

Prediction of air delivery, noise and power consumption of fan for TEFC electric motors

Rajgopal Desale* and N K Deshmukh
Crompton Greaves Ltd, R&D, Mumbai

Received 02 February 2005; revised 19 December 2005; accepted 03 February 2006

This paper describes the method for estimating air delivery, noise and power consumed by fan for TEFC (Totally Enclosed Fan Cooled) motor at design stage itself. It is a program based on semi-empirical relations with constants derived from extensive experiments. Program (in C++) is provided with inputs with regard to fan geometry, rotational speed and system resistance, the programme gives necessary output for air delivery, noise and power consumed. Optimisation is done based on the weightage factor provided by the user. Fans for different frame size were designed, developed, tested and results compared with predicted values and found to be within 10% for air delivery and noise. However, power consumption results were within 15%.

Keywords: Air delivery, Air velocity, Blade frequency, Noise level, Operating point, Power consumption, Pressure loss, Static pressure, System resistance, Velocity pressure, Volumetric loss

Introduction

TEFC (Totally Enclosed Fan Cooled) motor has a fan at its non-driving end for cooling motor. These are normally bi-directional, radial blades and mixed flow type of fans and have to provide adequate cooling without consuming too much power and producing high noise. The paper attempts to develop a design tool using theoretical formulae, empirical relations and experimental constants. The program is linked with optimization technique and is made simple to avoid the requirement of specialist to understand and use. User has to key in certain inputs related to geometry of the frame, for which fan is being designed, and the speed of rotation. User has to specify target values for output parameters and weightage factors. Provision has been made for the user to specify upper and lower limit for each parameter. Program will do iterations in a few minutes and provide optimum solution with maximum air delivery and minimum noise and power consumption.

Design Methodology for Air Delivery

Important factors taken into consideration for design of fan in TEFC motors (Fig. 1) are air delivery and geometry of the motor to meet noise and power consumption criteria. Secondary considerations are:

- i) Easy to manufacture, assemble and clean; ii) Economical; iii) Bi-directional; and iv) Mechanical strength.

Operating Point for the System

Operating point is the point where requirement of discharge is met after overcoming resistance to flow (Fig. 2). Fan performance will strongly depend upon actual operating point. System resistance can be calculated based on the dimensions of components in the flow path and degree of change in the direction of flow. For this, relevant dimensions of fan cover, frame, end shield, shaft and fan should be known. Air will enter from fan cover inlet and will be discharged at fan cover outlet. Energy imparted to air by the rotating blades of fan is split into static and velocity energy. Static energy is required to overcome system resistance. Velocity energy is the real desirable

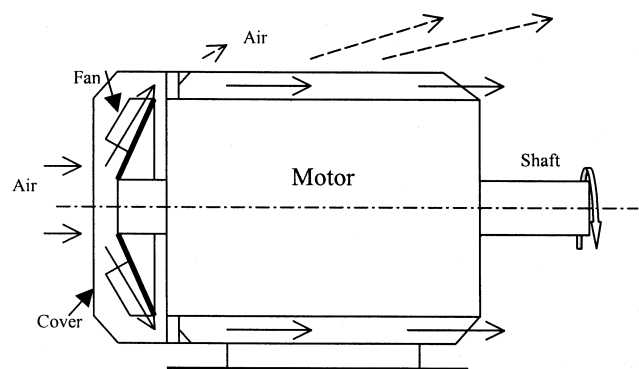


Fig. 1—Fan position and air

*Author for correspondence
Tel: 0241-2776142; Fax: 0241-2777508
E-mail: rajgopal.desale@cgl.co.in

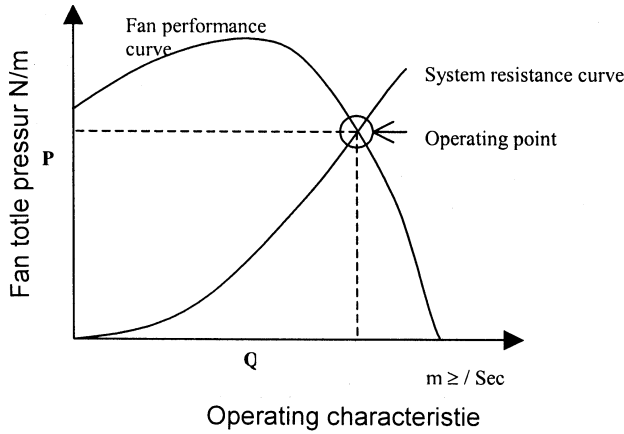


Fig. 2—Operating characteristic

outcome of the fan. While calculating the system resistance, flow through equivalent diam of pipe has been considered. Losses occur whenever there is change in cross section or change in direction of the flow. At each of these stages, pressure losses and volumetric losses have been considered. Following pressure losses have been taken into consideration: 1) Loss at entry to duct at fan cover, $\Delta P1$; 2) Loss due to change in cross section from fan outlet to annular area, $\Delta P2$; 3) Loss due to change in direction from annular area to fan cover outlet, $\Delta P3$; and 4) Loss due to change in cross section for flow at fan cover outlet due to fins, $\Delta P4$.

Relation for finding pressure loss for flow through pipes is as follows¹:

$$\Delta P1 = K1 \frac{1}{2} \rho V_i^2 \quad \dots(1)$$

where, $K1$ =constant 0.2-0.5; ρ =density of air 1.2 (kg/m^3); and V_i =velocity of air at fan inlet, m/s.

$$\Delta P2 = K2 \frac{1}{2} \rho V_o^2 \quad \dots(2)$$

where, $K2$ =constant $(1-A_{fo}/A_{anu})^2$; V_o =velocity of air at fan outlet, m/s; A_{fo} =fan outlet area, m^2 ; and A_{anu} = Annular area between fan cover and fan, m^2 .

$$\Delta P3 = K3 \frac{1}{2} \rho V_{anu}^2 \quad \dots(3)$$

where, $K3$ = constant 0.16-0.4; and V_{anu} =velocity of air at annular area, m/s.

$$\Delta P4 = K4 \frac{1}{2} \rho V_{cov}^2 \quad \dots(4)$$

where, $K4$ = constant 0.15-0.37; and V_{cov} =velocity of air at fan cover outlet, m/s; ΔP =total pressure

loss=system resistance=static pressure= $\Delta P1 + \Delta P2 + \Delta P3 + \Delta P4$.

Following volumetric losses have been taken into consideration:

(A) Loss due to clearance between fan and fan cover

$$QL = Cd \cdot \pi \cdot D1 \cdot \delta \cdot \sqrt{2 \cdot 0.66 \cdot P / \rho} \quad \dots(5)$$

where, Cd =coefficient of discharge, 0.6; δ =clearance between fan and fan cover, m; $D1$ =inlet diam, m; P =total pressure, N/m^2 ; and ρ =density of air, kg/m^3 .

(B) Discharge: Theoretical discharge, $Q_{th} = V_{mo} \cdot A_o$ and zero throttle discharge, $Q_{zt} = Q_{th} - QL$.

(C) Operating point: Operating point discharge, Q_{op} = interpolation of Q from fan characteristic curve at operating point

$$\text{Fan total pressure, } P = \rho \cdot U_o \cdot V_{uo} \quad \dots(6)$$

$$\text{Volume flow, } Q = A_o \cdot V_o \quad \dots(7)$$

Design Methodology for Noise

For fans, two types of noise related to frequency are important: i) Wide band noise, in which acoustic energy is continuously distributed over a frequency spectrum; and ii) Pure tones, in which the acoustic energy is concentrated over narrow bands in the frequency spectrum. The rating system is the "A weighted" sound pressure level (dBA) often used in determining compliance with regulations. The dBA rating is determined directly by a sound level meter equipped with a filtering system which de-emphasizes both the low and high frequency portions of the audible spectrum. Sound power level cannot be measured directly and must be calculated from sound pressure measurement, which is recorded at a distance of 1 m from the source and is used as the basic measurement for comparing noise levels of fans, as well as noise levels at different operating points of the same fan.

Following relation^{2,3} is used for estimating the fan noise:

$$L_w = K_w + 10 \cdot \log Q + 20 \cdot \log P + C \quad \dots(8)$$

where, L_w =estimated sound power level (dB re 1 pW); K_w =specific sound power level; Q =flow rate, m^3/sec ; P =fan pressure rise, N/m^2 ; and C =correction factor, dB.

Estimated sound power level is calculated for all seven bands with K_w selected from Table 1. The BFI

Table 1—Specific sound power levels and blade frequency increments

Fan type	Impeller size, mm	Sound power level, KW (dB)							
		Octave band center frequency, Hz							
Radial blade		63	125	250	500	1000	2000	4000	BFI
	>1000	48	42	45	42	40	35	33	8
	500-1000	58	51	51	48	48	43	41	
	< 400	66	60	61	53	47	42	42	

Table 2—Sound correction factors

Static efficiency, % of peak	Correction factor, dB
90-100	0
85-89	3
75-84	6
65-74	9
55-65	12
50-54	15

(blade frequency increment) is added to the octave band, in which the blade passage frequency falls. Sound correction factor is selected from Table 2.

Design Methodology for Fan Power

The output fan power P_{fo} is determined from product of the flow and total pressure rise.

$$P_{fo} = Q \cdot Pt \quad \dots(9)$$

where, P_{fo} =output power, W; Q =flow rate m^3/sec ; and Pt =fan total pressure, N/m^2

Fan input power is the ratio of fan output power to fan total efficiency

$$P_{fi} = Q \cdot Pt / \eta_t \quad \dots(10)$$

where, P_{fi} =input power W; and η_t =fan total efficiency, dimensionless ratio.

Verification method of Air Delivery Measurement

A Fan test rig (Fig. 3), used to verify air delivery results, contains: (i) Inlet section comprising mounting arrangement for fan and drive motor; (ii) Middle section comprising plenum chamber (3'x3' cross section) inside which are housed wooden models of motor body and its end shield to measure static pressure; and (iii) Outlet section comprising long duct and orifice plate to measure velocity pressure, throttle arrangement to control out flow, booster fan connected at the end of duct to overcome test system resistance and get points close to the operating point.

Small motors are put inside so that the system resistance is taken into account, keeping the fan inlet

outside the plenum chamber. Fan cover directs the flow leaving the fan around and along the fins of the actual or wooden motor body kept inside plenum chamber. A static pressure point is provided in this section of plenum. Flow then passes through the honeycomb section followed by wire mesh screen so that it is cleaned up of any swirl, etc. The flow leaves the plenum chamber through a streamlined bell mouth and enters the exit duct. An orifice metering system is incorporated in this duct to measure the flow rate. Diam of duct D is 9", length of straight duct upstream of orifice is 10 D , and length of straight duct downstream of orifice is 5 D .

Points (pressure and volume) for fan characteristics are obtained by controlling the throttle valve arrangement. Test start with booster fan OFF, throttle full open and gradually it is closed. The static tapping in plenum chamber is connected to precision alcohol (specific gravity, 0.8) manometer of least count 0.2 mm. The D and $D/2$ tapings of the orifice are connected to another precision manometer, the differential head giving a measure of the quantity of flow. From fully closed condition, it is again opened gradually with booster fan, which is put ON to overcome the test system resistance. Then the graph of pressure vs discharge is plotted and thus the discharge from fan is obtained at zero pressure head.

For measuring the flow, an orifice diam d is selected such that booster fan can absorb all test system losses. Following are the relations used for finding out the discharge:

$$Q = Cd \cdot A \cdot V_d \quad \dots(11)$$

where, Q =discharge, m^3/sec ; Cd =coefficient of discharge of orifice=0.6; A =area of orifice outlet= $\pi/4 \cdot d^2$, m^2 ; d =diam of orifice, m; V_d = velocity of discharge, $m/sec = \sqrt{2(pw \cdot g \cdot hw) / \sqrt{\rho_{air}[1-(d/D^4)]}}$; ρ_{pw} =density of water=1000 kg/m^3 ; ρ_{air} =density of air=1.09 kg/m^3 ; D =diam of duct, m; hw =head of water=0.8 x h_{alc} , m; h_{alc} =head of alcohol, m; and plot of H (Y-axis) vs Q (X-axis) will give fan discharge at $H=0$, where curve intersects at X-axis.

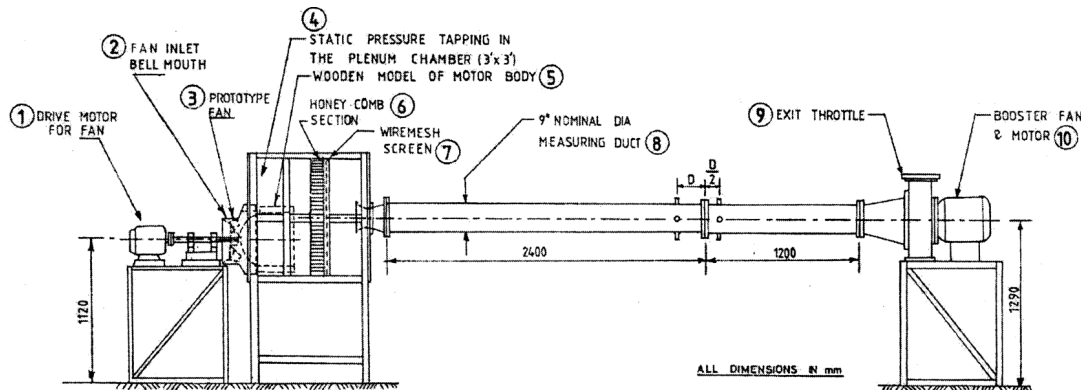


Fig. 3—Fan test rig for air delivery

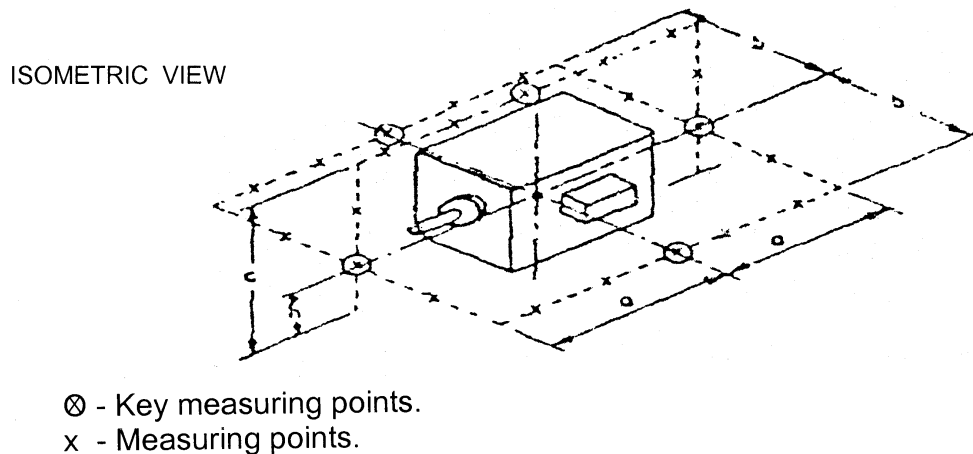


Fig. 4—Fan noise test method

Verification Method for Noise Measurement (As per IS12065)

Noise measurement is done in acoustic room where ambient noise level is 33dBA. Fig. 4 explains the positions of the measurement points around the motor. Generally, it is 1 m away from the body of the motor. A sound pressure level in dBA is obtained, which can give sound power level of the motor. To verify the predicted fan sound power levels, sound has to be measured with and without fan. By doing the logarithmic subtraction of the motor noise with fan and motor noise without fan, fan noise is obtained. Following relation explains the methodology:

$$L_w = L_{am} + 10 \log_{10} (2 \pi r s^2 / S_o) \quad \dots(12)$$

where, L_w =sound power level, dBA; L_{am} =av sound pressure level, dBA=arithmetic av of readings if the difference in the readings is less than 5 dBA; r =radius of the equivalent hemi-sphere, m= $\sqrt{a(b+c)/2}$; a =distance from center of motor to DE side location, m; b =distance from shaft center to

LHS/RHS of the motor location, m; c =distance from ground to top location (Position 5), m; and S_o =a reference area of 1 m².

Following expression shows how fan noise is determined:

Let, L_{wmf} =sound power level of motor with fan, dBA

L_{wm} =Sound power level of motor without fan, dBA

Then, L_{wf} =sound power level of fan, dBA= $10 \log_{10} [10^{L_{wm}/10} - 10^{L_{wmf}/10}]$

Verification Method of Measuring Power of the Fan

In a similar fashion, fan power is segregated from motor power. Reducing the voltage eliminates core losses from motor. Power consumed at different voltages with and without fan is measured. Power is linearly proportional to square of the voltage. Power (Y -axis) is plotted against square of voltage (X -axis). While reducing the voltage, care is taken that the speed shall not drop substantially. The difference in the values when the curve intersects Y -axis for with

Table 3—Comparison of predictions and actual results

h	Fan design				Air delivery, m ³ /sec			Noise, dBA			Fan power, W		
	OD	B	Z	RPM	A	P	$\%E$	A	P	$\%E$	A	P	$\%E$
90	150	20	6	3000	0.041	0.043	-4.6	72.22	78.26	7.7	29	28.87	-0.45
132	150	20	6	3000	0.070	0.065	-7.5	73.16	72.21	-1.3	39	41.12	5.43
	190	30	6	3000	0.120	0.110	-8.33	82.64	76.84	-7.55	102	103	0.98
	210	30	6	3000	0.125	0.126	0.8	83.98	79.86	-5.16	125	142.6	14.1
132	150	25	5	3000	0.06	0.062	3.33	65	70.61	8.6	23.0	21.4	-6.9
	165	25	5	3000	0.068	0.069	1.47	67	72.47	8.16	33.4	32.2	-3.6

h = Frame height, mm; OD = Outer diam, mm; B = Blade width, mm; Z = Number of blades; P = Predicted; A = Actual; $\%E$ = % Error.

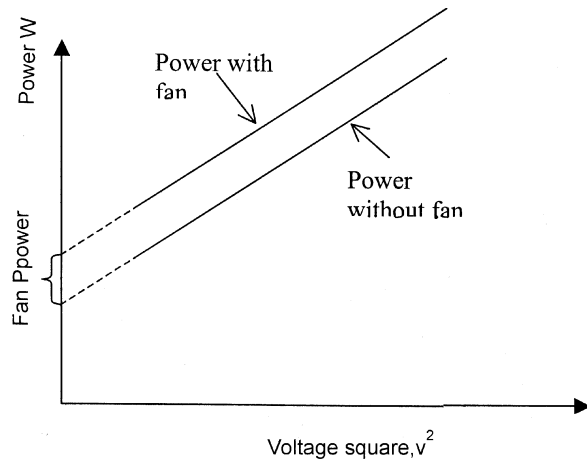


Fig. 5—Method of segregating the fan power

and without fan is attributed to fan power consumed (Fig. 5).

Optimisation

The well-known⁴ random walk optimisation method with direction exploitation is used for optimisation. The multi-objective function is constructed using weightage factors.

Results

Standards define the maximum noise levels, efficiency and the maximum temperature rise of motor. Values for various existing and few new fan designs were compared with actual and predicted results. The comparison of error bands for 95%

confidence interval of new method and actual results is shown in Table 3.

Conclusions

New methodology (applicable to radial flow bi-directional fans) of prediction from the program falls within $\pm 10\%$ of actual results for air delivery and noise. Designs with this program are more convenient, time saving and do not require any special skill. It gives total picture about fan performance with respect to important parameters like air delivery, noise and fan power. It facilitates the designer to evaluate various design parameters before hand so that one can choose the best solution. Linkage with optimization program provides the optimum solution doing thousands of iterations internally in few seconds.

Acknowledgement

The authors thank management of Crompton Greaves Ltd for their support for this project and permission to publish this paper.

References

- 1 Osborne W C, *Fans*, 2nd edn (Pergamon Press Ltd, Great Britain) 1982, 133-135.
- 2 *Fans Reference Guide*, 4th edn (Ontario Power Generation, Canada) 2001, 79-83.
- 3 Cyril M H, *Handbook of Acoustical Measurements and Noise Control*, 3rd edn (McGraw-Hill Inc., USA) 1991, 34.1-34.9.
- 4 Rao S S, *Optimization Theory and Application*, India, 10.