CHAPTER – 3

CENTRIFUGAL FAN DESIGN METHODOLOGIES

3.1 Introduction

Centrifugal fans and blowers are the turbo machines widely used in present industrial and domestic life. It is important to recognize that the design of any turbo machine 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. Manufacturing industries of fans and blowers seldom followed optimum design solution for individual fan/blower. Mostly their design and fabrication is based on series of successful past models or derived from fan laws and geometrical similarities.

During extensive literature review on design and performance evaluation of pumps, blowers and fans, it is observed that much research work has been carried out on local flow physics, aerodynamics and phenomena of energy transfer. Designing of these turbo machines requires computation with many variables and coefficients. It is also studied that the design methodologies suggested by different researchers differ widely. It has revealed lacuna of explicit design methodology which can give desired performance.

One of the objectives of present study is to design and analyze performance of explicit design methodologies as suggested in literature. Three systematic design methodologies for centrifugal fan are traced out after comprehensive literature review. Focusing on these three design methods, comparative assessment is made mathematically and then experiments are carried out to get optimum design solution. These design methodologies are summarized as under and discussed in detail subsequently.

3.2 Input Parameters for Centrifugal Fan Design

This research work is based on an industrial requirement for Fume Extraction Fan of SDS-9 texturising machine. Here variable flow is required at constant head under dust laden conditions. Radial blades are ideal for dust laden air or gas because they are less prone to blockage, dust erosion and failure and have self cleaning properties as observed by Mohamed Ali [20]. It has ideal zero slope in H-Q (head-discharge) curve to give variable discharge at constant head [9]. R. Ajithkumar [22] observed that the advantages of radial fans and blowers are having more stable operating range, ease of manufacturing with lower manufacturing and maintenance cost and high pressure development per stage. Wosika L. R. [37] has experimentally verified superior performance of radial vaned impellers.

Radial blades have lower unit blade stress for a given diameter and rotational speed hence lighter in weight. There is equal energy conversion in impeller and diffuser so it gives high-pressure ratio with good efficiency [9, 20]. Looking to these realization and facts radial blade fan is selected for this study.

The input parameters for the design of radial tipped centrifugal fan for fume extraction from SDS-9 texturing machine are summarized below.

Flow Discharge Q = 0.5 m^3/s

Static Suction Pressure = -196.4 N/m^2

Static Delivery Pressure = 784.8 N/m^2

Static Pressure Gradient ΔPs = 981.2 Pa

Speed of impeller rotation N = 2800 rpm

Air Density = 1.165 kg/m^3

Optimized number of blades z = 16 [135]

Outlet Blade Angle $\beta_2 = 90°$

Suction Temperature Ts = 30 °C = 303 K

Atmospheric Pressure $P_{atm} = 1.01325 \times 10^5$ Pa

Atmospheric Temperature T_{atm}= 30° C = 303° K

These parameters are kept identical for each design methodology prescribed here in.

3.3 Design of Radial Tipped Centrifugal Fan as per Fundamental Concepts [9, 10, 14, 28]

This design procedure is based on the fundamental principles of fluid flow with continuity and energy equations. The design follows the path from suction to discharge. To accelerate the flow at impeller inlet, converging section is designed after inlet duct. Energy balance is established at fan inlet, intermediate stage of impeller and outlet stage of volute/scroll casing.

During this process stage velocity, pressure and discharge at different stages are calculated. Flat front and back shrouds are selected for ease of impeller fabrication. Design procedure and calculations for above referred input parameters are presented below:

3.3.1 Design of Impeller

3.3.1.1 Impeller eye and inlet duct size

Let inlet duct size be 10% higher than impeller eye size or impeller inlet diameter. This will make conical insertion of inlet duct and flow acceleration at impeller eye or inlet.

$$\therefore D_{duct} = 1.1D_{eye} = 1.1D_1$$

Assuming no loss during 90º turning from eye inlet to impeller inlet, the eye inlet velocity vector will remain same as absolute velocity vector at the entry of impeller.

$$i.e. V_{eye} = V_1 = V_{ml}$$

Further let tangential velocity component be 10% higher than axial velocity component for better induction of flow.

So, Inlet Tip velocity

$$U_1 = 1.1V_1 = 1.1V_{ml}$$

$$Discharge, Q = \frac{\pi}{4} D_{eye}^2 \times V_1$$

$$Q = \frac{\pi}{4} \times (D_1)^2 \times V_1$$

$$V_1 = \frac{4Q}{\pi \times D_1^2}$$

$$U_1 = \frac{\pi D_1 N}{60} = 1.1V_1$$

$$\therefore \frac{\pi D_1 N}{60} = 1.1 \frac{4Q}{\pi D_1^2}$$

Here Q=0.5 m^3/s and speed of impeller rotation N=2800 rpm,

∴Impeller Inlet Diameter

D_1=0.168 m = D_{eye}

∴ Peripheral speed at inlet

$$U_1 = \frac{\pi D_1 N}{60} = 24.63\text{m/s} = 1.1V_1$$

$$V_1 = 22.45 \text{ m/s} = V_{m1} = V_{eye}$$

$$D_{duct} = 1.1D_1 = 1.1 \times 0.168 = 0.185\, m = 185\, mm$$

$$W_1 = \sqrt{U_1^2 + V_1^2} = \sqrt{24.63^2 + 22.45^2} = 33.37\, m/s$$

3.3.1.2 Impeller inlet blade angle (Refer Figure 3.5 for inlet velocity triangle)

$$\tan \beta_1 = \frac{V_1}{U_1} = \frac{22.45}{24.63}$$

$$\beta_1 = 42.35°$$

3.3.1.3 Impeller width at inlet

Here Z=16 and assumed blade thickness t = 2 mm

$$Q = [\pi D_1 - Zt] \times b_1 \times V_{ml}$$

$$0.5 = [(\pi \times 0.168) - (16 \times 2 \times 10^{-3})] \times b_1 \times 22.45$$

$$b_1 = 0.045\, m = 45\, mm$$

3.3.1.4 Impeller outlet parameters

The Fan Power $=\Delta P \times Q = 981.2 \times 0.5 = 490.6\, W$

Considering 10% extra to accommodate flow recirculation and impeller exit hydraulic losses.

So $1.1 \times the\, fan\, power = 1.1 \times 490.6 = 539.66\, W$

$$Power, P = \dot{m} \times W_s$$

$$\therefore \text{ Specific } Work\ done, W_s = \frac{539.66}{1.165 \times 0.5} = 926.45\, W/(\text{kg / s})$$

$$Eulerpower = \dot{m} V_{u2} U_2$$

$Taking\ V_{u2} = 0.8U_2$ (Assuming slip factor = 0.8 for radial blades) [9]

$$539.66 = 1.165 \times 0.5 \times 0.8U_2 \times U_2$$

$$U_2 = 34.03\, m/s$$

$$V_{u2} = 0.8 \times 34.03 = 27.22\, m/s$$

$$and\ U_2 = \frac{\pi D_2 N}{60}$$

$$\therefore D_2 = 0.232\, m$$

Taking width of blade at inlet = outlet blade width

$$\therefore b_1 = b_2$$

$$Q = [\pi D_2 - zt] \times b_2 \times Vm_2$$

$$0.5 = [(\pi \times 0.232) - (16 \times 2 \times 10^{-3})] \times 0.045 \times Vm_2$$

$$Vm_2 = 16\, m/s$$

$$W_{\mathrm{U2}} = 0.2U_2 = 6.81\, m/s$$

$$W_2' = \sqrt{W_{U2}^2 + Vm_2^2} = \sqrt{(0.2U_2)^2 + Vm_2^2}$$

$$W_2' = \sqrt{6.81^2 + 16^2} = 17.39\, m/s$$

$$V_2' = \sqrt{V_{U2}^2 + Vm_2^2}$$

$$V_2' = \sqrt{27.22^2 + 16^2} = 31.58\, m/s$$

$$\tan\alpha_2' = \frac{Vm_2}{V_{U2}} = \frac{16}{27.22} = 0.59$$

$$\alpha_2' = 30.45°$$

(Refer Figure 3.5 for outlet velocity triangle)

3.3.2 Design of Volute Casing

Analyzing steady flow energy equation at inlet and exit:

$$\frac{P_1}{\rho_1} + \frac{1}{2}V_1^2 + gz_1 + \mathrm{W_s} = \frac{P_2}{\rho_2} + \frac{1}{2}V_4^2 + gz_2$$

Neglecting potential difference,

$$V_4^2 = \frac{-2[P_2 - P_1]}{\rho_f} + V_1^2 + 2W_s$$

$$= \frac{-2(981.2)}{1.165} + 22.45^2 + 2(926.45)$$

Hence casing outlet velocity, $V_4 = 25.93\, m/s$

$$Q = A_v V_4$$

Where A_v is Exit area of volute casing = $A_v = b_v(r_4 - r_3)$

Allowing for 5 mm radial clearance between impeller and volute tongue,

$$r_3 = \frac{D_2}{2} + 5 = \frac{232}{2} + 5 = 121\, mm$$

$$D_3 = 2 \times 121 = 242\, mm$$

Width of volute casing (b_v) is normally 2 to 3 times b_1 [26]

Let us take it 2.5 times. Hence

$$b_v = 2.5 \times b_2 = 2.5 \times 45 = 112.5\, mm$$

$$Q = A_v V_4$$

$$0.5 = b_v(r_4 - r_3) \times 25.93$$

$$0.5 = 0.1125(r_4 - 0.121) \times 25.93$$

$$r_4 = 292\ mm$$

$$D_4 = 2 \times 292 = 584\ mm$$

Now incremental volute angle with respect to increase in radius of casing

$$r_\theta = r_3 + \frac{\theta}{360} \times \Delta r$$

$$\Delta r = (r_{at\ 360°} - r_{at\ 0°})$$

$$r_\theta = 0.121 + \frac{\theta}{360}(0.292 - 0.121)$$

The calculated volute radiuses at different volute angles are given in Table 3.1 as below.

Table 3.1 Radius of Volute at Different Angles

θ in Degree	Volute Radius r in m
0	0.121
60	0.150
120	0.178
180	0.207
240	0.235
300	0.264
360	0.292

Radius of Volute Tongue $r_t = 1.075 r_2 = 1.075 \times 0.116 = 0.125\ m$ [26]

Angle of Volute Tongue [26]

$$\theta_t = \frac{132 \log_{10}\left(\frac{r_t}{r_2}\right)}{\tan \alpha_2}$$

$$\theta_t = \frac{132 \times 0.0325}{0.59} = 7.26°$$

So angle of volute tongue = 7.26˚

3.3.3 Hydraulic, leakage and power Losses

3.3.3.1 Leakage loss [28]

$$Q_L = C_d \times \pi \times D_1 \times \delta \times \sqrt{\frac{2 \times P_s}{\rho}}$$

$$Here, P_s = \frac{2}{3}\Delta P_s$$

And coefficient of discharge C_d is 0.6 to 0.7,

$$\delta = Clearance\ between\ impeller\ eye\ inlet\ and\ casing$$

$= 2\ mm\ as\ per\ fabrication$ requirements.

$$= \pi \times 0.6 \times 0.169 \times 0.002 \times \sqrt{\frac{2 \times \left(\frac{2}{3}\right) \times 981.2}{1.165}}$$

$$= 0.0213\ m^3/s$$

3.3.3.2 Suction pressure loss [28]

$dp_{suc} = \frac{1}{2} \times k_i \times \rho \times V_{eye}^2$ Where k_i is a loss factor probably of the order of 0.5 to 0.8

$$= \frac{1}{2} \times 0.65 \times 1.165 \times 22.45^2$$

$$= 190.76\ Pa$$

3.3.3.3 Impeller pressure loss [28]

$$dP_{imp} = k_{ii} \times \frac{1}{2} \times \rho(W_1 - W_2')^2$$

At design point of maximum efficiency k_{ii} is in order of 0.2 – 0.3 for sheet metal blades and rather less for aerofoil section. Selecting $k_{ii} = 0.25$,

$$dP_{imp} = 0.25 \times \frac{1}{2} \times 1.165(33.37 - 17.39)^2$$

$$= 37.18\ Pa$$

3.3.3.4 Volute pressure loss [28]

$$dP_{VC} = k_{iii} \times \frac{1}{2} \times \rho(V_2' - V_4)^2$$

k_{iii} is of the order of 0.4 and will vary with deviation from design conditions.

$$= 0.4 \times \frac{1}{2} \times 1.165(31.58 - 25.98)^2$$

$$= 7.23\ Pa$$

3.3.3.5 Disc friction loss [28]

$$T_{df} = \pi f \rho \omega_2^2 \frac{r_2^5}{5} = \pi f \rho (U_2/r_2)^2 \frac{r_2^5}{5}$$

Where f is material friction factor in order of 0.005 for mild steel sheet metal [28]

$$T_{df} = \pi \times 0.005 \times 1.165 \times \frac{34.03^2}{0.116^2} \times \frac{0.116^5}{5}$$
$$\therefore T_{df} = 0.0066\ Nm$$

Hence, Power loss due to Disc friction

$$P_{df} = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 0.0066}{60} = 1.94\ W$$

3.3.4 Efficiencies

3.3.4.1 Hydraulic efficiency

$$\eta_{hy} = \frac{\Delta P}{\Delta P + dp_{suc} + dp_{imp} + dp_{vc}}$$
$$\eta_{hy} = \frac{981.2}{981.2 + 196.13 + 43.47 + 10.58} = 0.8065$$
$$\eta_{hy} = 80.65\%$$

3.3.4.2 Volumetric efficiency

$$\eta_{vol} = \frac{Q}{Q + Q_L} = \frac{0.5}{0.5 + 0.0213} = 0.959$$
$$\eta_{vol} = 95.9\%$$

3.3.4.3 Total efficiency

$$\eta_{total} = \eta_{hy} \times \eta_{vol} = 0.8065 \times 0.959 = 0.7736$$
$$\eta_{total} = 77.36\%$$

3.3.5 Ideal shaft power required to run the fan

$$= \frac{\left(\Delta P + dp_{suc} + dp_{imp} + dp_{vc}\right)(Q + Q_L)}{\eta_{total}} + Power\ loss\ due\ to\ disk\ friction$$
$$= \frac{(981.2 + 196.13 + 43.47 + 10.58)(0.5 + 0.0213)}{0.7736} + 1.94$$
$$= 821.7\ W$$
$$So, Torque = \frac{821.7 \times 60}{2\pi \times 2800}$$
$$= 2.804\ Nm$$

3.3.6 Shaft diameter [136]

$$D_s = \sqrt[3]{\frac{16T \times Factor\ of\ safety}{\pi\tau}}$$

$$= \sqrt[3]{\frac{16 \times 2.804 \times 4}{\pi \times 343 \times 10^5}} = 0.0119\, m = 11.9\, mm$$

3.3.7 Blade profile

Blade Profile is made by tangent arc method [26]. When this method is used, the impeller is divided into a number of assumed concentric rings, not necessarily equally spaced between inner and outer radii. The radius R_b of the arc is defining the blade shape between inner and outer radii.

$$R_b = \frac{r_2^2 - r_1^2}{2[r_1 \cos\beta_1 - r_2 \cos\beta_2]}$$

$$R_b = \frac{1}{2} \times \frac{\left(\frac{0.232}{2}\right)^2 - \left(\frac{0.168}{2}\right)^2}{\left(\frac{0.168}{2} \cos 42.27 - 0\right)} = 0.0512\, m = 51.2\, mm$$

After obtaining preliminary dimension of fan based on input data, further iterations are made to get optimum design parameters. Next stage of iteration is made after adding leakage and pressure losses of previous iteration in required design point discharge and pressure head across the impeller stage as given in input data. This is done to accommodate possible hydraulic, discharge and power losses at different flow passages. This process is carried out till there is no marginal dimensional change in impeller outer diameter and other important parameters. Figure 3.1 shows iterative design procedure algorithm. Summary of designed calculation including few iterations are given in Table 3.2, while Figure 3.2 to 3.5 gives dimensional drawings of the fan designed as per fundamental design.

The Matlab program has been developed for this design calculation and is given in Annexure A.

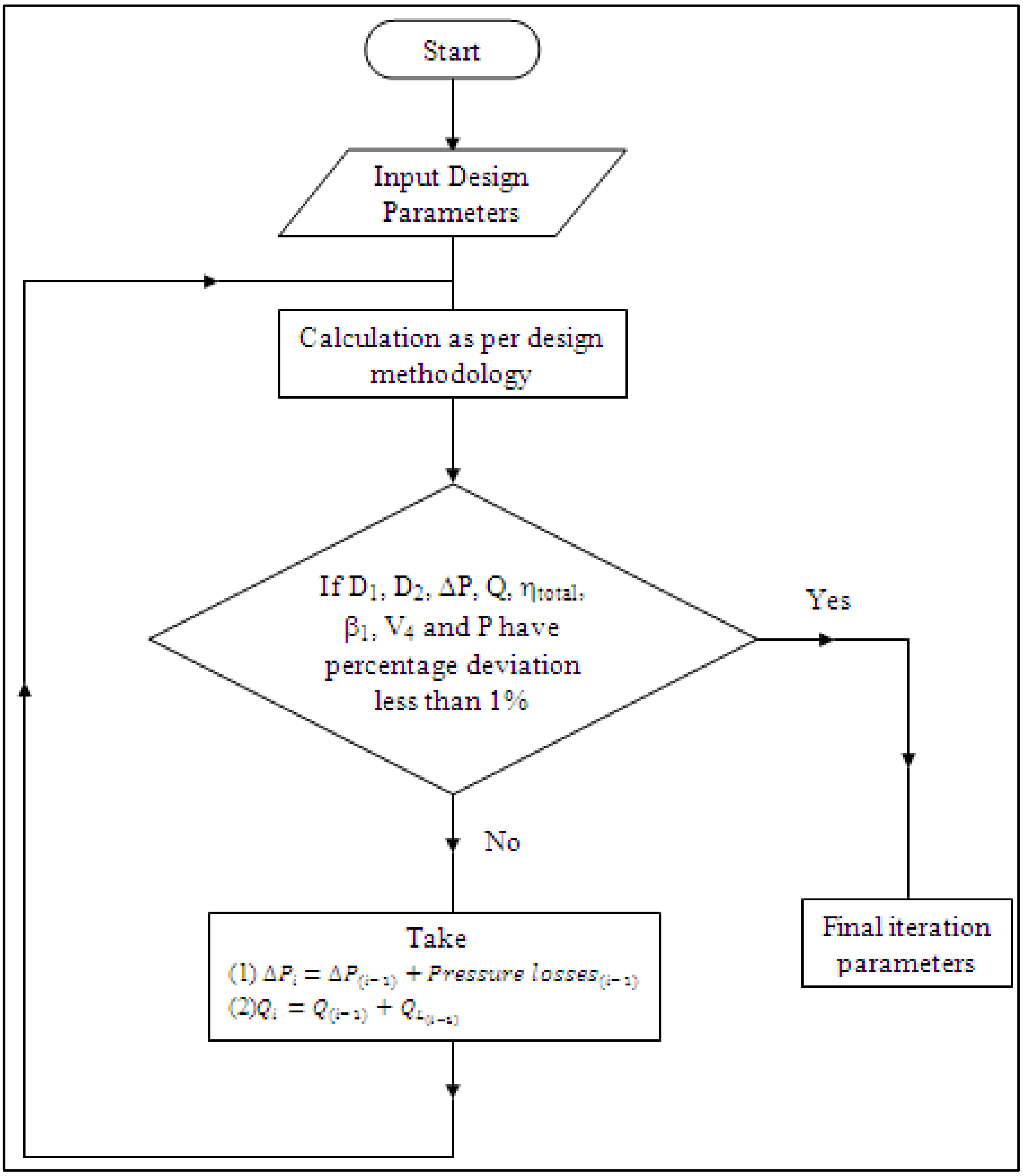

Figure 3.1 Iterative Design Process Algorithm

Table 3.2 Design Parameters as per Fundamental Design

Parameters as per Fundamental Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
At Impeller Inlet							
Inlet Duct Diameter	D_{duct}	mm	185	188	188	188	188
Eye Diameter	D_{eye}	mm	168	171	171	171	171
Eye Velocity	V_{eye}	m/s	22.45	22.76	22.80	22.80	22.80
Peripheral Velocity	U_1	m/s	24.63	25.04	25.08	25.08	25.08
Relative Velocity	W_1	m/s	33.37	33.83	33.89	33.90	33.90
Meridian Velocity	Vm_1	m/s	22.45	22.76	22.80	22.80	22.80
Absolute Velocity	V_1	m/s	22.45	22.76	22.80	22.80	22.80
Impeller Diameter	D_1	mm	168	171	171	171	171
Width of Blade	b_1	mm	45	45	45	45	45
Air Angle	α_1	Degree	90	90	90	90	90
Blade Angle	β_1	Degree	42.35	42.35	42.35	42.35	42.35
At Impeller Outlet							
Peripheral Velocity	U_2	m/s	34.03	37.89	38.12	38.14	38.14
Relative Velocity	W_2'	m/s	17.39	16.56	16.54	16.54	16.54
Swirl Velocity	V_{U2}'	m/s	27.22	30.31	30.50	30.51	30.52
Meridian Velocity	Vm_2	m/s	16.00	14.72	14.68	14.68	14.68
Absolute Velocity	V_2'	m/s	31.58	33.70	33.85	33.86	33.86
Impeller Diameter	D_2	mm	232	258	260	260	260
Width of Blade	b_2	mm	45	45	45	45	45
Air Angle	α_2'	Degree	30.45	25.90	25.71	25.69	25.69
Blade Angle	β_2	Degree	90°	90°	90°	90°	90°
At Volute/Scroll Casing							
Width of Casing	b_v	mm	112.5	114	114	114	114

Parameters as per Fundamental Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
Outlet Velocity of Casing	V_4	m/s	25.93	26.96	27.04	27.05	27.05
Scroll Radius at Inlet	r_3	mm	121	134	135	135	135
Scroll Radius at Outlet	r_4	mm	292	305	305	306	306
Scroll Height	Hs	mm	172	170	170	170	170
Radius of Tongue	r_t	mm	125	139	140	140	140
Angle of volute Tongue	θ_t	Degree	7	9	9	9	9
Pressure, Leakage and other Losses							
Pressure Losses in Suction	dP_{suc}	Pa	190.76	196.13	196.82	196.87	196.87
Pressure Losses in Impeller	dP_{im}	Pa	37.18	43.47	43.84	43.87	43.88
Pressure Losses in Scroll Casing	dP_{VC}	Pa	7.23	10.58	10.80	10.81	10.82
Leakage Loss	Q_L	m^3/s	0.0213	0.0240	0.0242	0.0242	0.0242
Disc Friction Torque	T_{df}	Nm	0.006626	0.011342	0.011691	0.01172	0.01172
Power Loss due to Disc Friction	P_{df}	W	1.94	3.33	3.43	3.44	3.44
Efficiencies							
Hydraulic Efficiency	η_{hy}	%	80.65	82.94	83.04	83.05	83.05
Volumetric Efficiency	η_{vol}	%	95.92	95.59	95.58	95.58	95.58
Total Efficiency	η_{total}	%	77.36	79.29	79.38	79.38	79.38
Other Parameters							
Pressure Head Across the Stage with Losses	ΔP	Pa	981.2	1216.6	1231.4	1232.8	1232.8

Parameters as per Fundamental Design		Unit	Initial Calculations	1^{st} iteration	13^{th} iteration	14^{th} iteration	Final iteration
Flow Discharge with Losses	Q	m^3/s	0.500	0.521	0.524	0.524	0.524
Blade Profile Radius	R_b	mm	51.2	74.5	75.8	75.9	75.9
Shaft Diameter	D_s	mm	11.9	11.8	11.8	11.8	11.8
Power Required to Run Fan	P	W	821.7	817.2	817.5	817.5	817.5

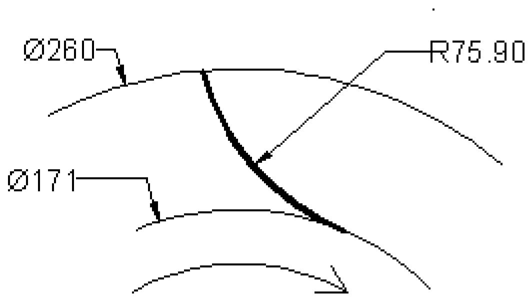

Figure 3.2 Blade Profile as per Fundamental Design

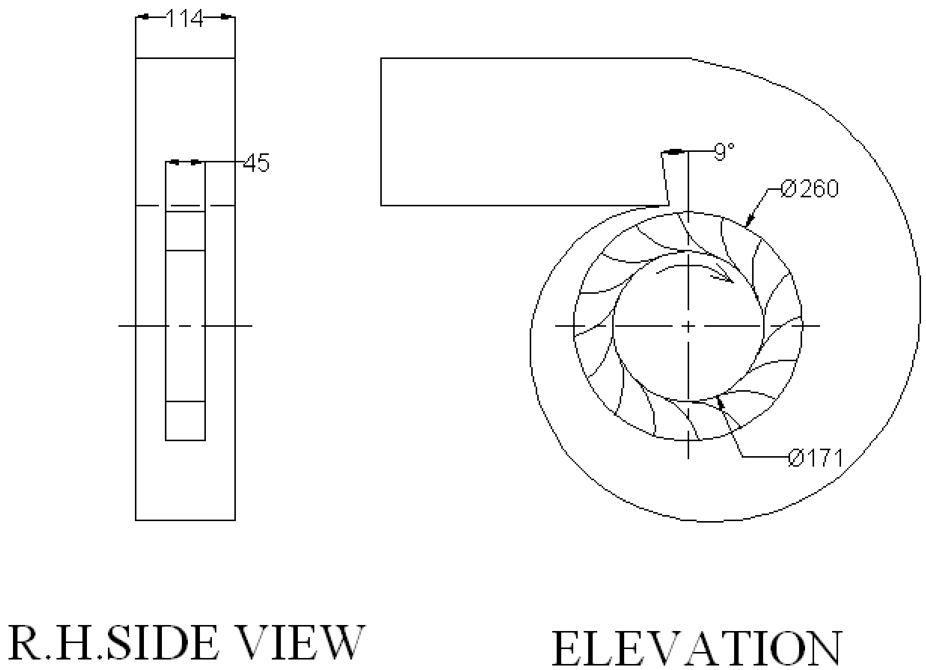

Figure 3.3 Impeller and Volute Casing Assembly as per Fundamental Design

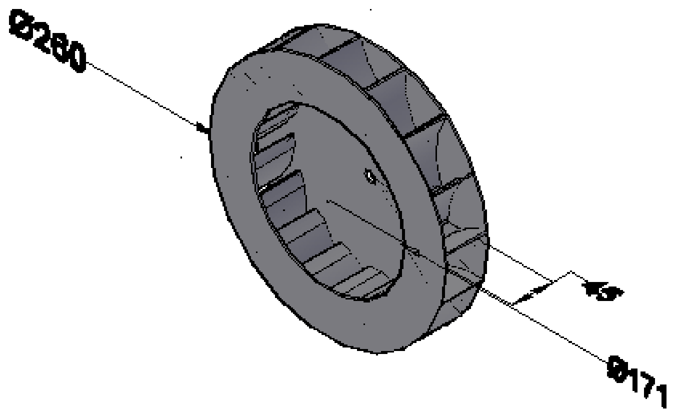

Figure 3.4 FCRT Impeller as per Fundamental Design

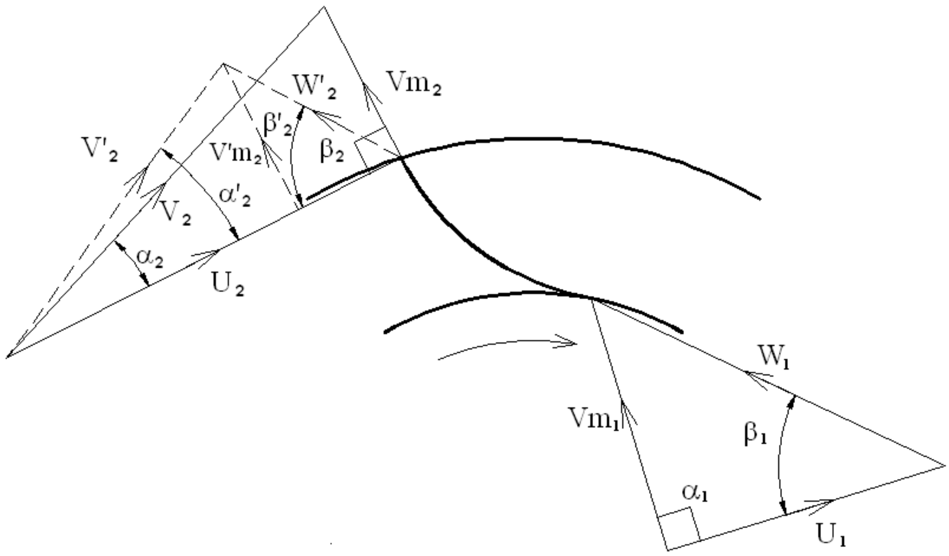

Figure 3.5 Theoretical and Actual Velocity Triangles as per Fundamental Design

3.4 Design of Radial Tipped Centrifugal Fan as Suggested By Austin Church [9, 26, 28]

Austin Church has done pioneering work to establish design methodology for pumps and blowers. He has presented his design with stage compressibility effect. He has also considered density changes at various flow sections with respect to change in temperature and pressure. Thus volume flow rate gets changed continuously. The dimensions of the air passage are calculated in accordance to this variation in volume flow. Stage pressure ratio between atmosphere to inlet eye, inlet eye to impeller inlet, impeller inlet to impeller outlet and impeller outlet to casing outlet are calculated individually. Church has used trapezoidal section which is difficult to fabricate for a single fan. Hence volute casing is designed as per principles of fundamental design as described earlier.

3.4.1 Effect of compressibility on design

In a centrifugal fan, air is compressible working fluid. If flow is steady, mass flow remains constant but density changes with respect to temperature and pressure. Hence volume flow Q gets changed. Taking compressibility effect into consideration:

Characteristic equation of gas is,

$$Density\ of\ atm.air, \rho_a = \frac{P_a}{RT_a}$$

$$= \frac{1.01325 \times 10^5}{287 \times 303} = 1.1652\ \frac{kg}{m^3}$$

$$Mass\ flow\ rate, \dot{m} = Q\rho_a = 0.5 \times 1.165 = 0.58259\ kg/s$$

3.4.2 Design of impeller

3.4.2.1 Impeller eye and inlet duct size

Assuming duct velocity at suction V_{duct} =16 m/s [26]

So, Diameter of suction duct

$$\mathrm{D_{duct}} = \sqrt{\frac{4Q}{\pi V_{duct}}}$$

$$= \sqrt{\frac{4 \times 0.5}{\pi \times 16}}$$

$$= 0.199\ m$$

The standard suction flange sizes are -1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 14, 16, 18 inch, Hence selecting Diameter of suction flange before duct $D_{suction}$ = 8 inch = 203 mm

Taking impeller eye velocity V_{eye} slightly higher than suction velocity= 18 m/s [26]

$$suction\ Velocity\ head\ at\ impeller\ inlet\ eye = \frac{V_{eye}^2}{2g}$$

$$= \frac{18^2}{2 \times 9.81} = 16.51\ m$$

$$Absolute\ Static\ Pressure\ at\ impeller\ eye, P_{\mathrm{eye}} = \mathrm{P_{atm}} - \mathrm{P_s} = 101325 - 196.5$$

$$P_{\text{eye}} = 1.01128.6 \times 10^5\ Pa$$

$$Pressure\ ratio\ of\ impeller\ eye\ to\ atmosphere = \frac{101128.6}{101325} = 0.9981$$

$$temperature\ at\ impeller\ eye, T_0 = T_a \times \epsilon_{peye}^{0.286}$$

$$= 303 \times 0.9981^{0.286}$$

$$T_0 = 302.83\ K$$

$$So, density\ at\ impeller\ eye, \rho_0 = \frac{P_0}{RT_0}$$

$$= \frac{1.011286 \times 10^5}{287 \times 302.83}$$

$$= 1.1636\ kg/m^3$$

$$Volume\ flow\ rate\ at\ impeller\ eye = {}^{\dot{m}}/_{\rho_0} = {}^{0.58259}/_{1.1636} = 0.50069\ m^3/s$$

Duct diameter is 0.199 m and impeller eye velocity is assumed 18 m/s.

Impeller eye diameter $= \sqrt{(4/\pi)} \times \sqrt{Q_{eye}/V_{eye}}$, it may be sloped slightly.

$$\mathrm{D_{eye}} = \sqrt{\left(\frac{4}{\pi}\right)} \times \sqrt{\frac{0.50069}{18}}$$

$$Impeller\ eye\ diameter, \mathrm{Deye} = 0.188\ m$$

3.4.2.2 Impeller inlet parameters

The main diameter at impeller inlet is made slightly greater than impeller eye diameter [26]. Let us take

$$1.02\mathrm{D_{eye}} = \mathrm{D_1} = 0.191\ \mathrm{m}$$

$$Peripheral\ velocity\ at\ inlet, U_1 = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.191 \times 2800}{60} = 28\ m/s$$

Now, impeller inlet velocity is assumed to be radial it means, it is slightly higher than V_{eye} [26],

$$\therefore V_1 = Vm_1 = 1.05 * \mathrm{V_{eye}} = 18.99\ \mathrm{m/s}$$

From inlet velocity triangle (Refer Figure 3.9 for inlet velocity triangle)

$$\tan \beta_1 = \frac{Vm_1}{U_1} = \frac{18.99}{28}$$

This should be increased 3% to care for the contraction of stream at inlet [26].

$$\beta_1 = 1.03 \times \tan^{-}(18.99/28) = 35.17°$$

Now, impeller inlet relative velocity, $W_1 = (U_1^2 + V_1^2)^{\frac{1}{2}}$

$$= (28^2 + 18.99^2)^{\frac{1}{2}}$$

$$= 33.84\ m/s$$

While calculating impeller inlet area, the flow must be increased because of leakage loss assumed to be 3%.

Impeller inlet area

$$A_1 = \frac{1.03 \times Q_0}{V_1} = \frac{1.03 \times 0.50069}{18.99} = 0.0272\ m^2$$

Assuming a vane thickness correction factor of έ = 0.925, the impeller inlet width:

$$b_1 = \frac{A_1}{\pi D_1 \varepsilon_1'} = \frac{0.0272}{\pi \times 0.191 \times 0.925} = 0.049\ m = 49\ mm$$

This width will be checked later when the number of vanes and their thickness have been fixed.

3.4.2.3 Impeller outlet parameters

Pressure ratio across the impeller, $\epsilon_P = \left({}^{102109.8}/_{101128.6}\right) = 1.00970$

Energy transfer by impeller$= \dot{m} g H_{ad} = \dot{m} C_p (T_2 - T_0)$

$\therefore$ Total adiabatic head across the impeller, $H_{ad} = (RT_0/0.286g)(\epsilon_P^{0.286} - 1)$

$$H_{ad} = \frac{287 \times 302.83}{0.286 \times 9.81}(1.00970^{0.286} - 1)$$

$$H_{ad} = 85.67\ m$$

Overall head to be developed by the centrifugal impeller by energy transfer is:

$$overall\ head\ to\ be\ developed, H_{ad} = \frac{K' U_2^2}{g}$$

$$where\ U_2 = \frac{\pi D_2 N}{60}$$

K' coefficient is used on account of the friction and turbulence occurring in the impeller. Church has found value of K' = 0.5 to 0.65 by experiments. The outside diameter of the impeller can be calculated by using following equation after assuming proper value of K'. Let us take K' = 0.6.

$$\therefore D_2 = \frac{60}{\pi N}\sqrt{\frac{gH_{ad}}{K'}} = \frac{60}{\pi \times 2800}\sqrt{\frac{9.81 \times 85.67}{0.6}} = 0.255\ m$$

Hence U_2 = 37.43 m/s

Radial outlet velocity Vm_2 is taken 15% less than Vm_1. This is due to sudden changes in velocity occurring within impeller passage.

$$So, Vm_2 = 0.85Vm_1 = 0.85 \times 18.99 = 16.14\, m/s = W_{2\, without\, slip}$$

Now, due to blade passage circulation effect (slip):

$$Vu_2' = U_2 - \frac{\pi U_2 \sin\beta_2}{Z} = U_2 - \frac{3.14 U_2}{16} = 0.80375 U_2 = 30.08\, m/s$$

$$Wu_2 = U_2 - Vu_2' = 7.35\, m/s$$

From velocity triangle,

$$Absolute\ velocity\ at\ outlet, V_2' = \sqrt{Vm_2^2 + Vu_2^2} = 34.14\, m/s$$

$$\tan\alpha_2' = \frac{Vm_2}{Vu_2'}$$

$$\alpha_2' = 28.22°$$

(Refer Figure 3.9 for outlet velocity triangle)

Now, outlet relative velocity with slip

$$W_2' = \sqrt{W_{U2}^2 + Vm_2^2} = 17.74 \frac{\mathrm{m}}{\mathrm{s}}$$

Virtual head developed by impeller

$$H_{virtual} = \frac{1}{2g}(U_2^2 - U_1^2 + W_1^2 - W_2^2) = 0.051(37.42^2 - 28^2 + 33.84^2 - 16.14^2)$$

$$H_{virtual} = 76.49\, m$$

Where

$$\frac{U_2^2 - U_1^2}{2} \rightarrow Head\ due\ to\ centrifugal\ action$$

$$\frac{W_1^2 - W_2^2}{2} \rightarrow Head\ due\ to\ change\ in\ relative\ velocities$$

It may be assumed that, owing to the circulatory flow, friction and turbulence in the impeller, 15 percent of this virtual head is lost [26].

Hence effective head $H_{effective} = 0.85 H_{virtual} = 0.85 \times 76.49 = 65.02\, m$

Impeller outlet pressure is based upon effective head so,

$$\therefore \epsilon_{Pimp}^{0.286} - 1 = \frac{0.286 g H_{effective}}{RT_0} = \frac{0.286 \times 9.81 \times 65.02}{287 \times 302.83} = 0.0021$$

Where ϵ_{Pimp} = Pressure ratio between impeller eye and impeller outlet base upon effective head.

$$\therefore \epsilon_{Pimp} = 1.00736$$

Impeller outlet pressure, $P_2 = \epsilon_{Pimp} \times P_0 = 1.00736 \times 101128.6 = 101872.6\, Pa$

The friction and turbulence losses will be transformed into heat which raises the temperature of air. The impeller outlet temperature may be based upon the adiabatic head in the impeller, neglecting losses.

$$\therefore \epsilon_{Pimp}^{0.286} - 1 = \frac{0.286 g H_{adiabatic}}{RT_0} = \frac{0.286 \times 9.81 \times 85.67}{287 \times 302.83} = 0.00276$$

$$\text{The outlet temperature}, T_2 = \epsilon_{Pimp}^{0.286} \times T_0 = 303.58\ K$$

$$\text{Impeller outlet density}, \rho_2 = \frac{P_2}{RT_2} = \frac{101872.6}{287 \times 303.58} = 1.1692$$

Considering 3% leakage loss and increasing flow at impeller exit by this 3%.

$$\text{Impeller exit volume flow,} \qquad Q_{out} = \frac{1.03 \times 0.58259}{1.1692} = 0.51321\ m^3/s$$

$$\text{Impeller outlet area}, A_2 = {Q_{out}}/{Vm_2} = {0.51321}/{16.14} = 0.031794\ m^2$$

$$\text{Outlet vane thickness factor}, \acute{\varepsilon}_2 = \frac{(\pi D_2 - Zt)/\sin\beta_2}{\pi D_2} = 1 - \frac{16 \times 0.002}{\pi \times 0.255} = 0.9601$$

$$\text{Outlet vane thickness}, b_2 = \frac{A_2}{\pi D_2 \acute{\varepsilon}_2} = 0.04124\ m = 41.3\ mm$$

3.4.3 Design of volute casing

Church has recommended trapezoidal section. This is difficult to fabricate for laboratory experimentation purpose. Hence rectangle section is chosen here as suggested in fan testing code [17]. Here b_3 is the width of volute casing and it is taken as:

$$bv = b_3 = 3b_2 = 3 \times 0.0413 = 0.124 \text{ m}$$

Volute base circle/inlet radius $r_3 = r_2 + 5 = 127.5 + 5 = 133$ mm

In determining cross sectional area of the volute at any point, the problem consists in finding the area of the section that will pass the volume $Q_\phi/360$ with a velocity $V_u \times r = c$. If friction is neglected, the flow through the differential section is:

$$dQ_\phi = b \times dr \times c/r$$

The total flow past the section becomes:

$$Q_\phi = \int_{r_2}^{r_\phi} DQ = c\int_{r_2}^{r_\phi} \frac{bdr}{r}$$

$$\therefore Instantaneous\ angle\ \emptyset^0 = \frac{360 r_2 V u_2}{Q_{out}} \int_{r_2}^{r_\theta} b\frac{dr}{r}$$

$$\emptyset^0 = \frac{360 \times 0.1275 \times 30.08}{0.51321} \int_{R2}^{R\theta} b\frac{dR}{R}$$

Volute casing design based on above equation is tabulated below in Table 3.3 [26].

Table 3.3 Volute Casing Design Parameters as per Church Design Method

R in m	ΔR	R_{ave} m	B_{ave} =(3*b_2) m	B ΔR/ R_{ave}	Δφ°	φ°	ΔA m^2	A m^2	Q x φ° / 360 m^3/s	V_{ave} m/s
0.1275						0		0	0.0000	0.00
	0.023	0.1388	0.124	0.02009	54.04		0.0028			
0.1500						54.04		0.0028	0.0751	26.93
	0.025	0.1625	0.124	0.01906	51.27		0.0031			
0.1750						105.3		0.0059	0.1463	24.86
	0.025	0.1875	0.124	0.01652	44.43		0.0031			
0.2000						149.7		0.009	0.2080	23.16
	0.025	0.2125	0.124	0.01457	39.21		0.0031			
0.2250						188.9		0.0121	0.2624	21.73
	0.025	0.2375	0.124	0.01304	35.08		0.0031			
0.2500						224		0.0152	0.3111	20.50
	0.025	0.2625	0.124	0.01180	31.74		0.0031			
0.2750						255.8		0.0183	0.3552	19.44
	0.025	0.2875	0.124	0.01077	28.98		0.0031			
0.3000						284.7		0.0214	0.3955	18.51
	0.025	0.3125	0.124	0.00991	26.66		0.0031			
0.3250						311.4		0.0245	0.4325	17.68
	0.025	0.3375	0.124	0.00918	24.68		0.0031			
0.3500						336.1		0.0276	0.4668	16.94
	0.01	0.3550	0.124	0.00349	9.39		0.0012			
0.3600						345.5		0.0288	0.4798	16.66
	0.01	0.3650	0.124	0.00339	9.13		0.0012			
0.3700						354.6		0.03	0.4925	16.40
	0.01	0.3750	0.124	0.00330	8.89		0.0012			
0.3800						363.5		0.0313	0.5048	16.14
0.3764						360		0.0310	0.5000	15.99

The values of all other parameters are received for cumulative **φ°** angle of 354.6^0 and 363.5^0. For cumulative **φ°** angle of 360^0, values of V_{ave}, r_4, r_t and θ_t are received by interpolation as given below:

V_{ave} at 360 ^{o} is = 15.99 m/s= V_4

Radius of volute at angle 360° = r_4 =0.376 m = 376 mm

$$D_4 = 2 \times 376 = 752\ mm$$

Radius of volute tongue, $\mathrm{r_t} = 1.075\mathrm{r_2} = 1.075 \times 127.5 = 137$ mm

Volute tongue angle, $\theta_\mathrm{t} = 132 \log_{10}(r_t/r_2)/\tan\alpha_2 = 7.7°$

3.4.4 Hydraulic, leakage and power losses

3.4.4.1 Leakage loss [26]

$$Q_L = 0.03Q$$
$$= 0.03 \times 0.5$$
$$= 0.015\ m^3/s$$

3.4.4.2 Suction pressure loss [28]

Friction and turbulence loss in impeller eye, $dp_{suc} = \frac{1}{2} \times k_i \times \rho \times V_{eye}^2$

Where k_i is a loss factor probably of the order of 0.5 to 0.8

$$= \frac{1}{2} \times 0.65 \times 1.1636 \times 18^2$$
$$= 122.52\ Pa$$

3.4.4.3 Impeller pressure loss [28]

$$dP_{imp} = k_{ii} \times \frac{1}{2} \times \rho_{eye} \times (W_1 - W_2')^2$$

At design point of maximum efficiency k_{ii} is in order of 0.2 – 0.3 for sheet metal blades and rather less for aerofoil section. Selecting k_{ii}= 0.25

$$dP_{imp} = 0.25 \times \tfrac{1}{2} \times 1.1636 \times (33.84 - 17.74)^2$$
$$dP_{imp} = 37.82\ Pa$$

3.4.4.4 Volute pressure loss [28]

$$dP_{VC} = k_{iii} \times \frac{1}{2} \times \rho_2 \times (V_2' - V_4)^2$$

k_{iii} is of the order of 0.4 and will vary with deviation from design conditions.

$$= 0.4 \times \frac{1}{2} \times 1.1692 \times (34.14 - 15.99)^2$$
$$= 71.63\ Pa$$

3.4.4.5 Disc friction loss [28]

$$T_{df} = \pi f \rho \omega_2^2 \frac{r_2^5}{5} = \pi f \rho (U_2/r_2)^2 \frac{r_2^5}{5}$$

(Where f is material friction factor in order of 0.005 for mild steel sheet metal)

$$T_{df} = \pi \times 0.005 \times 1.1692 \times \frac{37.42^2}{0.1275^2} \times \frac{0.1275^5}{5}$$

$$\therefore T_{df} = 0.0107\ Nm$$

Hence, Power loss due to Disc friction

$$P_{df} = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 0.0107}{60} = 3.14\ W$$

3.4.5 Efficiencies

3.4.5.1 Hydraulic efficiency

$$\eta_{hy} = \frac{\Delta P}{\Delta P + dp_{suc} + dp_{imp} + dp_{vc}}$$

$$\eta_{hy} = \frac{981.2}{981.2 + 122.52 + 37.8 + 71.63} = 0.8088$$

$$\eta_{hy} = 80.88\%$$

3.4.5.2 Volumetric efficiency

$$\eta_{vol} = \frac{Q}{Q + Q_L} = \frac{0.5}{0.5 + 0.03 \times 0.5} = 0.9709 = 97.1\%$$

3.4.5.3 Total efficiency

$$\eta_{total} = \eta_{hy} \times \eta_{vol} = 0.8088 \times 0.971 = 0.7647 = 78.52\%$$

3.4.6 Ideal Shaft power required to run the fan

$$= \frac{\left(\Delta P + dp_{suc} + dp_{imp} + dp_{vc}\right)(Q + Q_L)}{\eta_{total}} + Power\ loss\ due\ to\ disk\ friction$$

$$= \frac{(981.2 + 122.52 + 37.8 + 71.63)(0.5 + 0.015)}{0.7647} + 3.14 \quad = 798.8\text{W}$$

$$So, Torque = \frac{798.8 \times 60}{2\pi \times 2800}$$

$$= 2.7257\ Nm$$

3.4.7 Shaft diameter [136]

$$D_s = \sqrt[3]{\frac{16T \times Factor\ of\ safety}{\pi\tau}}$$

$$= \sqrt[3]{\frac{16 \times 2.7257 \times 4}{\pi \times 343 \times 10^5}} = 0.0117\ m = 11.7\ mm$$

3.4.8 Blade profile

Blade Profile is made by tangent arc method [26]. When this method is used, the impeller is divided into a number of assumed concentric rings, not necessarily equally spaced between inner and outer radii. The radius R_b of the arc is defining the blade shape between inner and outer radii.

$$R_b = \frac{r_2^2 - r_1^2}{2[r_1 \cos\beta_1 - r_2 \cos\beta_2]}$$

$$R_b = \frac{1}{2} \times \frac{\left(\frac{0.255}{2}\right)^2 - \left(\frac{0.191}{2}\right)^2}{\left(\frac{0.191}{2} \cos 35.17 - 0\right)} = 0.045\ m = 46\ mm$$

After obtaining preliminary dimension of fan based on input data further iterations are made to get optimum design parameters. Next stage of iteration is made after adding leakage and pressure losses of previous iteration in required design point discharge and pressure head across the impeller stage as given in input data. This is done to accommodate possible hydraulic, discharge and power losses at different flow passages. This process is carried out till there is no marginal dimensional change in impeller outer diameter and other important parameters. An iterative design procedure algorithm is shown in Figure 3.1. Summary of designed calculation including few iterations are given in Table 3.4, while Figure 3.6 to 3.9 gives dimensional drawings of the fan designed as per Church design.

The Matlab program has been developed for this design calculation and is given in Annexure A.

Table 3.4 Design Parameters as per Church Design

Parameters as per Church Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
At Impeller Inlet							
Inlet Duct Diameter	D_{duct}	mm	199	199	199	199	199
Eye Diameter	D_{eye}	mm	188	188	188	188	188
Eye Velocity	V_{eye}	m/s	18.00	18.00	18.00	18.00	18.00
Peripheral Velocity	U_1	m/s	28.00	28.00	28.00	28.00	28.00
Relative Velocity	W_1	m/s	33.84	33.84	33.84	33.84	33.84
Meridian Velocity	Vm_1	m/s	18.99	18.99	18.99	18.99	18.99
Absolute Velocity	V_1	m/s	18.99	18.99	18.99	18.99	18.99
Impeller Diameter	D_1	mm	191	191	191	191	191
Width of Blade	b_1	mm	49	49	49	49	49
Air Angle	α_1	Deg.	90	90	90	90	90
Blade Angle	β_1	Deg.	35.17	35.17	35.17	35.17	35.17
At Impeller Outlet							
Peripheral Velocity	U_2	m/s	37.43	41.60	41.56	41.56	41.56
Relative Velocity	W_2’	m/s	17.74	18.09	18.09	18.09	18.09
Swirl Velocity	V_{U2}’	m/s	30.08	33.43	33.40	33.40	33.40
Meridian Velocity	Vm_2	m/s	16.14	16.14	16.14	16.14	16.14
Absolute Velocity	V_2’	m/s	34.14	37.12	37.10	37.10	37.10
Impeller Diameter	D_2	mm	255.28	283.74	283.50	283.50	283.50
Width of Blade	b_2	mm	41.24	36.90	36.93	36.93	36.93
Air Angle	α_2’	Deg.	28.22	25.77	25.79	25.79	25.79
Blade Angle	β_2	Deg.	90°	90°	90°	90°	90°
At Volute/Scroll Casing							
Width of Casing	b_v	mm	124	111	111	111	111

Parameters as per Church Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
Outlet Velocity of Casing	V_4	m/s	15.99	19.71	19.68	19.68	19.68
Scroll Radius at Inlet	r_3	mm	133	147	147	147	147
Scroll Radius at Outlet	r_4	mm	376	376	376	376	376
Scroll Height	Hs	mm	243	229	229	229	229
Radius of Tongue	r_t	mm	137	153	152	152	152
Angle of volute Tongue	θ_t	^{o}	8	9	9	9	9
Pressure, Leakage and other Losses							
Pressure Losses in Suction	dP_{suc}	Pa	122.52	122.52	122.52	122.52	122.52
Pressure Losses in Impeller	dP_{im}	Pa	37.82	36.20	36.21	36.21	36.21
Pressure Losses in Scroll Casing	dP_{VC}	Pa	71.63	71.09	71.12	71.12	71.12
Leakage Loss	Q_L	m^3/s	0.0150	0.0150	0.0150	0.0150	0.0150
Disk Friction Torque	T_{df}	Nm	0.010714	0.018199	0.01812	0.01812	0.01812
Power Loss due to Disk Friction	P_{df}	Watts	3.14	5.34	5.31	5.31	5.31
Efficiencies							
Hydraulic Efficiency	η_{hy}	%	80.88	84.07	84.05	84.05	84.05
Volumetric Efficiency	η_{vol}	%	97.09	97.17	97.17	97.17	97.17
Total Efficiency	η_{total}	%	78.52	81.70	81.67	81.67	81.67
Other Parameters							
Pressure Head Across the Stage with Losses	ΔP	Pa	981.2	1213.2	1211.0	1211.1	1211.1

Parameters as per Church Design		Unit	Initial Calculations	1^{st} iteration	13^{th} iteration	14^{th} iteration	Final iteration
Flow Discharge with Losses	Q	m^3/s	0.500	0.515	0.515	0.515	0.515
Blade Profile Radius	R_b	mm	45.9	70.5	70.3	70.3	70.3
Shaft Diameter	D_s	mm	11.7	11.6	11.6	11.6	11.6
Power Required to Run Fan	P	Watts	798.8	768.7	769.0	769.0	769.0

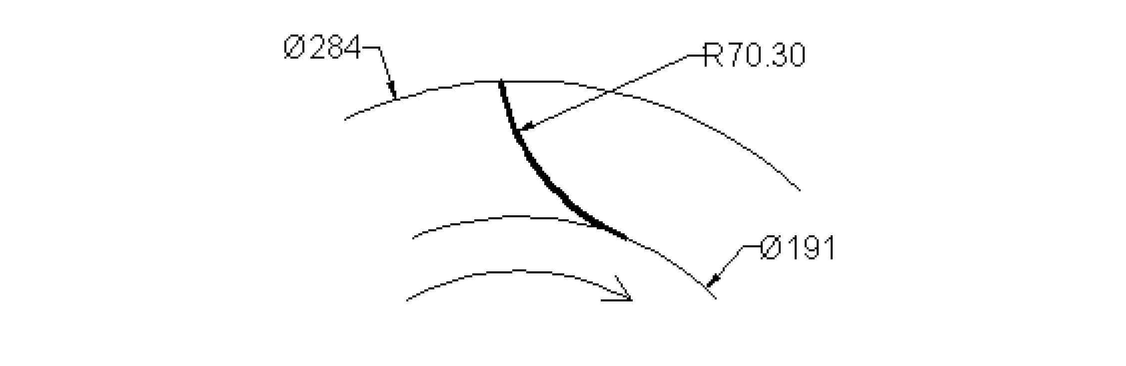

Figure 3.6 Blade Profile as per Church Design

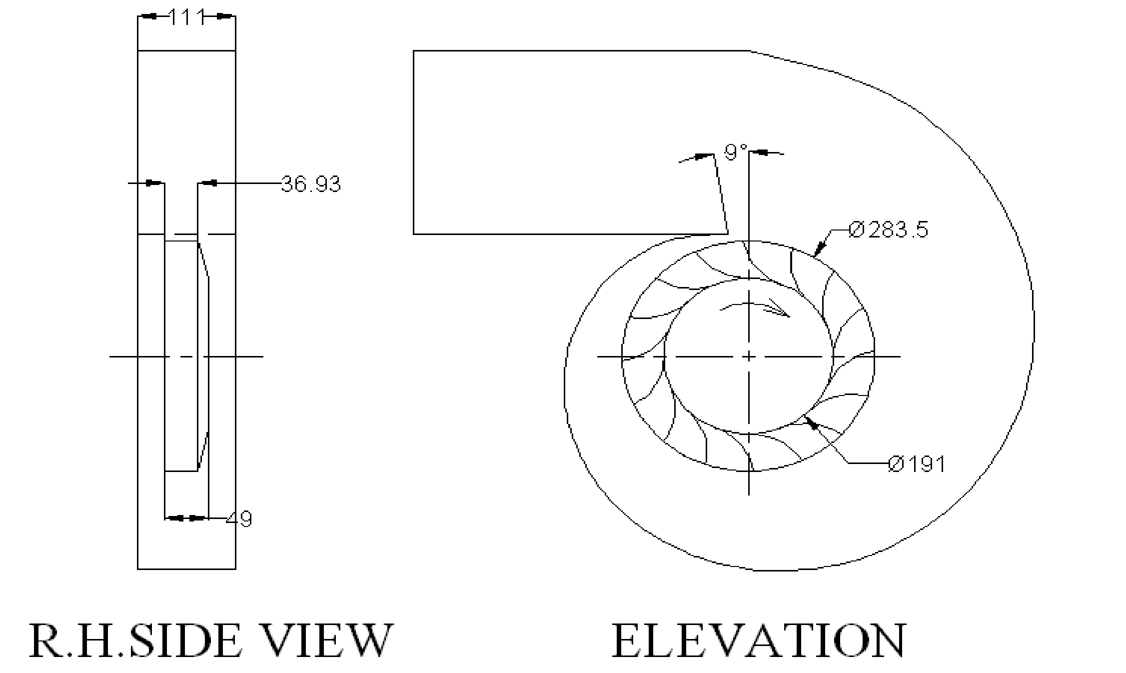

Figure 3.7 Impeller and Volute Casing Assembly as per Church Design

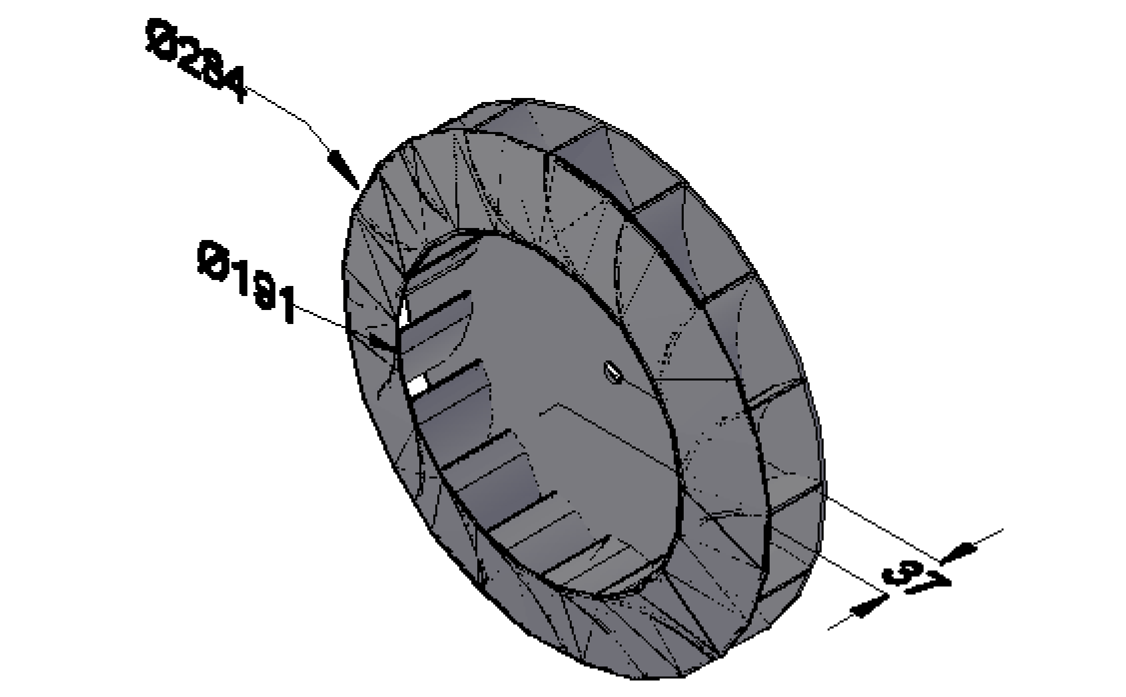

Figure 3.8 FCRT Impeller as per Church Design

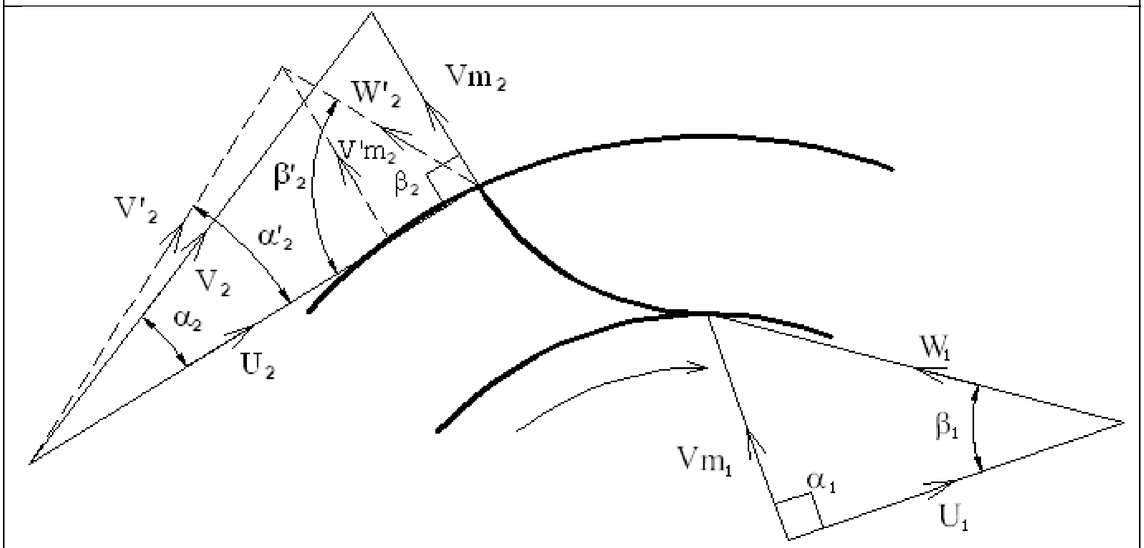

Figure 3.9 Theoretical and Actual Velocity Triangles as per Church Design

3.5 Design of Radial Tipped Centrifugal Fan as Suggested by William C. Osborne [9, 28]

William. C. Osborne has made very good attempt to use simple flow physics to design fans/blowers. He has used empirical relations for eye velocity, meridian velocity and casing velocity with respect to impeller tip peripheral velocity. Relative velocity is considered same for inlet and outlet conditions. This is one of the major limitations of this design. Suction, impeller, volute pressure losses and leakage losses are calculated separately. Design procedure and calculations for given input parameters are as per following.

Fan static pressure gradient, ΔP_s = 981.2 Pa.

$$\text{Fan exit velocity pressure} = \frac{1}{2}\rho V_4^2 \ (\text{Where } V_4 \text{ is casing outlet velocity})$$

$$\text{Fan total pressure developed in impeller as per Euler is } P = \rho U_2 Vu_2$$

$$\text{Due to inter blade circulation and considering slip, } P = \rho U_2 Vu_2'$$

3.5.1 Design of impeller

3.5.1.1 Impeller outlet parameters

Fan static pressure at exit P_{se} is fan total pressure minus the fan velocity pressure,

$$P_{se} = \rho U_2 Vu_2' - \frac{1}{2}\rho V_4^2$$

Vu_2 reduces to Vu_2' due to blade circulation known as slip in impeller,

$$\therefore Vu_2' = Vu_2 - \frac{\pi \sin\beta_2}{Z} U_2$$

$$= U_2 - \frac{\pi \sin\beta_2}{Z} U_2 \ (For\ radial\ blades, ideally\ U_2 = Vu_2)$$

$$\therefore P_{se} + \frac{1}{2}\rho V_4^2 = \rho U_2 \left[U_2 - \frac{\pi \sin\beta_2}{Z} U_2\right]$$

$$\therefore 981.2 + \frac{1}{2} \times 1.165 V_4^2 = \rho U_2^2 \left[1 - \frac{\pi}{16}\right]$$

Assuming initially outlet velocity of casing $V_4 = 0.3U_2$ [28]. Later it will be reconsidered as received from scroll casing design.

$$\therefore 981.2 + \frac{1}{2} \times 1.165(0.3U_2)^2 = 1.165U_2^2(0.80375)$$

$$\therefore 981.2 + 0.05242U_2^2 = 0.936U_2^2$$

$$Outlet\ peripheral\ velocity, U_2 = 32.87\ \frac{m}{s}$$

$$\text{and } Vu_2' = Vu_2 - \frac{\pi \sin\beta_2}{Z} U_2 = 27.10\ \text{m/s}$$

$$U_2 = \frac{\pi D_2 N}{60}$$

$$\therefore Impeller\ outlet\ diameter, D_2 = \frac{32.87 \times 60}{\pi \times 2800} 0.224\ m$$

$$= 224\ mm$$

Taking, $Vm_1 = Vm_2 = 0.2U_2 = 6.57$ m/s $= W_2$

From, outlet velocity triangle,

$$V_2' = \sqrt{Vm_2^2 + Vu'^2_2}$$

$$= \sqrt{6.57^2 + 27.10^2}$$

$$= 27.89\ m/s$$

$$\tan\alpha_2' = \frac{Vm_2}{Vu_2'} = \frac{6.57}{27.1} = 0.2424$$

$$\therefore \alpha_2' = 13.64°$$

3.5.1.2 Impeller inlet parameters

Ideally, the velocity through the impeller inlet/eye should not be more than V_{m1}. However, this cannot often be achieved since the inlet diameter becomes too large, and V_{eye} is generally of the order of twice V_{m1} or $0.4U_2$ [28].

Assuming $V_{eye} = 0.4U_2 = 13.15$ m/s

$$Impeller\ inlet\ diameter, D_1 = \sqrt{\frac{4Q}{\pi V_{eye}}}$$

$$= \sqrt{\frac{4 \times 0.5}{\pi \times 13.15}}$$

$$= 0.220\ m = 220\ mm$$

To avoid leakage between impeller eye and impeller inlet as being stationary and rotary geometry respectively, the gap must be minimum.

Taking $D_1 = 1.03\ D_{eye} = 1.03 \times 220 = 227$ mm

Note: Initial calculations are showing impeller inlet diameter D_1 large than impeller outlet diameter D_2. This is because of assuming less impeller eye velocity

V_{eye}. This will be corrected in further course of iterations by using other design parameters as received from successive design calculations.

$$Impeller\ Inlet\ peripheral\ velocity, U_1 = \frac{\pi D_1 N}{60}$$

$$= \frac{\pi \times 0.227 \times 2800}{60}$$

$$\therefore U_1 = 33.23\ m/s$$

$$Impeller\ inlet\ width, b_1 = \frac{Q}{\pi D_1 V m_1} = \frac{0.5}{\pi \times 0.227 \times 6.57} = 0.107\ m$$

$$Impeller\ outlet\ width, b_2 = \frac{Q}{\pi D_2 V m_2} = \frac{0.5}{\pi \times 0.224 \times 6.57} = 0.108\ m$$

From, inlet velocity triangle,

$$Inlet\ blade\ Angle, \beta_1 = \tan^{-1}\left(\frac{Vm_1}{U_1}\right) = \tan^{-1}\left(\frac{6.57}{33.23}\right) = 11.19°$$

(Refer Figure 3.13 for inlet and outlet velocity triangles)

$$Relative\ velocity\ at\ inlet, W_1 = \sqrt{Vm_1^2 + U_1^2} = \sqrt{6.57^2 + 33.23^2} = 33.87\ m/s$$

3.5.2 Design of volute casing

As assumed earlier, Casing Outlet Velocity, $V_4 = 0.3U_2$

$V_4 = 0.3 \times 32.87 = 9.861$ m/s

Width of Casing b_v for backward curved is 2.5 b_2, for forward curved it is 1.25 b_2 and for radial tipped blades it is considered $2b_2$ [28].

$\therefore b_v = 2b_2$

$\therefore b_v = 2 \times 108$

$= 216$ mm

$$\text{Casing Outlet Area} = \frac{Q}{V_4} = \frac{0.5}{9.861} = 0.0507\ m^2$$

$$Casingbasecircle/inletradius, r_3 = \frac{D_2}{2} + 5 = \frac{224}{2} + 5 = 117\ mm = 0.117$$

And $D_3 = 0.234$ m

Centrifugal fan casing instantaneous radius

$$r_\theta = r_3(1 + k\theta)$$

Where k = 0.0023 for backward curved blades,

k = 0.0020 for forward curved blades and

k = 0.00215 for radial blades

$$\therefore r_\theta = r_3 \times \left(1 + (0.00215 \times \theta)\right)$$

The calculated fan casing instantaneous radius at different volute angles are given in Table 3.5.

Table 3.5 Radius of Volute at Different Angles

θ in Degree	R_θ in meter
0	0.1185
60	0.1338
120	0.1491
180	0.1644
240	0.1796
300	0.1949
360	0.2102

$r_4 = R_{360} = 0.210$ m $\therefore$ $D_4 = D_{360} = 0.420$ m

$$Casing\ outlet\ height = H_s = r_4 - r_3 = 210 - 117 = 93\ mm$$

$$Radius\ of\ volute\ tongue, r_t = 1.075 \times r_2 = 121\ mm \quad [26]$$

$$volute\ tongue\ angle, \theta_t = \frac{132 \log\left(\frac{r_t}{r_2}\right)}{\tan \alpha_2} = 17° \quad [28]$$

3.5.3 Hydraulic, leakage and power losses

3.5.3.1 Leakage loss [28]

$$Q_L = C_d \times \pi \times D_1 \times \delta \times \sqrt{\frac{2P_s}{\rho}}$$

$$Here, P_s = \tfrac{2}{3}\Delta P_s$$

Where coefficient of discharge C_d is 0.6 to 0.7,

$$\delta = Clearance\ between\ impeller\ eye\ inlet\ and\ casing$$

$$= 2\ mm\ as\ per\ fabrication$$

$$= \pi \times 0.6 \times 0.227 \times 0.002 \times \sqrt{\frac{2 \times \left(\frac{2}{3}\right) \times 981.2}{1.165}}$$

$$= 0.0286\ m^3/sec$$

3.5.3.2 Suction pressure loss [28]

$$dp_{suc} = \frac{1}{2} \times k_i \times \rho \times V_{eye}^2$$

Where k_i is a loss factor probably of the order of 0.5 to 0.8

$$= \frac{1}{2} \times 0.65 \times 1.165 \times 13.15^2$$

$$= 65.46\ Pa$$

3.5.3.3 Impeller pressure loss [28]

$$dP_{imp} = k_{ii} \times \frac{1}{2} \times \rho(W_1 - W_2')^2$$

At design point of maximum efficiency k_{ii} is in order of 0.2 – 0.3 for sheet metal blades and rather less for aerofoil section. Selecting k_{ii}=0.25,

$$= 0.25 \times \frac{1}{2} \times 1.165(33.87 - 8.75)^2$$

$$= 91.92\ Pa$$

3.5.3.4 Volute pressure loss

$$dP_{VC} = k_{iii} \times \frac{1}{2} \times \rho_2 \times (V_2' - V_4)^2$$

k_{iii} is of the order of 0.4 and will vary with deviation from design conditions.

$$= 0.4 \times \frac{1}{2} \times 1.165(27.89 - 9.861)^2$$

$$= 75.71\ Pa$$

3.5.3.5 Disc friction loss [28]

$$T_{df} = \pi f \rho \omega_2^2 \frac{r_2^5}{5} = \pi f \rho (U_2/r_2)^2 \frac{r_2^5}{5}$$

(Where f is material friction factor in order of 0.005 for mild steel sheet metal)

$$T_{df} = \pi \times 0.005 \times 1.165 \times \frac{32.87^2}{0.112^2} \times \frac{0.112^5}{5}$$

$$\therefore T_{df} = 0.0055719\ Nm$$

Hence, Power loss due to Disc friction

$$P_{df} = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 0.0055719}{60} = 1.63\ W$$

3.5.4 Efficiencies

3.5.4.1 Hydraulic efficiency

$$\eta_{hy} = \frac{\Delta P}{\Delta P + dp_{suc} + dp_{imp} + dp_{vc}}$$

$$\eta_{hy} = \frac{981.2}{981.2 + 65.46 + 91.92 + 75.71} = 0.808$$

$$\eta_{hy} = 80.8\%$$

3.5.4.2 Volumetric efficiency

$$\eta_{vol} = \frac{Q}{Q + Q_L} = \frac{0.5}{0.5 + 0.0286} = 0.9458$$

$$\eta_{vol} = 94.58\%$$

3.5.4.3 Total efficiency

$$\eta_{total} = \eta_{hy} \times \eta_{vol} = 0.808 \times 0.9458 = 0.7643$$

$$\eta_{total} = 76.43\%$$

3.5.5 Ideal shaft power required to run the fan

$$= \frac{\left(\Delta P + dp_{suc} + dp_{imp} + dp_{vc}\right)(Q + Q_L)}{\eta_{total}} + Power\ loss\ due\ to\ disk\ friction$$

$$= \frac{(981.2 + 65.46 + 91.92 + 75.51)(0.5 + 0.0286)}{0.7643} + 1.63$$

$$= 841.5\ W$$

$$So, Torque = \frac{841.5 \times 60}{2\pi \times 2800}$$

$$= 2.871\ Nm$$

3.5.6 Shaft diameter [136]

$$D_s = \sqrt[3]{\frac{16T \times Factor\ of\ safety}{\pi\tau}}$$

$$= \sqrt[3]{\frac{16 \times 2.871 \times 4}{\pi \times 343 \times 10^5}} = 0.01177\ m = 11.77\ mm$$

3.5.7 Blade profile

Blade Profile is made by tangent arc method [26]. When this method is used, the impeller is divided into a number of assumed concentric rings, not necessarily equally spaced between inner and outer radii. The radius R_b of the arc is defining the blade shape between inner and outer radii.

$$R_b = \frac{r_2^2 - r_1^2}{2[r_1 \cos\beta_1 - r_2 \cos\beta_2]}$$

$$R_b = \frac{1}{2} \times \frac{\left(\frac{0.224}{2}\right)^2 - \left(\frac{0.227}{2}\right)^2}{\left(\frac{0.227}{2}\cos 11.19 - 0\right)} = -0.0012\, m = -1.2\, mm$$

Note: During initial calculations, we have assumed $V_4 = 0.3U_2$, so we are getting value of D_1 greater than D_2 and negative blade profile radius R_b. While iterating further, V_4 is taken as actual received from volute design. Hence there after design parameters gets corrected.

After obtaining preliminary dimension of fan based on input data further iterations are made to get optimum design parameters. Next stage of iteration is made after adding leakage and pressure losses of previous iteration in required design point discharge and pressure head across the impeller stage as given in input data. This is done to accommodate possible hydraulic, discharge and power losses at different flow passages. This process is carried out till there is no marginal dimensional change in impeller outer diameter and other important parameters. An iterative design procedure algorithm is shown in Figure 3.1. Summary of designed calculation including few iterations are given in Table 3.6, while Figure 3.10 to 3.13 gives dimensional drawings of the fan designed as per Osborne design.

The Matlab program has been developed for this design calculation and is given in Annexure A.

Table 3.6 Design Parameters as per Osborne Design

Parameters as per Osborne Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
At Impeller Inlet							
Inlet Duct Diameter	D_{duct}	mm	242	224	218	215	215
Eye Diameter	D_{eye}	mm	220	203	198	196	196

Parameters as per Osborne Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
Eye Velocity	V_{eye}	m/s	13.15	16.30	17.11	17.52	17.52
Peripheral Velocity	U_1	m/s	33.23	30.69	29.97	29.57	29.57
Relative Velocity	W_1	m/s	33.87	31.75	31.17	30.84	30.84
Meridian Velocity	Vm_1	m/s	6.57	8.15	8.56	8.76	8.76
Absolute Velocity	V_1	m/s	6.57	8.15	8.56	8.76	8.76
Impeller Diameter	D_1	mm	227	209	204	202	202
Width of Blade	b_1	mm	107	99	96	95	95
Air Angle	α_1	Degree	90	90	90	90	90
Blade Angle	β_1	Degree	11.19	14.87	15.93	16.50	16.50
At Impeller Outlet							
Peripheral Velocity	U_2	m/s	32.87	40.74	42.78	43.79	43.79
Relative Velocity	W_2’	m/s	8.75	10.84	11.38	11.65	11.65
Swirl Velocity	V_{U2}’	m/s	27.10	33.59	35.27	36.10	36.10
Meridian Velocity	Vm_2	m/s	6.57	8.15	8.56	8.76	8.76
Absolute Velocity	V_2’	m/s	27.89	34.57	36.30	37.15	37.15
Impeller Diameter	D_2	mm	224	278	292	299	299
Width of Blade	b_2	mm	108	74	67	64	64
Air Angle	α_2’	Degree	13.64	13.64	13.64	13.64	13.64
Blade Angle	β_2	Degree	90°	90°	90°	90°	90°
At Volute/Scroll Casing							
Width of Casing	b_v	mm	216	149	135	128	128
Outlet Velocity of Casing	V_4	m/s	9.861	31.93	33.58	34.40	34.40
Scroll Radius at Inlet	r_3	mm	117	144	151	154	154

Parameters as per Osborne Design		Unit	Initial Calculations	1^{st} iteration	13^{th} iteration	14^{th} iteration	Final iteration
Scroll Radius at Outlet	r_4	mm	210	255	268	274	274
Scroll Height	Hs	mm	93	111	117	119	119
Radius of Tongue	r_t	mm	121	149	157	161	161
Angle of volute Tongue	θ_t	Degree	17	17	17	17	17
Pressure, Leakage and other Losses							
Pressure Losses in Suction	dP_{suc}	Pa	65.46	100.56	110.88	116.18	116.18
Pressure Losses in Impeller	dP_{im}	Pa	91.92	63.67	56.99	53.62	53.62
Pressure Losses in Scroll Casing	dP_{VC}	Pa	75.71	18.96	4.45	1.77	1.77
Leakage Loss	Q_L	m^3/s	0.0286	0.0294	0.0281	0.0276	0.0276
Disk Friction Torque	T_{df}	Nm	0.0055719	0.016299	0.02081	0.02338	0.023382
Power Loss due to Disk Friction	P_{df}	W	1.63	4.78	6.10	6.86	6.86
Efficiencies							
Hydraulic Efficiency	h_{hy}	%	80.80	86.89	87.11	87.05	87.05
Volumetric Efficiency	h_{vol}	%	94.58	94.73	94.95	95.03	95.03
Total Efficiency	h_{total}	%	76.43	82.31	82.71	82.72	82.72
Other Parameters							
Pressure Head Across the Stage with Losses	ΔP	Pa	981.2	1214.3	1164.4	1152.8	1152.8
Flow Discharge with Losses	Q	m^3/s	0.500	0.529	0.529	0.528	0.528
Blade Profile Radius	R_b	mm	-1.2	41.3	55.2	62.7	62.7
Shaft Diameter	D_s	mm	11.9	11.5	11.5	11.5	11.5
Power Required to Run Fan	P	W	841.5	753.7	742.6	742.2	742.2

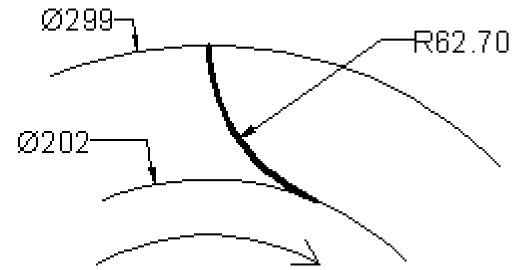

Figure 3.10 Blade Profile as per Osborne Design

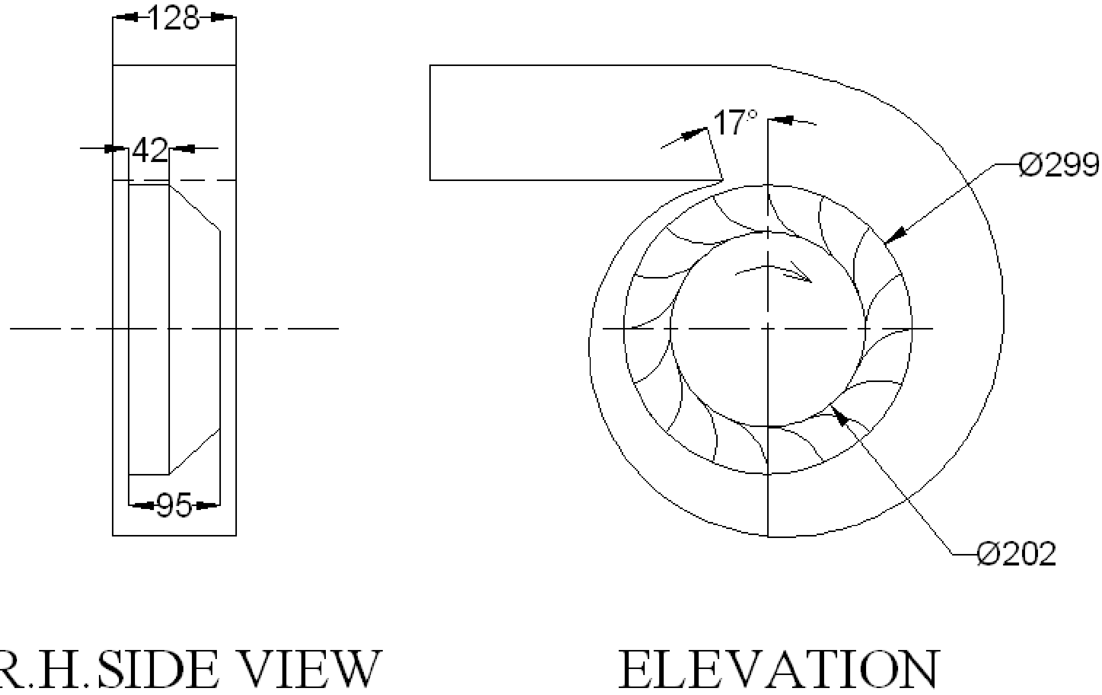

Figure 3.11 Impeller and Volute Casing Assembly as per Osborne Design

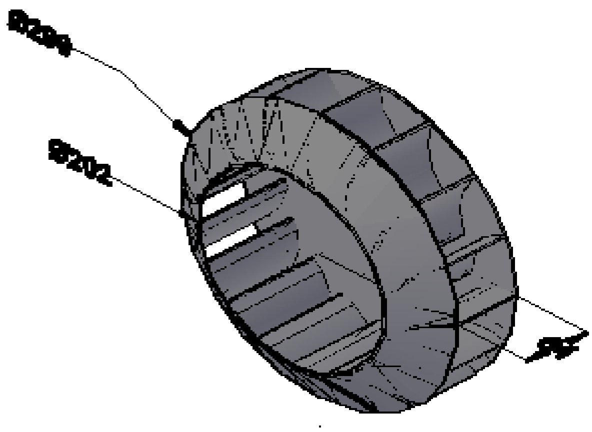

Figure 3.12 FCRT Impeller as per Osborne Design

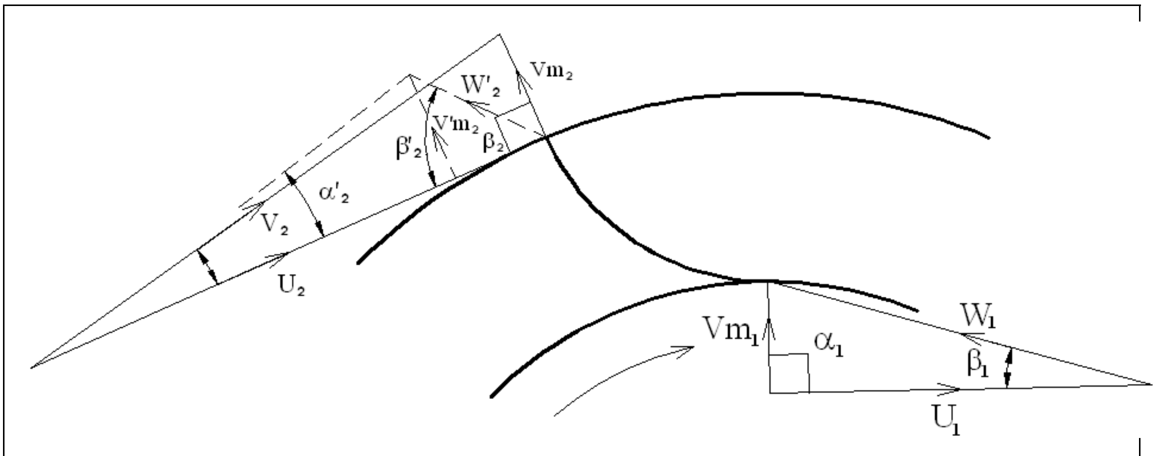

Figure 3.13 Theoretical and Actual Velocity Triangles as per Osborne Design

3.6 Comparative Assessment of Explicit Design Methodologies

Table 3.7 shows comparison of various formulae and assumptions used in case of each design methodology.

Table 3.7 Comparisons of Formulae and Assumptions for CF Fan Design Parameters

Para-meters	Fundamental Design	Church Design	Osborne Design
Impeller Design	$D_{duct} = 1.1D_{eye}$ $= 1.1D_1$ $U_1 = 1.1V_1 = \frac{\pi D_1 N}{60}$ $V_1 = V_{ml} = V_{eye}$ $W_1 = \sqrt{U_1^2 + V_1^2}$ $D_1 = 2.8066\sqrt[3]{\frac{Q}{N}}$ $b_1 = \frac{Q}{(\pi D_1 - zt)V_{ml}}$ $\sigma = 0.80$ $V_{u2} = 0.8U_2$ $P = mV_{u2}U_2$ $D_2 = \frac{60U_2}{\pi N}$ $W_2' = \sqrt{W_{U2}^2 + Vm_2^2}$ $V_2' = \sqrt{V_{U2}^2 + Vm_2^2}$ $\tan\alpha_2' = \frac{Vm_2}{V_{U2}}$	$Mass\ flow\ rate,$ $\dot{m} = Q\rho_a$ Assume, V_{duct}= 16 m/s $D_{duct} = \sqrt{\frac{4Q}{\pi V_{duct}}}$ $D_{eye} = \sqrt{\frac{4Q_{eye}}{\pi V_{eye}}}$ $1.02D_{eye}= D_1$ $V_1= Vm_1=1.055V_{eye}$ $\tan\beta_1 = \frac{Vm_1}{U_1}$ $W_1 = \sqrt{U_1^2 + V_1^2}$ $b_1 = \frac{A_1}{\pi D_1 \varepsilon_1'}$ $D_2 = \frac{60}{\pi N}\sqrt{\frac{gH_{ad}}{K'}}$ K'=0.5 to 0.65 $Vu_2' = U_2 - \frac{\pi U_2 \sin\beta_2}{Z}$ $V_2' = \sqrt{Vm_2^2 + Vu_2^2}$ $\tan\alpha_2' = \frac{Vm_2}{Vu_2'}$ $W_2' = \sqrt{W_{U2}^2 + Vm_2^2}$	Fan Static Pressure = Fan Total Pressure - Fan Dynamic Pressure $P_{se} = \rho U_2 Vu_2' - \frac{1}{2}\rho V_4^2$ $U_2 = \frac{\pi D_2 N}{60}$ Vu_2' $= Vu_2 - \frac{\pi \sin\beta_2}{Z}U_2$ $V_4=0.3U_2$ $Vm_1 = Vm_2 = 0.2U_2$ $V_{eye} = 0.4U_2$ $D_1= 1.03 D_{eye}$ $D_1 = \sqrt{\frac{4Q}{\pi V_{eye}}}$ $b_1 = \frac{Q}{\pi D_1 Vm_1}$
β_1	$\beta_1 = \tan^{-1}\frac{V_1}{U_1}$		
b_2	$b_1 = b_2$	$b_2 = \frac{A_2}{\pi D_2 \acute{\varepsilon}_2}$	$b_2 = \frac{Q}{\pi D_2 Vm_2}$

Para-meters	**Fundamental Design**	**Church Design**	**Osborne Design**
		Where, ε_2'= Outlet Vane Thickness Factor given by Church as: $\varepsilon_2' = \frac{(\pi D_2 - Zt)/\sin\beta_2}{\pi D_2}$	
Blade Profile	By Circular Arc Method, $R_b = \frac{r_2^2 - r_1^2}{2[r_1\cos\beta_1 - r_2\cos\beta_2]}$		
Volute Radius at Different Angles, r_θ	$r_\theta = r_3 + \frac{\theta}{360} \times \Delta r$ $where, \Delta r = [r_{360°} - r_{0°}]$ $r_3 = \frac{D_2}{2} + 5$ $Q = A_v V_4$ $A_v = b_v(r_4 - r_3)$	Instantaneous angle of volute, $\emptyset^0 = \frac{360 r_2 V u_2}{Q_{out}} \int_{r_2}^{r_\theta} b \frac{dr}{r}$ here $\emptyset^0$ = Differential radius from tabular integration method	$r_\theta = r_3(1 + k\theta)$ Where k = 0.0023 for backward curved blades, k = 0.0020 for forward curved blades and k = 0.00215 for radial blades $r_\theta = r_3(1 + 0.00215\theta)$
Width of Casing, B_v	$b_v = 2.5 b_2$	$b_v = 3 b_2$	$b_v = 2 b_2$
r_t	Volute Casing tongue radius, $r_t = 1.075 r_2$		
θ_t^o	Volute Casing tongue angle, $\theta_t = \frac{132\log_{10}\left(\frac{r_t}{r_2}\right)}{\tan\alpha_2}$		
Casing Outlet Velocity, V_4	By Steady Flow Energy Equation $V_4^2 = \frac{-2[P_2 - P_1]}{\rho_f} + V_1^2 + 2W_s$	By Interpolation	Assuming V_4=0.3V_2
Shaft Diameter	$D_s = \sqrt[3]{\frac{16T \times Factor\ of\ safety}{\pi\tau}}$		
Leakage Loss	$Q_L = C_d \pi D_1 \delta \sqrt{\frac{2P_s}{\rho}}$	Assuming 3% leakage losses while designing impeller inlet area	$Q_L = C_d \pi D_1 \delta \sqrt{\frac{2P_s}{\rho}}$

Para-meters	Fundamental Design	Church Design	Osborne Design
	$where\ P_s = \frac{2}{3}\Delta P$ $\delta = 2$ mm $C_d = 0.6$		$where, P_s = \frac{2}{3}\Delta P$
Disc Friction Torque	$T_{df} = \pi f \rho \omega_2^2 \frac{r_2^5}{5}$ Here f = Friction Factor, $\omega_2 = \frac{U_2}{r_2}$		
Hydraulic Efficiency	$\eta_{hy} = \frac{\Delta P}{\Delta P + dp_{suc} + dp_{imp} + dp_{vc}}$		
Volumetric Efficiency	$\eta_{vol} = \frac{Q}{Q + Q_L}$		
Total efficiency	$\eta_{total} = \eta_{hy} \times \eta_{vol}$		
Power required to run the fan	$= \frac{(\Delta P + dp_{suc} + dp_{imp} + dp_{vc})(Q + Q_L)}{\eta_{total}} + Power\ loss\ due\ to\ disk\ friction$ Here Power loss due to Disc friction $P_{disc} = \frac{2\pi N T_{disc}}{60}$		

Optimum geometrical and other parameters obtained by fundamental, Church and Osborne design methodologies after final iteration are given in Table 3.8. This is for identical input data as mentioned in sub topic 3.2. It is clearly observed by comparative assessment as given in Table 3.8, that there is wide variation in many of the parameters, achieved under these design methodologies.

Table 3.8 Optimum Design Parameters as per Fundamental, Church and Osborne Design

Comparison of Fan Design Optimum Parameters		Unit	Fundamental Design	Church Design	Osborne Design
At Impeller Inlet					
Inlet Duct Diameter	D_{duct}	mm	188	199	215
Eye Diameter	D_{eye}	mm	171	188	196
Eye Velocity	V_{eye}	m/s	22.80	18.00	17.52
Peripheral Velocity	U_1	m/s	25.08	28.00	29.57
Relative Velocity	W_1	m/s	33.90	33.84	30.84
Meridian Velocity	Vm_1	m/s	22.80	18.99	8.76
Absolute Velocity	V_1	m/s	22.80	18.99	8.76
Impeller Diameter	D_1	mm	171	191	202
Width of Blade	b_1	mm	45	49	95
Air Angle	α_1	Degree	90	90	90
Blade Angle	β_1	Degree	42.35	35.17	16.50
At Impeller Outlet					
Peripheral Velocity	U_2	m/s	38.14	41.56	43.79
Relative Velocity	W_2’	m/s	16.54	18.09	11.65
Swirl Velocity	V_{U2}’	m/s	30.52	33.40	36.10
Meridian Velocity	Vm_2	m/s	14.68	16.14	8.76
Absolute Velocity	V_2’	m/s	33.86	37.10	37.15
Impeller Diameter	D_2	mm	260	283.50	299

Comparison of Fan Design Optimum Parameters		Unit	Fundamental Design	Church Design	Osborne Design
Width of Blade	b_2	mm	45	36.93	64
Air Angle	α_2'	Degree	25.69	25.79	13.64
Blade Angle	β_2	Degree	90º	90º	90º
At Volute/Scroll Casing					
Width of Casing	b_v	mm	114	111	128
Outlet Velocity of Casing	V_4	m/s	27.05	19.68	34.40
Scroll Radius at Inlet	r_3	mm	135	147	154
Scroll Radius at Outlet	r_4	mm	306	376	274
Scroll Height	Hs	mm	170	229	119
Radius of Tongue	r_t	mm	140	152	161
Angle of volute Tongue	θ_t	Degree	9	9	17
Pressure, Leakage and other Losses					
Pressure Losses in Suction	dP_{suc}	Pa	196.87	122.52	116.18
Pressure Losses in Impeller	dP_{im}	Pa	43.88	36.21	53.62
Pressure Losses in Scroll Casing	dP_{VC}	Pa	10.82	71.12	1.77
Leakage Loss	Q_L	m^3/s	0.0242	0.0150	0.0276
Disk Friction Torque	T_{df}	Nm	0.011723	0.01812	0.023382
Power Loss due to Disk Friction	P_{df}	W	3.44	5.31	6.86

Comparison of Fan Design Optimum Parameters		Unit	Fundamental Design	Church Design	Osborne Design
Efficiencies					
Hydraulic Efficiency	h_{hy}	%	83.05	84.05	87.05
Volumetric Efficiency	h_{vol}	%	95.58	97.17	95.03
Total Efficiency	h_{total}	%	79.38	81.67	82.72
Other Parameters					
Pressure Head Across the Stage with Losses	ΔP	Pa	1232.754	1211.1	1152.8
Flow Discharge with Losses	Q	m^3/s	0.5242	0.5150	0.5276
Blade Profile Radius	R_b	mm	75.9	70.3	62.7
Shaft Diameter	d_s	mm	11.8	11.6	11.5
Power Required to Run Fan	P	W	817.5	769.0	742.2

Radial tipped centrifugal fans are fabricated for each design methodology as per optimum parameters given in Table 3.8. These fans are tested as per test standard IS: 4894-1987, Indian Standard Specification for Centrifugal Fans, 1994 [17]. Precise and calibrated instruments are used to get accurate results. Observations are made for static pressure distribution along the flow path and for variable flow rate at outlet, accompanied with other associated parameters. This was done at designed and off design rotational speeds. Experimental set up, procedure and results are discussed in detail in subsequent chapters.

The performance of fans fabricated as per individual and explicit design methodologies suggested by Church, Osborne and retrieved from fundamental principles of fluid flow having minimum assumptions, is critically evaluated. Based on experimental results obtained, it is observed that there exists a wide performance difference amongst fans under study. It concludes that each design method, as an individual, is not performing as marked. It has also revealed that there is a need to develop unified design methodology.

3.7 Unified Design Methodology

Successful outcomes of fundamental design [9, 10, 14, 28], Church design [9, 26, 28] and Osborne design [9, 28] are incorporated together and a new design methodology for radial tipped centrifugal fan is developed. It is to be noted here that the unified design is derived, based on non- dimensional parameters with inclusion of compressibility effect which makes it quite generalized for application to wide range of radial tipped centrifugal machines. This is named as unified design methodology for radial tipped centrifugal fan. Input parameters are identical as considered in previous designs. This work is presented in three sections.

Non- dimensional parameters

Impeller design

Scroll casing design

3.7.1 Non- dimensional parameters

[14, 28, 56]

The turbo machine designer is often faced with the basic problem of deciding what type of turbomachine will be the best choice for a given duty. Usually the designer is provided with some preliminary design data such as the head H, the volume flow rate Q and the rotational speed N when a pump, fan or compressor design is under consideration. For the design, comparison, and critical assessment of all fans, dimensionless coefficients have been used [14, 56]. These coefficients must be dimensionless so that the numerical values can be independent of the actual increases in pressure, mass flow, and other physical properties.

Important non- dimensional coefficients used in literature are specific speed, pressure coefficient, volume coefficient, power coefficient, speed coefficient, diameter coefficient and noise coefficient. These parameters are briefly explained below.

- **Specific speed Ns**

A non-dimensional parameter called the specific speed, N_s, referred to and conceptualized as the shape number, is often used to facilitate the choice of the most appropriate machine. This parameter is derived from the non-dimensional groups in such a way that the characteristic diameter D of the turbo machine is eliminated. The value of N_s gives the designer a guide to the type of machine that will provide the normal requirement of high efficiency at the design condition.

Specific speed is a criterion at which a fan of unspecified diameter would run to give unit volume flow and pressure. This applies to all fans in a homologous series. This is derived from the fan laws by elimination of the diameter term, thus:

$$N_s = \frac{NQ^{\frac{1}{2}}}{(gH)^{\frac{3}{4}}} \quad usually\ for\ power\ absorbing\ machines$$

$$N_s = \frac{N(P/\rho)^{\frac{1}{2}}}{(gH)^{\frac{5}{4}}} \quad usually\ for\ power\ producing\ machines$$

- **Pressure coefficient ψ**

The fan impeller has a typical outside diameter d_2, and a typical peripheral velocity u_2. As a possible basis of comparison for the pressure produced by the impeller, the dynamic pressure of the peripheral velocity $\rho u_2^2/2$ is considered. This coefficient affects the fan noise and has a dependent relationship with the other coefficients. For example, if designer wants to design a minimal noise-generation fan, then the maximum value of pressure coefficient must be selected. Thus the pressure coefficient Ψ can be defined as the ratio of the pressure produced by the impeller and the pressure of the peripheral velocity.

$$\psi = \frac{\Delta P}{\left(\frac{\rho}{2}\right) {U_2}^2}$$

- **Volume coefficient Φ**

Volume coefficient Φ is a ratio of actual volume flow to that of theoretical one. The volume flow Q per second requires a similar basis of comparison. So the circular area of the impeller $\pi d_2^2/4$ can be employed, through which the flowing medium passes with a velocity U_2. Thus, the volume flow can be expressed as $Q = (U_2 \pi d_2^2)/4$. Naturally, this volume differs from the actual volume: therefore the volume coefficient Φ can be obtained from the difference between the actual volume and the theoretical volume. This coefficient will affect the capacity and the noise of the fan. The maximum value of the volume coefficient will cause the maximum capacity, while the minimum value will cause the minimum noise. The volume coefficient is expressed as follows:

$$\Phi = \frac{Q}{((U_2 \pi D_2^2)/4)} = \frac{\pi \times D_2 \times b_2 \times V_{m2}}{(U_2 \pi D_2^2)/4)} = \frac{(Vm_2) \times (4b_2)}{(U_2) \times (D_2)}$$

- **Power coefficient λ**

Power coefficient is also known as output coefficient or coefficient of performance. This is associated with driving shaft power input to head and volume flow generated by the impeller. This is expressed mathematically as follows:

$$\lambda = \frac{\psi \times \Phi}{\eta}$$

- **Speed coefficient σ and diameter coefficient δ**

In practice, pressure and volume coefficients have been found inadequate for numerical evaluation of the important characteristics of the centrifugal impeller. For any given volume flow Q and a given pressure increase Δp can be produced by various fans which are widely different in their dimensions. The price of a fan is often dependent upon its size, and the importance of this from the point of view of the designer cannot be over-emphasized. Moreover, the designer often requires a specified speed, so coefficients relating to the dimensions and speed of a fan are necessary.

The diameter coefficient δ indicates the number of times greater the diameter of an impeller is, in relation to the diameter of a model impeller with Ψ = 1 and Φ = 1. These coefficients are related to head, flow rate, outlet diameter, and motor speed. Therefore, if the designer selects those values, the speed and diameter coefficient can be calculated. The coefficients are expressed as follows:

$$\text{Speed coefficient}, \sigma = n_{rps} \sqrt[4]{\frac{Q^2}{(2gH)^3}} \{\sqrt[2]{\pi}\} = \frac{1}{28.5} \times Q^{\frac{1}{2}} \times \left(\frac{\Delta P}{\rho}\right)^{\frac{-3}{4}} \times N_{rpm}$$

$$\text{Speed coefficient}, \sigma = 0.379 \times n_{rps} \sqrt[4]{\frac{Q^2}{(H)^3}} = \frac{0.379 N_{rpm} Q^{1/2}}{60 H^{3/4}}$$

$$\text{Diameter coefficient}, \delta = 1.865 \times d_2 \times \sqrt[4]{\frac{H}{Q^2}}$$

Also, these coefficients are dependent on each other and related to the pressure and volume coefficients. Therefore, these coefficients can be expressed in the form of pressure and volume coefficients.

$$Pressure\ coefficient, \psi = {}^{1}/_{\sigma^2\delta^2}$$

$$Volume\ coefficient, \Phi = {}^{1}/_{\sigma\delta^3}$$

- **Noise coefficient**

Nowadays, investigations of fan noise are extremely important, due to advances in air-conditioning and ventilation techniques. Fan noise can be expressed as a function of the volume coefficient Φ and the pressure coefficient ψ. Variation from the optimum volume coefficient and pressure coefficient either due to an increase or decrease in volume and pressure, results in an increase or decrease in sound level. Therefore, to design a minimum-noise fan, the designer should select the minimum value of the volume coefficient and the maximum value of the pressure coefficient. The noise coefficient τ can be expressed as follows:

$$\tau = \frac{\Phi^2}{\psi}$$

Referring to input design data for radial tipped centrifugal fan given in section 3.2, unified design procedure using non- dimensional coefficients is as per following:

$$Specific\ speed, N_S = \frac{\omega Q^{\frac{1}{2}}}{(gH)^{\frac{3}{4}}}$$

$$= \frac{\frac{2\pi N}{60} \times Q^{\frac{1}{2}}}{(gH)^{\frac{3}{4}}}$$

$$= \frac{\frac{2\pi \times 2800}{60} \times 0.5^{\frac{1}{2}}}{(9.81 \times 85.59)^{\frac{3}{4}}}$$

$$\therefore N_S = 1.3292$$

$$Speed\ Coefficient,\ \sigma_1 = \frac{0.379NQ^{1/2}}{60H^{3/4}} = 0.4443$$

Cordier has given the relation between $\sigma_1$and δ in graphical form as shown in Figure 3.14.

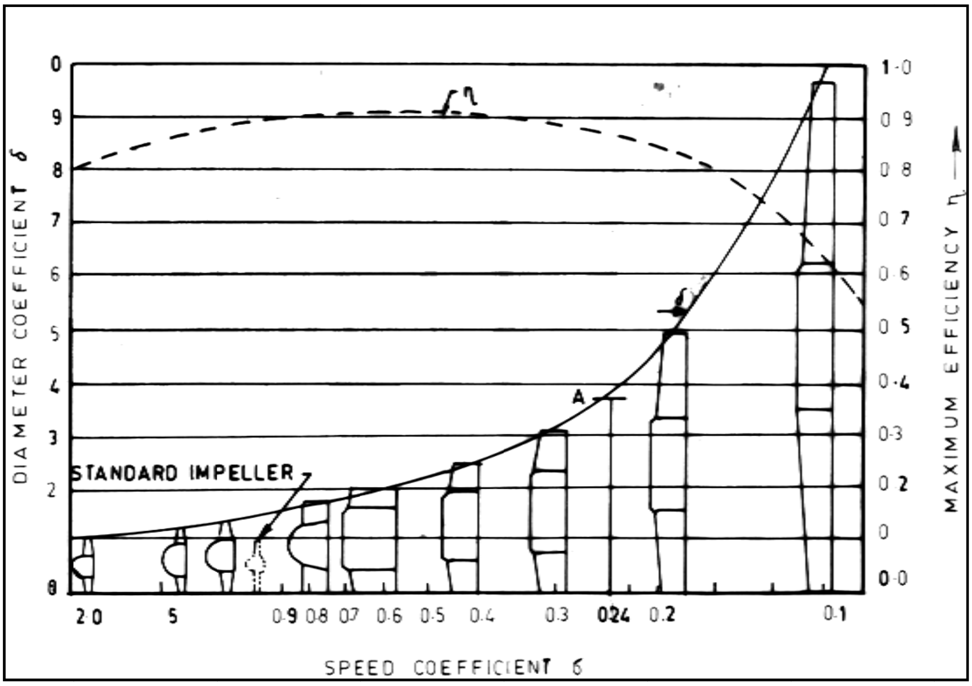

Figure 3.14 Cordier's Diagram Correlating Speed Coefficient σ Vs Diameter Coefficient δ for Pump and Fan Selection [14]

This graphical form is converted into mathematical form by weighted residue method [13]. The resultant equation for δ for different stages is obtained as:

Diameter coefficient, δ	=	a σ_1^{-b}
Here a	=	0.99 where $0.1 < \sigma_1 < 0.4$
	=	1.5 where $0.4 < \sigma_1 < 2$
b	=	0.995 where $0.1 < \sigma_1 < 0.4$
	=	0.5866 where $0.4 < \sigma_1 < 1$
	=	0.505 where $1 < \sigma_1 < 2$

Thus, for $\sigma_1 = 0.4443$, we will get $\delta = 2.4137$

$$Pressure\ coefficient, \psi = \frac{1}{\sigma_1^2 \times \delta^2} = 0.86899 = \frac{\Delta P}{(\rho/2)({U_2}^2)}$$

$$Volume\ coefficient, \emptyset = \frac{1}{\sigma_1 \times \delta^3} = 0.163 = \frac{(Vm_2)(4b_2)}{(U_2)(D_2)}$$

3.7.2 Effect of compressibility on design

[26]

If flow is steady, mass flow $\dot{m}$ remains constant but density will change at different flow sections due to change in temperature and pressure. Hence volume flow Q will change. This is compressibility effect. From characteristic equation of gas:

$$Density\ at\ atm.air, \rho_a = \frac{P_a}{RT_a} = \frac{1.01325 \times 10^5}{287 \times 303} = 1.1652\ kg/m^3$$

$$mass\ flow\ rate, \dot{m} = \rho_a Q = 0.58259\ kg/m^3$$

$$Density\ at\ suction, \rho_0 = \frac{P_0}{RT_0} = \frac{1.011286 \times 10^5}{287 \times 303} = 1.1629\ kg/m^3$$

$$So, Q_s = {}^{\dot{m}}/_{\rho_s} = 0.50098\ m^3/s$$

$$Density\ at\ delivery, \rho_d = \frac{P_d}{RT_a} = \frac{1.021098 \times 10^5}{287 \times 303} = 1.1742\ kg/m^3$$

$$\Delta P = 981.2\ Pa$$

$$\Delta P = \rho_m g H$$

$$where\ \rho_m = \frac{\rho_s + \rho_d}{2} = 1.1686$$

$$So\ head\ devloped\ in\ impeller, H = 85.5928\ m$$

3.7.3 Design of impeller

3.7.3.1 Impeller outlet parameters

Here,

$$Density\ at\ delivery, \rho_d = \frac{P_d}{RT_a} = \frac{1.021098 \times 10^5}{287 \times 303} = 1.1742\ kg/m^3$$

$$\Delta P = 981.2\ Pa$$

And,

$$Pressure\ coefficient, \psi = \frac{1}{\sigma_1^2 \times \delta^2} = 0.86899 = \frac{\Delta p}{(\rho/2)(U_2{}^2)}$$

$$\therefore\ U_2 = 44.03\ m/sec$$

$$U_2 = \frac{\pi D_2 N}{60}$$

$$\therefore\ D_2 = 0.300\ m$$

Now,

$$Volume\ coefficient, \emptyset = \frac{1}{\sigma_1 \times \delta^3} = 0.163 = \frac{(Vm_2)(4b_2)}{(U_2)(D_2)}$$

Assuming $Vm_1 = Vm_2 = 0.2U_2 = 8.81$ m/s = $W_{2\,without\ slip}$ [28]

$$\therefore\ b_2 = 0.061\ m = 61\ mm$$

Parallel shroud plates are taken for impeller, hence impeller width at inlet and outlet will be equal. $\therefore b_1 = b_2$. This parameter will be verified at the end of impeller stage calculations and major dimension will be considered.

Now, due to blade passage circulation effect (slip):

$$Vu_2 = U_2 - \frac{\pi U_2 \sin\beta_2}{Z} = U_2 - \frac{3.14U_2}{16} = 0.80375U_2 = 35.38\ m/s$$

From velocity triangle,

$$Absolute\ velocity\ at\ outlet, V_2 = \sqrt{Vm_2^2 + Vu_2^2} = 36.46\ m/s$$

$$\tan\alpha_2' = \frac{Vm_2}{Vu_2}$$

$$\alpha_2' = 13.98°$$

$$W_{u2} = U_2 - V_{U2} = 8.65\ m/s$$

Now, outlet relative velocity with slip, $W_2' = \sqrt{W_{U2}^2 + Vm_2^2} = 12.34\ m/s$

3.7.3.2 Impeller inlet parameters

Assuming $V_{eye} = 0.4U_2 = 0.4 \times 44.03 = 17.61$ m/s [28]

$$Impeller\ eye\ diameter, D_{\mathrm{eye}} = \sqrt{\frac{4Q}{\pi V_{eye}}}$$

$$= \sqrt{\frac{4 \times 0.5}{\pi \times 17.54}}$$

$$= 0.190\ m = 190\ mm$$

Taking $1.1\ D_1 = D_{eye}$

$\therefore\ D_1 = 191/1.1 = 173$ mm

$$Impeller\ Inlet\ peripheral\ velocity, U_1 = \frac{\pi D_1 N}{60}$$

$$= \frac{\pi \times 0.174 \times 2800}{60}$$

$$\therefore U_1 = 25.34\ m/s$$

$$V_1 = V_{m1} = 8.81\ m/s$$

$$W_1 = \sqrt{U_1^2 + V_1^2} = \sqrt{25.5^2 + 8.77^2} = 26.82\ m/s$$

$$\tan\beta_1 = \frac{V_1}{U_1} = \frac{8.77}{25.5}$$

$$\beta_1 = 19.16°$$

(Refer Figure 3.18 for inlet and outlet velocity triangles)

Virtual head developed by impeller,

$$H_{virtual} = \frac{1}{2g}(U_2^2 - U_1^2 + W_1^2 - W_2^2) = 95.01\ m$$

Where

$$\frac{U_2^2 - U_1^2}{2} \rightarrow Head\ due\ to\ centrifugal\ action$$

$$\frac{W_1^2 - W_2^2}{2} \rightarrow Head\ due\ to\ change\ in\ relative\ velocitites$$

It may be assumed that, owing to the circulatory flow, friction and turbulence in the impeller, 15 percent of this virtual head is lost.

Hence effective head $H_{effective} = 0.85 H_{virtual} = 0.85 \times 95.01 = 80.78\ m$

Impeller outlet pressure is based upon effective head so,

$$\therefore \epsilon_{Pimp}^{0.286} - 1 = \frac{0.286 g H_{effective}}{RT_0} = \frac{0.286 \times 9.81 \times 80.78}{287 \times 302.79} = 0.00261$$

Where $\in_{Pimp}$= Pressure ratio between impeller eye and impeller outlet base upon effective head.

$$\therefore \epsilon_{Pimp} = 1.00916$$

Impeller outlet pressure, $P_2 = \epsilon_{Pimp} \times P_0 = 1.00916 \times 101128.6 =$ $102054.9\ Pa$

The friction and turbulence losses will be transformed into heat which raises the temperature of air. The impeller outlet temperature may be based upon the adiabatic head in the impeller neglecting losses.

$$\therefore \epsilon_{Pimp}^{0.286} - 1 = \frac{0.286 g H_{adiabatic}}{RT_0} = \frac{0.286 \times 9.81 \times 85.59}{287 \times 302.83} = 0.00276$$

The outlet temperature, $T_2 = \epsilon_{Pimp}^{0.286} \times T_0 = 303.61\ K$

Impeller outlet density, $\rho_2 = \frac{P_2}{RT_2} = \frac{102046.9}{287 \times 303.61} = 1.1711\ \text{kg/m}^3$

Impeller exit volume flow, $\qquad Q_{out} = \frac{0.5825}{1.1711} = 0.4974\ m^3/s$

$$\text{Impeller outlet area}, A_2 = {Q_{out}}/{Vm_2} = {0.4974}/{8.81} = 0.0565\ m^2$$

Taking optimized numbers of blades, Z = 16 and blade thickness t = 2 mm,

$$\therefore Impeller\ outlet\ width, b_2 = {Q_{out}}/{(\pi D_2 - Zt)Vm_2} = 0.0619\ m = 62\ mm$$

Considering parallel shrouded impeller, so $Impeller\ inlet\ width\ b_1 = Impeller\ outlet\ width\ b_2 = 62$ mm

It is seen that impeller width received from dimensionless coefficient and other one based on impeller exit area have very marginal difference. Here major dimension is selected.

3.7.4 Scroll casing design

[137]

The various air streams leaving the blade tips are collected in the scroll housing and reunited into a single air stream that leaves the unit at right angle to the axis.

The spiral shape of casing is approximated by three circular sections as shown in Figure 3.14.

SCr_{31} =71.2% of D_2 = 214 mm

SCr_{32} =83.7% of D_2 = 251 mm

SCr_{33} =96.2% of D_2 = 289 mm

This method of scroll casing design is known as 4 point method. The radii at different cross section are correlated with impeller exit diameter.

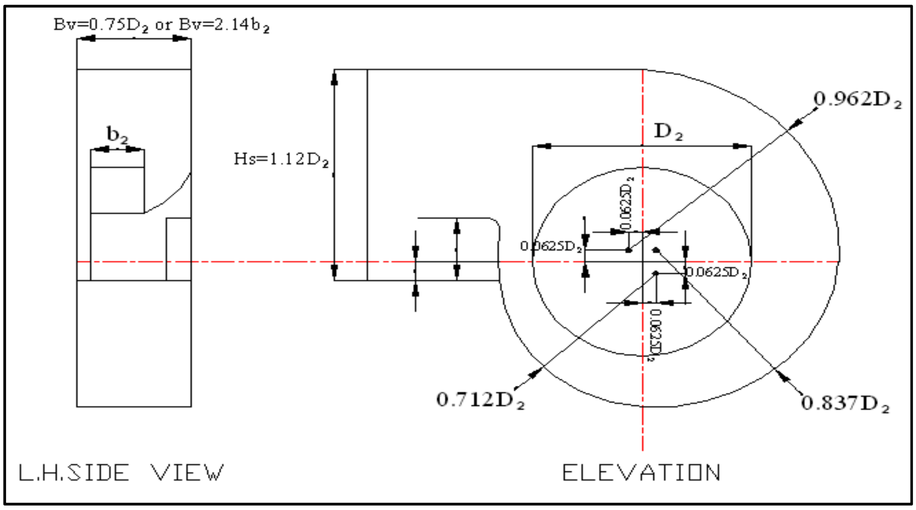

Figure 3.15 Scroll Casing Design by 4 Point Method [137]

Centre of these three are located off the centre lines by interval of 6.25% of wheel dia. = 18.7 mm

Width of housing is 75% of D_2 = 225 mm

Height of housing outlet is 112% of D_2= 336 mm

$$Radius\ of\ volute\ tongue, r_t = 1.075 \times r_2 = 161.25\ mm \text{ [26]}$$

$$volute\ tongue\ angle, \theta_t = \frac{132 \log\left(\frac{r_t}{r_2}\right)}{\tan \alpha_2} = 16.65° \quad \text{[26]}$$

At the housing outlet, two piece of cut off is used which is placed only on the inlet side of housing which is called as recirculation shield. The purpose of recirculation shield is to minimize the recirculation of the bypassing air (which of course, is a loss in air volume and efficiency) without producing excessive noise due to a small cutoff clearance. In other words, the 5 to 10% cutoff clearance is a compromise between prevention of recirculation and quiet operation. The recirculation shield protrudes into housing outlet by 30 to 35% of outlet height. Here 35% of D_2 exit shield is considered. We know that,

Impeller exit volume flow, $Q_{out} = 0.4974\ m^3/s$

Same volume flow must leave volute casing at exit. Exit volute area

$$A_V = (1 - 0.35)H_V \times b_V = 0.65 \times 0.336 \times 0.225 = 0.049\ m^2$$

$$\therefore V_4 = \frac{Q_{out}}{A_V} = \frac{0.4974}{0.049} = 10.11\ m/s$$

3.7.5 Hydraulic, leakage and power losses

3.7.5.1 Leakage loss [26]

$$Q_L = 0.03Q = 0.03 \times 0.5 = 0.015\ m^3/s$$

3.7.5.2 Suction pressure loss [28]

Friction and turbulence loss in impeller eye

$dp_{suc} = \frac{1}{2} \times k_i \times \rho \times V_{eye}^2$ where k_i is a loss factor probably of the order of 0.5 to 0.8

$$= \frac{1}{2} \times 0.65 \times 1.1629 \times 17.61^2$$

$$= 117.28\ Pa$$

3.7.5.3 Impeller pressure loss [28]

$$dP_{imp} = k_{ii} \times \frac{1}{2} \times \rho(W_1 - W_2')^2$$

At design point of maximum efficiency k_{ii} is in order of 0.2 – 0.3 for sheet metal blades and rather less for aerofoil section. Selecting $k_{ii} = 0.25$

$$dP_{imp} = 0.25 \times \frac{1}{2} \times 1.1686 \times (26.83 - 12.34)^2$$

$$dP_{imp} = 30.62\ Pa$$

3.7.5.4 Volute pressure loss [28]

$$dP_{VC} = k_{iii} \times \frac{1}{2} \times \rho(V_2' - V_4)^2$$

k_{iii} is of the order of 0.4 and will vary with deviation from design conditions.

$$= 0.4 \times \frac{1}{2} \times 1.1742 \times (36.46 - 10.11)^2$$

$$= 162.69\ Pa$$

3.7.5.5 Disc friction loss [28]

$$T_{df} = \pi f \rho \omega_2^2 \frac{r_2^5}{5} = \pi f \rho (U_2/r_2)^2 \frac{r_2^5}{5}$$

(Where f is material friction factor in order of 0.005 for mild steel sheet metal)

$$T_{df} = \pi \times 0.005 \times 1.1686 \times \frac{44.03^2}{0.150^2} \times \frac{0.150^5}{5}$$

$$\therefore T_{df} = 0.02414\ Nm$$

Hence, Power loss due to Disc friction

$$P_{df} = \frac{2\pi NT}{60} = \frac{2\pi \times 2800 \times 0.02414}{60} = 7.08\ W$$

3.7.6 Efficiencies

3.7.6.1 Hydraulic efficiency

$$\eta_{hy} = \frac{\Delta P}{\Delta P + dp_{suc} + dp_{imp} + dp_{vc}}$$

$$\eta_{hy} = \frac{981.2}{981.2 + 117.28 + 30.62 + 162.69} = 0.7596$$

$$\eta_{hy} = 75.96 = 76\%$$

3.7.6.2 Volumetric efficiency

$$\eta_{vol} = \frac{Q}{Q + Q_L} = \frac{0.5}{0.5 + 0.015} = 0.971 = 97.1\%$$

3.7.6.3 Total efficiency

$$\eta_{total} = \eta_{hy} \times \eta_{vol} = 0.7596 \times 0.971 = 0.7375 = 73.75\%$$

3.7.7 Ideal shaft power required to run the fan

$$= \frac{\left(\Delta P + dp_{suc} + dp_{imp} + dp_{vc}\right)(Q + Q_L)}{\eta_{total}} + Power\ loss\ due\ to\ disk\ friction$$

$$= \frac{(981.2 + 117.28 + 30.62 + 162.69)(0.5 + 0.015)}{0.7375} + 7.08$$

$$= 909.18\ Watts$$

$$So, Torque = \frac{909.18 \times 60}{2\pi \times 2800}$$

$$= 3.102\ Nm$$

3.7.8 Shaft diameter [136]

$$D_S = \sqrt[3]{\frac{16T \times factor\ of\ safety}{\pi\tau}}$$

$$= \sqrt[3]{\frac{16 \times 3.102 \times 4}{\pi \times 343 \times 10^5}} = 0.01223\ m = 12.26\ mm$$

3.7.9 Blade profile

Blade Profile is made by tangent arc method [26]. When this method is used, the impeller is divided into a number of assumed concentric rings, not necessarily equally spaced between inner and outer radii. The radius R_b of the arc is defining the blade shape between inner and outer radii.

$$R_b = \frac{r_2^2 - r_1^2}{2[r_1 \cos\beta_1 - r_2 \cos\beta_2]}$$

$$R_b = \frac{1}{2} \times \frac{\left({}^{0.300}/_{2}\right)^2 - \left({}^{0.173}/_{2}\right)^2}{\left(\frac{0.173}{2}\cos 19.16 - 0\right)} = 0.09235\ m = 92.35\ mm$$

After obtaining preliminary dimension of fan based on input data further iterations are made to get optimum design parameters. Next stage of iteration is made after adding leakage and pressure losses of previous iteration in required design point discharge and pressure head across the impeller stage as given in input data. This is done to accommodate possible hydraulic, discharge and power losses at different flow passages. This process is carried out till there is no marginal dimensional change in impeller outer diameter and other important parameters. An iterative design procedure algorithm is shown in Figure 3.1. Summary of designed calculation including few iterations are given in Table 3.9, while Figure 3.16 to 3.19 gives dimensional drawings of the fan designed as per Unified Design.

The Matlab program has been developed for this design calculation and is given in Annexure A.

Table 3.9 Design Parameters as per Unified Design

Parameters as per Unified Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
At Impeller Inlet							
Inlet Duct Diameter	D_{duct}	mm	209	206	205	205	205
Eye Diameter	D_{eye}	mm	190	187	187	187	187
Eye Velocity	V_{eye}	m/s	17.61	18.67	18.82	18.84	18.84
Peripheral Velocity	U_1	m/s	25.34	24.98	24.88	24.86	24.86
Relative Velocity	W_1	m/s	26.83	26.66	26.60	26.59	26.59
Meridian Velocity	Vm_1	m/s	8.81	9.34	9.41	9.42	9.42
Absolute Velocity	V_1	m/s	8.81	9.34	9.41	9.42	9.42
Impeller Diameter	D_1	mm	173	170	170	170	170
Width of Blade	b_1	mm	62	55	54	54	54
Air Angle	α_1	Degree	90	90	90	90	90

Parameters as per Unified Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
Blade Angle	β_1	Degree	19.16	20.49	20.72	20.75	20.75
At Impeller Outlet							
Peripheral Velocity	U_2	m/s	44.03	46.68	47.04	47.10	47.10
Relative Velocity	W_2'	m/s	12.34	13.08	13.19	13.20	13.20
Swirl Velocity	V_{U2}'	m/s	35.38	37.51	37.81	37.86	37.86
Meridian Velocity	Vm_2	m/s	8.81	9.34	9.41	9.42	9.42
Absolute Velocity	V_2'	m/s	36.46	38.66	38.96	39.01	39.01
Impeller Diameter	D_2	mm	300	318	321	321	321
Width of Blade	b_2	mm	62	55	54	54	54
Air Angle	α_2'	Deg.	13.98	13.98	13.98	13.98	13.98
Blade Angle	β_2	Deg.	90°	90°	90°	90°	90°
At Volute/Scroll Casing							
Width of Casing	b_v	mm	225	239	241	241	241
Outlet Velocity of Casing	V_4	m/s	10.10	8.98	8.84	8.82	8.82
Scroll Radius at Inlet	r_3	mm	214	227	228	229	229
Scroll Radius at Outlet	r_4	mm	289	306	309	309	309
Scroll Height	Hs	mm	336	357	359	360	360
Radius of Tongue	r_t	mm	161.25	171	172.5	172.5	172.5

Parameters as per Unified Design		**Unit**	**Initial Calculations**	**1st iteration**	**13th iteration**	**14th iteration**	**Final iteration**
Angle of volute Tongue	θ_t	Degree	16.65	16.66	16.66	16.66	16.66
Pressure, Leakage and other Losses							
Pressure Losses in Suction	dP_{suc}	Pa	117.28	131.82	133.91	134.25	134.25
Pressure Losses in Impeller	dP_{im}	Pa	30.62	26.93	26.26	26.16	26.16
Pressure Losses in Scroll Casing	dP_{VC}	Pa	162.69	206.35	212.62	213.63	213.63
Leakage Loss	Q_L	m^3/s	0.0150	0.0150	0.0150	0.0150	0.0150
Disk Friction Torque	T_{df}	Nm	0.02414	0.03236	0.03366	0.03387	0.03387
Power Loss due to Disk Friction	P_{df}	W	7.08	9.49	9.87	9.93	9.93
Efficiencies							
Hydraulic Efficiency	h_{hy}	%	75.96	77.96	78.31	78.37	78.37
Volumetric Efficiency	h_{vol}	%	97.09	97.09	97.09	97.09	97.09
Total Efficiency	h_{total}	%	73.75	75.70	76.04	76.09	76.09
Other Parameters							
Pressure Head Across the Stage with Losses	ΔP	Pa	1291.8	1291.8	1346.3	1355.2	1355.2
Flow Discharge with Losses	Q	m^3/s	0.5150	0.5150	0.5150	0.5150	0.5150

Parameters as per Unified Design		Unit	Initial Calculations	1st iteration	13th iteration	14th iteration	Final iteration
Blade Profile Radius	R_b	mm	92.3	113.3	116.8	117.4	117.4
Shaft Diameter	D_s	mm	12.3	12.3	12.3	12.3	12.3
Power Required to Run Fan	P	W	909.2	925.4	926.9	927.2	927.2

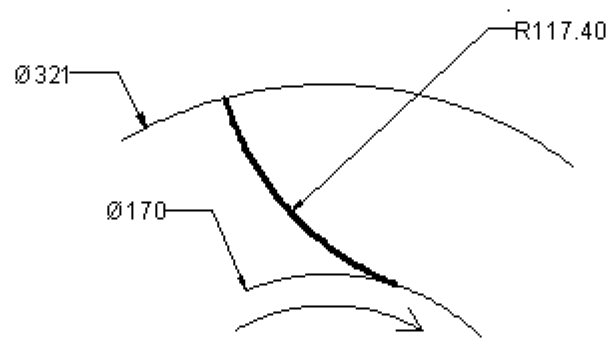

Figure 3.16 Blade Profile as per Unified Design

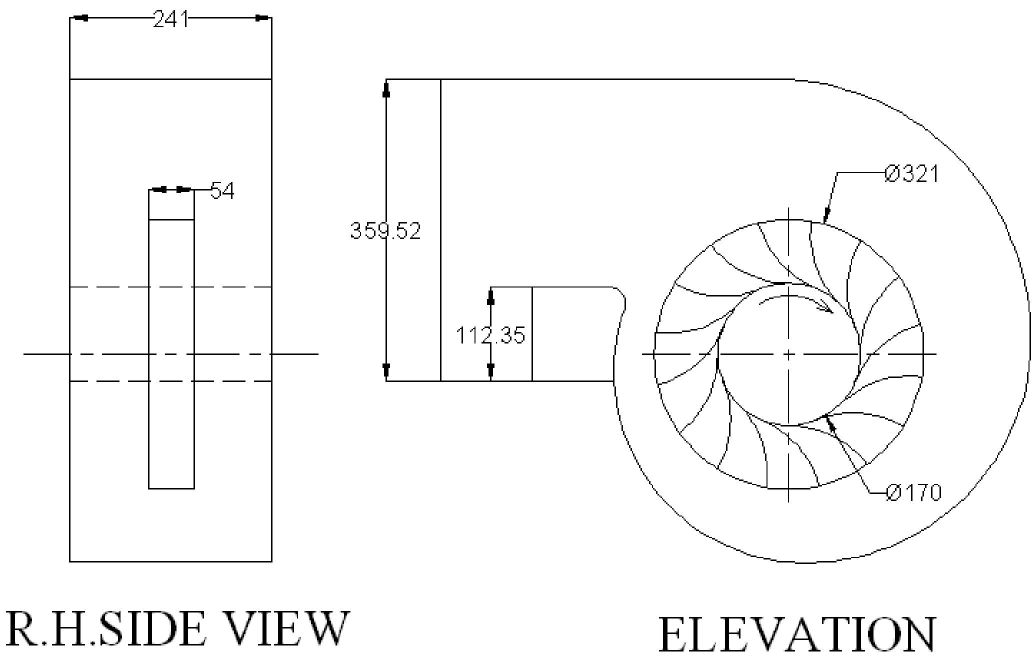

Figure 3.17 Impeller and Volute Casing Assembly as per Unified Design

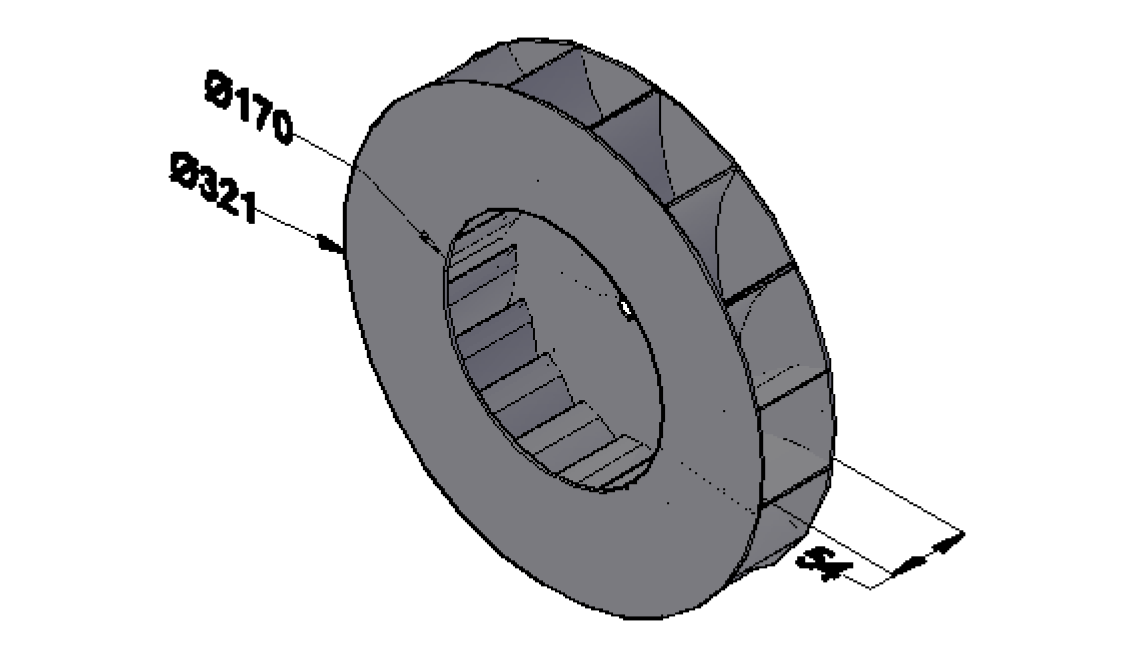

Figure 3.18 FCRT Impeller as per Unified Design

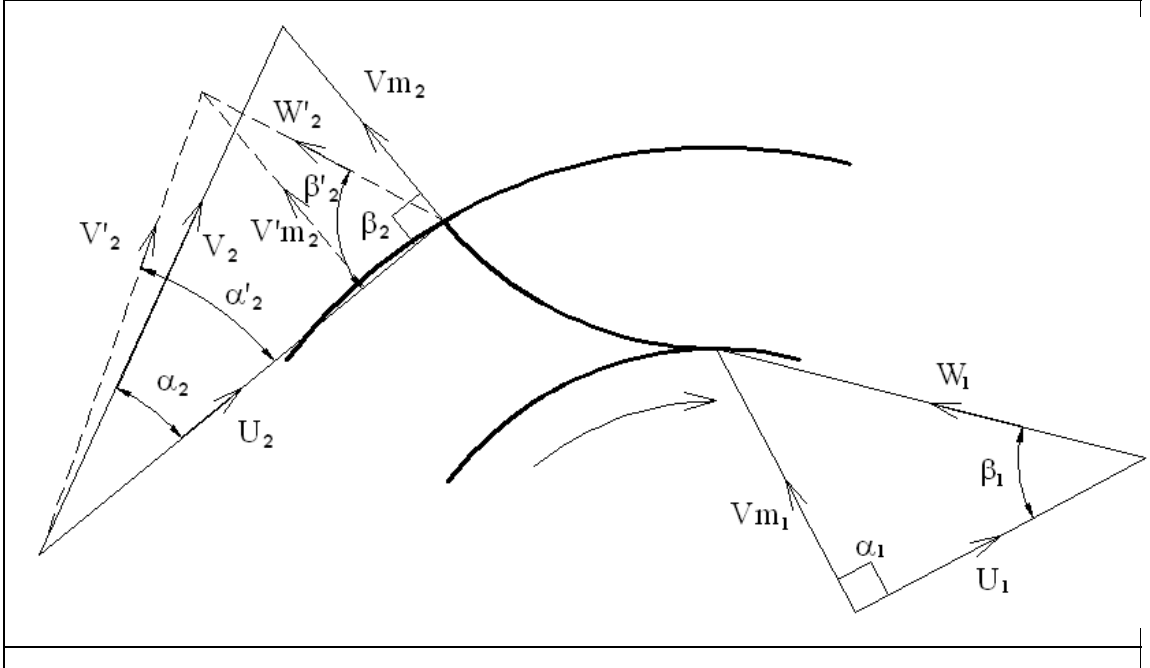

Figure 3.19 Theoretical and Actual Velocity Triangles as per Unified Design

3.8 Snapshots of Visual Basic Computer Program for Unified Design

Making of design calculations and iterations is cumbersome process. Number of variables and coefficients should be taken into account while designing. It needs number of iterations before reaching to optimum parameters. In present era of computerization, it can be made faster by writing sequential program in various softwares like Matlab, Visual Basic and Turbo C. Microsoft Visual Basic 6.0 is the latest and the greatest incarnation of old Basic Language. It gives a complete window base application development system in one package. Visual Basic writes, edits and tests in single window application. VB includes tools that can be used to write and compile help files, active X controls and even internet applications. When we write a statement in Visual Basic language, the statement never has multiple meaning within the same context.

User-friendly software in Visual basic is prepared for designing radial tipped centrifugal fan based on proposed unified design methodology. Few frames from this software are presented in Figure 3.20 to 3.25.

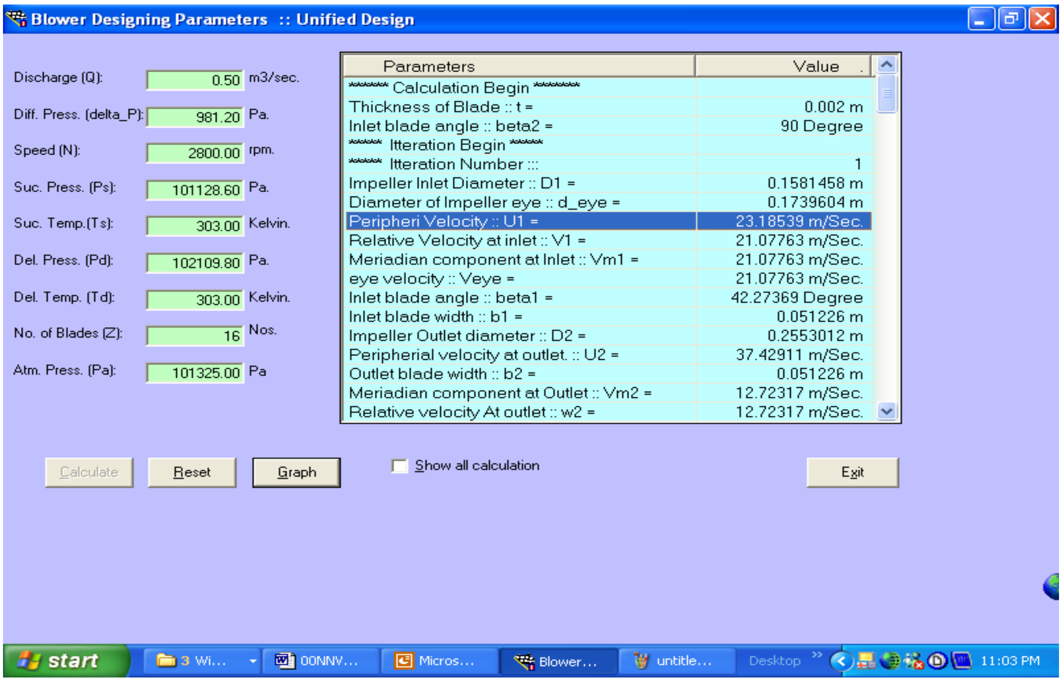

Figure 3.20 Input Data Entry Screen with Start of Calculations

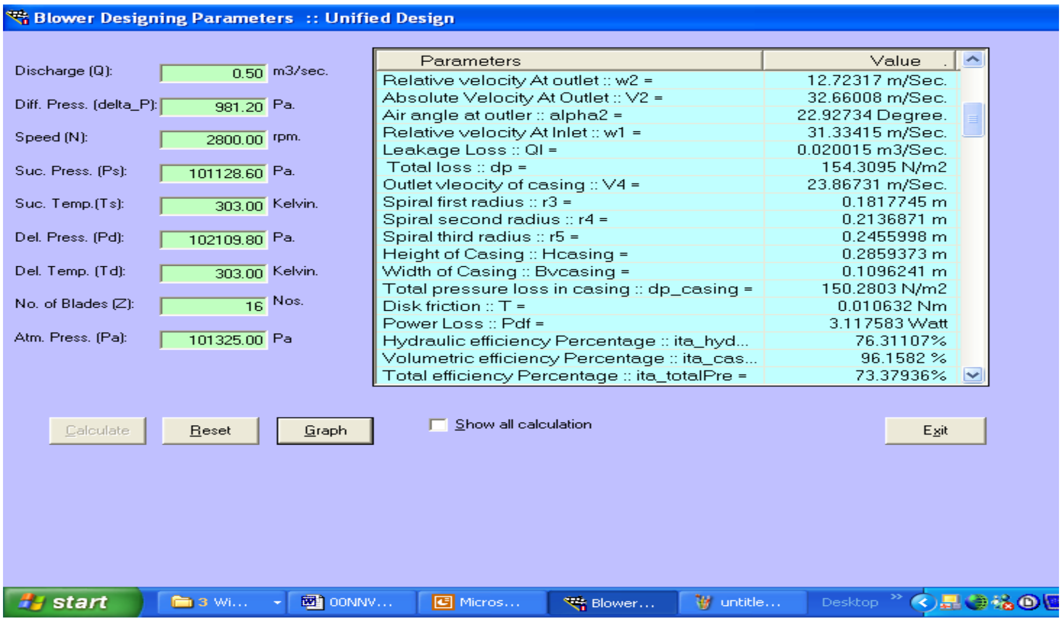

Figure 3.21 Output Parameters

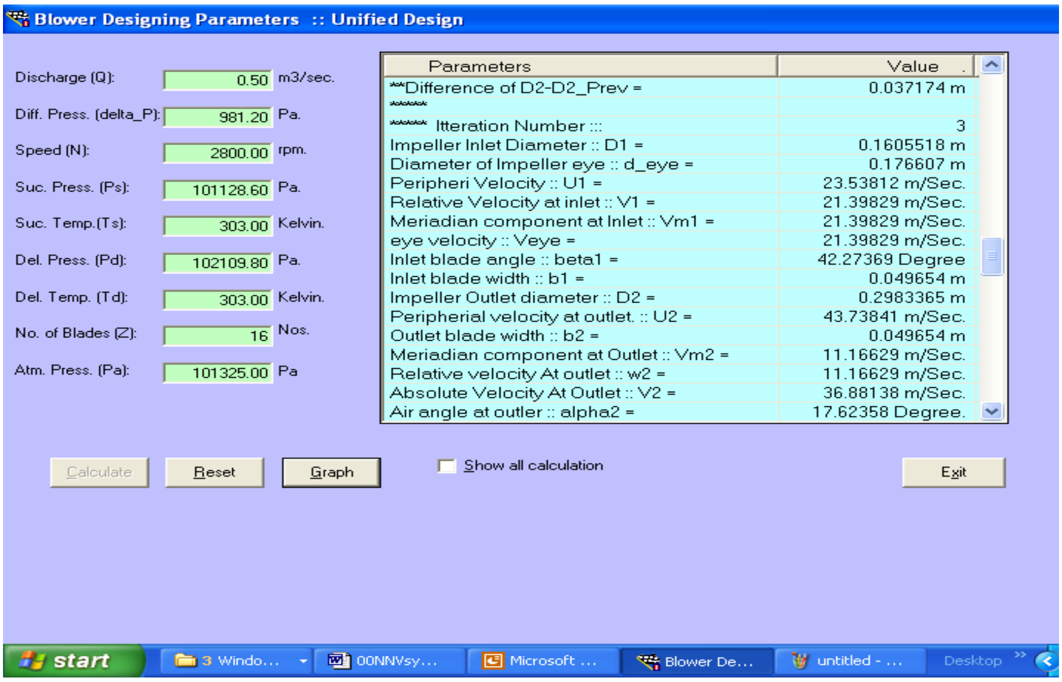

Figure 3.22 Output Parameters at 3rd Iteration

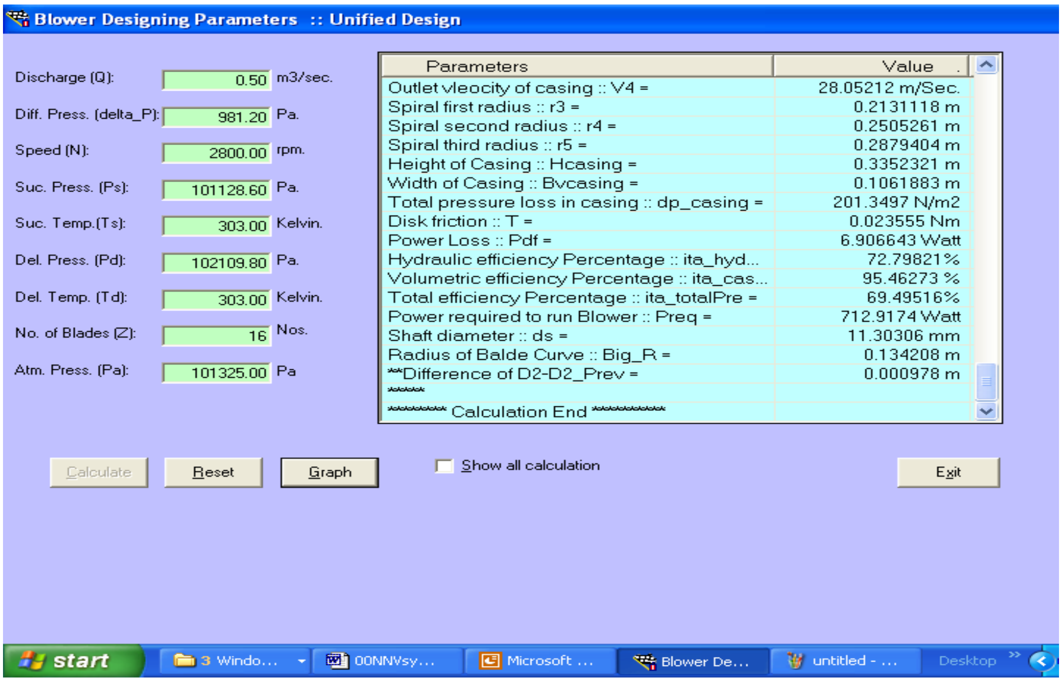

Figure 3.23 Output Parameters at End of Calculations

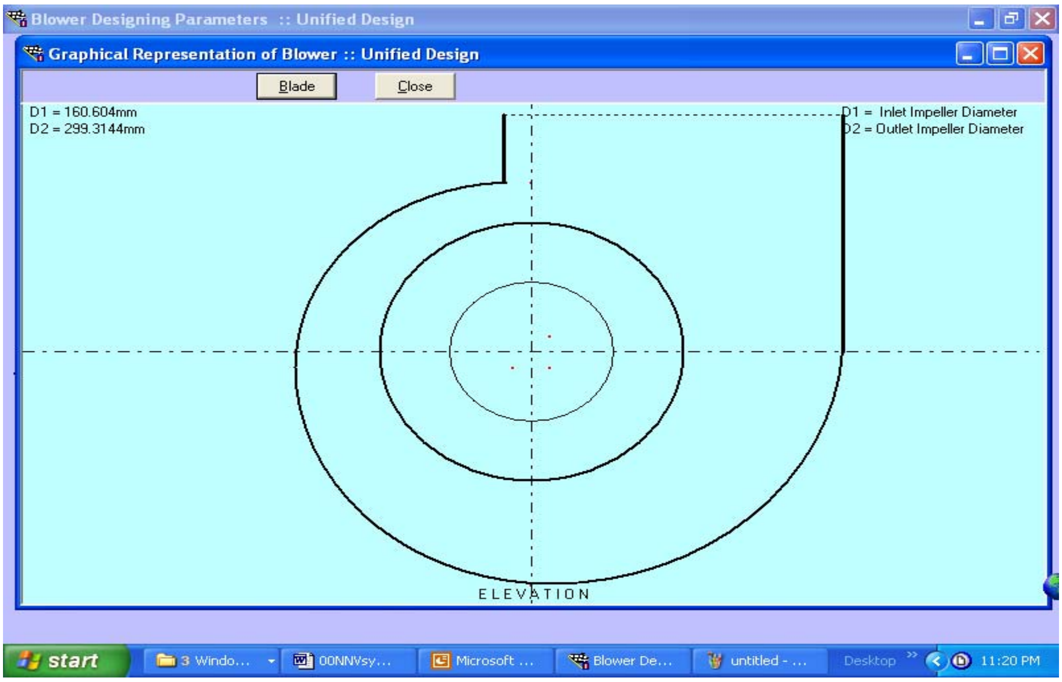

Figure 3.24 Volute Casing Design and Fabrication Drawing

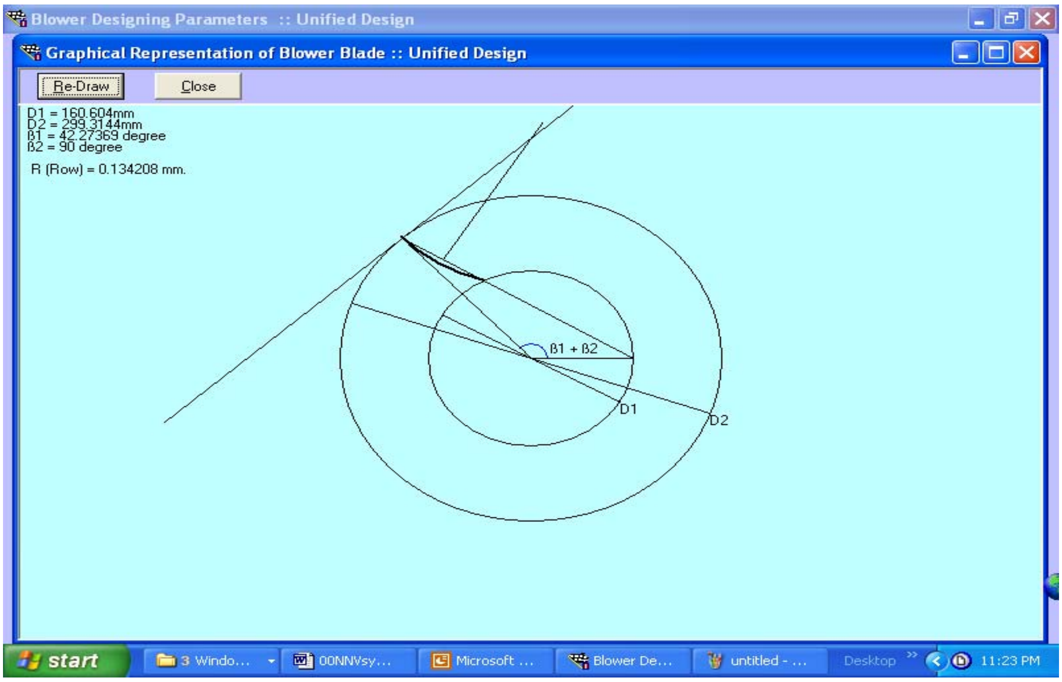

Figure 3.25 Blade Profile Design