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Effect of Tongue Geometry on the Performance of Cross Flow Fan

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***Abstract* - Experimental investigations were carried out to study the effect of casing geometry on the performance cross flow fans. Four casing geometries with different tongue configuration were tested. The experiment program includes performance studies of impellers with different tongue configurations at different speeds. Probe measurements at the inlet and exit of the impeller were also carried out. The efficiency is found to be higher for tongue II configuration. The performance found to sensitive to the tongue geometry. The vortex formation and its movement to impeller centre with decrease in flow coefficient could be clearly observed from flow visualisation photographs. The circumferential variation of span wise mass averaged total pressure indicates that the value of total pressure is maximum around at θ=2400 and reduces towards the inlet.**

Keywords – Cross flow fan, Flow visualization, Performance, Tongue geometry.

INTRODUCTION

Cross Flow Fan (CFF) is an axial or centrifugal turbo machine that works differently than an axial machine. The flow runs the length of the impeller and then exits via a different sector. Without causing any aerodynamic difficulties, the volume flow of CFF may be increased simply by expanding the impeller. CFF is appropriate where space is limited since the inlet has a rectangular shape. Geometrical characteristics, on the other hand, have a significant impact on CFF's performance and stability. The diameter of the impeller, its breadth, blade angles, casing form, tongue shapes, and clearance are all geometrical characteristics that impact the CFF.

The flow structure and vortex movement inside die impeller was studied using experimental as well as flow visualization techniques. Noise level of CFF is higher than that of centrifugal or axial flow fans and is influenced by the geometry and tongue clearance [1-2]. The performance of CFF by varying various geometrical parameters. Now a days CFFs are finding various and industrial applications like, (l) Central air conditioners (2) Air curtains, (3) Computer CPU coolers, (4) Hair dryers, (5) Vacuum cleaners, (6) Heaters and (7) Sprayers [3-5].

Even now the design of a CFF is dependent on experience, as its internal flow, as complicated as axial or centrifugal fans, is not fully understood. The CFF's loss models were not created in full. Therefore, due to the absence of experimental data and numerical forecasts, it is difficult to anticipate the operation of the machine for a particular shape [9-9]. A systemic investigation on fans of cross-flows with different major characteristics such as blade angles, the radius ratio and the lingual shape is not performed from the current literature review. The aim of this research is to carefully examine the influence of these factors on the effect of these parameters on the performance and flow field.

EXPERIMENTAL FACILITY

The impeller is driven by a D.C. motor, whose speed can be varied. Blades are of circular arc sheet and are fabricated from 2 mm thick mild steel sheet. Impeller is placed inside the casing. The casing is of 170 mm width and is rigidly fixed to stand. Casing exit is of rectangular cross section and is connected to the circular delivery duct through a transition duct. The delivery duct is of 300 mm in diameter. At the end of delivery duct, a throttle cone is fixed for controlling the volume flow rate. A pre-calibrated orifice plate is housed in the duct for measurement of volume flow rate through the CFF. A pre-calibrated three hole probe were used for measuring the flow angles, total and static pressure at inlet and exit of the rotor at a fixed radius of 265 mm. All the top-pings from the delivery duct and three hole probe are connected to a micrometer via a scanning box measurement of pressures. A stroboscope and a SLR Camera are used for flow visualization.

EXPERIMENTAL RESULTS

The experimental results of present investigations are presented in the following order.

1. Non-dimensional plots to verify similarity laws.
2. Performance characteristics (Non-Dimensional plots) of three impellers for four tongue geometries.
3. Flow visualization results for two impellers I and III with two tongue geometries III and IV at two speeds viz., N = 900 and 1100 rpm

1. Performance Characteristics

Performance characteristics of four tongue geometries at four speeds viz., N = 900, 1000, 1100 and 1200 rpm are determined by varying the volume flow rate in fifteen steps. At a constant speed, volume flow rate is changed by regulating the throttle valve. The throttle valve is always moved from high volume flow rate to low volume flow rate. Data obtained are processed through computer and the results are presented in the form of pressure coefficient, fluid power coefficient, input power coefficient, and efficiency.

2. Effect of RPM on Performance Characteristics

Literature survey reveals that the similarity laws are followed only for a particular region of flow coefficient. But in the present study, non-dimensional plots of pressure coefficient, fluid power coefficient, input power coefficient and efficiency did not vary with rpm, Figure 1. This figure shows that similarity laws are followed for the entire range of flow coefficient. Hence, graphs for a particular rpm (1100 rpm) only are presented here for the comparisons. Test were conducted at four all speeds basically to veri-fy the similarity laws. However, it must be remembered that cross flow fans employed in many applications are required to run at different speeds.

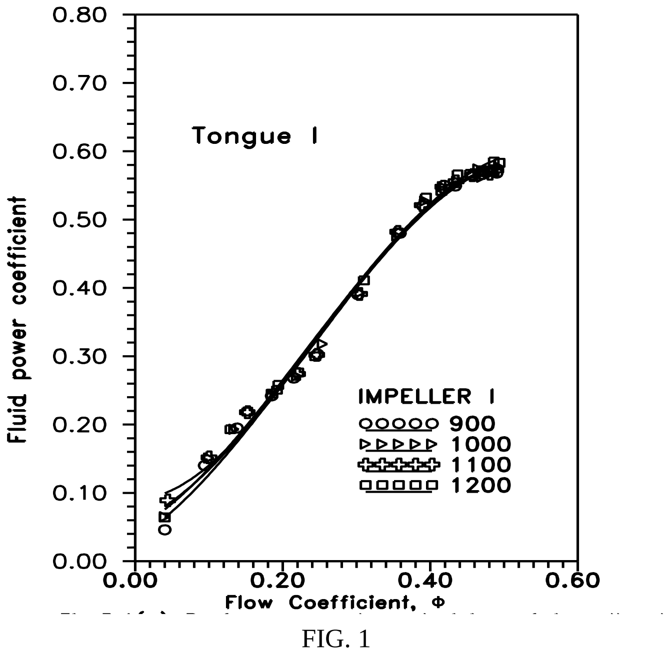

FIG. 1

FLUID POWER VS FLOW COEFFICIENT

An abstract of all the graphs is given in the Table I. These contains the following information.

1. Maximum and minimum pressure coefficient
2. Flow coefficient at maximum and minimum pressure coefficient
3. Maximum and minimum input power coefficient
4. Flow coefficient at maximum and minimum flow coefficient
5. Maximum efficiency
6. Flow coefficient at maximum efficiency
7. Maximum fluid power
8. Flow coefficient at maximum fluid power

TABLE 1

IMPORTANT PERFORMANCE PARAMETERS OF VARIOUS TONGUES

Non dimensional coefficient	900				1000			
	Tongue				Tongue			
	I	II	III	IV	I	II	III	IV
ψ (max)	2.20	1.70	2.20	1.50	2.20	1.70	2.20	1.50
ϕ at ψ (max)	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
ψ (min)	1.20	0.85	1.05	0.60	1.20	0.85	1.05	0.60
ϕ at ψ (min)	0.50	0.20	0.48	0.44	0.50	0.20	0.48	0.44
γ (max)	1.80	1.50	2.00	1.05	1.80	1.50	2.00	1.05
ϕ at γ (max)	0.04	0.04	0.04	0.44	0.04	0.04	0.04	0.44
γ (min)	1.10	0.60	1.00	0.40	1.10	0.60	1.00	0.40
ϕ at γ (min)	0.22	0.20	0.26	0.18	0.22	0.20	0.26	0.18
η(max)	0.36	0.44	0.35	0.40	0.36	0.44	0.35	0.40
ϕ at η(max)	0.46	0.38	0.44	0.38	0.46	0.38	0.44	0.38
Fluid power(max)	0.55	0.75	0.50	0.38	0.55	0.75	0.50	0.38
ϕ at Fluid power(max)	0.48	0.50	0.46	0.44	0.48	0.50	0.46	0.44

From this Table 1, it is clear that effects of rpm on the above parameters are negligible as explained earlier. Maximum pressure coefficient of 2.20 is achieved for tongue geometry I and III. Minimum pressure coefficient of 0.60 is achieved for tongue geometry IV. Input power is more for tongue geometry III and less for tongue geometry IV. Maximum peak efficiency of 44% and maximum fluid power coefficient of 0.75 are achieved for tongue geometry II where as minimum peak efficiency of 35% is achieved for tongue geometry III and minimum fluid power coefficient of 0.38 is got from tongue geometry IV.

3. Pressure Coefficient vs Flow Coefficient

Figure 2 shows the pressure coefficient (ψ) vs. flow coefficient (ϕ) for four tongue shapes. Pressure coefficient is high at the minimum flow coefficient and decreases as the flow coefficient is increased, reaching a minimum value at about ϕ=0.20, ϕ = 0.22, and ϕ = 0.13 for tongue I, tongue II and tongue IV respectively. The pressure coefficient increases further with increase in flow coefficient for tongue II whereas for tongue I, pressure coefficient reaches its second peak at about ϕ = 0.40, but for tongue IV it is constant after ϕ=0.30. For tongue III, the pressure coefficient remains more or less same after reaching its minimum at ϕ=0.20.

Tongue I produces higher pressure coefficient for the entire operating range followed by tongue III, tongue II and tongue IV respectively. The impeller achieving different pressure coefficients and different maximum flow rates for different tongue geometries is mainly due to two reasons. (i) Clearance between the tongue and impeller periphery. (ii) the vortex, its shape, its position and magnitude. The vortex formation and its movement with tongue geometry and flow coefficient will be discussed later. The maximum flow rate is high for tongue II where as it is low for tongue IV.

In general pressure coefficient is maximum at low flow coefficient and it increases after reaching a minimum value as the flow coefficient is increased. This is due to vortex formation which is clearly seen from flow visualisation photograph near tongue region (ie. around θ=180°) at full volume flow rate which prevent the leakage. Since it move away from the tongue as the flow coefficient decreases, fluid is escaping through the gap between the impeller and tongue. At low flow coefficient, the vortex centre is at about θ = 215° and extends upto the centre of the impeller and tongue. This leads to less leakage at low flow coefficient also. Though impellers produce higher pressure coefficient at low flow coefficient, efficiency is very less due to the larger size vortex which consumes more power. Impeller II with tongue IV geometry, pressure coefficient is less for the entire flow coefficient range when compared to other impellers and tongue geometries due to less deflection and less guidance to the fluid.

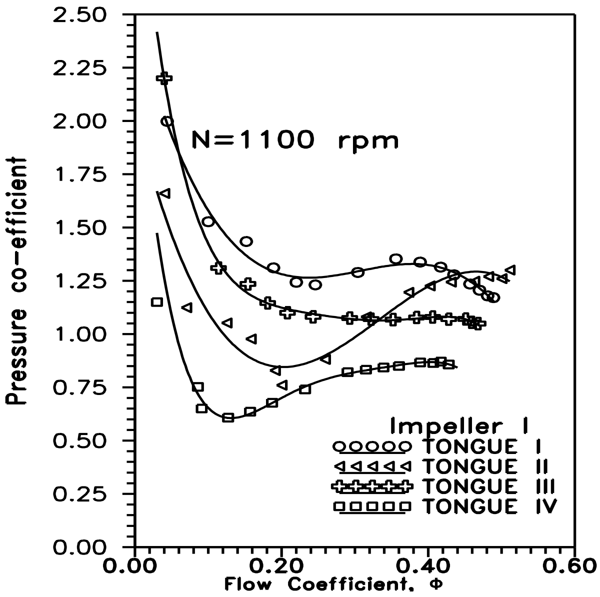

FIG. 2
PRESSURE VS FLOW COEFFICIENT

4. Fluid Power Coefficient vs Flow Coefficient

Figure 3 show the fluid power coefficient vs. flow coefficient. The fluid power developed by the impeller increases more or less linearly with the flow coefficient.

Fluid power developed is more for tongue II followed by tongue I, tongue III and tongue IV.

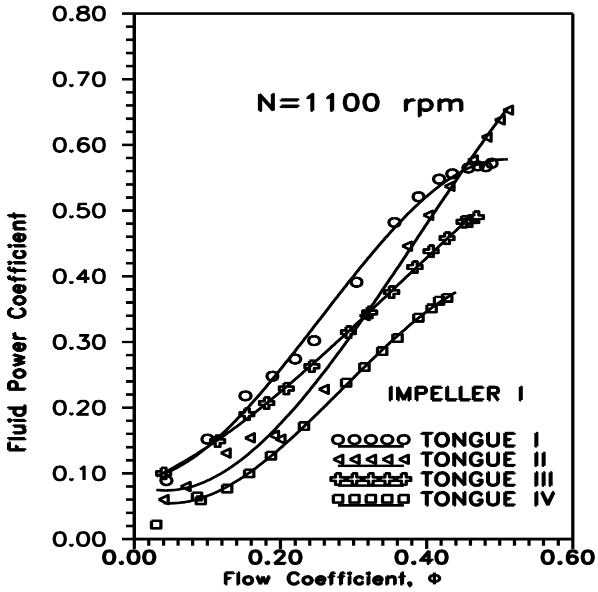

FIG. 3
FLUID POWER T VS FLOW COEFFICIENT

5. Input Power Coefficient vs Flow Coefficient

The power consumed by the impellers is shown in Figure 4. Input power decreases from maximum and again it increases after reaching a minimum value almost at mid flow coefficient. The variation of input power coefficient with flow coefficient is in contrast to axial or centrifugal fans where input power more or less increases with flow coefficient and may decrease in the higher flow coefficient range. The vortex size in cross flow fan is very large at low flow coefficient and decreases, as the flow coefficient is increases. Porter and Markland (1970) reported that the area belonging to re-circulating flow (vortex flow) is proportional to the power input in the re-circulating area. They revealed that in the vicinity of best efficiency point, 25 -50% of power input is required to maintain recirculation and a close relation exists between the power input and circulation around it. It can be concluded that the power input to maintain re-circulating area at very low flow coefficient could be more than 50% of input power. Hence, input power coefficient is highest at very low flow coefficient. The input power coefficient increases after reaching its minimum due to the variation in pressure coefficient and flow coefficient.

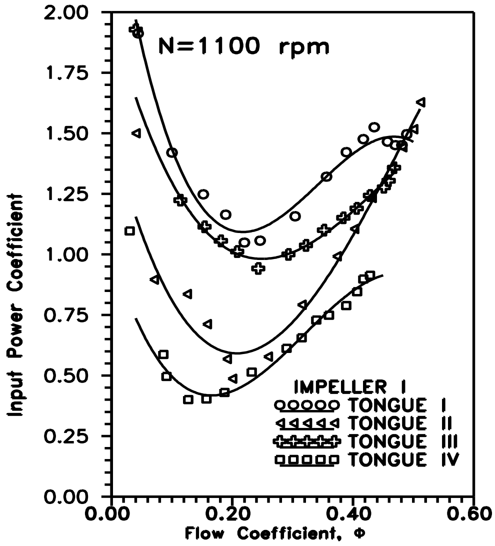

FIG. 4
INPUT POWER VS FLOW COEFFICIENT

6. Efficiency Vs Flow Coefficient

Figure 5 improves efficiency with the flow coefficient and reduces when the maximum value is reached. In the low-flow band due to its high input power and low fluid power the efficiency is extremely poor. After its apex, the decreased efficiency is attributed to more losses in friction at a higher flow rate. Refriction losses are proportional to the square volume flux for turbulent flow. Cross fans are often less efficient than their counterparts. The primary reasons for the steady operation are I the vortex consumes a substantial input power. (ii) There are very high shock (incidence) losses among fans of cross flow. They are present at the entrances to the suction and supply areas. The flow passes the inside of the impeller after exiting the suction zone. In this way, the vortex structure comes into the delivery area after the flow, resulting in significant incidence losses (shock) both at positive and negative angles. The total effect is that, in contrast to other blade channels, the blade channels that receive flux without any influence conceive maximal energy for fluid. In contrasts to centrifugal or axial fans, where every blade also contributes to the transfer of energy.

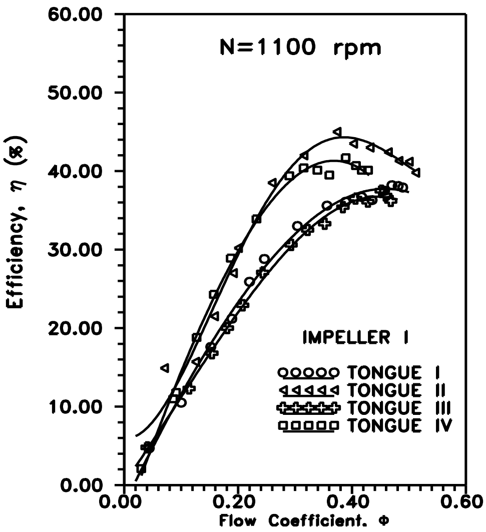

FIG. 5

EFFICIENCY VS FLOW COEFFICIENT

FLOW VISUALIZATION

In order to get the qualitative understanding of vortex formation, its size and movement, flow visualisation has been conducted with the aid of woolen tufts. Woolen tufts were stitched to a thin yellow paper and the paper was pasted the interior of the back shroud. Care was taken to have more tufts in the region of tongue clearance and inside the vane channels. The tufts are weightless and hence, take the shape of the flow pattern when the impeller is running. Flow visualisation was done with video camera. Uniform intensity of lights posed several problems as the lights kept in front of the impeller to avoid shadows produced reflections. The shutter speed of the camera is 1/2000 s. For each throttle setting, video coverage was done for about 10 minutes. After viewing it on the TV monitors, the footage of about 2 minutes was selected. As the output was in analogue form, it was later converted to digital form. The entire flow visualisation data runs into several mega bytes of memory. Using media player software, frame by frame analysis was done and the best frames in terms of quality and presentation were selected and printed. Flow visualisation was conducted for impeller I and III for tongue III and tongue IV respectively.

Some of the observations made earlier in connection with the variation of pressure coefficient, fluid power coefficient, input power coefficient and efficiency can be explained with the aid of flow visualisation studies.

The flow visualization studies of tongue III at 900 and 1100 rpm reveals the formation of vortex near the tongue region (θ = 1800) is clearly visible, which prevents the leakage of fluid at high volume flow rate. In absolute as well as in relative coordinates the true flow in CFF is unstable. Therefore, the circulation of the blade changes around the impeller perimeter. Following a change in the circulation of the blades, vortices are removed on the trailing rims, making the flow of the impeller rotating. The work carried out changes along the blades, such that the overall pressure for the individual streamlines differs, i.e. the flow that passes the blades is rotating. The eccentric vortex that dominates the flow pattern in the cross fans is generated by the vortex. But at low volume flow rate, though the vortex is away from the tongue region, size of the vortex is sufficiently large to seal the gap between the impeller and tongue that cause higher pressure coefficient at low volume flow rate also. Bigger size of this vortex consumes more than 50% of the input power that cause very low efficiency at low volume flow rate. At mid flow region, the vortex is away from the tongue and also size is not sufficiently large to seal the gap between the impeller and tongue causing the leakage of fluid. Flow leaks after impeller does the work on the fluid. Hence, leakage flow not only causes loss in volume flow rate but also in power. The position and size of the vortex was observed to de sensitive to tongue geometry. Size of the vortex is found to be small when compared to the previous geometry. This leads to more leakage than tongue geometry III. Due to these reasons, same flow coefficient for different tongue geometries could not be achieved for the same throttle settings. The throttle settings have to be changed in order to get the required flow coefficient. When the leakage is more, the throttle has to be moved forward. Impeller sucks more fluid but considerable part leaks to the atmosphere after receiving energy. Hence, input power is high and efficiency is low. That is why, different tongue geometries for a given impellers. For the same reasons, the maximum volume flow rate delivered is different for different tongue geometries. At minimum flow rate vortex is almost situated at about θ = 2700. For this reason pressure coefficient is not much affected at minimum flow coefficient. At full volume flow rate, vortex is formed at tongue region (θ = 1800) as usual, but the size of this vortex is small and it is formed within the blade region and so sealing effect also less when compared to tongue geometry III. This is the reason for less pressure coefficient for tongue geometry IV than III, for the given flow coefficient. Size of this vortex increases and also moves towards centre of the impeller, which cause leakage of fluid as the flow coefficient decreases. At low flow coefficient vortex centre is at about (θ = 2800) and directed towards the bottom of the impeller and so the gap between the impeller and tongue periphery is not covered by the vortex. This leads to more leakage at minimum flow coefficient also causing. This vortex is necessary to sustain stable flow through a cross flow fan and the positions and size of this vortex determine the pressure coefficient and efficiency respectively.

CONCLUSIONS

Based on the present experimental investigations the following broad conclusions are drawn.

The performance of the impellers is found to be sensitive to the tongue geometry but it fallow the similarity laws. Efficiency is maximum for the tongue. Volume flow rate is minimum for tongue geometry IV. It is observed that high volume flow accompany with higher losses thereby decrease in efficiency.

The variation of pressure coefficient and input power coefficient with flow coefficient is found to be same most of the cases. Pressure rise and efficiency is contradicting each other. If the impeller produces higher fluid power, its efficiency is found to be minimum.

The fluid power for all impellers increases as the volume flow rate is increased, further, the fluid power also increases with the speed. Fluid power and input power are more for impeller I with tongue II than for other two impellers. Impeller I with tongue II is slightly more efficient than the other two impellers at higher volume flow rates.

For tongue I, tongue III and tongue IV impeller III gives maximum power and maximum efficiency.

Flow visualization photographs showing flow patterns on the back shroud of all impellers clearly revealed the vortex formation, its size and movement with flow coefficient. Flow leakage through the tongue was clearly evident to decrease in pressure rise at low flow coefficient for the impellers III. In some part of the outlet flow, the woolen tufts followed the blade curvature whereas in other parts the flow

REFERENCES

[1] Chen, P.H., & Lee, D.S., Effect of Tongue Shape on the Performance Curve and the Acoustic Noise of Cross Flow Fan, *J. Chin. Soc. Mech. Eng.,* 1995, 16, No. 5, pp. 445–455.

[2] Takushima, A., Shinobu, Y., Tanaka, S., & Matsuki, K. (1990). Flow measurement by laser doppler velocimeter in a cross-flow fan for airconditioning use. *ASHRAE Transactions, 96*(1), 497-501.

[3] Tsurusaki, H., Tsujimoto, Y., Yoshida, Y., & Kitagawa, K. (1997). Visualization measurement and numerical analysis of internal flow in cross-flow fan. *J. Fluids Eng.*, 119, pp. 633–638.

[4] Porter, A.M., & Markland, E. (1970). A Study of the Cross Flow Fan, *J. of Mechanical Engineering Science*, Vol. 12, No. 6, pp. 421- 431.

[5] Murata, S., & Nishihara, K. (1976). An Experimental Study of Cross Flow Fan (1st Report, Effects of Housing Geometry on the Fan Performance) *Bulletin of JSME*, Vol. 19, No. 129, pp. 314-321.

[6] Murata, S., & Nishihara, K. (1976). An Experimental Study of Cross Flow Fan (2nd Report, Movement of Eccentric vortex inside Impeller) *Bulletin of JSME*, Vol. 19, No. 129, pp. 322-329.

[7] Chen, A., Li, S., & Huang, D. (2008). Numerical Analysis of Aerodynamic Noise Radiated from Cross Flow Fan. *J. Front. Energy,* Power *Eng.,* Vol. 2, No. 4, pp. 443–447.

[8] Murata, S., & Nishihara, K. (1976). An Experimental Study of Cross Flow Fan (1st Report, Effects of Housing Geometry on the Fan Performance) Bulletin *of JSME*, Vol. 19, No. 129, pp. 314-321.

[9] Murata, S., & Nishihara, K. (1976). An Experimental Study of Cross Flow Fan (2nd Report, Movement of Eccentric vortex inside Impeller) *Bulletin of JSME*, Vol. 19, No. 129, pp. 322-329.