ISSN 2250-3153

Prediction of Centrifugal Pump Performance on Theoretical and Experimental Observation at Constant Speed of Impeller

Aung Zaw Lynn[*1], Cho Cho Khaing[2], Nay Win Sein[***3]**

[*1] Department of Mechanical Engineering, Technological University (Mandalay), Myanmar
[**2] Department of Mechanical Engineering, Technological University (Mandalay), Myanmar
[***3] Department of Mechanical Engineering, Technological University (Mandalay), Myanmar

DOI: 10.29322/IJSRP.9.08.2019.p92146
http://dx.doi.org/10.29322/IJSRP.9.08.2019.p92146

Abstract- This paper presents the design of impeller and it's performance describes under the prediction of theoretical and experimental observations of single-suction centrifugal pump. The designed pump is single stage centrifugal pump with closed type and it is capability deliver 0.015 m^3/s of water at a head of 20 m. The designed parameters of impeller are 99 mm of inlet diameter and 250 mm of outlet diameter, by mean of vane angles at 20˚ inlet and 23˚outlet. The number of vanes is 6 and input shaft power is 6 hp. The inlet width and outlet width are 20 mm and 12 mm respectively. The discharge diameter is 80 mm to operate the designed head and capacity. The performance analysis of the pump is also presented on design speed. According to theoretical observation, the predicted maximum efficiency is nearly 65% and the expressed actual efficiency of designed pump is 61%. Also experimental results are 66.67% of maximum efficiency without consideration overall losses and actual capacity of designed pump is 0.0215 m^3/s at the same condition. Therefore, the designed efficiency has a satisfactory value. The designed single-suction centrifugal pump can fulfill the requirements of water pumping system for irrigation, and agricultural application.

Index Terms- head, flow rate, speed, efficiency

I. Introduction

A pump is a device which lifts water from a lower level to a higher level at the expense of mechanical energy. It consists an impeller rotating within a volute casing. Radial flow pumps are centrifugal pumps in which the fluid is pumped perpendicularly to the pump shaft. The flow mechanism in a centrifugal pump can generally be described as follows: Through a suction flange the liquid flows through the suction hub into the rotating impeller due to an energy fall. The pump unit absorbs mechanical energy from a drive motor through a shaft. The blades of the impeller which is permanently fixed on the shaft exert a force on the fluid and increase its angular moment. Pressure and absolute speed increase as a result. Consequently energy is being transferred to the fluid. The energy which is present in kinetic form as an increased absolute speed is usually converted into additional static pressure energy by a diffuser device. Pumps are widely used for irrigation and are most common where pumping from surface water supplies such as river, lakes and streams and rising water to a higher level.

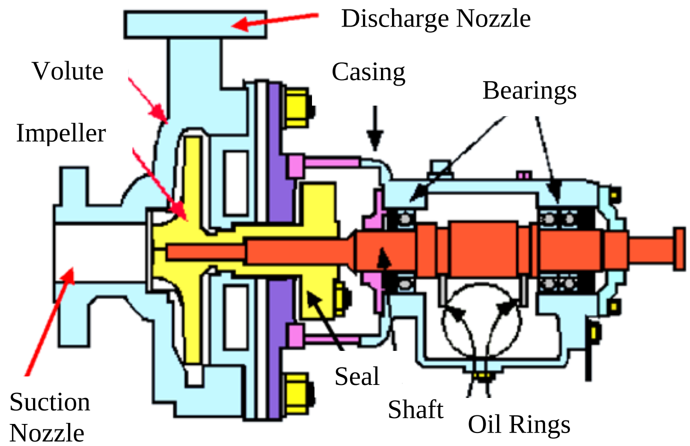


Fig.1 Single Suction Centrifugal Pump

II. Design Of Centrifugal Pump

The two main components of centrifugal pump are impeller and casing. The impeller is enclosed in a water tight casing that the kinetic energy of water is converted into pressure energy before the water leaves the casing. The other components are suction pipe, discharge pipe, shaft, bearing, wear rings, stuffing box, mechanical seal and various types of valves and gauges.

When the overall design of pump is considered, the shape of an impeller is the most important for optimum efficiency. Impeller design should be in such a way that, losses must be as low as possible. The design of a pump's impeller can be divided into two parts. The first is the selection of proper velocities and vane angles needed to obtain the desired performance with the best possible efficiency. The second is the layout of the impeller for the selected angles and areas.

The specifications of pump that will be designed are:

Pump head,	H	= 20 m
Discharge,	Q	= 0.9 m^3/min
	Q_s	= (Q/60) m^3/s

= 0.015m^3/s
Rotational Speed, n = 1800 rpm
Density of water, ρ = 1000 kg/m^3

A. Design of impeller

Specific speed is an essential criterion to determine the impeller shapes. It is mathematically expressed as

$$n_s = \frac{n \times \sqrt{Q}}{H^{3/4}} \tag{1}$$

In this design, calculated value of specific speed based on required head and capacity is 180 rpm and it is within the range of low specific speed pump that is greater than 80 and less than 600. So, end-suction type single stage centrifugal pump with closed impeller is chosen.

Pump efficiency, η is assumed by using Fig. A1. and also the diameter of suction pipe D_s can be estimated from this chart. The discharge pipe diameter D_d is usually selected equal to or one size smaller than that of the suction pipe. Thus, velocities in these pipes are given by

$$V_s = \frac{Q_s}{\pi \frac{D_s^2}{4}}, V_d = \frac{Q_s}{\pi \frac{D_d^2}{4}} \tag{2}$$

Input power of centrifugal pump can be determined by following equation.

$$L = \frac{\rho Q_s g H}{\eta} \tag{3}$$

For charge condition of the pump work, maximum shaft power or rated output of an electric motor L_r (kW) is decided by using Equation (4).

$$L_r = \frac{(1+F_a) \times L}{\eta_{tr} \times 1000} \tag{4}$$

Where, F_a is the allowance factor, and 0.1~ 0.4 for an electric motor and larger than 0.2 for engines And then, η_{tr} is the transmission efficiency, and 1.0 for direct coupling and 0.9 ~ 0.95 for belt drive.

The shaft diameter at hub section of impeller is

$$d_s = \sqrt[3]{\frac{16\,T}{\pi\,\tau}} \tag{5}$$

Where, T is the torsional moment and it can be estimated by

$$T = \frac{60\,L_r}{2\,\pi\,n} \tag{6}$$

Allowable shear stress of material of shaft, τ is 24.5 MPa because the main shaft is made of S30C. The estimated shaft diameter will be increased because it is difficult to predict the bending moment at this time.

The hub diameter, D_h is usually taken from 1.5 to 2.0 times of the shaft diameter and the hub length, L_h is from 1.0 times to 2.0 times of the shaft diameter.

The diameter of impeller eye, D_o is calculated by

$$D_o = \sqrt{\frac{4Q_s'}{\pi V_{mo}} + D_h^2} \tag{7}$$

Where, the flow rate through the impeller, Q'_s is Q/η_v and volumetric efficiency η_v is estimated by

$$\eta_v = \frac{1}{1 + \frac{1.124}{n_s^{2/3}}} \tag{8}$$

For Equation (7), the velocity at the eye section is given by

$$V_{mo} = K_{mo}\sqrt{2gH} = (1.5 \sim 3.0) \leq V_{m1} \tag{9}$$

$$K_{mo} = (0.07 \sim 0.11) + 0.00023\,n_s \tag{10}$$

For a fluid flowing through the rotating impeller, u is the tangential velocity, V is the absolute velocity and v is the relative velocity of a fluid particle to impeller rotation. The angle between V and u is α and the angle between v and u is β and it is the angle made by tangent to the impeller vane and a line in the direction of motion of the vane. The tangential component and radial component of absolute velocity V are V_u and V_r respectively. The outlet velocities triangle with solid lines represents the actual diagram.

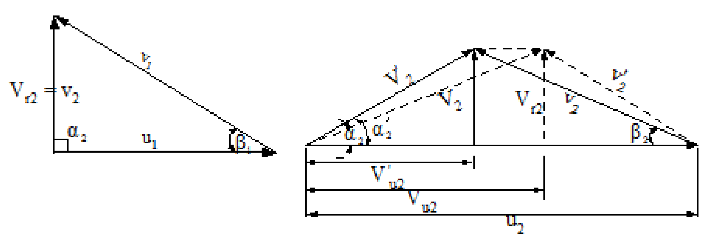


Fig. 2 Impeller Inlet and Outlet Velocity Diagrams

The parameters K_u (speed constant), K_{m1}, K_{m2}, and D_1/D_2 are obtained on the value of specific speed in Fig A2.

The outlet diameter D_2,

$$D_2 = \frac{u_2 \times 60}{\pi \times n} \tag{11}$$

Where, the peripheral velocity at impeller outlet is

$$u_2 = K_u\sqrt{2gH} \tag{12}$$

The peripheral velocity at the inlet is also expressed by

$$u_1 = \frac{\pi D_1 n}{60} \tag{13}$$

And then, flow velocities at the inlet and outlet are

$$V_{r1} = K_{m1}\sqrt{2gH} \quad \text{and} \quad V_{r2} = K_{m2}\sqrt{2gH} \tag{14}$$

If the incoming flow has no pre-rotation, the blade angle β_1 (deg) is given by

$$\beta_1 = \tan^{-1}[\frac{K_{b1}V_{r1}}{u_1}] \approx \tan^{-1}[\frac{V_{r1}}{u_1}] + (0 \sim 6) \tag{15}$$

Where, K_{b1} = 1.1~1.25

The amount of outlet angle β_2 usually has between 15˚ and 35˚. So, the vane outlet angle is assumed that β_2 = 23˚ in this design. From the velocity triangles, inlet and outlet relative velocities are

$$v_1 = \frac{u_1}{\cos\beta_1} \quad \text{and} \quad v_2 = \frac{V_{r2}}{\sin\beta_2} \tag{16}$$

The virtual tangential component V_{u2} of V_2 is

$$V_{u2} = u_2 - \frac{V_{r2}}{\tan\beta_2} \tag{17}$$

For radial-type impellers, the slip factor, η_∞ varies between 0.65 and 0.75and it is assumed that η_∞ = 0.7 average. Thus, the actual tangential component V'_{u2} of V'_2 is

$$V'_{u2} = V_{u2}\eta_\infty \tag{18}$$

Thus, the actual outlet is found by

$$\tan\alpha'_2 = \frac{V_{r2}}{V'_{u2}} \tag{19}$$

The absolute outlet velocity from outlet velocity diagram is

$$V'_2 = \sqrt{V_{r2}^2 + V'^2_{u2}} \tag{20}$$

The number of blades, Z is decided by using the Plfeiderer formula.

$$Z \approx 6.5\frac{D_2 + D_1}{D_2 - D_1}\sin[\frac{\beta_1 + \beta_2}{2}] \tag{21}$$

In this design, blade thickness and shroud thickness are taken as 2.5 mm and 3.0 mm respectively for $D_2$is greater than 200 mm.

The inlet passage width b_1 and outlet passage width b_2 are calculated by

$$b_1 = [\frac{Q'_s}{\pi D_1 V_{r1}}][\frac{\pi D_1}{\pi D_1 - S_1 Z}] \quad \text{and} \quad b_2 = [\frac{Q'_s}{\pi D_2 V_{r2}}][\frac{\pi D_2}{\pi D_2 - S_2 Z}] \tag{22}$$

Where, S_1 is (δ_1/sin β_1), S_2 is (δ_2/sin β_2), and δ_1 and δ_2 are blade thicknesses near the leading edge and trailing edge respectively. Moreover, S_2 can also be determined by the following relationship equation.

$$\frac{\pi D_1}{(\pi D_1 - S_1 Z)} = \frac{\pi D_2}{(\pi D_2 - S_2 Z)} \tag{23}$$

The impeller blade is drawn by three circular arcs method with solid work software.

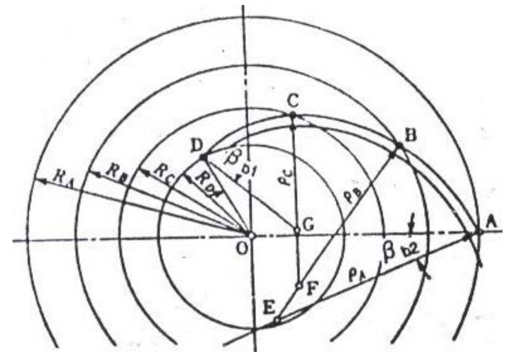


Fig. 3 Curvature of Impeller Blade

$$\rho_A = \frac{(R_A^2 - R_B^2)}{2(R_A\cos\beta_2 - R_B\cos\beta_B)},$$
$$\rho_B = \frac{(R_B^2 - R_C^2)}{2(R_B\cos\beta_B - R_C\cos\beta_C)} \quad \text{and}$$
$$\rho_C = \frac{(R_C^2 - R_D^2)}{2(R_C\cos\beta_C - R_D\cos\beta_1)} \tag{24}$$

Where, R_A, R_B, R_C and R_D are base circle radii, R_A=D_2/2 and R_D = D_{1h}/2.

$$R_B = R_A - \frac{R_A - R_D}{3} \quad \text{and} \quad R_C = R_B - \frac{R_A - R_D}{3} \tag{25}$$

The angles between β_1 and β_2 are divided into three angles.

B. Design of volute casing

Design of volute casing is calculated depending on the D_2 and the basis of constant average flow velocity in volute casing. The volute casing increases proportionally in size from cut water to the discharge nozzle. In rear velocities distribution, across volute section is not uniform. Volute angle is read from volute constant chart shown in Fig. A3 and in this design, the volute angle, α_v is 8˚ based on n_s value.

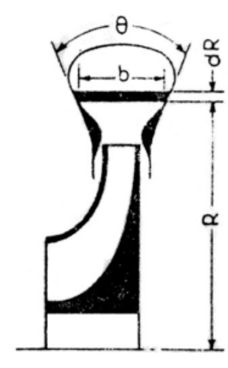


Fig. 4 Section through Volute Casing [4]

The width of the volute at any point may be calculated from

$$b = b_3 + 2x \times \tan(\theta/2) \tag{26}$$

Where, x is the distance between any radius R and impeller outside radius R_2. The volute is designed by determining the angle Φ˚ measured from and assumed radial line by tabular integration of Equation (27).

$$\Phi^{o} = \frac{360\ R_2 V'_{u2}}{Q} \int_{R_2}^{R_\Phi} \frac{bdR}{R} = \frac{360 R_2 V'_{u2}}{Q} \sum_{R_2}^{R_\Phi} b \frac{\Delta R}{R} \qquad (27)$$

The tongue angle of volute casing is determined by

$$\Phi_t^{o} = \frac{132\ \log_{10}\ R_t / R_2}{\tan\ \alpha'_2} \qquad (28)$$

Volute wall thickness is chosen according to suction pipe diameter and it is taken as 6 mm since the suction pipe diameter is within 100 and 150 mm in this design.

No	Descriptions	Symbols	Results
1	Input Power	L	6 hp
2	Shaft diameter	d_s	34 mm
3	Hub diameter	D_h	51 mm
4	Hub length	L_h	68 mm
5	Impeller eye diameter	D_o	97 mm
6	Impeller inlet diameter	D_1	99 mm
7	Impeller outlet diameter	D_2	250 mm
8	Inlet angle of impeller blade	β_1	20°
9	Outlet angle of impeller blade	β_2	23°
10	Impeller passage width at inlet	b_1	20 mm
11	Impeller passage width at outlet	b_2	12 mm
12	Number of impeller blades	Z	6 blades
13	Base width of volute casing at D_2	b_3	24 mm
14	Volute tongue angle	Φ°_t	15.51°
15	Discharge nozzle diameter	D_d	264 mm

A. *Modelling of Centrifugal Pump*

The three dimensional centrifugal pump is created by using Solidworks Software.

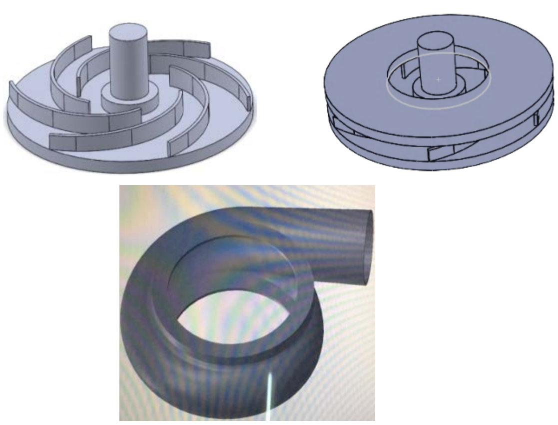


Fig.5 Three Dimensional View of Impeller and Casing

IV. Impeller Pattern Making Processes

The design of wood pattern of impeller for 6 hp output makes by using OPTIMUM F105(CNC MILLING MACHINE). After that it uses in the sand moulding to do the foundry process. Finally, the impeller casting produce from foundry was machining with Lathe machine.

III. Designed Results Of Centrifugal Pump

A. *Calculated Results*

The calculated results for both impeller and casing design of centrifugal pump are clearly expressed in Table I.

Table I

Calculated Results of single-suction Centrifugal Pump Design

Fig.6 (a) 3 Axis CNC Milling Machine, (b) Impeller Core, (c) Final Product of Impeller

V. Description Experimental Set Up

(a) (b)

(c)

Fig. 6 (a) 3-axis CNC Milling Machine, (b) Impeller Core, (c) Final Product of Impeller

VI. Description Of Experimental Set Up

The description of experimental set up is single suction centrifugal pump with coupling a 18 hp diesel engine. It consists of main components such as impeller, volute casing, gate valve, discharge pressure gauge, suction pressure gauge, suction pipe, 200 gallons fiber tank and supply lake. A speed measurement device (Tachometer) use to measure in pump designed speed operation. Also stop watch is as a timer to take time in filling water tank with discharge water.

Fig.7 Description of Experimental Observation

VII. Performance Analysis At Constant Speed Of Impeller

Performance Characteristic curves of actual head and efficiency on capacity are presented at the design speed of impeller in Fig 8. The actual head is achieved by subtracting of shock losses, diffusion losses, friction losses, circulatory flow effect, leakage losses and mechanical losses from the theoretical head.

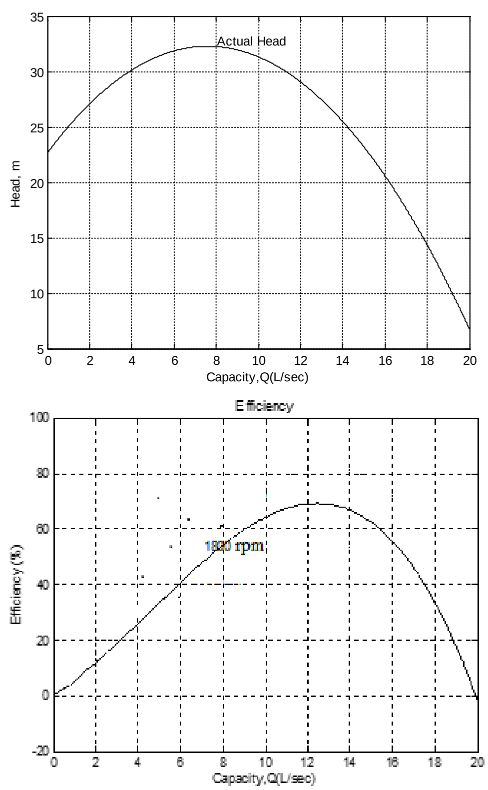


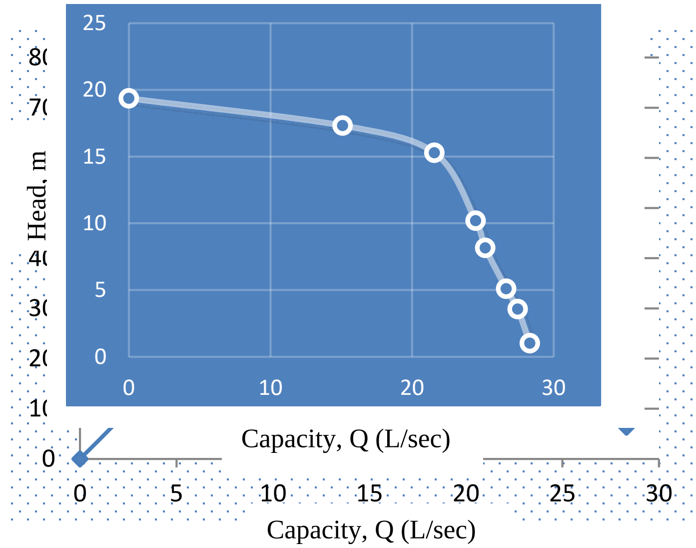
Fig. 8 Performance Characteristic Curves of Actual Head and Efficiency on Capacity in designed Speed

VIII. Performance Results At Constant Speed Of Impeller

Table. II

. Experimental Results In Eight Points Gate Opening

Sr. No	Gate Opening	Discharge Pressure MN/m^2	Head (m)	Time (Sec)	Q (Lit/ sec)	Efficiency (%)
1	Fully Opened	0.010	1.019	32	28.317	6.323
2	6/7	0.035	3.5680	33	27.459	21.47
3	5/7	0.050	5.0970	34	26.651	29.77
4	4/7	0.080	8.1550	36	25.171	44.98
5	3/7	0.100	10.194	37	24.490	54.71
6	2/7	0.150	15.291	42	21.575	66.67
7	1/7	0.170	17.329	60	15.102	57.35
8	Fully Closed	0.190	19.368	0	0	0



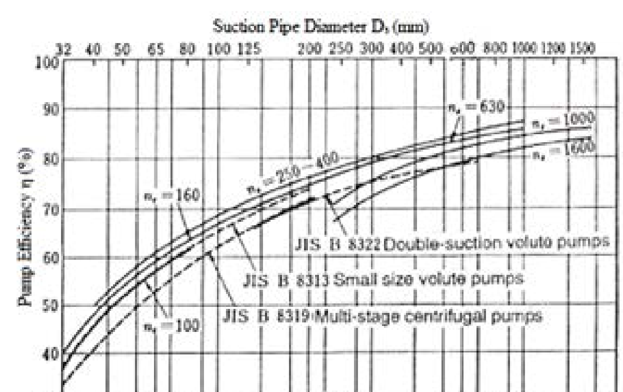
ig. 9 Performance Characteristic Curves of Head and Efficiency on Capacity in designed Speed

IX. Conclusion

he designed pump is intended to use in agricultural application especially for river pumping project. The clearance between impeller and tongue of volute is 3 mm. The diameter of discharge flange is 80 mm. The thickness of volute casing to withstand the discharge pressure, 6 mm is selected depending upon the suction pipe diameter. If the performance is predicted, the maximum efficiency has nearly 65%. From Fig 8, the designed centrifugal pump satisfies for head of 20 m and capacities of 15 L/sec at speed of 1800 rpm. This pump is observed at design speed 1800 rpm by setting up the throttling valve in discharge nozzle with 8 points opening. In each point, impeller operates at constant speed. Under the test run the pump head is reached to 19 m of head at fully closed position. Before the maximum shut of head at 1/7 gate opening, the capacity of water is being in design flow rate. According to the experimental observation, the maximum pump efficiency is 66.67% at 15.29 m of head and 21.575 L/sec of capacity. Therefore this paper recommends the designed pump is capable to do nearly in designed head and flow rate.

Appendix

A.Figure



http://dx.doi.org/10.29322/IJSRP.9.08.2019.p92146

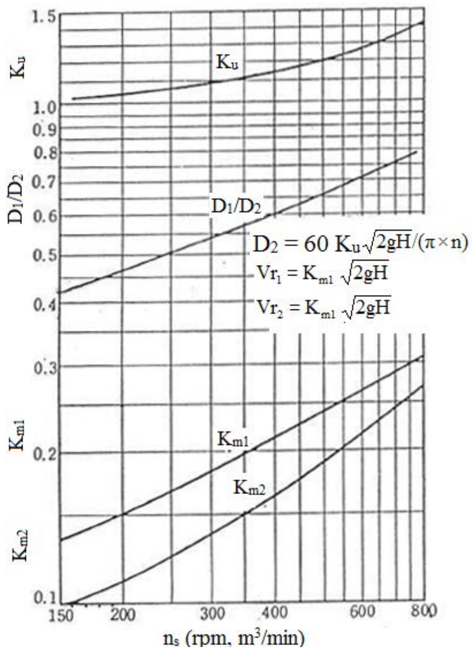


Fig. A2 Stepanoff Chart

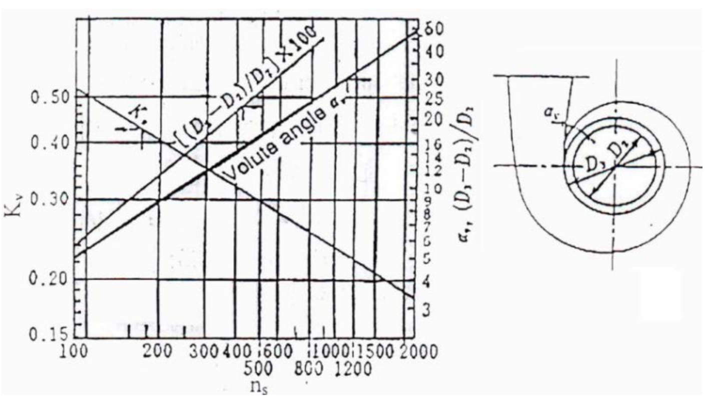


Fig. A3 Fig. A3 Design Parameters for Volute Casing [3]

ACKNOWLEDGMENT

The author would like to express his special thanks to Nay Win Sein, Associate Professor, Department of Mechanical Engineering, Technological University (Mandalay) for his valuable experience in production area, Cho Cho Khaing, Associate Professor, Department of Mechanical Engineering, Technological University (Mandalay) for her advice and some under graduate students for collaboration in observation test.

REFERENCES

[1] EBARA Corporation, "Application of Pumps in Agriculture" Tokyo, Japan, November, 1999.
[2] Kyushu Institute Technology, Training Course, "Fluid Mechanics of Turbomachinery", Japan, 1996.
[3] Stepanoff. A. J, "Centrifugal and Axial Flow Pumps", 1957.
[4] Khin Maung Aye, U, "Fluid Machinery for Mechanical Engineers", December, 2000.
[5] Touzson, J, "Centrifugal Pump Design", John Wiley and Sons, Inc, 2000.
[6] Ausstin, H. Church, "Centrifugal Pump and Blower", 1972, John Wiley and Sons, Inc, New York.
[7] Christian Brix Jacobsen, "Centrifugal Pump", Department Head, Structural and Fluid Mechanics, R&T,

AUTHORS

First Author -Aung Zaw Lynn, Ph.D (Mech), Technological University (Mandalay), aungzawlynn.mdy@gmail.com
Second Author - Cho Cho Khaing, M.E (Mech), Technological University (Mandalay), khaingcho999@gmail.com
Third Author - Nay Win Sein, M.E (Mech), Technological University (Mandalay), naywinsein@gmail.com