip clearance and ovality. Messers P.B. Vilekar and S.V. Paradkar of M/s. Parag helped in carrying out the experiments on test rig and further analysis of the data. Also, many mills have helped us by sending the details as requested. The authors take a note of all of them with deep gratitude for their contribution. Authors are also thankful to the Director, ATIRA, and Chairman - Managing Director of M/s. Parag Fans & Cooling Systems Ltd., for permitting this paper to be presented in this conference.

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