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UNIFIED DESIGN AND COMPARATIVE PERFORMANCE EVALUATION OF FORWARD AND BACKWARD CURVED RADIAL TIPPED CENTRIFUGAL FAN

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ABSTRACT
It is important to recognize that the design of any turbomachine is an interdisciplinary process, involving aerodynamics, thermodynamics, fluid dynamics, stress analysis, vibration analysis, the selection of materials, and the requirements for manufacturing. Though centrifugal fans have been developed as highly efficient machines, design is still based on various empirical and semi empirical rules proposed by fan designers. The objective of present study is to analyze performance of explicit design methodologies and tracing of unified design to get design point performance. Further the scope of work was extended towards performance evaluation of unified design along with comparative assessment of forward and backward leaned radial tipped blades. The unified design procedure developed in present work is outlined after performance evaluation of explicit designs (which differs widely), suggested by Church, Osborne and retrieved from fundamental principles of fluid flow having minimum assumptions made. The comparative assessment of forward and backward leaned radial tipped centrifugal fans as designed by this unified method is also presented to authenticate the process.

Keywords: Present design methodologies, Development of unified design methodology, Comparative assessment of forward and backward leaned radial tipped blades.

1. INTRODUCTION
The word turbo or turbines is of Latin origin and implies that spins or whirls around. Radial blade with curved surface is known as radial tipped blade. It is a forward curved radial tipped, if curvature of blade is towards the direction of rotations but opposite to direction of rotation signifies backward curved radial tipped blade. These are clear and self explanatory by Fig.1.

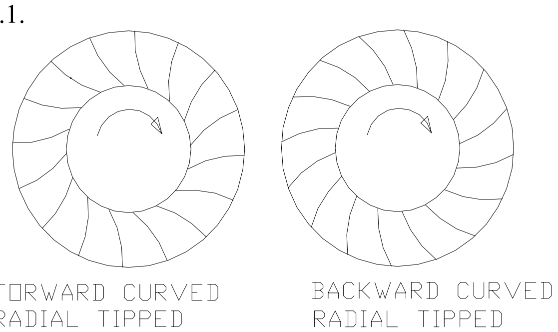

Fig. 1 Types of Blade Leaning

Primary attempt to focus on the design methodology for centrifugal pumps and blowers was made by Church [1]. This design methodology for centrifugal, axial and cross flow fans was extended by Eck Bruno and stated that the optimum number of blades of a radial impeller can only be truly ascertained by experiments [2]. Following successive researchers have studied some facets of design process.

- Bela Mishra had studied and made an attempt to compile design methodology, for the radial tipped centrifugal blower.[3]
- Nitin Vibhakar tested for the experimental validity of this design methodology and found that 16 no. of blades give the optimum performance.[4]
- Deepak Sharma carried out the experimental work on transparent blower to measure different velocities at impeller exit with the help of 3-point wedge probe and presented experimental value of slip factor.[5]
- C.C.Patel made experimental comparative assessment of design methodologies, suggested by Church, Osborne and method laid down from Fundamental Principles of fluid flow and energy

transfer.[6]

The distinguish study for forward and backward leaned blades have revealed following interesting results;

- Wasika experimentally verified that radial and backward leaning vanes give slightly better efficiencies than forward vanes.[7]
- Balje observed that optimum efficiency of a centrifugal blower occurred for slightly backward leaning vanes.[8]

The survey of literature shows that there still exists a need to device unified design methodology for radial tipped centrifugal fan/blower, which has been validated through experiments. There is also a need emerges to make comparative performance evaluation of forward and backward curved radial tipped centrifugal blower/fan. Looking to this need, here an attempt is made;

- To evolve a unified design methodology which is based more on fundamental concepts and involving minimum assumptions.
- To validate the unified design through performance evaluation.
- To make a comparative assessment of backward and forward leaned radial tipped centrifugal fans/blowers based on their relative performances.

2. UNIFIED DESIGN METHODOLOGY FOR RADIAL TIPPED CENTRIFUGAL FAN/BLOWER

Much research work is done towards more and more systematic design approach for centrifugal blower/fan. Different authors have suggested different procedures, and each has used differing concepts. Based on design methodologies, it is recognized that the design of radial tipped centrifugal blower/fan is well-traced area by different researchers. But these designs provide an open ended problem, there being no 'correct' procedure and considering the compromises usually involved, there is no 'correct' solution. So, there is need to develop unified design which can give design point performance.

Design procedure suggested here is an essence of Fundamental, Church [1] and Osborne [9] design concepts.

Here the design procedure described during the course of this work is presented in three main sections.

- Non dimensional parameters
- Impeller design
- Scroll Casing design

Occurrence of losses in various flow passages is also considered. Considering the losses, iterations are made to get optimum geometry at minimum losses. Volute casing is taken of spiral shape.

2.1 Non Dimensional Parameters

The Non-dimensional parameters are having considerable assistance to manufacturers and users of the fans/blowers. General users are least concerned with the theoretical aspects of designs but they can use effective tool of non dimensional parameters to select proper fan/blower. Following are widely used non-dimensional parameters.

Specific Speed = $$\omega Q^{1/2} / (gh)^{3/4} \qquad (1)$$

Speed Coefficient $\sigma = (0.379 * N * Q^{1/2}) / (h)^{3/4}$ (2)

Diameter Coefficient δ can be found out from Cordier diagram ($\delta \rightarrow \sigma$) [10].

Pressure Coefficient $\psi = 1/[\sigma^2 \delta^2]$ (3)

Volume Coefficient, $\phi = 1/[\sigma \delta^3]$ (4)

2.2 Number of Blades

Nitin Vibhakar [4] carried out series of experiments to optimize no. of blades in a radial tipped centrifugal fan/blower. As an outcome of his work, he has suggested the optimum performance of the blower is achieved for 16 no. of blades. This was confirmed under varying suction pressure and varying speed of rotation conditions. In unified design methodology, 16 no. of blades are selected as optimum.

2.3 Effect Of Compressibility On Design

In blower/fan, air is compressible working fluid. Within different flow passages of a fan, density of air changes with respect to change in temperature & pressure.

In unified design, it is taken into consideration that the weight flow of air per unit of time passing at a point in a blower/fan is constant when the flow is steady. The volume flow will not be constant since the specific weight varies with changes in temperature and pressure of air. The dimensions of the air passage must be calculated in accordance with this variation in volume flow. So, volume flow Q will not remain constant, but mass flow rate m remains constant. Considering compressibility effect in a flow path, fan is designed by taking stage pressure ratios between atmosphere to impeller eye, impeller eye to impeller inlet, impeller inlet to impeller outlet and impeller outlet to casing outlet, to make geometrical design of a blower/fan.

2.4 Impeller Design

Impeller design is heart of any turbo machine as energy transfer takes place in impeller passages. It must be designed with maximum care and to get optimum efficiency. Steps involved for impeller design are presented herewith in flow process path.

Impeller Design

From, Characteristics Gas Equations

ρ_a, ρ_s, ρ_d Kg/m^3

$\Delta P = \rho * g * Q * h$ h in meter (5)

Impeller Inlet Dimensions As Per Fundamental Concept Of Flow Through Pipe:

Assuming $U1 = 1.1 * V_{m1} = 1.1 * V_1$ (6)

And $Q_1 = \pi/4 * D_{eye}^2 * V_1$ where $D_{eye} = 1.1 * D_1$ so,

$Q_1 = \pi/4 * (1.1)^2 * D_1^{\ 2} * U_1 / 1.1$ (7)

Here $U1 = \pi * D_1 * N / 60 * 1.1$

So D_1, U1, D_{eye}, V_{eye}, V1 = Vm1 can be received.

From Bernoulli's Equation and Considering Adiabatic Compression Work,

$$H = \frac{\gamma}{\gamma - 1} RT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma}{\gamma - 1}} - 1 \right] \quad (8)$$

$$\therefore \varepsilon p^{0.283} - 1 = \frac{0.283 * H}{RTa}$$

$\varepsilon_p = P_a/P_o$ & $\varepsilon_p^{0.283} = T_a/T_o$ $\therefore P_o, T_o, \rho_o$

From Inlet Velocity .Triangle $\therefore W_1$ in m/s and β_1 °

$Q = [\pi * D_1 - Z*t] * b_1 * V_{m1}$ $\therefore b_1$ in meter (9)

Impeller Outlet Diameter As Per Church [1]

Pressure Head $h = k' U_2^2 / g$ (10)

where k' = (0.5-0.65) $\therefore D_2$ in meter

Taking Flat Shroud, i.e. $b_1 = b_2$

$Q = [\pi * D_2 - Z*t] * b_2 * V_{m2}$ $\therefore V_{m2} = W_2$ in m/s (11)

From, Outlet Velocity Triangle $\therefore V_2$ in m/s and α_2'

Virtual Pressure Head Developed By Impeller =

$$\frac{1}{2g} \left[U_2^2 - U_1^2 + W_1^2 - W_2^2 \right] \quad (12)$$

Taking Compressibility Effect Between Impeller Eye And Impeller Outlet,

$\therefore \in_{P1}^{0.283} - 1 = 0.283 * H_{vir} * \alpha_p / R*T_0$ (13)

Where $\in_{P1}$ = Pressure Ratio Between Impeller Eye & Impeller Outlet Base Upon Full Head

$\therefore T_2 = \varepsilon_{P1}^{0.283} * T_0$ (14)

$\in_{P2}$ = P.R. Between Impeller Eye & Impeller Outlet Based Upon Effective Head

2.5 Losses In Impeller:

The actual performance of a centrifugal blower/fan at the design point of operation differs from that predicted by Euler's equation for virtual head ,

$$H = V_{u2} * U_2/g - V_{u1} U_1/g.$$

Part of this difference can be accounted by an adjustment for inter blade circulation which results in a reduction of the work done by the impeller.

Losses occur in both, the stationary as well as moving parts of the blower. The various losses are;

2.5.1 Impeller Leakage Loss [9]

Let C_d=discharge coefficient, δ = Radial clearance between impeller inlet and casing, Ps = Static pressure difference across impeller and volute casing on suction side, so

$$Q_1 = \pi C_d D_1 \delta \sqrt{\frac{2 Ps}{\rho}} \quad (15)$$

2.5.2 Impeller Pressure Losses [1]

Friction and turbulence in impeller eye,

$dP_i = ½ * k_i * \rho_o * V_{eye}^2$ (16)

Turbulence and friction at impeller vane passage,

$dP_{ii} = ½ * k_{ii} * \rho_o * (W_1 - W_2)^2$ (17)

Turbulence at impeller vane inlet,

$dP_{iii} = ½ * k_{iii} * \rho_o * (W_1)^2$ (18)

Here k_i, k_{ii} and k_{iii} = 0.1 to 0.2

2.5.3 Disk Friction Loss

There may also be a power loss due to fluid drag on the reverse surface of the impeller back plate. The loss may be estimated as

Disk Friction = $T = \dfrac{\pi f \rho W_2^2 r_2^5}{5}$ (19)

Where f - friction Factor = 0.005

2.6 Scroll Casing Design

The design concept adopted [11] here is well known as 4-point method. The spiral shapes obtained by three circular sections, as shown in fig.2 are;

- r_3 =71.2 % of D_2 ,
- r_4 =83.7 % of D_2 ,
- r_5 =96.2 % of D_2 .

Center of these three arcs are located from the center lines taking interval of 6.25% of impeller diameter D_2.

- Width of housing is 2.14 times b_2.
- Height of housing outlet is 112 % of D_2.
- Radius of tongue is taken 5 to 10 % of D_2.

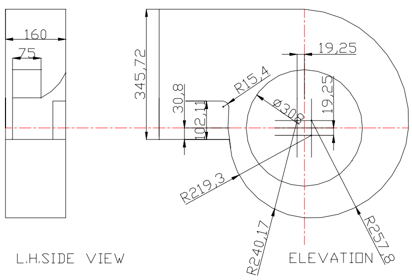

Fig.2 Scroll Casing By 4-Points Method

2.7 Casing Pressure Losses

The function of casing is to collect the flow coming out from impeller exit and also to diffuse the flow i.e. partially converting its kinetic energy into pressure energy. During this process volute casing creates following overall loss [1];

Friction and turbulence in casing

$dP_{v} = \frac{1}{2} * \rho_2 * (V_2 - V_3)^2 * k_v$ (20)

At maximum efficiency K_v is of the order of 0.4.

2.8 Efficiencies

The head in a centrifugal fan is generated by the impeller . The rest of the parts contribute nothing to the head but invites losses like hydraulic, mechanical and leakage.

All leading losses which are taking place between the stage (fan inlet and outlet) are considered for the hydraulic efficiency. These include skin friction losses along the fluid path from inlet to the discharge losses due to sudden change in area or direction of flow , and all losses due to eddies or whatever their causes .Considering all these pressure and leakage losses in the fan/blower .

2.8.1 Hydraulic Efficiency

$$\eta_{hy} = \frac{\Delta Ps}{\Delta Ps + dpi + dpii + dpiii} \tag{21}$$

2.8.2 Volumetric Efficiency

$$\eta_{vol} = \frac{Q}{Q + Ql} \tag{22}$$

2.8.3 Total efficiency

$$\eta_{total} = \eta_{hy} \ x \ \eta_{vol} \ x \ 100 \tag{23}$$

2.9 Input Power

Power Required running a fan is the ratio of output power developed by the fan to the electric motor or prime mover efficiency. So input power is;

$$= \frac{(\Delta p + dpi + dpii + dpiii)(Q + Ql)}{\eta motor} + \text{Power loss due to disk friction} \tag{24}$$

3. GEOMETRICAL PARAMETERS

Input data for design calculations is based on industrial application of fan, used for fume extraction from SDS-9 texturizing machine. They are;

Discharge Q = 0.5 m^3/s
Suction pressure = - 196.4 N/m^2
Delivery pressure = 784.8 N/m^2
Static Pressure difference ΔPs = 981.2 Pa
Speed N = 2800 rpm
Air Density ρ = 1.165 kg/m^3
Atmospheric Pressure (Pa) =1.01325x 105 N/m^2
Atmospheric Temperature = 30 ^{o} C = 303 ^{o} K
Nature Of Medium: Atmospheric air.

Following table shows final geometric and aerodynamic details of fan inlet, outlet and scroll casing. These dimensions are received after series of iterations made. Table also shows theoretical values of major losses and different efficiencies estimated for fan performance.

Table 1: Final Dimensions

At Final Iteration		**Abb.**	**Design**
Impeller Outlet Dimensions			
Peripheral Velocity	m/s	U_2	43.67
Relative Velocity	m/s	W_2	11.05
Meridian Velocity	m/s	V_{m2}	11.05
Absolute Velocity	m/s	V_2	36.80
Impeller Diameter	mm	D_2	298
Width Of Blade	mm	b_2	52
Air Angle		α_2	17.48º
Blade Angle		β_2	90º
Impeller Inlet Dimensions			
Eye Diameter	mm	D_o	176
Eye Velocity	m/s	V_{eye}	21.36
Peripheral Velocity	m/s	U_1	23.44
Relative Velocity	m/s	W_1	31.68
Meridian Velocity	m/s	V_{m1}	21.31
Absolute Velocity	m/s	V_1	21.31
Impeller Diameter	mm	D_1	160
Width Of Blade	mm	b_1	52
Air Angle		α_1	90°
Blade Angle		β_1	42.27°
Leakage Loss	m3/s	Q_l	0.0204
Pressure Losses In Impeller	Pa	D_{pim}	165.01
Scroll Casing			
Width Of Casing	mm	B_v	111.28
Outlet Velocity Of Casing	m/s	V_3	27.86
Scroll Radius	mm	r_3	212.18
Scroll Radius	mm	r_4	249.43
Scroll Radius	mm	r_5	286.68
Scroll Height	mm	H_s	333.76
Radius of tongue	mm	R_t	29.8
Casing Pressure Losses	Pa	d_p	199.98
Disk Friction	N-m		0.0231
Power Loss In Disk Friction	Watts	P_{df}	6.78
Hydraulic Efficiency	%	η_{hy}	72.89
Volumetric Efficiency	%	η_{vol}	96.08
Total Efficiency	%	η_t	70.03
Power Required To Run Fan		Watt	707.42

4. EXPERIMENTAL SET UP, MEASUREMENTS & INVESTIGATIONS

For evaluation and measurements of fan performance, the experimental set up was made as per IS-4894:1994. Line diagram of experimental setup is shown in fig.3.

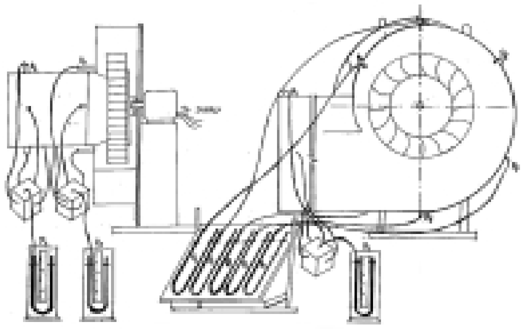

Fig.3 Experimental set-up line diagram

Photographic view of experimental set up with different pressure tapings is shown in fig.4.

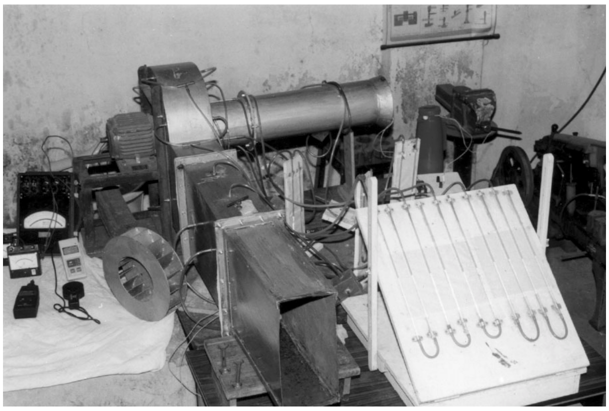

Fig.4 Photographic view of experimental set up

The investigations were designed to study the fan performance under varying speed and damping position. This appraisal was optimized for forward and backward curved radial tipped impellers. The final set of observations comprises actual stage pressure head developed across the fan, average air discharge, shaft power, static airpower developed and static efficiency.

5. RESULTS AND DISCUSSIONS

The performance obtained under unified design methodology for forward and backward curved radial tipped impellers are very much encouraging. Major performance parameters achieved are on higher side of design point. This is useful in establishing best operating range, which is essential for any turbo machine, especially fan/blower.

Table 2: Optimum Performance Data

Design Type	Damp	Fan ΔP Pa	Avg. Air Discharge m^3/Sec	Static Air Power Watts	Shaft Power Watts	Static Eff. %
Design point	0%	981	0.5	490.5	609	86.0
Funda-f [6]	75%	837	0.161	135	211	64.0
Church-f [6]	75%	1698	0.149	253	360	70.3
Osborne-f[6]	25%	1796	0.517	929	1506	61.7
Unified –f*	50%	1735	0.834	1448	1654	87.5
Unified–b*	50%	1550	0.757	1174	1250	93.9

**Here f stands for forward curved radial tipped centrifugal fan and b stands for backward curved radial tipped centrifugal fan.*

Table 2 presents optimum performance data for various design methodologies along with design point.

Comparative assessments of forward and backward curved radial tipped centrifugal fans are presented herewith graphically in fig.5 and fig.8.

Here it can be observed that the pressure head generated by forward curve radial tipped centrifugal fan is obviously higher than that for backward curved radial tipped centrifugal fan and this justifies that for the requirement of higher pressure head, forward curved radial tipped centrifugal fan can be selected.

At 50% & 75% damping conditions, efficiencies of backward curved radial tipped centrifugal fan are 93.9% and 81%, while that for forward curved radial tipped centrifugal fan, these values are 87.5% and 66% respectively at 2800 rpm. This also confirms that efficiency of backward curved radial tipped centrifugal fan is higher than forward curved centrifugal fan for the same speed and same damping conditions.

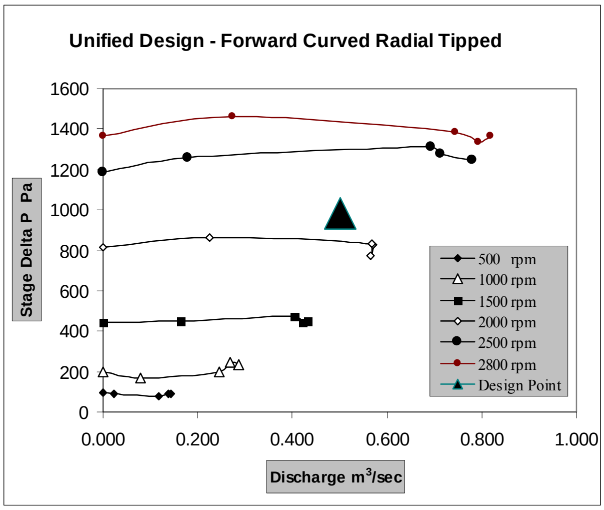

Fig.5 Discharge v/s Stage Pressure Rise for FCRT blades

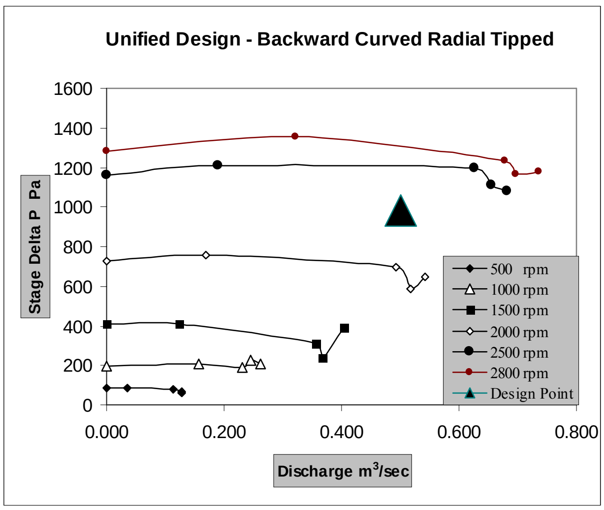

Fig.6 Discharge v/s Stage Pressure Rise for BCRT blades

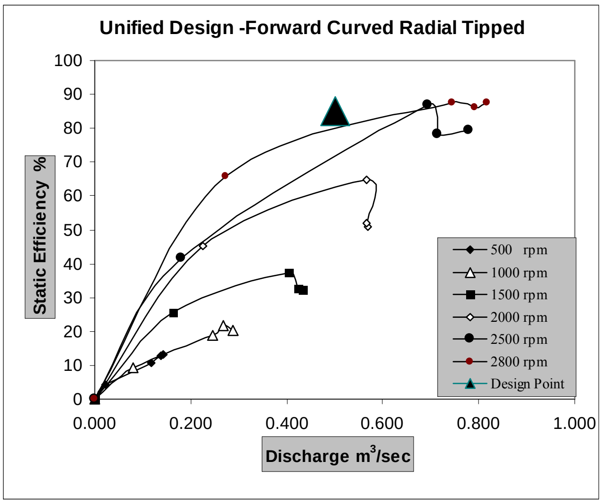

Fig.7 Discharge v/s Static Efficiency for FCRT blades

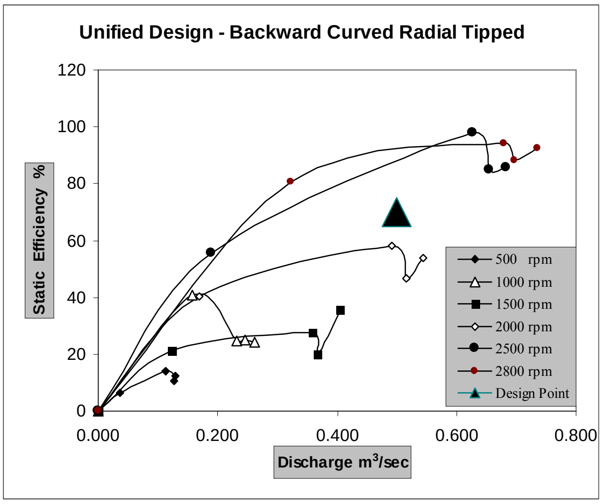

Fig.8 Discharge v/s Static Efficiency for BCRT blades

These tabular and graphical performance parameters of different design methodologies justify validity and superiority of presented unified design methodology.

6. CONCLUSIONS

Both the fans meet the design point performance and even higher level of performance at 2800 rpm in the damping range of 50% to 75%. For the same speed and same damping condition one has to go for forward curved impeller if the requirement is of higher pressure head and can go for backward curved impeller if the requirement is for the higher efficiency. Unified design methodology offers the highest efficiency levels of the order of 87.5% and 93.9% for forward and backward curved radial tipped centrifugal fan at 50% damping & at the speed of 2800 rpm.

It may be stated that unified design methodology outlined during the course of present work may be accepted as the experimentally validated design for radial tipped centrifugal fan/blower which can confidently offer the design point performance.

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7. NOMENCLATURE

Symbol	*Meaning*	*Unit*
ω	Angular Velocity	rad/s
σ	Speed Coefficient	
h	Head	m
g	Acceleration due to gravity	m/s^2
H_{ad}	Adiabatic head	m
H_{eff}	Effective pressure head	m
m	Mass flow rate	Kg/s
K	Pressure coefficient	
R	Characteristic gas constant	J/Kg-K
Z	Number of blades	
C_d	Discharge coefficient	
ε_p	Pressure Ratio	
δ	Radial clearance	mm
P_s	Static Pressure	Pa
f	Friction factor	
Subscript	**Meaning**	
a	Atmospheric condition	
s	Suction condition	
d	Delivery Condition	
o	Eye condition	
1	Inlet condition	
2	Exit condition at the impeller	
4	Exit condition of casing	