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ASSESSMENT OF DESIGN METHODOLOGY AND THREE DIMENSIONAL NUMERICAL (CFD) ANALYSIS OF CENTRIFUGAL BLOWER

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ABSTRACT

Case study for assessment of design methodology and three dimensional numerical (CFD) analysis of forward curved blade type centrifugal blower. Input data selected for design is based on industrial requirement for texturising machine to study forward curved blade type centrifugal blower.

Keywords: Design of Forward Curved Blade Type Centrifugal Blower; Numerical Simulation CFD; Flow Characteristics;

I. INTRODUCTION

Much research has gone into more and more systematic design of centrifugal blower. Different authors have been suggested different procedures, although each has a slightly different method of calculation, the broad underling principles all are similar. Impeller design procedure suggested in this paper is as per Eck Bruno. Casing design and Calculation of losses in this design is suggested by author W C Osborne. Volute is taken as spiral shape. Iterations are made to design calculations to get optimum geometry at minimum losses. The design procedure described in three main sections are 1) Non dimensional parameters 2) Impeller design 3) Volute design. The Non dimensional parameters will be of considerable assistance to manufacturers and users of fans who are not concerned with the theoretical aspects of designs. Input data is selected common for each design & is based on industrial requirement for texturising machine. The selected input data is, Discharge V =0.5 m^3/sec, Differential Pressure ΔP=981.2 Pa, Speed N=2800 rpm, Delivery Pressure P_d= 784.8 Pa, Suction Pressure P_s= -196.4 Pa, Suction Temperature T_s=30 °C = 303 K, Atmospheric Pressure=1.01325 x 10^5 Pa, Atmospheric Temperature =30°C = 303 K. The numerical analysis carried out for forward curved blade type centrifugal blower using Ansys CFX software.

II. DESIGN OF FORWARD CURVED CENTRIFUGAL BLOWER

The design procedure described in three main sections are 1) Non dimensional parameters 2) Impeller design 3) Volute design. The Non dimensional parameters will be of considerable assistance to manufacturers and users of fans who are not concerned with the theoretical aspects of designs.

2.1 Nondimentional Parameters

For the design, comparison, and critical assessment of all fans, one employs dimensionless coefficients. These coefficients must be dimensionless so that the numerical values which arise are independent of the actual increase in pressure, the mass flow, and other physical properties. There are in the fortunate position of being able to recommend a number of dimensionless coefficients which are the results of extensive study and which will most probably become standardized.

Specific speed $= \frac{w\,V^{1/2}}{(g\,H)^{3/4}} = \frac{\frac{2\pi N}{60}\times V^{1/2}}{(g\,H)^{3/4}} = \frac{\frac{2\pi\times 2800}{60}\times (0.5)^{1/2}}{(9.8\times 85.94)^{3/4}} = 1.326$

speed coeffcient σ $= \frac{0.379\times N\times V^{1/2}}{H^{3/4}} = \frac{0.379\times 2800\times (0.5)^{1/2}}{(85.94)^{3/4}} = 0.443$

Coordinator has given the relation between σ and δ in graphical form .the graphical form converted into mathematical form by Weighted Residue Method. The resultant equation for δ in different Σ stages obtained bellow

$\delta = a\,\sigma^{-b}$ where, δ is diameter coefficient

Here, a = 0.99 where $0.1 < \sigma < 0.4$

= 1.5 where $0.4 < \sigma < 2$

= 0.995 where $0.1 < \sigma < 0.4$

b = 0.5866 where $0.4 < \sigma < 1$

= 0.505 where $1 < \sigma < 2$

$\delta = 2.418$ for $\sigma = 0.443$

Pressure coefficient $\psi = 1/_{\sigma^2 \times \delta^2} = 0.872$, Volume coefficient $\phi = 1/_{\sigma \times \delta^3} = 0.159$

2.2 Impeller Design

Now, $\psi = \frac{\Delta p}{\rho/_2 \times u_2^2}$ ρ = air density, u_2 = peripheral velocity at outlet of impeller

$0.872 = \frac{981.2}{1.165/_2 \times u_2^2}$ Hence, $u_2 = 43.95\ m/s$

$\phi = \frac{v}{\pi/_4 \times d_2^2 \times u_2}$ v = volume flow rate, d_2 = outlet diameter of impeller

$0.159 = \frac{0.5}{\pi/_4 \times d_2^2 \times 43.95}$ Hence, $d_2 = 0.302\ m$

Formula for the volume coefficient ϕ is given in Eck Bruno's book, i..e

$\emptyset = \tan\beta_1 \times (d_1/d_2)^3 \times \frac{1-v^2}{\partial} \times \frac{t-a}{t}$ $v = \frac{d_1}{d_2}$, where d_1 = hub diameter, t = blade thickness and β_1 = inlet blade angle of impeller.

Taking $1 - v^2 \approx 1$, $\frac{t-a}{t} \approx 1$, $\partial = 1.2$,

$\emptyset = \tan\beta_1 \times (d_1/d_2)^3 \times \frac{1}{\partial}$ Taking values $\beta_1 = 30$ and the above calculated $d_2 = 0.302$,

$0.159 = \tan 30 \times (d_1/0.302)^3 \times \frac{1}{1.2}$ Hence, $d_1 = 0.208\ m$

Number of blades in impeller is

$$z = \frac{4\pi}{1.5} \times \frac{\sin\beta_2}{(1-r_1/r_2)}$$ β_2 =outlet blade angle of impeller

Thus the number of blades depends only on the β_2 and the radial ratio r_1/r_2. The formula gives an approximate indication of number of blades required for normal radial impellers, while multivane impellers require more specialized treatment. However, the optimum number of blades of radial impeller can only be truly ascertained by experiment.

Number of blades $z = 16$ optimized experimentally by Mr. Nitin Vibhakar for above taken input data.

Thus for $z = 16$

$16 = \frac{4\pi}{1.5} \times \frac{\sin\beta_2}{(1-0.104/0.151)}$ Hence, $\beta_2 = 36.30$

In the impeller separation of flow at the bend must be prevented. The most effective measure to combat separation at this point is to accelerate the main stream.

"Therefore the impeller entry area $\pi d_1 b_1$ must be smaller than the intake opening $(\pi/4)d_1^2$. This change in area will be designed by ∂"

Thus, $\frac{\pi}{4} \times d_1^2 = \partial \times \pi \times d_1 \times b_1$ where, b_1 = blade width at inlet of impeller

With 20 % acceleration, i.e. $\partial = 1.2$

$\frac{\pi}{4} \times 0.208^2 = 1.2 \times \pi \times 0.208 \times b_1$ Hence, $b_1 = 0.0433\ m$

No information is yet available to enable the discharge width b_2 to be determined from one definite aspect. In practice, one finds both parallel as well as tapered shrouds enclosing the blades of an impeller.

The shape of the shroud will depend on the shape of the blade. The decisive factor is the blade passage, not the meridional area. The mean velocity reduces from w_1 to w_2 in the blade passage. This deceleration is a very important factor in the design of an impeller. In the absence of reliable experimental data about the separation of flow in rotating passage, one makes use of an analogy with stationary diffusers.

Accordingly care must be exercised to ensure that the tapering angle dose not exceed 9° to 12°.

Therefore taking the 10° tapering angle, the formula will be

$\tan\theta = \frac{b_1 - b_2}{r_2 - r_1}$ where, b_1 = blade width at inlet of impeller and b_2 = blade width at outlet of impeller

$\tan 10^\circ = \frac{0.433 - b_2}{0.151 - 0.104}$ Hence, $b_2 = 0.035\ m$

Now velocity at inlet, $u_1 = \frac{\pi d_1 N}{60}$ where,

u_1 = peripheral velocity at inlet of impeller N = speed of impeller in rpm

$u_1 = \frac{\pi \times 0.208 \times 2800}{60}$ Hence, $u_1 = 30.49\ m/s$

From the inlet velocity triangle, $\alpha_1 = 90$,

Therefore, $w_1 = \frac{u_1}{\cos\beta_1}$ where, w_1 = Relative velocity at inlet of impeller

$w_1 = \frac{43.95}{\cos 30}$ Hence, $w_1 = 35.212 \frac{m}{s}$

Now, $c_1 = w_1 \times \sin\beta_1$ where, c_1 = absolute velocity at inlet of impeller

$c_1 = 35.212 \times \sin 30$ Hence, $c_1 = 17.606\ m/s$

One plots the cross- sectional area of the shroud or diameter of an equivalent circle against the mean velocity of the stream lines so that the enlargement of the blade passage area can be easily examined. The shorter the blade passages, i.e. a larger value for d_1/d_2, the less area. In general, one can relate the permissible enlargement of the blade passages, which is dependent on w_2/w_1, to the diameter ratio. Therefore as an approximation

$$\frac{w_2}{w_1} \geq \frac{d_1}{d_2}$$

So assumed that $\frac{w_2}{w_1} = \frac{d_1}{d_2}$ where, w_1 = Relative velocity at inlet of impeller

$\frac{w_2}{35.212} = \frac{0.208}{0.302}$ Hence, $w_2 = 24.252\ m/s$

From outlet velocity diagram,

$c_{2m} = w_2 \times \sin\beta_2$ where, c_{2m} = Tangential component of Absolute Velocity

$c_{2m} = 24.252 \times \sin 36.30$ Hence, $c_{2m} = 14.358\ m/s$

$AB = w_2 \times \cos\beta_2$

$AB = 24.252 \times \cos 36.30$ Hence, $AB = 19.545\ m/s$

$c_{2u} = u_2 - AB$

$c_{2u} = 43.95 - 19.545$ Hence, $c_{2u} = 24.405\ m/s$

Now, $c_2^2 = c_{2u}^2 + c_{2m}^2$ where, c_2 = absolute velocity at outlet of impeller

$c_2^2 = 24.405^2 + 14.358^2$ Hence, $c_2 = 28.323\ m/s$

Air angle at outlet is, $\tan\alpha_2 = \frac{c_{2m}}{c_{2u}} = \frac{14.358}{24.405}$ Hence, $\alpha_2 = 30.46$

The Fan Power = $\Delta P \times V = 981.2 \times 0.5 = 490.6\,Watts$

Consider power factor 1.1

Therefore, Power factor $1.1 \times$ the fan power $= 1.1 \times 490.6 = 539.66$ Watts

P = m x W *therfore*, $W = \frac{P}{\rho \times V} = \frac{539.66}{1.165 \times 0.5} = 926.45$ watt/kg/s

2.3 Design of Volute Casing

Analyzing steady flow energy equation at inlet and exit

$$\frac{P_1}{\rho_1} + \frac{1}{1}c_1^2 + gz_1 + w_\phi = \frac{P_2}{\rho_2} + \frac{1}{1}c_3^2 + gz_2$$

$$c_3^2 = \frac{-2\,(P_1 - P_2)}{\rho} + c_1^2 + 2w = \frac{-2\,(981.2)}{1.165} + 17.606^2 + 2(926.45) \quad Hence, \quad c_3 = 21.87 m/s$$

1. Casing Outlet Velocity $c_3 = 21.87 m/s$
2. Width of volute casing (Bv), Backward curved is 2.5 b_2 , Forward curved is 1.25 b_2

   And For Radial tipped is taken as 1.875 b_2 = 2 b_2

   Here Bv = 1.8 b_2 = 2 x 0.0433 = 0.055 m

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3. Centrifugal Fan Casing $r_\theta = r_3(1 + k\theta)$ Where k = 0.0023 for backward blade impeller
= 0.0020 for forward blade impeller
= 0.00215 for radial blade impeller

$$r_3 = \frac{d_2}{2} + 0.005 = \frac{0.302}{2} + 0.005 = 0.1515$$

Therefore for forward blade impeller $r_\theta = 0.1515\,(1 + 0.0020\theta)$

So we get,

θ in °	0	60	120	180	240	300	360
r in m	0.1515	0.16968	0.18786	0.20604	0.22422	0.2424	0.26058

4. Radius of volute tongue $r_t = 1.075 \times r_2 = 1.075 \times 0.151 = 0.16232\ m$

5. Angle of volute tongue $\theta_t = \dfrac{132\log(\frac{r_t}{r_2})}{\tan\alpha_2} = \dfrac{132\log(\frac{0.163}{0.151})}{\tan 30.46} = 7^\circ$

2.4 Losses

1. Leakage loss (V_1), Here, We are dealing with sharp-edged openings we shall assume that a coefficient of contraction $\mu = 0.7$ is applicable to gap.

   Thus volume leakage through the gap is $V_1 = V \times \frac{d_1}{d_2} \times \frac{\mu}{\varphi\, u_2} \times 4 \times \sqrt{\frac{2}{3}}$

$$V_1 = 0.5 \times \frac{0.204}{0.302} \times \frac{0.7}{0.159 \times 43.95} \times 4 \times \sqrt{\frac{2}{3}} = 0.11266\ m^3/s$$

2. Entry losses (Δp_1), this type of Losses may arise from a change of direction in the impeller, i.e. upon entry into the impeller intake; the air is diverted through an angle of approximately 90 before entry into cascade. These losses, which are comparable to losses at bends, are dependent upon the values of c_1 and c_2. If one relates this loss in the usual manner to the dynamic pressure of the maximum velocity,

   Then $\Delta p_1 = \delta_1 \times \frac{\rho}{2} \times c_1^2$ Where, $\delta_1 \approx 0.15 - 0.25$

   $$\Delta p_1 = 0.15 \times \frac{1.165}{2} \times 17.606^2 = 27.08\ pa$$

3. Friction loss in the impeller ($\Delta p_{impeller}$), the greatest losses arise from the passage of a fluid through an impeller. Length of blade curve is $l = \frac{r_2 - r_1}{\sin\beta_m}$ Where,

   $\beta_m = (\beta_1 + \beta_2)/2$ and $\beta_m = 33.15$

   $$l = \frac{0.151 - 0.104}{\sin 33.15} = 0.0859\ \text{m}$$

   $$w = w_1 \frac{1 + (w_2/w_1)}{2} = 35.212\,\frac{1 + (24.252/35.212)}{2} = 29.732\ m/s$$

   $$\Delta p^1 = c_f \frac{[2\, z\, l\, b_m + \pi(r_2^2 - r_1^2)2](\rho/2)w^3}{w_1 \sin\beta_1\, \pi d_1 b_1}$$ Where, $b_m = \frac{b_1 + b_2}{2} = 0.03915$

   $$= 0.004 \times \frac{[2 \times 16 \times 0.0859 \times 0.03915 + \pi(r_2^2 - r_1^2)2](1.165/2)29.732^3}{35.212 \times \sin 30 \times \pi \times 0.208 \times 0.0433} = 22.48\ pa$$

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$$\Delta p_{impeller} = \Delta p^1 + 0.15\ \frac{\rho}{2}\ u_2^2\ \left(\frac{d_1}{d_2}\right)^2 \frac{1-(w_2/w_1)^2}{cos^2 \beta_1}$$

$$= 22.48 + 0.15\ \times \frac{1.165}{2}\ \times 43.95^2\ (\frac{0.208}{0.302})^2\ \frac{1-(35.212/24.252)^2}{cos^2 30} = 78.60\ pa$$

4. Volute Casing Pressure Loss(Δp_{volute})

$$\Delta p_{volute} = kiii\ \times \frac{\rho}{2}\ (c_2 - c_3)^2 = 0.4 \times \frac{1.165}{2}\ (28.323 - 21.87)^2 = 10\ pa$$

5. Disk Friction(T)

$$\mathrm{T} = \frac{\pi\ f\ \rho\ u_2^3 r_2^3}{5} = \frac{\pi \times 0.005 \times 43.95^2 \times 0.151^3}{5} = 0.0066\ \mathrm{Nm}$$

Other Parameters:

1. Power Loss InWatts Due To Disk Friction

$$P_{loss} = \frac{2\pi NT}{60} = \frac{2\times\pi\times 2800\times 0.0066}{60} = 1.94\ wat$$

2. Hydraulic Efficiency

$$\eta_{hy} = \frac{\Delta p}{\Delta p + \Delta p_1 + \Delta p_{impeller} + \Delta p_{volute}} = \frac{981.2}{981.2+27.08+78.60+10} = 0.8945 \approx 90\%$$

3. Volumetric Efficiency $\eta_v = \frac{V}{V+V_1} = \frac{0.5}{0.5+0.1126} = 0.8161 \approx 82\%$

4. Total efficiency

$$\eta_{total} = \eta_v \times \eta_{hy} = 0.82 \times 0.90 = 0.738 \approx 74\%$$

5. Power Required to Run the impeller

$$= \frac{(\Delta p + \Delta p_1 + \Delta p_{impeller} + \Delta p_{volute}) \times (V - V_1)}{\eta_{total}} + \ power\ loss\ due\ to\ disk\ friction$$

$$= \frac{(981.2 + 27.08 + 78.60 + 10) \times (0.5 - 0.11266_1)}{0.74} + 1.94 = 584.84\ watt$$

So Torque = $\frac{585\times 60}{2\times\pi\times 2800} = 1.995\ Nm$

6. Shaft Diameter $d_s = \sqrt[3]{\frac{16\ T\ fs}{\pi\sigma_{shear}}} = \sqrt[3]{\frac{16\times 1.995\times 4}{\pi\times 343\times 10^5}} = 0.01069\ m$

7. Blade Profile by Circular arc Method

$$r_b = \frac{r_2^2 - r_1^2}{2(r_1 \cos\beta_1 - r_2 \cos\beta_2)} = \frac{0.151^2 - 0.104^2}{2(0.104\ \cos 30 - 0.151\ \cos 36.3)} = 0.189$$

Table 2.1 shows the values as per 0-iteration. Now, further calculations are done by adding leakage loss and pressure losses in discharge and differential pressure given in design input data respectively. 1st and 2nd iterations are shown in table 2.2, table 2.3 respectively. The calculation summaries listed bellow in tables.

Table 2.1

0^{th} ITERATION	At $\Delta p = 981.2\ pa$ and $V = 0.5\ m^3/s$	0^{th} ITERATION	At $\Delta p = 981.2\ pa$ and $V = 0.5\ m^3/s$
Non dimensional parameters		*Volute Casing*	
Speed coefficient σ	0.443	Width Of Casing	0.055 m
diameter coefficient δ	2.418	Outlet Velocity Of Casing	21.87 m/s
Pressure coefficient ψ	0.872	Diameter Of Casing at 0°	0.1515 m
Volume coefficient ϕ	0.159	Diameter Of Casing at360°	0.2605 m
Impeller inlet Dimensions		Volute Tongue Angle	7
Peripheral Velocity	30.49 m/s	Radius of Tongue	0.1623 m
Relative Velocity	35.212 m/s	Casing Pressure Losses	10 pa
Meridian Velocity	17.606 m/s	Disk Friction	0.0066 N m
Absolute Velocity	17.606 m/s	Power Loss Disk Friction	1.995 watt
Impeller Diameter	0.208 m	Power Required To Run impeller	584.8watt
Width Of Blade	0.0433	Hydraulic Efficiency	89.45%
Air Angle	90	Volumetric Efficiency	81.61%
Blade Angle β_1	30	Total Efficiency	73.8%
Impeller Outlet Dimensions		Shaft Diameter	0.01069 m
Peripheral Velocity	43.95m/s	Blade Profile Radius	0.1894 m
Relative Velocity	24.252 m/s		
Meridian Velocity	14.385 m/s		
Absolute Velocity	28.323 m/s		
Impeller Diameter	0.302 m		
Width Of Blade	0.035 m		
Air Angle	30.46		
Blade Angle	36.30		
Leakage Loss	0.113 m^3/s		
Entry loss	27.08 pa		
Pressure Losses In Impeller	78.60 pa		

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Table 2.2

1st ITERATION		At $\Delta p = 1096.88\ pa$ and $V = 0.612\ m^3/s$	1st ITERATION	At $\Delta p = 1096.88\ pa$ and $V = 0.612\ m^3/s$
Non dimensional parameters			*Volute Casing*	
Speed coefficient	σ	0.4508	Width Of Casing	0.059 m
diameter coefficient	δ	2.3937	Outlet Velocity Of Casing	22.5 m/s
Pressure coefficient	ψ	0.860	Diameter Of Casing at 0°	0.1605 m
Volume coefficient	ϕ	0.162	Diameter Of Casing at360°	0.2760 m
Impeller inlet Dimensions			Volute Tongue Angle	7
Peripheral Velocity		32.7 m/s	Radius of Tongue	0.172 m
Relative Velocity		37.76 m/s	Casing Pressure Losses	15.52 pa
Meridian Velocity		17.86 m/s	Disk Friction	0.0072 N m
Absolute Velocity		17.86 m/s	Power Loss Disk Friction	2.11 watt
Impeller Diameter		0.2225	Power Required To Run impeller	803.6watt
Width Of Blade		0.0465	Hydraulic Efficiency	89.3%
Air Angle		90	Volumetric Efficiency	82.71%
Blade Angle	β_1	30	Total Efficiency	73.86%
Impeller Outlet Dimensions			Shaft Diameter	0.01170 m
Peripheral Velocity		46.79 m/s	Blade Profile Radius	0.1933 m
Relative Velocity		26.34 m/s		
Meridian Velocity		15.32 m/s		
Absolute Velocity		29.80 m/s		
Impeller Diameter		0.320 m		
Width Of Blade		0.038 m		
Air Angle		30.99		
Blade Angle		35.37		
Leakage Loss		0.128 m^3/s		
Entry loss		28.31 pa		
Pressure Losses In Impeller		87.03 pa		

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Table 2.3

2nd ITERATION	At $\Delta p = 1112.06\ pa$ and $V = 0.6128\ m^3/s$	**2nd ITERATION**	At $\Delta p = 1112.06\ pa$ and $V = 0.6128\ m^3/s$
Non dimensional parameters		*Volute Casing*	
Speed coefficient σ	0.4465	Width Of Casing	0.059 m
diameter coefficient δ	2.4071	Outlet Velocity Of Casing	22.5 m/s
Pressure coefficient ψ	0.866	Diameter Of Casing at 0°	0.1610 m
Volume coefficient ϕ	0.161	Diameter Of Casing at360°	0.2769 m
Impeller inlet Dimensions		Volute Tongue Angle	7
Peripheral Velocity	32.75 m/s	Radius of Tongue	0.1723 m
Relative Velocity	37.87 m/s	Casing Pressure Losses	15.52 pa
Meridian Velocity	17.86 m/s	Disk Friction	0.00725 N m
Absolute Velocity	17.86 m/s	Power Loss Disk Friction	2.112 watt
Impeller Diameter	0.223	Power Required To Run impeller	803.6watt
Width Of Blade	0.0465	Hydraulic Efficiency	89.4%
Air Angle	90	Volumetric Efficiency	82.71%
Blade Angle β_1	30	Total Efficiency	73.86%
Impeller Outlet Dimensions		Shaft Diameter	0.01170 m
Peripheral Velocity	47.01 m/s	Blade Profile Radius	0.1964m
Relative Velocity	26.34 m/s		
Meridian Velocity	15.355 m/s		
Absolute Velocity	29.86 m/s		
Impeller Diameter	0.321 m		
Width Of Blade	0.0385 m		
Air Angle	30.94		
Blade Angle	35.66		
Leakage Loss	0.1286 m^3/s		
Entry loss	28.31 pa		
Pressure Losses In Impeller	87.84 pa		

After this diameter of impeller & the parameter is unchanged. So, there is no need to do further iteration.

III. NUMERICAL ANALYSIS RESULTS

The numerical analysis carried out for forward curved blade type centrifugal blower using Ansys CFX software is presented herewith. This section presents qualitative and quantitative simulation results of the flow in forward curved blade type centrifugal blower with 0.5 m3/s discharge, 2800 rpm rotational speed and 16 numbers of blades in impeller. Efficient energy transfer in a centrifugal blower depends upon proper blade profile, gradual change in area of volute casing and smooth surface finish. For such energy transfer Flow lines must be parallel to each other and should generate streamlined flow within guided three dimensional passages.

Smooth and parallel streamlines within and around impeller region confirms well guided path for flow offered by this design. Flow leaves impeller smoothly to enter in volute casing. Volute casing progressively transmit flow up to outlet of the centrifugal blower. The flow just before tongue is recirculating towards impeller and after tongue it generates small vortex.

Following figure shows the streamline pattern for forward curved centrifugal blower with 0.5 m3/s discharge, 2800 rpm rotational speed and16 number of blades in impeller

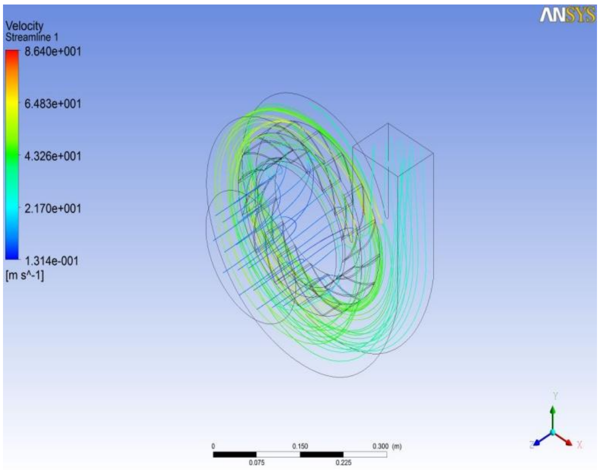
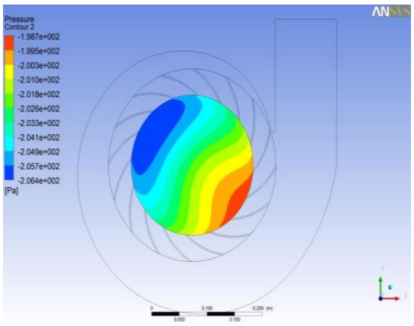
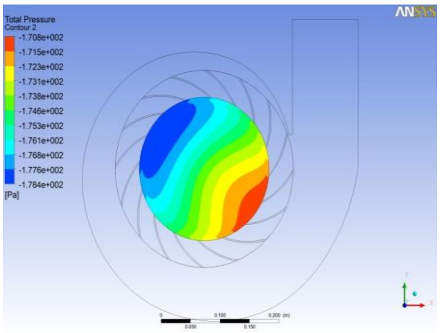
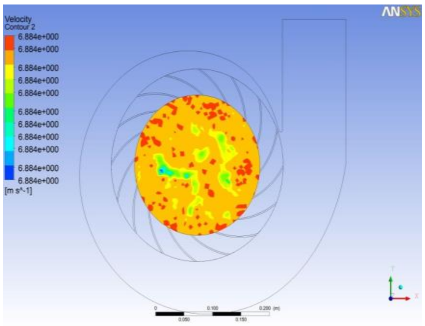


Figure: Streamlines of Designed Centrifugal Blower

Contours of static pressure, total pressure and velocity magnitude at inlet of inlet duct of forward curved blade centrifugal blower

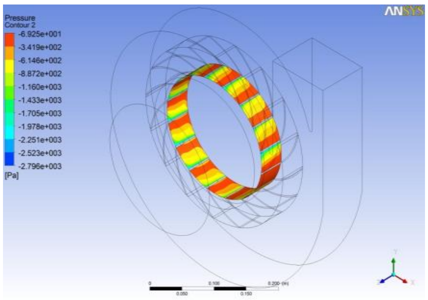


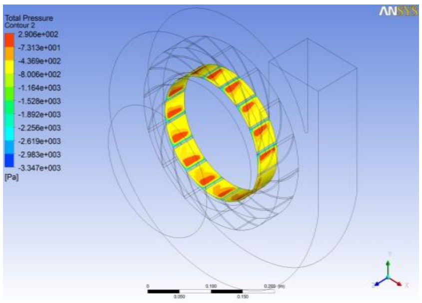


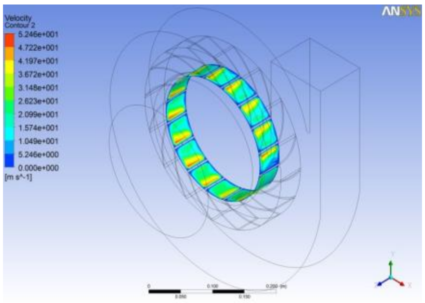


Range for static pressure = -206.4 to -198.7 Pa Average static pressure = -203.102 pa	Range for total pressure = -178.4 to -170.8 Pa Average total pressure = -175.039 pa	Range for velocity magnitude = 0 to 6.884 m/s Average velocity magnitude = 6.88213 m/s

Contours of static pressure, total pressure and velocity magnitude at inlet of impeller of forward curved blade centrifugal blower







Range for static pressure = -2796 to –69.25 Pa

Average static pressure = -482.403 pa

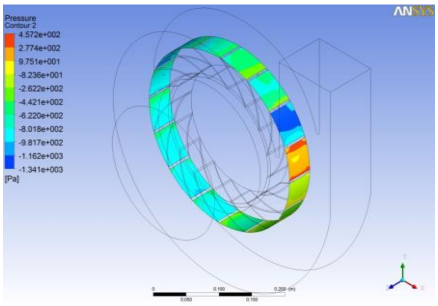
Range for total pressure = -3347 to 290.6Pa

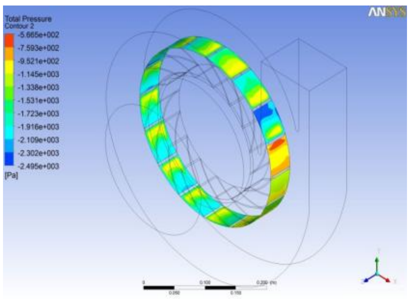
Average total pressure = -491.022 pa

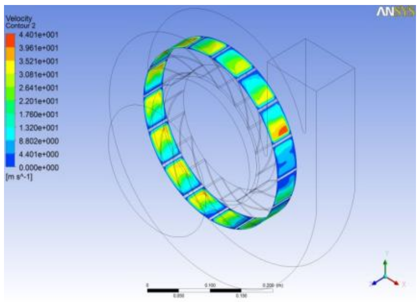
Range for velocity magnitude = 0 to 52.46 m/s

Average velocity magnitude = 22.8954 m/s

Contours of static pressure, total pressure and velocity magnitude at outlet of impeller of forward curved blade centrifugal blower







Range for static pressure = 457.2 to -1341 Pa

Average static pressure = -613.294 pa

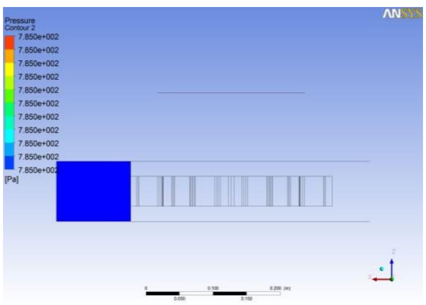
Range for total pressure = -2495 to -56.65Pa

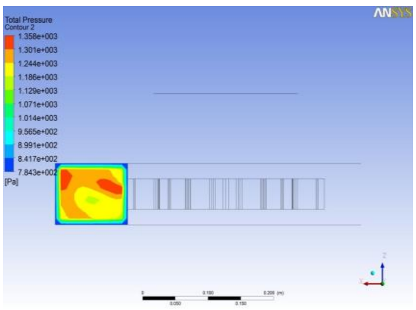
Average total pressure = -1421.74 pa

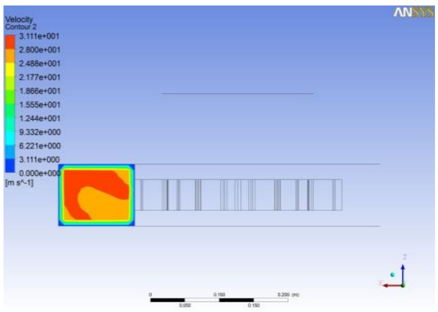
Range for velocity magnitude = 0 to 44.46 m/s

Average velocity magnitude = 27.6072 m/s

Contours of static pressure, total pressure and velocity magnitude at outlet of casing of forward curved blade centrifugal blower







Range for static pressure = 457.2 to -1341 Pa

Average static pressure = -613.294 pa

Range for static pressure = 457.2 to -1341 Pa

Average static pressure = -613.294 pa

Range for static pressure = 457.2 to -1341 Pa

Average static pressure = -613.294 pa

Taken input data for design of blower,

i.e. Discharge = 0.5 m3/s, Differential pressure =981.2, Rotational speed = 2800 rpm,
Inlet pressure at inlet duct = -196.4 pa and Outlet pressure at casing = 784.8 pa

And simulation of flow in forward curved blade type blower at rotational speed = 2800 rpm and at discharge =0.5 m3/s gives following results,

i.e. Inlet pressure at inletduct = -203.102 pa, Outlet pressure at casing = 785.085 pa,
and Differential pressure = 988.187 pa

So we can say error between simulated results and in design of blower is $Error = \frac{988.187-981.2}{988.187} \times 100 =$ 0.707%

IV. CONCLUSION

1. The theoretical and numerical analysis (CFD) is closer to design point conditions in centrifugal blower under study.
2. The flow phenomenon of recirculation near tongue region is confirmed by numerical analysis as shown by stream line diagram.
3. Pressure pulsations are observed at impeller outlet near tongue region caused by obstruction of tongue. Hence design of tongue is very important in blower design to reduce back flow and recirculation.
4. The mean pressure distribution around the volute is not uniform even at the design flow rate. Jet and wakes are observed in the vicinity of tongue region.
5. The nature of curves obtained after simulation closely follows trend of standard fan performance curves
6. Low and high pressure regions along suction and pressure side respectively of a blade are visualized by numerical analysis. Energy transfer from impeller to fluid is also confirmed by pressure and velocity contours within blade passage.

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