COMPUTER AIDED DESIGN OF CENTRIFUGAL PUMP

Aravind K G*, Giridhar B G*, Jay Raj Kumar Singh Sodhi*, Suresh J*

Mahendra K V**, S Ramamurthy***

ABSTRACT:

The design of centrifugal pump involves a large number of empirical data derived from experimental evaluation of pumps. For any small change in input details, it requires a complete rework of design procedure. Even after following this detailed, time consuming procedure, it was very difficult to explain the geometry of the pump elements. This paper discusses the optimisation and computerisation of the design procedure of single stage centrifugal pumps. The design procedure was developed using Graphical User Interface software with Visual Basic 5.0 and interfaced with a specially developed AutoLISP program in AutoCAD R13 environment. An algorithm for generating the Cartesian co-ordinates of the designed pump elements was evolved and incorporated in the program, which are used to show the geometrical shapes of the pump elements and also used as an input data for the AutoLISP program for generating three-dimensional views of the pump components. The options and utilities made available through a single program enables the user to design, view, redesign and optimize the pump geometry till it reaches a satisfactory value. The developed software is user-friendly and can be used as a design tool to configure an optimum pump geometry for a given duty like discharge and head.

* 8th Semester students of Mechanical Engineering, Dayananda Sagar College of Engineering, Bangalore-78.

** Lecturer, Department of Mechanical Engineering, Dayananda Sagar College of Engineering, Bangalore-78.

*** Deputy Director, Scientist-F, Propulsion Division, National Aerospace Laboratories, Bangalore-17.

INTRODUCTION

The hydraulic machine that converts mechanical energy into pressure energy by means of centrifugal force acting on the fluid is called Centrifugal Pump. A centrifugal pump consists of an impeller with vanes rotating in a suitably shaped casing which has an inlet at the centre and usually a spiral 'volute' terminating in an outlet branch of circular cross-section to suit a pipe.

Stephen Lazarkiewicz and Adam Troskolanski. T [1] has included analytical formulae and experimental research values extensively. They differentiate clearly on every design procedure to standardise the pump design procedure. They rely upon past experience during blade selection. Dr. Bruno Eck [2] has suggested an analytical formula for blade selection, which helps in minimizing the frictional losses and optimising the Pump performance. He assumes the credentials of the user to be his equal and is very complex on many aspects. Relies heavily on his past experience on deciding many turbo machine parameters. B. Neumann [3] has extensively dealt with the analysis of designed Pump elements. Every formula has been analytically derived and defined. The formulae require many parameters during calculation. A De Kovats and Desmur .G [4] do not differentiate between various impeller types. The flow of design procedure is not continuous and hence very complex. They assume the final values before designing almost all Pump elements by using past experience.

As none of the books presented an overall standard view of the design procedure, an extensive study was made into all the available literature on this subject other than these standard literatures and a comprehensive and standardised design procedure incorporating a thoroughly analytical format was developed.

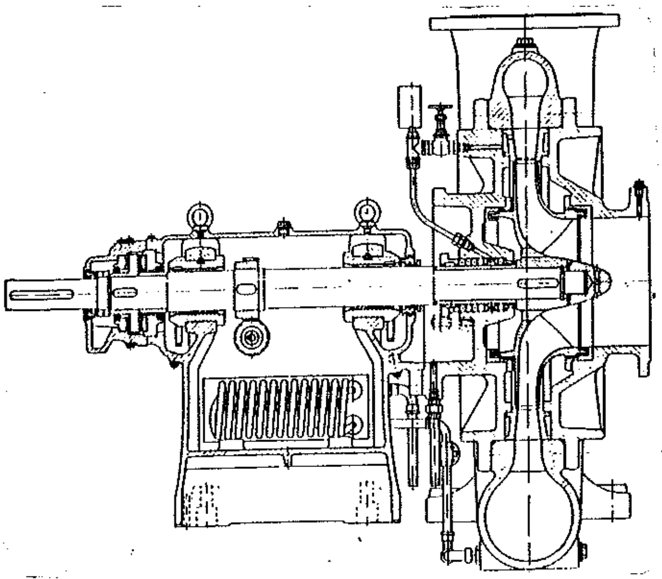


Fig. 1 : General profile of single entry, single stage centrifugal pump with guide vanes.

This paper describes the design, optimisation and computerisation of a single stage, single entry, end suction general purpose Centrifugal Pump with guide vanes and vertically split volute casing.Fig.1 shows the general profile of the same [1].

As the manual design of centrifugal pumps was a laborious, it resulted in loss of precious time in redesign for every small change of input data. As a result, a standard design procedure was evolved incorporating pump duty namely, discharge and head and the same was transferred to a computer program intended for optimising the pump design and geometry.

NOMENCLATURE

b	breadth	C_v	flow velocity	α^*	vane inclination angle	i	impeller
cl	clearance	D_B	bigger diameter	β	relative flow angle	k	key
d	diameter	D_S	small diameter	β^*	vane angle	s	shaft
e	height of passage	H	head	η	efficiency	se	seal
g	acceleration due to gravity	Kcm	velocity co-efficient	ε	velocity co-efficient	sei	seal inside
l	length	Mm	moment of momentum	ρ	radius	0	impeller eye
n_{sf}	dimensionless shape number	Mt	torque transmitted	Σ	angle of deviation	1	impeller inlet
n_{SQ}	kinematic specific speed	Rf	fillet factor	τ_y	shear stress	2	impeller outlet
r	radius	Su	blade projection			3	diffuser inlet
A	area	U	peripheral velocity	**SUBSCRIPTS :**		4	diffuser outlet
C	absolute velocity	W	relative velocity	A	base circle	5	volute
Cm	meridional velocity	W*	relative vane velocity	B,d	diffuser		
Cp	pfleiderer's correction ratio	Z	number of blades	h	hub		
Cu	tangential velocity	α	absolute flow angle	hi	hub inlet side		

PUMP DESIGN

Here the procedures followed for the design of Pump elements like impeller, diffuser and volute casing along with design of shaft, key, inlet suction passage (reducer), seal and also selection of standard commercial steel pipe and bearing selection are discussed. The input data required for design Total head H (m), Discharge required Q (m^3/s), Rated speed n (rpm), Pump efficiency η_p (%).

Dimensionless Shape Number, n_{SF} leads to standardisation of Pump selection. Using the value of n_{SF} a suitable impeller profile can be selected. This dimensionless shape number is given by, $\mathbf{n_{SF} = 3 \cdot n_{SQ}}$, where Kinematic specific speed, $\mathbf{n_{SQ} = n \cdot (Q)^{1/2} / H^{3/2}}$

From the pump duty that is required namely discharge and head, the motor output power required to run the pump and in turn the torque transmitted by the shaft is calculated. By selecting a suitable shaft material and its corresponding allowable shear stress (τ_{ys}) in N/mm^2, the shaft diameter [7], $\mathbf{d = (16 \cdot Mt / \pi \cdot \eta_k \cdot \tau_{ys})^{1/3}}$ where, η_k is the Key way factor is estimated. The Shaft diameter thus obtained is rounded off for the next higher Standard diameter.

For a shaft diameter of d mm, a standard key of breadth b_k and height h_k was selected from the standard design handbook. The key length can be taken roughly as, $\mathbf{l_k = d \text{ to } (4 \cdot d)}$

Different types of bearings like journal, ball, roller, etc., are used in centrifugal pumps. Generally, ball bearings the most common antifriction bearings are used on centrifugal pumps. A suitable ball bearing for the required duty is selected from the standard handbook.

The impeller is the rotating part of a Centrifugal Pump mounted on a rotating shaft. The diameter of impeller hub depends upon the diameter of shaft. Generally, the impeller hub extends into the impeller eye.

The impeller inlet geometry is suitably calculated from the given input conditions. The radial velocity[1] $\mathbf{C_{m1} = Kcm_1 \cdot (2 \cdot g \cdot H)^{1/2}}$ where Kcm_1, a velocity coefficient is a function of n_{SF} whose value is obtained from the fig. 2 for a given value of n_{SF}.

The inlet velocity usually lies within the limits 1.5 to 6 m/s although it can be as high as 12m/s in pumps with high positive suction heads. For end suction pumps, inlet velocity [1], $\mathbf{C_0 = (0.9 \text{ to } 1.0) \cdot C_{m1}}$
The total area $\mathbf{A_1 = A_0 + A_h}$ where A_0 is the free area available for flow at the eye and A_h is the hub

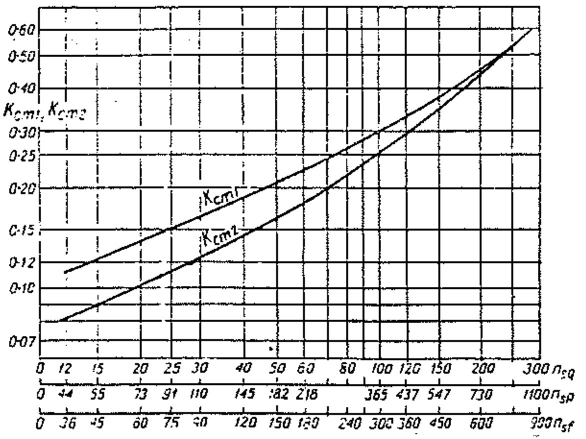


Fig. 2 : Velocity co-efficients Kcm_1 and Kcm_2 in relation to specific speed.

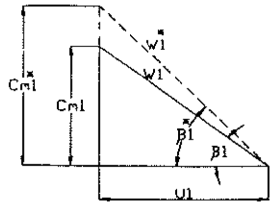


Fig. 3 : Impeller inlet velocity triangle

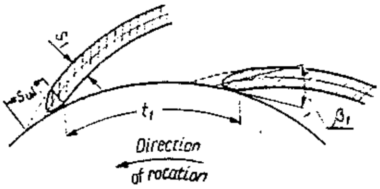


Fig. 4 : Blade shape at impeller inlet

cross sectional area, using which the inlet eye diameter is calculated. The mean blade inlet diameter $\mathbf{d_1 = (4.0 \text{ to } 4.5) \cdot (Q / n)^{1/3} + d_{hi}}$.
After calculating the inlet flow conditions, the Vane angle β_1^* is calculated using the relation [3], $\mathbf{\tan \beta_1^* / \tan \beta_1 = 1.05 \text{ to } 1.25}$. The inlet velocity triangle is as shown in fig. 3. Limiting the angle of camber to 5^0, outlet vane angle[3] $\mathbf{\beta_2^* = \beta_1^* + 5}$.

For optimum number of blades [2] the empirical correlation incorporating the diameter ratio is given by, $\mathbf{Z_i = 4 \cdot \pi \cdot \sin\beta_2^* / 1.5 \cdot (1 - d_0 / d_2)}$

The impeller blade thickness [3] is found using $\mathbf{S_1 / d_2 = 0.018}$. The typical shape of the leading edge of the blade is shown in fig. 4 where S_{u1} is given by[1], $\mathbf{S_{u1} = S_1 / \sin \beta_1^*}$. The impeller inlet breadth in the meridional plane, $\mathbf{b_1 = A_1 / \pi \cdot d_1}$ where A_1 is the impeller inlet area.

At outlet, the radial velocity [1] $\mathbf{Cm_2 = Kcm_2 \cdot (2 \cdot g \cdot H)^{1/2}}$ where Kcm_2 is a velocity co-efficient. The Peripheral velocity [1] $\mathbf{U_2 = (C_{m2} / 2 \cdot \tan \beta_2^*) + [(C_{m2} / 2 \cdot \tan \beta_2^*)^2 + (g \cdot H (1 + C_p) / \eta_h)]^{1/2}}$ where (H / η_h) is the theoretical head (H_{th}), C_p is a function a function based on head co-efficient and diameter ratio as given by Pfleiderer for finite number of blades and η_h is the hydraulic efficiency.
The impeller outside diameter d_2 is calculated on the basis of rotational speed.

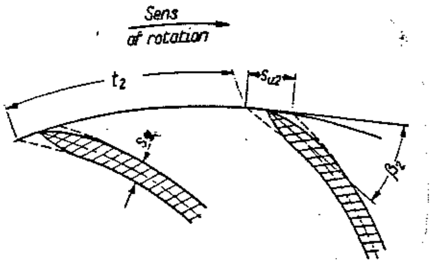


Fig. 5 : Blade shape at impeller outlet

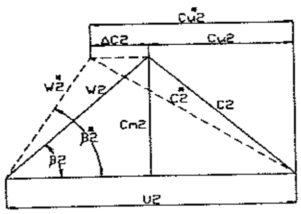


Fig. 6 : Impeller outlet velocity triangle

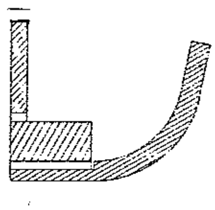


Fig. 7 : Seal

Now with this value of d_2, the diameter ratio d_2 / d_0 is once again calculated which will be usually slightly different from the diameter ratio assumed. Thus with this new value C_p and U_2 are recalculated.
The typical shape of the trailing edge of the blade is shown in fig. 5 where $\mathbf{S_{u2} = S_1 / \sin\beta_2^*}$. The impeller outlet breadth in the meridional plane, $\mathbf{b_2 = A_2 / \pi \cdot d_2}$ where A_2 is the impeller outlet area.

The theoretical zero slip value Cu_2^* is calculated from $\mathbf{\tan\beta_2^* = Cm_2 / (U_2 - Cu_2^*)}$. The absolute vane velocity, $\mathbf{C_2^* = Cm_2 / \sin\alpha_2^*}$.The absolute vane angle $\mathbf{\tan\alpha_2^* = Cm_2 / Cu_2^*}$. The velocity co-efficient [3] $\mathbf{\varepsilon = 1 - \{5 \cdot [0.5 \cdot U_2 - W_2^* \cdot \cos\beta_2^*] / Z_i \cdot Cu_2^*\}}$ is used to calculate the tangential velocity. The absolute flow angle measured from peripheral direction (α_2) is given by, $\mathbf{\tan\alpha_2 = Cm_2 / Cu_2}$ .Thus radial flow angle β_2 is calculated using the formula, $\mathbf{\tan\beta_2 = Cm_2 / (U_2 - Cu_2)}$. After calculating all the outlet flow values, one can construct the velocity triangle at outlet as shown in fig. 6.

The shape of the blade plays a very important role in a pump. When the flow is turned from axial to radial at the inlet, the velocity is no longer uniform from hub to shroud. At the hub, the velocity is lower in magnitude than near the shroud, which can cause different incidence angles during the flow. This can be avoided by configuring the blade with different diameters from hub to shroud. The criterion, which decides the blade inlet diameter at shroud and hub, is the angle of deviation (Σ) which directly depends on the Total Head supplied.

At the shroud, blade inlet diameter, $\mathbf{d_{1S} = d_1 + (\sin\Sigma \cdot b_1)}$. Thickness of the blade at shroud side is made slightly less than that at the mean (uniformly tapered).

The radial (meridional) velocity at inlet will be same as that at the mean (C_{m1}). The absolute velocity is calculated as, $\mathbf{C_{1S} = Cm_1 \cdot [1 + (2 \cdot e^{-b1/Rf})] / 3}$ where Rf is known as fillet factor given by, $\mathbf{Rf = 0.1625 \cdot d_2}$. All the outlet parameters at shroud will be same as the outlet parameters at mean.

At the hub, blade inlet diameter, $\mathbf{d_{1h} = d_1 - (\sin\Sigma \cdot b_1)}$ .Thickness of the blade at hub side is made slightly higher than that at the mean (uniformly tapered).

It is assumed that radial (meridional) velocity at inlet will be same as that at the mean (C_{m1}). The absolute velocity is calculated by using the formula, $\mathbf{C_{1h} = 2 \cdot Cm_1 - C_{1S}}$. All the outlet parameters at shroud will be same as the outlet parameters at mean since there is no radius change.

From Mark's standard handbook [6], a standard commercial steel pipe of inside diameter greater than that of inlet diameter is selected and its wall thickness and outside diameter are noted. The end of the standard steel pipe is connected to the volute casing at the inlet diameter by a convergent passage called 'Reducer'. The convergent angle is within 0 to 8 degrees and is made flat topped to avoid formation of air pockets due to flow separation.

Since the pump is shrouded, seals are required to be placed between the casing and the impeller parts of pumps. For many small / general-purpose centrifugal pumps, mechanical seals as shown in fig. 7 are provided. The outside diameter of impeller eye is given by, $\mathbf{d_{10} = d_0 + 2 \cdot t_{S1}}$ where t_{S1} is the shroud thickness near the eye. The inside diameter of the seal is calculated as, $\mathbf{d_{sei} = d_{10} + clearance}$ where the clearance depends on the shaft deflection and vibration level. The height of the seal (h_{se}) is taken as 3% of the impeller outlet diameter. The length of the seal is , $\mathbf{l_{se} = (0.12 \text{ to } 0.16) \cdot d_{sei}}$

The vaned diffuser ring is a stationary element having a series of symmetrically spaced vanes forming gradually widening passages, in which the velocity of flow is reduced and the kinematic energy at impeller outlet converted into useful static pressure head.

In order to avoid resonance in the pump, the number of blades and vanes should not have a common divisor (except unity). It has been proved beneficial to the efficiency of the pump to use a number of diffuser vanes not much larger than the number of the impeller blades. Thus number of diffuser vanes[1], $\mathbf{Z_d = Z_i + 1}$

A small clearance is provided between impeller and the vaned diffuser to allow the flow to mix out and become uniform. Usually it is 1 to 4 % of the impeller tip diameter.

The inlet velocity triangle of the diffuser can be drawn as shown in the fig. 8 after necessary calculations.

The ratio of the diffuser outlet diameter to impeller outlet diameter is usually around $\mathbf{d_4/d_2 = 1.3}$ **to** $\mathbf{1.5}$ to provide sufficient pressure recovery. The angle of inclination of diffuser vane, α_3^* is calculated using the relation [1,2], $\mathbf{\tan\alpha_3^* = t_3/(t_3 - S_{U3}) \cdot \tan\alpha_3}$ where t_3 is the inlet pitch. Inlet area of diffuser passage, $\mathbf{A_d = Q/C_V}$ where $\mathbf{C_V = Kcv(2 \cdot g \cdot H)^{1/2}}$ and Kcv is an empirical co-efficient.

The distance between the diffuser vanes can be calculated from the formula[1], $\mathbf{e_3 = [(\pi \cdot d_3 \cdot \sin\alpha_3) / Z_d] - S_3}$. For height of passage e_3, the breadth of diffuser passage [1], $\mathbf{b_3 = a_3/e_3}$. Radius of curvature of diffuser blade at throat, $\mathbf{r_B = r_3 + e_3 + S_3}$.

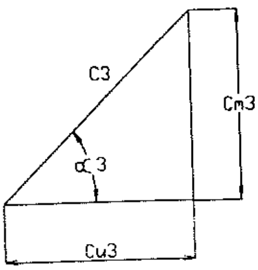


Fig. 8 : Diffuser inlet velocity triangle

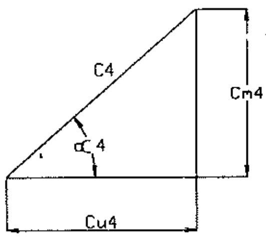


Fig. 9 : Diffuser outlet velocity triangle

The outlet area, $\mathbf{A_4 = h \cdot b_3 \cdot Z_d}$. The relative velocity at diffuser outlet, Cm_4 is equal to the relative velocity at diffuser inlet, Cm_3. Absolute flow angle, α_4 is, $\mathbf{\sin\alpha_4 = Cm_4 / C_4}$. The outlet velocity triangle is as shown in fig. 9. The length of the diffuser [1], $\mathbf{l_2 = (3\ to\ 4)\ e_3}$. Assuming angle of divergence of diffuser (fixing [4] $\mathbf{\delta = 10^0}$) the height of the volute passage at exit, e_4 is calculated from the relation, $\mathbf{\tan(\delta/2) = (e_4 - e_3) / 2 \cdot l_d}$

A correctly designed volute with smooth walls, combined with a conical diffuser, is the perfect element for conducting the liquid from the impeller. When it is used, the highest efficiencies encountered in impeller pump are attained. These good results are due to the simple shape of the volute, which produces small hydraulic losses. The basic function of the volute is collecting the liquid leaving the diffuser and partially converting the kinetic energy into pressure energy. For better performance, symmetrical type, radial discharge volutes are selected.

The radius of the base circle of the volute [1], $\mathbf{r_A = r_4 + cl}$ where cl is the clearance between the diffuser ring and volute beginning. The inlet breadth of the volute b_5 should be greater than the outlet breadth of the diffuser b_3. This is essential in order to account for any inaccuracy. From the ratio [1] $\mathbf{b_5/b_3 = 1.4}$ **to** $\mathbf{1.8}$, b_5 is calculated. The minimum clearance possible without efficiency loss between volute tongue and diffuser outlet can be taken [1] as $\mathbf{cl = 0.04 \cdot d_4}$.

The following different parameter [1] are calculated at each turn angle $\theta = 45^0, 90^0, 135^0, 180^0, 225^0, 270^0, 315^0$, and 360^0.

a. Radius, $\rho = \theta^0/C + (2 \cdot r_A \cdot \theta^0/C)^{1/2}$ where $\mathbf{C = 720 \cdot \pi \cdot Mm/Q}$, $\mathbf{Mm = Cu_4 \cdot r_4}$ = moment of momentum at diffuser outlet.

b. $\mathbf{a = r_A + \rho}$

c. $\mathbf{r_B = (2 \cdot \rho) + r_A}$

The final change of kinetic energy into pressure energy is performed by the conical diffuser. The angle of taper $2\delta t$ depends on the velocity of flow, to avoid breakaway from the walls, which produces considerable energy losses.

The diffuser, which is in the shape of truncated cone is as shown in fig. 12. The smaller diameter, $\mathbf{D_S = 2 \cdot \rho}$ (at 360^0) . Length of diffuser, $\mathbf{l_d = r_B + r_B/4}$ (at 90^0). Bigger diameter (at outlet), D_B is calculated using the relation,

$\mathbf{\tan\delta_t = (D_B - D_S)/(2 \cdot l_d)}$. Area at bigger diameter, $\mathbf{A_D = (\pi/4) \cdot D_B^2}$

GEOMETRIC DESIGN

This deals with the formulation of general equations, which are put in a computer program to find the X, Y, Z co-ordinates of the designed pump elements which are stored in various data files used by "Visual Basic" and "AutoCAD" to generate 2D and 3D views of centrifugal Pump elements. This makes the analysis of Pump easier. Any necessary change needed to optimise the Pump geometry can be done with ease.

The aerodynamic design gives the overall geometry of the Pump such as impeller eye diameter & outlet diameter, blade thickness & angle, diffuser inlet & outlet diameter, volute diameter & thickness etc. These data will be used to generate co-ordinates for impeller and diffuser meridional view as well as blade profiles and the volute cross-section .

Using co-ordinate geometry equations, the impeller data is used to generate the overall co-ordinates of the impeller and its blades. The different corners & fillets encountered in the profile are standardised in accordance to the calculated parameters. A 2-arc blade is selected as shown in fig. 10 on account of its ease of manufacture and optimum performance. All co-ordinates are calculated and stored in respective data-files.

The equations for co-ordinate generation of the first arc are :

$\mathbf{X = C_1 + i \cdot len3 \cdot Cos\,(\gamma_C)}$

$\mathbf{Y = R \cdot Sin\,(\phi_C - Angle)}$

$\mathbf{Z = R_C \cdot Cos\,(\phi_C - Angle)}$

where $\mathbf{R = OC}$

$\mathbf{C_1 = 0.273 \cdot d_2 -}$ **hub thickness** $\mathbf{- b_1 \cdot Cos\,(\sigma_C)}$

$\mathbf{i = 0 \text{ to } n}$

$\mathbf{\angle EOC = Angle = Cos^{-1}\,[((\delta/2)^2 + R^2 - \rho_1^2) / (2 \cdot (\delta/2) \cdot R)]}$

len3, γ_C and σ_C are taken from meridional view.

The equations for co-ordinate generation of the second arc are :

$\mathbf{X = C_1 + i \cdot len3 \cdot Cos\,(\gamma_C)}$

$\mathbf{Y = R \cdot Sin\,(Angle1 - (\alpha_C - \psi_C))}$

$\mathbf{Z = R \cdot Cos\,(Angle1 - (\alpha_C - \psi_C))}$

where $\mathbf{R = OB}$

$\mathbf{\angle GOB = Angle1 = Cos^{-1}\,[((d_{is}/2)^2 + R^2 - \rho_2^2) / (2 \cdot (d_{is}/2) \cdot R)]}$

In the case of the diffuser the vane profile is that of a logarithmic spiral to aid the increase in pressure energy of the liquid leaving the impeller as shown in fig. 11. The side walls of the diffuser are parallel. A suitable set of equations are applied to calculate the polar co-ordinates of the vane from which the X,Y,Z co-ordinates are calculated and stored. The side wall co-ordinates are found using co-ordinate geometry equations and stored.

The equations for co-ordinate generation of the diffuser vane are :

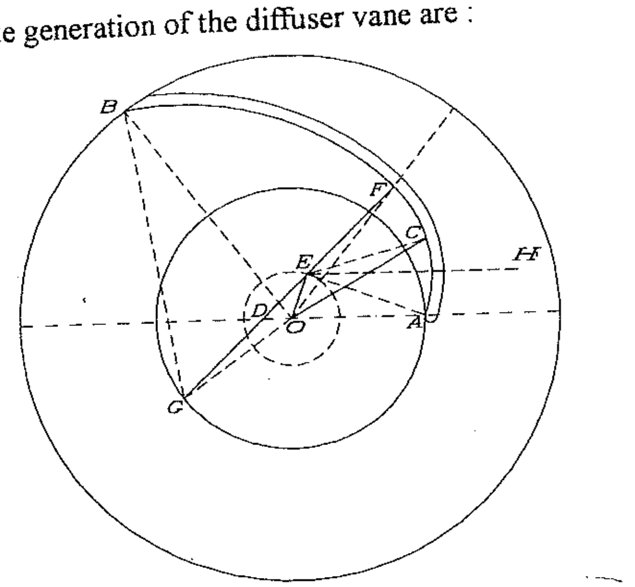
$\mathbf{X = 0.273 \cdot d_2}$

$\mathbf{Z_N = r_N \cdot Cos\ (\theta_N)}$

$\mathbf{Y_N = r_N \cdot Sin\ (\theta_N)}$

Where, $\mathbf{r_N = (r_4 \cdot r_B)^{1/2}}$

$\mathbf{\theta_N = \theta_d/2}$



$OA = D_{IS}/2$
$EA = \rho_1$
$GF = \rho_2$
$OF = R_F$
$OC = R$
$OB = R$
$\angle OAE = \beta_1{}^*$
$OE = \delta/2 = D_{IS} \cdot SIN(\beta_1{}^*)/2$
$\angle AOE = \phi_C$
$\angle AOF = \theta_C$
$\angle EFO = \beta_F$
$\angle FEH = \alpha_C$
$\angle OGD = \psi_C$

Fig. 10 : Impeller blade profile

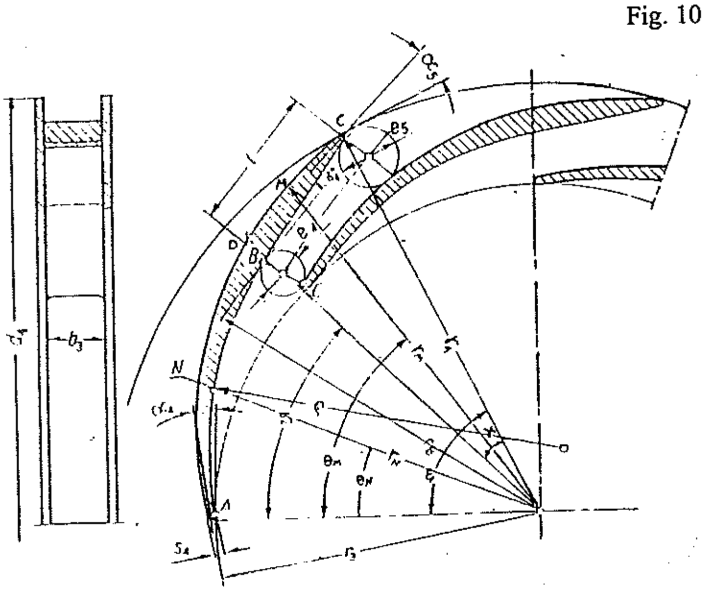


Fig 11 : Diffuser ring

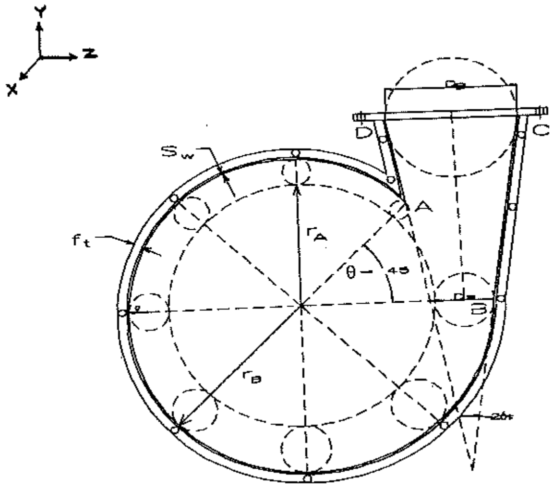


Fig. 12 : Cross-section of volute

In the case of the volute the cross-section follows a logarithmic spiral as shown in fig. 12 and hence constantly varying in its area. Therefore, the data obtained in polar co-ordinates as explained in mechanical design are utilised and suitably converted to the standard Cartesian form using necessary equations and stored separately in a file.

SOFTWARE DEVELOPMENT

The design procedures were incorporated into a software, whose main functions were to provide a user friendly interface, to accept and validate necessary input details, design the pump elements and provide the designed data. The software developed can be run iteratively till a satisfactory design is obtained. The outputs from the program include values of designed parameters, the velocity triangles, data-files, 2D and 3D figures. The software generates data-files that can be used for generating the 3D figures or for machining purposes. The software has many divisions in its organisation. The first is the Graphic User Interface phase developed using Visual Basic 5.0. The second is the 3D-geometry generation using a specially developed AutoLISP program in AutoCAD R13 environment. The third phase involved the complete drafting of the pump assembly using Auto Mechanical Desktop 1.2.

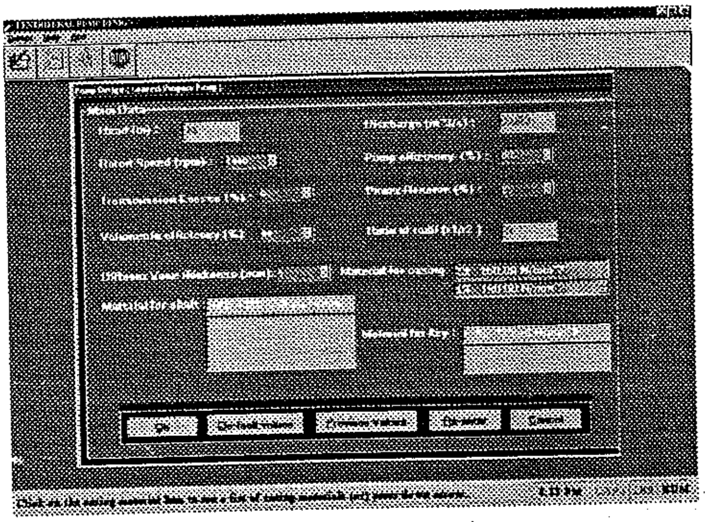


Fig. 13 : Main form and main_data form

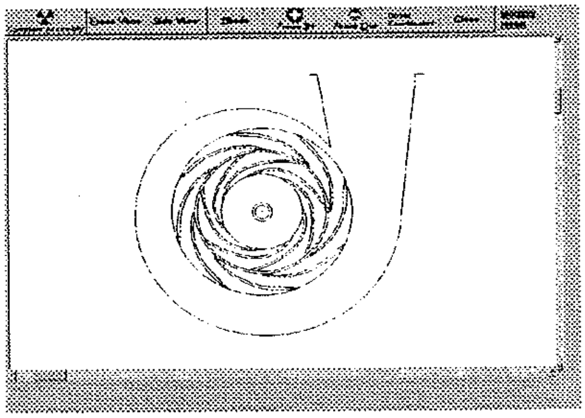


Fig. 14 : Geo_cord Form

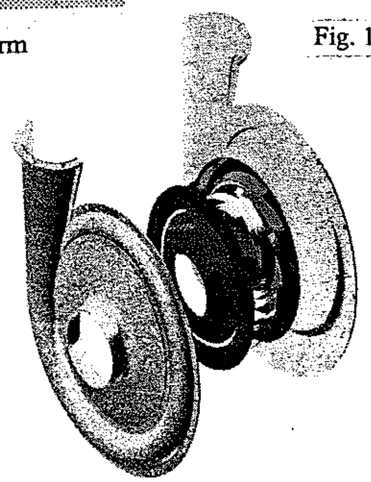


Fig. 15 : Exploded view of centrifugal pump

The Graphic User Interface was developed using Visual Basic 5.0 from Microsoft. Forms / windows have been developed which accepts input like head, discharge, rated speed, blade thickness, power reserve and starting efficiency as shown in fig. 13. the standard motor speeds, casing, shaft, and key materials are automatically loaded and presented. The software then follows a thorough design procedure and calculates all the aerodynamic and geometric parameters. It then presents the user with the details of impeller, diffuser and volute in separate windows. Velocity triangles drawn to a suitable scale are also created along with the data-files for all the elements. The software provides 2D views of the pump elements as front and sectional views as shown in fig. 14. A provision is also made to allow the user to zoom in on to particular sections and view the drawings with as close precision as possible.

The AutoCAD section is mainly responsible for the 3D views of the pump elements. A LISP program has been written which reads the data-files and generates the profiles of the impeller with its blades, diffuser with its vanes. The software directly interfaces with AutoCAD and loads the required program.

The complex volute geometry has been developed using Auto Mechanical Desktop 1.2, a 3D solid, surface and assembly modeling software. This process involved sweeping the circular cross-sectional elements in a circular path. The AutoLISP generated profile was Xreferenced with the volute resulting in a complete pump assembly as shown in fig. 15.

RESULTS AND DISCUSSIONS

A working pump parameters were chosen and the evolved design procedure was applied to the same to carry out the design of the pump in a detailed manner to redefine and to redesign the pump element

This design procedure was cross-checked, verified and then transferred to a computer program to computerise the design procedure. As shown in fig.17 the software program optmises the pump input parameters and displays the designed pump element parameters in a comprehensive manner.

An algorithm was developed for generating the Cartesian co-ordinate of the designed pump elements for viewing the 2D and 3D (interfacing the design program with AutoCAD) figures of the pump elements.

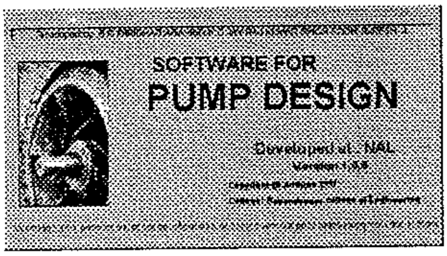


Fig. 16 : Open_screen form

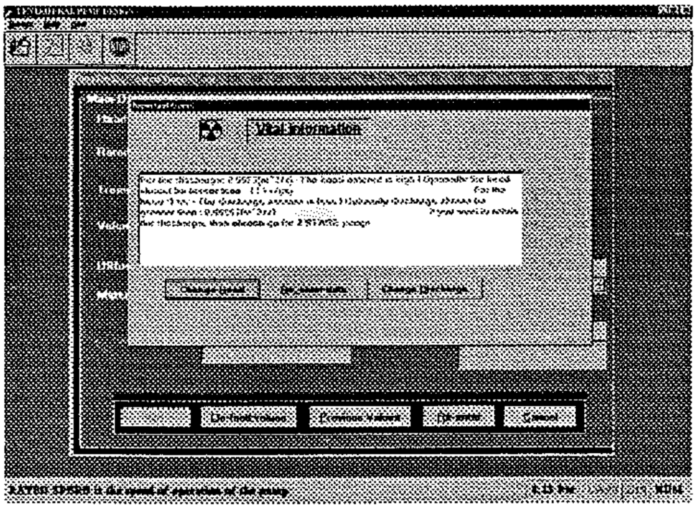


Fig. 17 : Frmquery form

CONCLUSION

As the interface is user friendly, even a layman can use the software with relative ease. The software optimises the output data providing the user greater liberty to modify his design. By providing velocity triangles, the user is given a detail idea regarding the flow pattern, pressure distribution and velocity distribution in the pump elements. The 2D and 3D figures of pump elements provided by the software helps designer to analyse and visualise the pump geometry. The pump accessories like shaft, key, bearings, inlet suction pipe are selected as per market standards, there by helping the manufacturers. This aspects will go a long way in helping a designer with the pump design.

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