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COMPUTER AIDED DESIGN FOR THE IMPELLER OF THE CENTRIFUGAL PROCESS PUMP

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Abstract

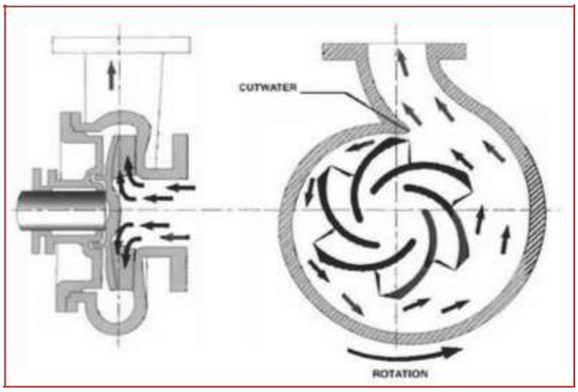
The centrifugal process pump is as important in industry as a heart in the human body. In today's world of cut-throat competition, customer's expectations are very high and changed very rapidly. Hence, it is very essential to develop efficient centrifugal process pump to satisfy customer's requirements. An impeller is a critical part of a centrifugal process pump. It should be designed in such a way that the pump will consume less power and give maximum head at given discharge. This work is tedious; chances of errors are there if done manually. Latest developments in the field of the computer can be employed to solve the problem. In the present work, an actual case is studied to design an impeller with the computer assistance. The dimensions of the impeller are calculated with a programming environment (i.e. Visual Basic) as a Graphical User Interface (GUI). It is attempted to develop a user-friendly program that depends on the Microsoft Access database to extract stored data. Finite Element Analysis (FEA) is done for the component and validated with analytical method.

Keywords: Process pump, impeller, Visual Basic, GUI, FEA

INTRODUCTION

The centrifugal pump is a kinetic device to move a liquid from one location to another location. It consists an impeller, immersed in a liquid, rotates within a casing. An impeller is having number of blades and is mounted on a shaft projected outside of the casing. As shown in Figure 1, the liquid enters the pump near the impeller axis, and the rotating impeller sweeps the liquid out toward the ends of impeller blade at high pressure. Head and capacity are two important parameters for any pump.

Figure.1 Impeller as a component of centrifugal process pump



Source: Engineer's world

Due to influence of impeller geometry, designers often face a problem while selecting a number of independent geometrical parameters for achieving the desired performance from the pump. As per S. Yedidiah S. (1996), the blade geometry has a significantly greater effect on the developed head than the outlet angle of the impeller blades. The blade geometrise are chosen in such a manner that the theoretical components increased linearly from the inlet tips to their outlet tips. Dupont P.(2002) has stated that the cavitation is the limiting factor in the pump design. It is better to keep outlet angle low and restrict difference between outlet and inlet angles to avoid instability. As per Anderson, increasing the impeller speed increases the efficiency of centrifugal pumps.

DESIGN CRITERIA

In the present study, the dimensions for the impeller with the radial ribs and partial shroud at the back are calculated for the discharge 20 m^3 /hr (5.56 lit/sec), specific speed of 2900 rpm and 20 meter of total head. The material for impeller is considered as 04Cr17Ni12MO2 (AISI-316) having ultimate stress (σu) 490 N/mm^2 and

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HB 192. The given value of head and discharge are taken from company catalogue. (i.e., PPI Pump, Ahmedabad). Flow to inlet (eye) of the impeller will be normal to the line of the motion of the relevant portion of vane without pre whirl. The calculation of various parameters of the vane profile is explained by Austin C. (1972, pp: 93-96). Actual flow (Qa) considering leakage loss 5 % is as per equation (1),

$$Qa = 1.05 \quad 20 \tag{1}$$

$$= 21 \text{ m}^3 \text{/hr}$$

$$= 0.0058 \text{ m}^3 \text{/sec}$$

Suction pipe diameter is taken as 25 mm to calculate suction area (Asuc) as per equation (2)

$$Asuc = \left(\frac{\pi}{4} \times 0.025^2\right) \tag{2}$$

$$= 0.00049 \text{ m}^3$$

From that, suction velocity,

$$Vsuc = Qa/Asuc$$

$$=0.058/0.00049$$

$$=11.89 \text{ m/sec}$$

But recommended velocity at the inlet of impeller is 2 m/sec to 5 m/sec. So, it is reduced by provided divergent in the suction flange of the casing. Let the velocity at eye Vo is equal to 2.5 m/sec. For calculation of eye diameter (Do), the value of actual discharge has been utilized into the equation (3), assume nut diameter (Di) is as 22 mm, at the eye.

$$Qa = Vo\left(\frac{\pi}{4} D_0{}^2 - \frac{\pi}{4} D_i{}^2\right) \tag{3}$$

$$= 2.5\left(\frac{\pi}{4} D_1{}^2 - \frac{\pi}{4} 0.022^2\right)$$

$$D0 = 0.059 \text{m}$$

Let velocity at entry (Vf1),

$$Vf1 = Vo \times 1.05$$

$$= 2.5 \times 1.05 \text{ m/sec}$$

$$= 2.625 \text{ m/sec}$$

From that, Radius of impeller eye, R1 =22.85 mm Tangential velocity at outlet (U2) is found from equation (4).

$$U2 = (2 \times g \times H)^{0.5} \tag{4}$$

$$= (2 \times 9.81 \times 20)^{0.5}$$

$$= 19.81 \text{ m/sec}$$

The value of tangential velocity at outlet (U2) from the equation (4) is to be put in equation (5) to calculate overall diameter of impeller disc (D2).

$$U2 = \frac{\pi \times D_2 \times N}{60} \tag{5}$$

$$19.81 = \frac{\pi \times D_2 \times 2900}{60}$$

$$D2 = 0.131 \text{ m}$$

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Velocities of flow Vf1 and Vf2 are radial at inlet as well as at outlet. The entrance angle (vane inlet angle) β1 should be within limits 15^0 to 50^0. The discharge angle (vane outlet angle) β2 should be within limits 15^0 to 35^0. To find, vane angle, calculation of tangential (peripheral) velocity of impeller at inlet (U1) is required as per stated by equations (6)

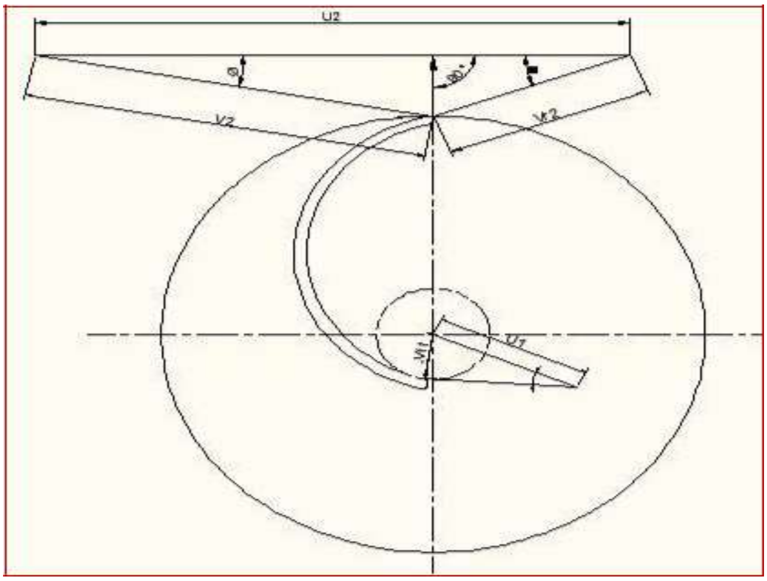
$$U1 = \frac{\pi \times D_0 \times N}{60} \quad (6)$$

$$= \frac{\pi \times 0.059 \times 2900}{60}$$

$$= 8.92 \text{ m/sec}$$

From Figure 2, inlet as well outlet velocity triangles are explained by Austin C. (1972, pp: 25).

Figure 2 Velocity triangle of an Impeller



Vane angle (β1) at inlet can be calculated from equation (7),

$$\tan\beta 1 = \frac{V_{f1}}{U1} \quad (7)$$

$$= \frac{2.625}{8.92}$$

$$= 0.29$$

$$\beta 1 = 16.39^0$$

As it is preferable to have β2 > β1, β1 is assumed 17.5^0, velocity of whirl at outlet (Vw2) can be found using equation (8),

$$Vw2 = U2 = 19.81 - \frac{V_{f1}}{\tan\beta 1} \quad (8)$$

$$= 19.81 - \frac{2.625}{\tan 17.5}$$

$$= 11.48 \text{ m/sec}$$

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To find angle at which the water leave the impeller (ɸ), equation (9) will be useful,

$$\phi = \frac{V_{f_2}}{V_{w_2}} \tag{9}$$

$$= \frac{2.625}{11.48}$$

$$=12.88^0$$

Number of vanes (Z) can be calculated using equation (10),

$$z = 6.5 \times \frac{D_2 + D_0}{D_2 - D_0} \tag{10}$$

$$= 6.5 \times \frac{131 + 59}{131 - 59}$$

$$=5$$

Impeller passage width at inlet (b1) can be calculated using inlet correction factor (ε1) as suggested by Austin C. (1972, pp: 109) using equation (11), assuming vane thickness tv as 5.5 mm as per V. Labanoff (2012).

$$\varepsilon 1 = \frac{\pi \times D_0 - \left[\frac{Z \times tv}{\sin \beta m}\right]}{\pi \times D_0} \tag{11}$$

$$= \frac{\pi \times 0.058 - \left[\frac{5 \times 5.5}{\sin 16.59}\right]}{\pi \times 0.058}$$

$$=0.49$$

$$b1 = \frac{Qa \times 1000}{\pi \times D_0 \times vf1 \times \varepsilon}$$

$$= \frac{0.00583 \times 1000}{\pi \times 0.587 \times 2.62 \times 0.49}$$

$$=24.62 \text{ mm}$$

Impeller passage width at outlet (b2) can be calculated using outlet correction factor (ε2) as per equation (12),

$$\varepsilon 2 = \frac{\pi \times D_2 - \left[\frac{Z \times tv}{\sin \beta m}\right]}{\pi \times D_2} \tag{12}$$

$$= \frac{\pi \times 0.131 - \left[\frac{5 \times 5.5}{\sin 16.59}\right]}{\pi \times 0.131}$$

$$=0.77$$

$$b2 = \frac{Qa \times 1000}{\pi \times D_0 \times vf2 \times \varepsilon}$$

$$= \frac{0.00583 \times 1000}{\pi \times 0.131 \times 2.62 \times 0.77}$$

$$=7.04 \text{ mm}$$

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Table 1 describes summary of the calculated parameters for impeller design as per following for given actual discharge (Qa) 0.0058 m^3/sec and head (H) 20 m.

Table: 1 Summary of parameters to be calculated for impeller design

Description	Remark
Impeller type	Semi open
Impeller material	04Cr17Ni12M02
Velocity of water at suction flange vsuc	4.64 m/sec
Eye diameter Do	0.059 m
Impeller diameter D2	0.131 m
Tangential (peripheral) velocity of impeller at inlet U1	8.92 m/sec
Tangential (peripheral) velocity of impeller at outlet U2	19.81 m/sec
Radial component of velocity of flow at inlet Vf1	2.63 m/sec
Radial component of velocity of flow at outlet Vf2	2.63 m/sec
Tangential component of V1 of impeller (velocity of whirl at inlet) Vw1	0
Tangential component of V2 of impeller (velocity of whirl at outlet) Vw2	11.48 m/sec
Vane angle at inlet β1	16.39^0
Vane angle at outlet B2	17.5^0
Angle at which the water leave the impeller ϕ	12.88^0
Number of vane Z	5
Vane thickness tv	5.5 mm
Impeller passage width at inlet b1	24.62 mm
Impeller passage width at inlet b2	7.04 mm

After calculated important parameters for impeller, power required to drive the pump is calculated as suggested by Labanoff V. (2013).

For Specific speed NSS,

$$Nss = \frac{N \times Q^{0.5}}{H^{0.75}} = \frac{2900 \times 0.005^{0.5}}{20^{0.75}} = 22.86 \text{ rpm} \tag{13}$$

From that pump disc friction Pdf,

$$Pdf = \frac{10.89}{NSS^{5/3}} = \frac{10.89}{22.86^{5/3}} = 0.06 \tag{14}$$

Hydraulic efficiency ηH,

$$\eta H = 1 - \frac{0.071}{Q^{0.25}} \tag{15}$$

$$\frac{0.071}{0.005^{0.25}}$$

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= 1-

= 0.74

To calculate overall efficiency (η) according to Karassik I.(1976), volumetric efficiency (ηH) and mechanical loss (Pm) are assumed as 0.85 and 0.15 respectively.

$$\eta = \frac{1}{\left[\frac{1}{\eta H} \times \frac{1}{\eta V}\right] + Pdf + Pm} \quad (16)$$

$$= \frac{1}{\left[\frac{1}{0.74} \times \frac{1}{0.85}\right] + 0.06 + 0.15}$$

= 0.56

Motor power (P) to drive the pump can be calculated using equation (17)

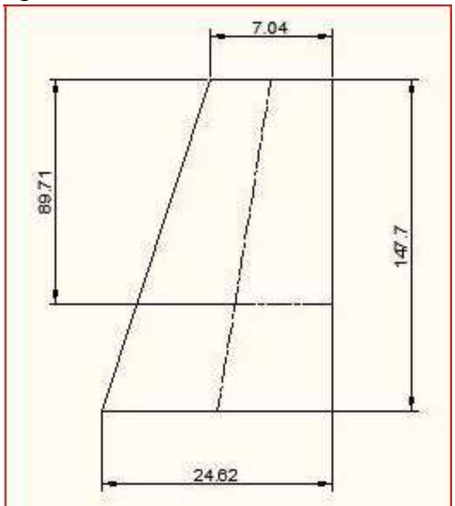
$$P = \frac{Q \times H \times g}{\eta \times 3600} \quad (17)$$

$$= \frac{20 \times 20 \times 9.81}{0.56 \times 3600}$$

= 1.95 kW

After calculating all the important criteria, it is required to check stresses in the impeller. It is noted that bending stress occurs due to centrifugal forces and affected at the root of blade. The approximated shape of blade is considered as a trapezoid. As per figure 3, the vane is considered as a rotating mass.

Figure 3: Stress consideration in the vane



To calculate area of blade (A) using equation (18),

$$A = 0.5 \times [(b1 + b2) \times lb] \quad (18)$$

$$= 0.5 \times [(24.62 + 7.04) \times 147.7]$$

= 2339.57 m^2 Similarly to calculate mass of blade equation (19) is used,

$$m = \rho \times A \times tv \quad (19)$$

= 7860 $\times$ 2339.57 5.5

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= 0.1 Kg

To find centrifugal force (F) equation (19) is used as per following,

$$\bar{Y} = \frac{[a1 \times y1] + [a2 \times y2]}{[a1 + a2]}$$

$$= \frac{\left[147.7 \times 7.04 \times \frac{147.7}{2}\right] + \left[0.5 \times (24.62 - 7.04) \times 147.7 \times \frac{147.7}{3}\right]}{[(147.7 \times 7.04) + (0.5 \times (24.62 - 7.04) \times 147.7)]}$$

$$= 60.21 \text{ mm}$$

$$y = \bar{Y} + (DO/2)$$

$$= 89.71 \text{ mm}$$

$$F = m\omega^2 y \quad (20)$$

$$= 0.1 \times (2 \times 2900/60 \times 3.14)^2 \times (89.71/1000)$$

$$= 826.52 \text{ N}$$

Now the vane is considered as a cantilever to calculate bending stress as per equation (20),

$\tan\theta$ = (b1-b2)/lb

= (24.62-7.04) / 147.7

= 0.12

From the geometry of the vane,

$\tan\theta$ = (b1-h)/

0.12 = (25.49-h)/

h = 17.45 mm

From that, z = lb $xh^2/6$

= 147.7

= 744.99 mm^3

Considering point load,

Bending moment will Mb = F h /2

= 826.52

= 7211.39 N-mm

Hence, σb = Mb /z (21)

= 7211.39 / 744.99

= 9.62 N/mm^2

Now considering shut off H = 30 m

P = 0.3 N/mm^2

Force no blade at shut off F = P/ A

= 0.3

= 701.88 N

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Bending moment will be	Mb	=	$701.88 \times 2339.57 \times h/2$	
		=	$701.88 \times \frac{60.21}{2}$	
		=	21130.1 N-mm	
From that	σb	=	Mb /z	(22)
		=	21130.1 / 701.88	
		=	30.1 N/mm^2	

Maximum permissible stress for vane is 80 N/mm^2. Calculated stress is less than maximum allowable stress. Hence, vane is safe. As impeller is very much important part of the pump, deflection of the

vane is checked as per following calculations. Now, the deflection (δ) is to be within permissible range. The angular speed and equivalent radius are calculated as mentioned in equation (23).

$$\omega = \frac{2 \times \pi \times N}{60}$$

$$= \frac{2 \times \pi \times 2900}{60}$$

$$= 303.53\ rad / sec$$

$$R = \sqrt{R_i \times R_o}$$

$$= \sqrt{0.011 \times 0.065}$$

$$= 0.027 \text{ m}$$

From that, deflection (δ) can be calculated using equation (21)

$$\delta = \left[\frac{\rho \times \omega^2 \times R_O}{E \times 10^6}\right]\left(R_i^{\,2} + R_O^{\,2} - \left(\left(\frac{1+\gamma}{1-\gamma}\right)\left[\frac{R_i^2 \times R_O^2}{R_O^2}\right] - \left(\frac{1+\gamma}{3+\gamma}\right) \times R_O^{\,2}\right)\right) \quad (23)$$

$$= \left[\frac{7850 \times 303.53^2 \times 0.065}{2.1 \times 10^5 \times 10^6}\right]\left(0.011^2 + 0.065^2 - \left(\left(\frac{1+0.3}{1-0.3}\right)\left[\frac{0.011^2 \times 0.065^2}{0.065^2}\right] - \left(\frac{1+0.3}{3+0.3}\right)0.065^2\right)\right)$$

$$= 0.62 \times 10^{-6} \text{ m}$$

As stated earlier, there is a limited freedom to pick outlet angle only. For that, there are two restrictions, outlet angle should be greater than inlet angle and it should be less than 35^0. Though, the vane number should be in prime and in between 04 to 13 as a result. The manual procedure is iterative and to avoid chances of errors, software is developed using Visual Basic (VB) as a GUI. The structure of the programming language is very simple, particularly as to the executable code. Hence, Visual Basic is adopted as a programming language because it is not only a language but primarily an integrated, interactive development environment.

SOFTWARE FOR PUMP DESIGN

The software consists two parts: 1) pump design and 2) selection of the pump. The flowchart is discussed in Figure 4 as per following. Inlet angle, outlet angle and blade height are main parameters to design impeller.

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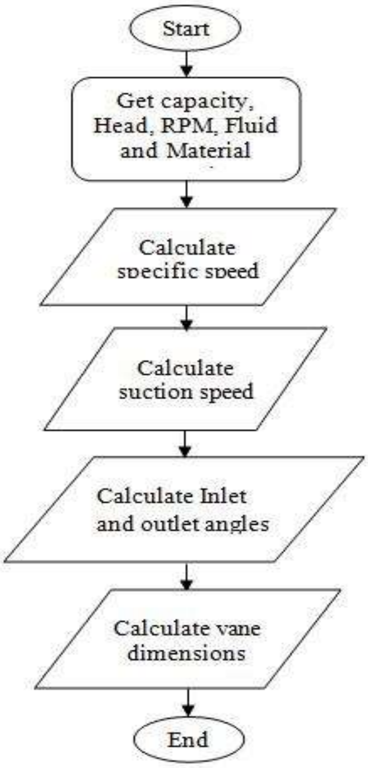
Change in one parameter causes change in entire dimensions of the impeller. So, design of impeller is typical, time consuming and there is always chances of errors, if done manually. Hence, there is a need for software to design centrifugal pump to make all calculation more accurate and faster. Very few parameters are needed to program the impeller design. Visual basic is adopted as a programming language. Dr. Liew (2017) has mentioned that Visual Basic has very good pre-build interface elements as a Graphical User Inter5face (GUI). The language is user friendly and avoids lengthy program writing. Facility of drop down (POP up) commands is also available. Only several commands and instruction lines are required to build program.

Figure 4: Flowchart for impeller design



Following figures 5 (a) and (b) show screen shot of the software for the pump design. By clicking, ‘File’ from menu bar and then clicking ‘New design’, new design can be calculated by using the software.

Figure 5 Screenshots of output of software for pump design

(a)

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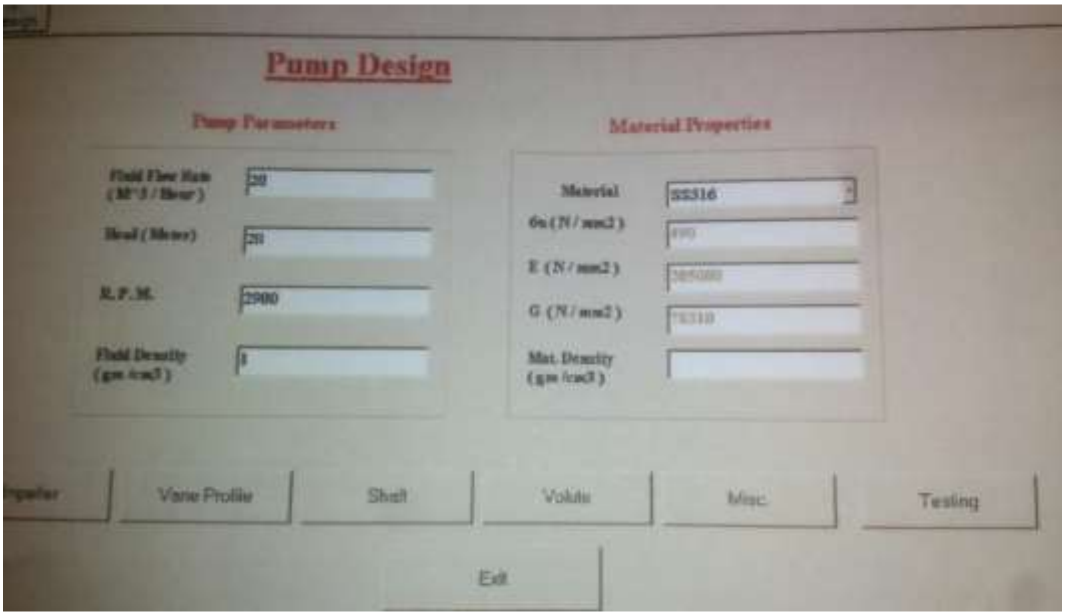
(b)



User can enter the value of discharge (Q), Head (H), and RPM. Material can also be selected and displayed properties of the selected material are not possible to manipulate. The software is user friendly to operator as it provides advice to avoid absurd result by suggesting maximum and minimum head for the specific discharge. The software is easy to use and results generated by the software are same as per calculated result. Hence, the software is error free. For better understanding, AutoCAD drawings are embedded in the software with the pump design.

SOFTWARE FOR PUMP SELECTION

Selection of the pump for given head at specific discharge is time consuming. There is always chance of error to select the pump for the required data from the catalogue charts. It is also difficult for semi skill or unskilled

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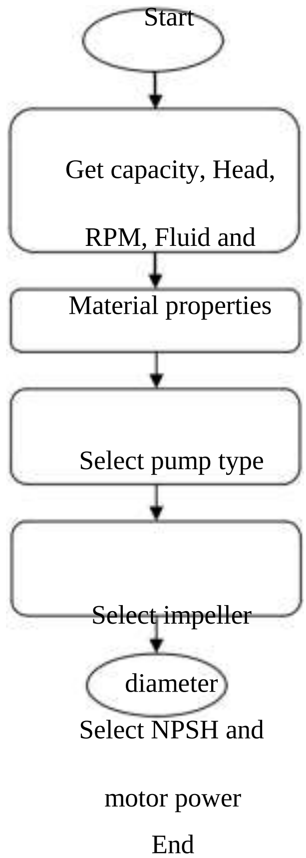
worker. In general, impeller, motor power and NPSH charts are refer to select the model. The same efforts are taken care by the software developed using Visual Basic due to better GUI experience. The flowchart is discussed in Figure 6 as per following. Each database table is having upper and lower limits of head H (H1 and H2) for given discharge Q. The software compares given value of the discharge Q and head H from the table to find specific model.

Figure 6 Flowchart for pump selection



Following figures 7 (a) shows screen shot of the software for the pump selection. When software is operated, it searches appropriate database from the specific table for the given discharge (Q) and head

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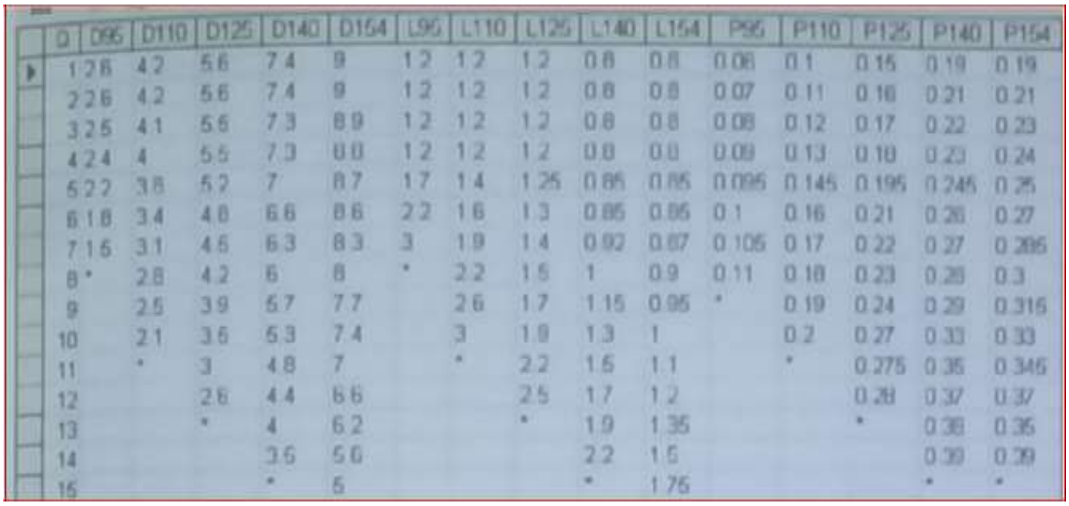
(H). Each database table is having upper and lower limits of head H (H1 and H2) for given discharge Q. The software compares given value of the discharge Q and head H from the table to find specific model. Figure 7 (b) shows one of the created databases for impeller diameter, NPSH and motor RPM. The values of discharge (Q) and head (H) is to be entered. 'Model' button is only activated after all value entered. After clicking 'Model' button, if model is not available as per customer requirements, message will be displayed that "Model is not available as per requirement" otherwise message will be displayed that "Model is available as per requirement". Graphically representation is possible by the chart as shown in figure 7 (e).

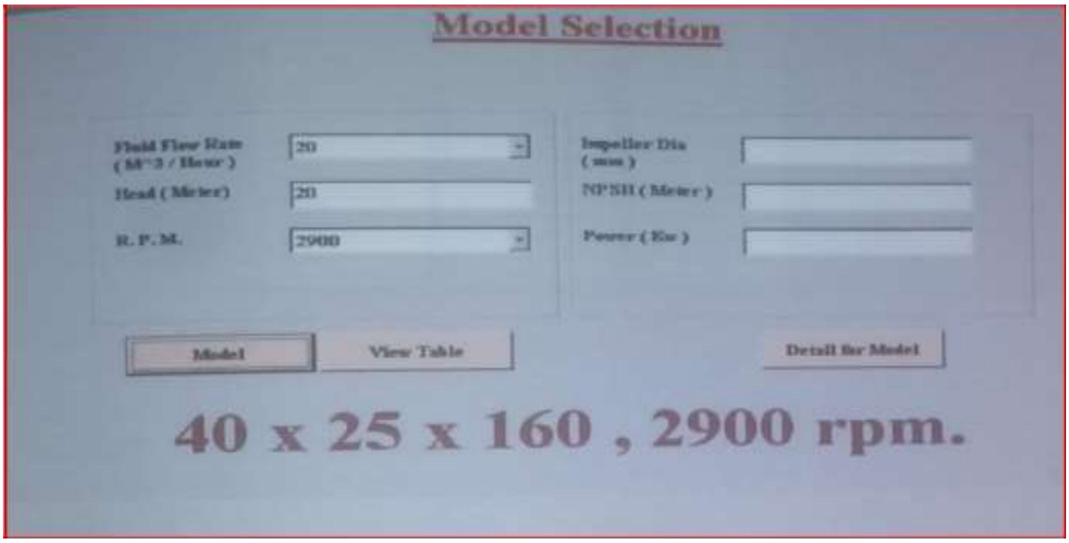
Figure 7 Screenshots of output of software for pump design

(a)

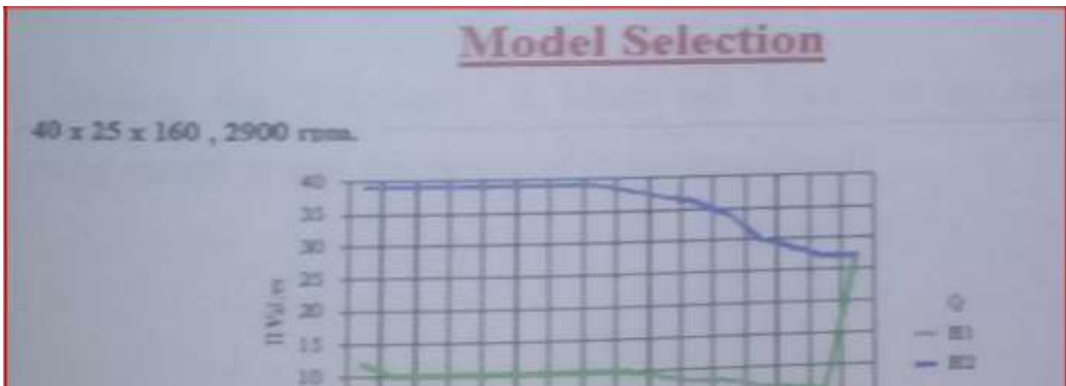


Q	D96	D110	D125	D140	D164	L96	L110	L125	L140	L154	P96	P110	P125	P140	P154
1	2.8	4.2	5.6	7.4	9	1.2	1.2	1.2	0.8	0.8	0.06	0.1	0.15	0.19	0.19
2	2.6	4.2	5.6	7.4	9	1.2	1.2	1.2	0.8	0.8	0.07	0.11	0.16	0.21	0.21
3	2.5	4.1	5.6	7.3	8.9	1.2	1.2	1.2	0.8	0.8	0.08	0.12	0.17	0.22	0.23
4	2.4	4	5.5	7.3	8.8	1.2	1.2	1.2	0.8	0.8	0.09	0.13	0.18	0.23	0.24
5	2.2	3.8	5.2	7	8.7	1.7	1.4	1.25	0.85	0.85	0.095	0.145	0.195	0.245	0.25
6	1.8	3.4	4.8	6.6	8.6	2.2	1.6	1.3	0.85	0.86	0.1	0.16	0.21	0.26	0.27
7	1.6	3.1	4.6	6.3	8.3	3	1.9	1.4	0.92	0.87	0.106	0.17	0.22	0.27	0.286
8	*	2.8	4.2	6	8	*	2.2	1.5	1	0.9	0.11	0.18	0.23	0.28	0.3
9		2.5	3.9	5.7	7.7		2.6	1.7	1.15	0.95	*	0.19	0.24	0.29	0.316
10		2.1	3.6	5.3	7.4		3	1.9	1.3	1		0.2	0.27	0.33	0.33
11		*	3	4.8	7		*	2.2	1.6	1.1		*	0.275	0.35	0.346
12			2.6	4.4	6.6			2.5	1.7	1.2			0.28	0.37	0.37
13			*	4	6.2			*	1.9	1.35			*	0.38	0.35
14				3.6	5.6				2.2	1.5				0.39	0.39
15				*	5				*	1.75				*	*

(b)



(c)



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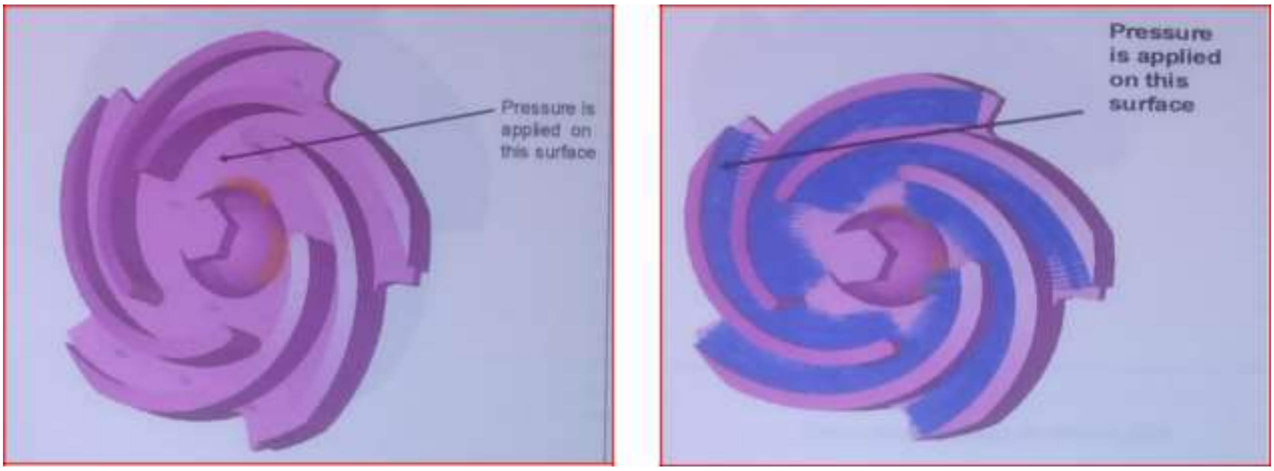
After development of the software, the finite element analysis is done for impeller for the radial load is applied on the surface of vane profile and shut off pressure is 0.3 N/mm^2 (30 m Head) is applied on the pump disk and results are narrated. Figure 8 (a) and (b) show modelling of front and back views of an impeller respectively. As suggested by Gokhale N. (2008), solid tetra nodes having parabolic edge with boundary conditions for given degree of freedom have been employed It is to be noted that total nodes and finite elements generated are 21848 and 11425 respectively after that pressure applied on disk and vane surfaces of impeller.

Figure 8 Screenshots of finite elements analysis for impeller

(a) ***(b)***



After generating finite elements, defining boundary condition and applying pressures, analysis is done for the bending stress and deflections as shown in figure 9(a) and (b).

Figure 9 Screenshots of bending stress and deflections analysis for impeller

(a)

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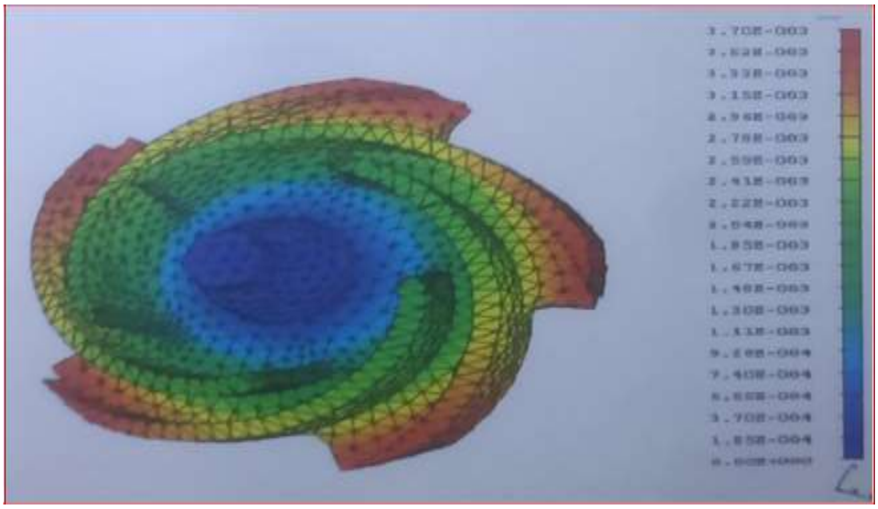
(b)



The results for bending stress and deflection obtain from finite element analysis are matched reasonably with calculations obtained by equations (20).

CONCLUSIONS

Impeller is crucial part of a centrifugal process pump. In present work, it is attempted to develop efficient centrifugal process pump to meet customer's needs. As an impeller is a critical part of a centrifugal process pump, the important parameters of an impeller is designed. The dimensions of an impeller are calculated with a programming environment (i.e. Visual Basic) as a Graphical User Interface (GUI). Softwares are developed to calculate pump design and to select available pump model to make tedious task error free on the basis of actual case study. Program is user-friendly with the help of Microsoft Access database to extract stored data. Stress considering bending moment for the vane is calculated (i.e., 9.62 N/mm^2) and validated by the Finite Element Analysis (FEA) method.

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