Design Comparison of Forward and Backward Curved Radial Tip Blades for Centrifugal Blower

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Abstract- The objective of this paper is to analyze performance of different blades design to get design point performance. This paper is extended towards the comparative assessment of forward and backward leaned radial tipped blades. It is important to recognize that the design of any turbomachine is an interdisciplinary process, involving aerodynamics, thermodynamics, fluid dynamics, stress analysis, vibration analysis, the selection of materials, and the requirements for manufacturing. Though centrifugal blowers have been developed as highly efficient machines, design is still based on various empirical and semi empirical rules proposed by fan or blower designers. The design procedure developed in present work is outlined after performance evaluation of explicit designs which differs widely, suggested by Church, Osborne and retrieved from fundamental principles of fluid flow having minimum assumptions made.

Indexed Terms- backward, centrifugal blowers, forward, stress analysis.

I. INTRODUCTION

The principle involved in the design of a blower is almost very important aspect as that of a centrifugal pump except for the fact that the term "centrifugal pump" is often associated with liquid as its working fluid while the blower is meant to work on air. The blower can therefore be described as a device, which converts 'driver' energy to kinetic energy in a fluid by accelerating it. The key idea here is that the energy created is kinetic energy. Addison established that the faster the impeller revolves or the bigger the impeller is, the higher will be the velocity of the fluid at the vane tip and the greater the energy imparted to the fluid. Blowers are utilized widely in all kinds of engineering applications. There are three main types of blowers used for moving air: axial, centrifugal, and mixed flow. Among the three main types, centrifugal is commonly applied in engineering fields. Depending on the blade design, centrifugal blowers can be categorized into forward curved, backward-curved, radial, and airfoil types. Radial blade with curved surface is known as radial tipped blade. It is a forward curved radial tipped, if curvature of blade is towards the direction of rotations but opposite to direction of rotation signifies backward curved.

Systems that require air flow are normally supplied by one or more blower of various types, drive by a motor. In term if air movement, Bernoulli's theorem states that static pressure plus velocity pressure at upstream is equal to that of the downstream in the direction of airflow plus the friction and dynamic losses between the two measuring point due to fraction with the walls changes in the direction of flow (due to elbows and other fittings) as well as air losses through unintentional leaks.

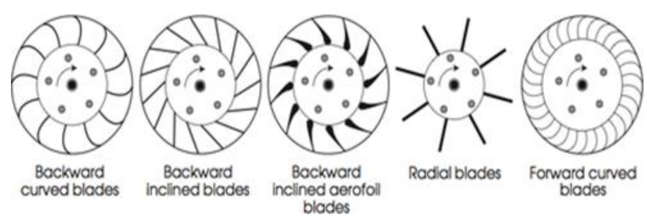

Fig.1 Various Types of Blade

II. DESIGN PROCEDURE OF IMPELLER

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

1) Non dimensional parameters
2) Impeller design
3) Scroll Casing design

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

III. NON DIMENSION PARAMETER

In designing, the losses are difficult to predict and are usually estimated by coefficient base upon tests and experience. Following are widely used non-dimensional parameters.

A. Specific Speed

Specific speed (Ns) is a non-dimensional design index that identifies the geometric similarity of blowers. Blowers of the same Ns but of different size are considered to be geometrically similar, one blower being a size factor of the other.

$$N_s = \frac{\omega\sqrt{Q}}{(gh)^{3/4}} \tag{1}$$

B. Flow Coefficients

The actual volume flow through the compressor is expressed as a non-dimensional parameter flow coefficient, defined as the ratio of radial velocity to the peripheral velocity at the impeller exit.

$$\varphi = \frac{Q}{\pi D_2 b_2 U_2} \tag{2}$$

Where, D2 = impeller tip diameter
b2 = width at the impeller exit

IV. IMPELLER DESIGN

Impeller design is heart of any turbomachine and it must be designed with maximum care to get optimum efficiency. The impeller design is based on the theory of Centrifugal Pumps and Blowers by Austin H. Church and the formal set of ideas from the National Imperial College of Engineering.

A. Velocity Triangles

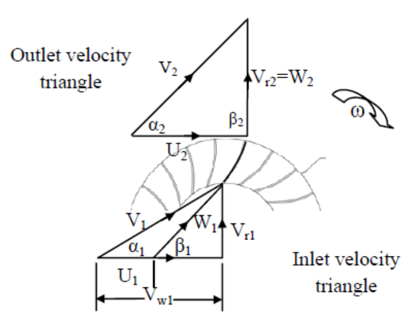

Fig. 2 Velocity Diagram of Backward Curved Radial Tip Impeller

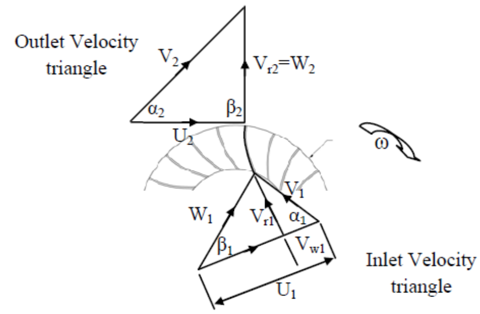

Fig. 3 Velocity Diagram of Forward Curved Radial Tip Impeller

Where, V = absolute velocity
W = relative velocity
U = blade velocity
Vr = radial velocity
α = air angle
β = blade angle

In actual, the inlet air is axial direction in both type of impeller. So the whirl velocity Vw1 = 0, V1=Vr1 and the inlet air angle α1= 0. Therefore, the design procedure for both types of impellers is not different.

B. Adiabatic Head Air Power and Mass Flow Rate

Impeller designed for industrial application of blower, used for fume extraction from texturizing machine is illustrated and clarified.

The overall pressure ratio can be calculated by,

$$\varepsilon_p = \frac{P_d}{P_s} \tag{3}$$

The total adiabatic head can be calculated by,

$$H = RT_a \frac{\gamma}{\gamma-1}\left[\left(\frac{P_d}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{4}$$

The adiabatic air power can be calculated by,

$$P = \rho \times g \times Q \times H \tag{5}$$

The Plant Engineering Lab and P.G. Student (S.V. National Institute of Technology) has been modifying the adiabatic air power to;

$$P = P_s Q\left(\frac{\gamma}{\gamma-1}\right) \times \left[\left(\frac{P_d}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{6}$$

From continuity equation mass flow rate is,

$$m = \rho \times Q \tag{7}$$

C. Impeller Inlet Dimensions and Vane Angles

Assuming the velocity through the impeller eyes V0, the velocity head at the impeller eye is,

$$H = \frac{V_0^2}{2g} \quad (8)$$

$$H = RT_a \frac{\gamma}{\gamma-1}\left(\varepsilon_p^{\frac{\gamma-1}{\gamma}} - 1\right) \quad (9)$$

$$\varepsilon_p = \frac{P_a}{P_0} \text{ and } \varepsilon_p^{0.283} = \frac{T_a}{T_0} \quad (10)$$

$$\rho_0 = \frac{P_0}{RT_0} \text{ and } Q_0 = \frac{\dot{m}}{\rho_0} \quad (11)$$

The blade width at inlet is,

$$b_1 = \frac{Q_0}{\pi \times D_1 \times \varepsilon_1 \times V_1} \quad (12)$$

Where,

ε_p = pressure ratio

T_a, P_a = ambient condition (K and N/m^2)

T_0, P_0 = condition at the impeller eye

ρ_0 = density of air at the impeller eye (kg/m^3)

ε_1 = thickness factor (0.85 to 0.95)

The shaft diameter Ds is based on torque and bending,

$$D_s = \sqrt[3]{\frac{16T}{\pi S_s}} \quad (13)$$

Where,

Ss = shear stress (27.5×10^6 N/m^2 for steel)

T = torque (N-m)

Then, hub diameter DH is made slightly larger than it. The impeller eye diameter is,

$$D_0 = \sqrt{\frac{4Q_0}{\pi V_0} + D_H^2} \quad (14)$$

The vane inlet diameter D1 is may be slightly made greater than the eye diameter. The inlet blade velocity is,

$$U_1 = \frac{\pi D_1 N}{60} \quad (15)$$

The absolute velocity is radial. So, v1 = vr1, vw1 = 0 and α1= 90^o. According to the practical test from Imperial College of Engineering, impeller inlet velocity V1 is less than at eye due to the rise in static pressure. From inlet velocity triangle relative velocity is,

$$W_1 = \sqrt{V_1^2 + U_1^2} \quad (16)$$

Inlet blade angle is,

$$\tan \beta_1 = \frac{V_1}{U_1} \quad (17)$$

D. Impeller Outlet Dimensions and Vane Angles

Impeller outlet design is based on the formal set of ideas of 'Church' and the 'National Imperial College of Engineering'. The outlet blade velocity is,

$$U_2 = \sqrt{\frac{Hg}{K'}} \quad (18)$$

The impeller outlet diameter is,

$$D_2 = \frac{60U_2}{\pi N} \quad (19)$$

Where,

H = pressure head (m)

Pd,Ps = delivery and suction pressure (N/m^2)

γ = isentropic exponent for air (1.4)

K' = the pressure coefficient (0.50 to 0.65)

Outlet blade angle, β2 = 90°

(Radial tipped centrifugal blower)

Neglecting the compressibility effect between impeller eye and impeller outlet, the volume flow rate is constant. The radial velocity at the outlet is,

$$V_{r2} = \frac{Q}{\pi D_2 b_2 \varepsilon_2} \quad (20)$$

Where, ε_2 = thickness factor (0.95 to 0.99)

From the outlet velocity triangle the relative velocity at the outlet is, Vr2 = W2. The absolute velocity is,

$$V_2 = \sqrt{V_{r2}^2 + U_2^2} \quad (21)$$

The outlet air angle is,

$$\tan \alpha_2 = \frac{V_{r2}}{U_2} \quad (22)$$

Taking the flat shroud, blade width b1 = b2. So V2, W2, β2, α2 and Vr2 can be determined.

E. Taking Compressibility Effect between Impeller Eye and Impeller Outlet

The virtual pressure head develop in the impeller is,

$$H_{vir} = \frac{1}{2g}(U_2^2 - U_1^2 + W_1^2 - W_2^2) \quad (23)$$

$$H_{vir} = RT_a \frac{\gamma}{\gamma-1}\left(\varepsilon_{P1}^{\frac{\gamma-1}{\gamma}} - 1\right) \quad (24)$$

$\varepsilon_{p1} = \frac{P_2}{P_0}$ and $\varepsilon_p^{0.283} = \frac{T_2}{T_0}$ (25)

$\rho_2 = \frac{P_2}{RT_2}$ and $Q_2 = \frac{\dot{m}}{\rho_2}$ (26)

So the impeller outlet condition P2, T2, ρ2 and the flow rate Q2 can be obtained.

V. NUMBER OF BLADES

The number of blade in a centrifugal blower can vary from 2 to 64 depending on the application type and size. The number of blade is,

$z = 6.5 \left[\frac{D_2+D_1}{D_2-D_1}\right] \sin\frac{1}{2}(\beta_1 + \beta_2)$ (27)

But the optimum number of blades of a radial impeller can only be truly ascertained by experiments.

VI. LOSSES

By accounting for the stage losses, the actual performance of a blower can be predicted. Losses occur in the both the stationary and moving parts of the centrifugal blower. The various losses are,

A. Pressure Losses in Impeller

Due to friction and turbulence in impeller eye,

$dP_i = \frac{1}{2} \times k_i \times \rho_0 \times V_{eye}^2$ (28)

Turbulence and friction at impeller vane passage,

$dP_{ii} = \frac{1}{2} \times k_{ii} \times \rho_0 \times (W_1 - W_2)^2$ (29)

Turbulence at impeller vane inlet,

$dP_{iii} = \frac{1}{2} \times k_{iii} \times \rho_0 \times W_1^2$ (30)

Here, ki, kii and kiii = 0.1, 0.15 and 0.2

B. Impeller Leakage Loss

To reduce the leakage, wearing rings are fitted to the impeller and casing. These rings are designed with specified clearances.

Head loss due to leakage is,

$H_L = \frac{3}{4}\frac{(U_2^2-U_1^2)}{2g}$ (31)

Pressure due to leakage is,

$P_s = \rho \times g \times H_L$ (32)

The leakage across the ring is,

$Q_L = \pi \times \varphi \times D_L \times \delta\sqrt{\frac{2P_s}{\rho}}$ (33)

Where,
ϕ = flow coefficient
Ps = pressure due to leakage (N/m^2)
DL = mean clearance diameter (m)
δ = diametrical clearance. (m)

A well-designed blower usually comes with a diametral
Clearance of 0.002 and 0.004.

C. Power Loss from Disk Friction

The power required to rotate a disk in a fluid in a fluid is known as the disk friction. This loss occurs due to fluid drag on the reverse surface of the impeller back plate.

$P_{df} = \frac{\pi \times f \times \rho_a \times \omega^2 \times r_2^5}{5}$ (34)

Where,
ρa = density of air (kg/m^3)
ω = speed of blower (rad/s)
f = friction factor (0.005)

D. Diffuser and Volute Losses

The function of casing is to collect the flow and also to diffuse the flow i.e. partially converting its kinetic energy into pressure energy. During this process volute casing creates following overall loss.

$dP_v = \frac{1}{2} \times \rho_2 \times (V_2 - V_2)^2 \times K_v$ (35)

Where,
V2 = absolute velocity at impeller outlet (m/s)
V3 = absolute velocity at casing outlet (m/s)
ρ2 = density at impeller outlet (kg/m^3)

VII. EFFICIENCY

The head in a centrifugal blower is generated by the impeller. The rest of the parts contribute nothing to the head but invites losses like hydraulic, mechanical and leakage. Considering all these pressure and leakage losses in designing blower.

A. Hydraulic Efficiency

The internal losses in the impeller and in the casing due to friction and separation, are sometime called hydraulic losses and it is,

$$\eta_h = \frac{\Delta P_s}{\Delta P_s + dP_i + dP_{ii} + dP_{iii} + dP_v} \quad (36)$$

B. Volumetric Efficiency

The volumetric efficiency in the case of centrifugal pumps and especially fans or blower is,

$$\eta_v = \frac{Q}{Q + Q_L} \quad (37)$$

C. Overall Efficiency

This is the product of hydraulic and volumetric efficiency.

$$\eta_0 = \eta_h \times \eta_v \quad (38)$$

VIII. DESIGN SPECIFICATION AND RESULTS

TABLE I
SPECIFICATIONS FOR DESIGN CALCULATION

Quantity	Symbol	Unit	Value
Discharge	Q	m^3/s	0.5
Suction pressure (abs)	P_s	kPa	101
Delivery pressure (abs)	P_d	kPa	102
Static Pressure difference	ΔP_s	kPa	1.04
Speed	N	rpm	2800
Air Density	P	kg/m^3	1.16
Atmospheric Pressure	P_a	kPa	101.32
Atmospheric Temperature	T_a	K	303
Nature of medium - atmospheric air			

TABLE II
RESULT TABLE FOR IMPELLER INLET DIMENSIONS

Quantity	Symbol	Unit	Value
Blade Velocity	U_1	m/s	23.7
Relative Velocity	W_1	m/s	32.7
Absolute Velocity	V_1	m/s	22.6
Radial Velocity	V_{r1}	m/s	22.6
Impeller Diameter	D_1	mm	160
Impeller width	b_1	mm	51
Eye diameter	D_0	mm	159
Hub diameter	D_H	mm	9
Eye velocity	V_0	m/s	25
Flow rate at eye	Q_0	m3/s	0.508
Air angle	α_1	Degree	90
Blade angle	β_1	Degree	43.6

TABLE III
RESULT TABLE FOR IMPELLER OUTLET DIMENSIONS

Quantity	Symbol	Unit	Value
Blade Velocity	U_2	m/s	42.2
Relative Velocity	W_2	m/s	11.2
Radial Velocity	V_{r2}	m/s	11.2
Absolute Velocity	V_2	m/s	43.5
Impeller Diameter	D_2	mm	287.8
Impeller width	b_2	mm	51
Air angle	α_2	Degree	14.9
Blade angle	β_2	Degree	90
Impeller outlet flow rate	Q_2	m^3/s	0.5
Number of blade	Z	-	21
Power require	P	W	529

TABLE IV
RESULT TABLE OF LOSSES AND EFFICIENCIES

Quantity	Symbol	Unit	Value
leakage Losss	Q_L	m^3	0.01
power Loss from Disk Friction	P_{df}	W	5.
casing Pressure Loss	dP_v	Pa	11
impeller Pressure Loss	dP_{imp}	Pa	201.54
hydraulic Efficiency	η_h	%	83.04
volumetric Efficiency	η_v	%	97.01
overall Efficiency	η_o	%	80.5

IX. DESIGN OF VANES

For smooth flow, we must design the vane such that this angle increases smoothly from β1 to β2. To obtain intermediate values of radii corresponding to intermediate values of the position angle) plotting β, Vr, and W against vane radius, r, for the entrance and outlet stations by using MATLAB program. Referring to the figure,

$$\theta^0 = \frac{180}{\pi}\int_{r_1}^{r} \frac{dr}{r \tan\beta} \quad (39)$$

Using the MATLAB performing the integration with sufficiently close spacing of *r* to obtain angle to get smooth vanes shape. With the result data the impeller vane shape can be design.

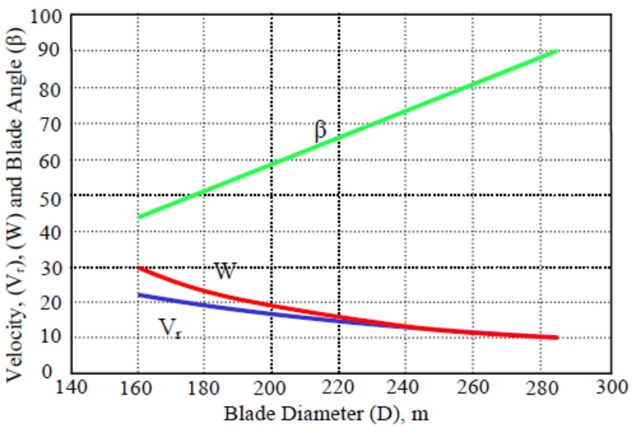

Fig. 4 Plot of Velocities and Vane Angles Against Impeller Diameters

TABLE V
RESULT TABLE TO DESIGN IMPELLER VANE SHAPE

Radius, r (mm)	Blade Angles, β (Degree)	θ (Degree)
80	43.6	0
90	50.87	5.49
100	58.16	9.24
110	65.43	11.74
120	72.72	13.29
130	79.97	14.09
140	87.27	14.30
141.5	88.34	14.32
143.8	90	14.32

X. SCROLL CASING DESIGN

The design concept adopted here is well known as 4-point method. The spiral shapes obtained by three circular sections, as shown in figure below are;

1) R3 =71.2 % of D2,
2) R4 =83.7 % of D2,
3) R5 =96.2 % of D2.

Center of these three arcs are located from the center lines taking interval of 6.25% of impeller diameter D2.

1. Width of housing is 2.14 times b2.
2. Height of housing outlet is 112 % of D2.
3. Radius of tongue is taken 5 to 10 % of D2.

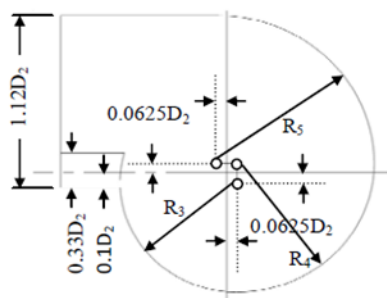

Fig. 5 Scroll Casing

TABLE VI
RESULT TABLE FOR SCROLL CASING

Quantity	Symbol	Unit	Value
Scroll Radius	R3	mm	204.9
Scroll Radius	R4	mm	240.9
Scroll Radius	R5	mm	276.9
Width Of Casing	Wc	mm	109.1
Scroll Height	Hs	mm	322.34
Radius of tongue	Rt	mm	28.7
Center lines interval	-	mm	17.99

XI. DISCUSSION, CONCLUSION AND RECOMMENDATIONS

Advantages of designing the forward and backward radial tip blade are that the velocities triangles for both types are same. So they can be designed without much difficulty. Therefore, comparative assessments of forward and backward curved radial tipped centrifugal blowers are presented with the help of practical test from 'Imperial college of Engineering'. It is representing by graphically in Fig.6, Fig.7 and Fig.8. It can be observed that the pressure head generated by forward curve radial tipped centrifugal blower is obviously higher than that of backward curved radial tipped centrifugal blower but efficiency and power output of the backward blower is at a more advance level than forward curved blower.

The performance obtained for forward and backward curved radial tipped impellers are very much encouraging. This is useful in establishing best operating range, which is essential for any turbo machine, especially fan or blower. At specific point of damping conditions, efficiencies of backward curved radial tipped centrifugal blower are 92% while that for forward curved radial tipped centrifugal blower, these values are 83% respectively at 2800 rpm. For the same speed and same damping condition, one has to go for forward curved impeller if the requirement is of higher pressure head and can go for backward curved impeller if the requirement is for the higher efficiency. It may be stated that design methodology outlined during the period of study and present work may be accepted as the experimentally validated design.

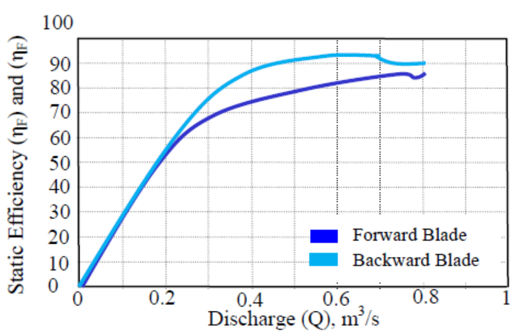

Fig. 6 Discharge versus Efficiency of Forward And Backward Curved Radial Tip Blades

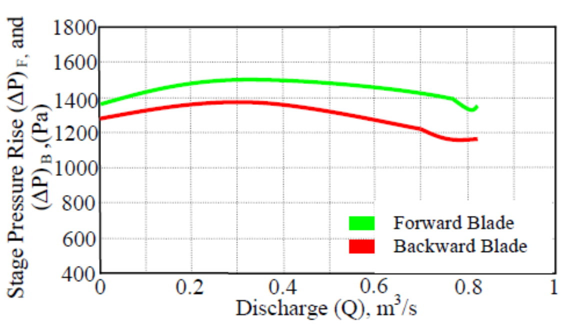

Fig. 7 Discharge versus Stage Pressure Rise of Forward and Backward Curved Radial Tip Blades

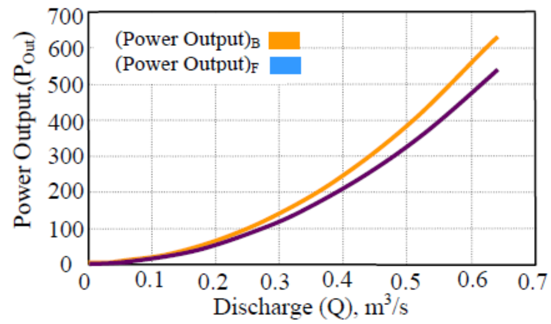

Fig. 8 Discharge versus Power Output of Forward And Backward Curved Radial Tip Blades

REFERENCES

[1] Marco Antonio Rodrigues Cunh and Helcio Francisco Villa Nova, "Cavitation modelling of a centrifugal pump impeller," 22nd International Congress of Mechanical Engineering, Ribeirao Petro, Sao Paulo, Brazil, 2013.

[2] Oyelami A. T., Adejuyibe S. B., Ogunkoya A. K., Analysis of Radial- Flow Impellers of Different Configurations, The pacific Journal of Science and Technology, Vol.13 No. 1, May, 2012.

[3] Singh O P, Khilwani R., Sreenivasulu T., Kannan M., Parametric study of centrifugal fan performance: Experiments and Numerical Simulation, International Journal of Advances in Engineering & Technology, Vol. 1, Issue 2, pp.33-50, May 2011.

[4] P.Usha Shri ans C.Syamsundar, "computational analysis on performance of a centrifugal pump impeller," Proceedings of the 37th National & 4th International Conference on Fluid Mechanics and Fluid Power, IIT Madras, Chennai, 2010.

[5] Mohammed Khudhair Abbas, "cavitation in centrifugal pumps," Diyala Journal of Engineering Sciences, pp. 170-180, 2010.

[6] Performance Analysis and Optimized Design of Backward-curved Airfoil Centrifugal Blowers, Mu-En Hsieh, Department of Mechatronic Engineering, National Taipei University of Technology, Taipei, Taiwan. 2009.

[7] Sharma N. Y., Karanth K. V., Numerical Analysis of Centrifugal Fan for Improved Performance using Splitter Vanes, World Academy of Science, Engineering and Technology Vol.36, 2009.

[8] Liu X., Dang Q., Xi G., Performance Improvement of Centrifugal Fan by Using CFD, Engineering Application of Computational Fluid Mechanics, Vol.2 No.2,pp.130-140, 2008.

[9] E.C. Bacharoudis, A.E. Filios, M.D. Mentzos and D.P. Margaris, “Parametric Study of a Centrifugal Pump Impeller by Varying the Outlet Blade Angle,” The Open Mechanical Engineering Journal, no 2, 75-83, 2008.

[10] S.M.Yayha “Turbines compressors and fans” by Tata McGraw Hill publishing company limited, Delhi, 2005.

[11] Rad S. Z., Finite Element, Model Testing and Modal Analysis of a Radial Flow Impeller, Iranian Journal of Science & Technology, Transaction B, Engineering, Vol. 29, No. B2, 2005.