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Fully Parameterized Design of Hydrodynamic Torque converter

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ABSTRACT: *The design methodology of hydrodynamic torque converter was investigated, a 31-parameter design model was proposed, and a program code was developed. First, based on the 25 input data, 6 blade angles were determined by optimization method. Second, the center stream surface was designed. Third, the blade was design. Next, semi-automatically modeling technology was applied. After that, by using mesh generation technology and numerical simulation technology, the torque of each converter wheel was obtained. Finally, the speed ratio at the design point taken into account, the peak efficiency of torque converter was obtained. The investigation results show that the 31-parameter design model is feasible and the fully parameterized design of torque converter has been realized successfully. By using the program code, each converter wheel model can be acccomplished within 8 minutes, mash generation can be completed within 12 minutes, and the most vital performance parameter, peak efficiency, can be obtained by flow feild data within several hours. Therefore, the research and development cycle of hydrodynamic torque converter can be shorten greatly.*

KEY WORDS: *Hydrodynamic Torque Converter, Fully Parameterized Design, Semi-automatically Modeling, Flow Field Simulation, Peak Efficiency*

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I. INTRODUCTION

A hydrodynamic torque converter can automatically change the rotational speed from the engine to transmission and also multiply the torque according to the change of external load, which makes the vehicle have good adaptability. In addition, due to the vibration absorbing and damping functions, the lifespan of the vehicle's transmission components can be prolonged. However, the main drawback of a torque converter is that its efficiency is not high enough, which affects the fuel economy of the vehicle. Over the years, many scholars have made great efforts to improve the performance of torque converters. As early as the end of the last century, Masaaki Kubo et al. took two design parameters (the turbine bias angle and the contraction ratio of the pump flow passage) as special examples and described the relationship between the design parameters used to define the geometry and resultant efficiency [1]. Zhu et al. parameterized blade angles and took the peak efficiency as the optimization objection [2]. Kawashima et al. outlined the mapping method and the quantitative sensitivities about each of the parameters [3]. Saravanakumar et al. evaluated the performance of torque converter by changing the angle of stator blades [4]. Chen et al. proposed parametric design method which realizes the quick modification of a torque converter[5]. Wu et al. established the parametric flow passage model of a torque converter, carried out three-dimensional flow field simulation [6]. Venkateswaran et al. took four main parameters (density, viscosity, inlet velocity and temperature) into consideration, created orthogonal arrays for the parameter design[7]. Guan et al. illustrated a method to generate the parameters and three dimensional blade automatically [8]. These results have promoted the progress of parameterized design for torque converters, however, fully parameterized design has not yet been achieved. In this paper, a 31-parameter design model was proposed, and the fully parameterized design of torque converter has been achieved.

II. PARAMETERIZED DESCRIPTION OF TORQUE CONVERTER

Because of the complex geometry, a torque converter needs be presented by using multiple parameters. These parameters can be classified into three different categories. The first category includes original parameters, the second includes design parameters, and the third includes automatically generating parameters.

2.1 Original parameters

The original parameters are those that were specified in design task file, and directly serve as input quantities of the program.

•***Nominal Diameter*:** The nominal diameter D is the most important parameter.

•***Design Speed Ratio*:** Usually used speed ratio at design point is i_d=0.7, or 0.75, o 0.8.

•***Pump Rotational Speed*:** The pump rotational speed n_p equals its engine's rotational speed n_e. It should be pointed out that the engine's rotational speed n_e takes a positive value if the engine rotates counterclockwise; otherwise, the n_e takes a negative value.

•***Pump Torque*:** The pump wheel torque T_P is equal to the engine torque T_e. According to the engine characteristic curved line, the torque corresponding to the above rotational speed is determined.

2.2 Design Parameters

Design parameters refer to parameters that can be chosen, changed and determined by a designer. These design parameters also serve as input quantities of the program.

•***Minimal Diameter*:** The minimal diameter D_0 and the diameter ratio m are related. The diameter ratio of a torque converter is defined by $m=D_0/D$ After the minimum diameter is taken on, the diameter ratio can be calculated.

•***Clearance Between Pump and Turbine*:** In general, the clearance between pump and turbine is taken on δ_{PT}=2-3mm.

•***Fluid Density and Viscosity*:** Fluid density occurs in torque formulas, naturally it should serve as an important parameter. Fluid viscosity does not occur in the design formula, but it plays an important role in the simulation of flow field. Consequently it should be used as an important parameter.

•***Polar Angles*:** It should be pointed out that the entrance and exit radius of each converter wheel directly appears in its torque formula; while the entrance and exit radii are related to the entrance and exit polar angles (θ_{P1},θ_{P2},θ_{T1}, θ_{T2},θ_{S1} and θ_{S2}). The geometric meanings of 6 polar angles are shown in Fig. 1, 2 and 3.

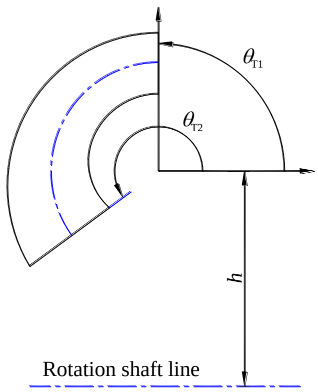

Figure 1: Polar angles of turbine

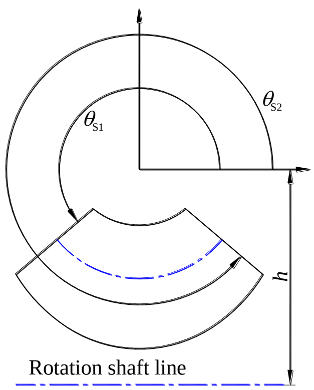

Figure 2: Polar angles of stator

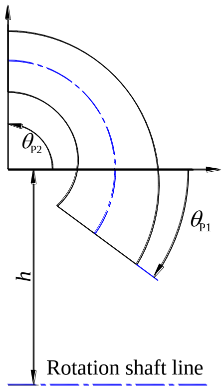

Figure 3: Polar angles of pump

•***Blade Number*:** The blade number for each converter wheel plays an important role in the fluid flow inside the torque converter.

•***Blade Thickness*:** The widely used pump blades and turbine blades of automotive torque converter are usually stamped with thin steel sheets. Each pump blade thickness is a constant t_P, and each turbine blade thickness is a constant t_T as well.

For the stator, the blade thickness refers to the maximal thickness t_m. Because the stator withstands a large variation of the attack angle for the various operating points, and therefore it is important that the losses do not increase too much by a possible separation. Usually, each stator blade is designed to be as airfoil or hydrofoil profile[9], as shown in Fig. 4.

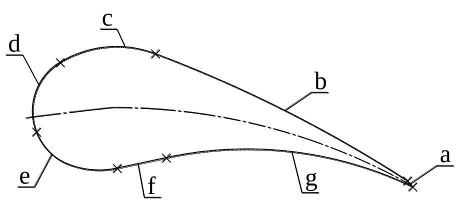

Figure 4: Thickness profile of stator blade

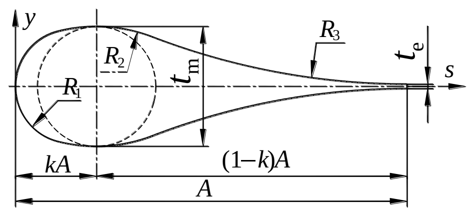

Figure 5: Calculation model of stator blade thickness

In order to calculate blade thickness conveniently, the camber line of the stator blade can be straighten and the cross section contour line of the stator blade can be replace with different radius arcs (Fig. 5). For this blade, parameter t_m, k, R_1 and t_e serve as input quantities of the program.

•***Area Ratio*****:** The area ratio τ is the ratio of through-flow area to the circle area represented by nominal diameter. A large number of statistical data shows that the optimal through-flow area is 23% of the circle area represented by nominal diameter[10].

•***Meridional Velocity***: The meridional velocity plays a vital role and is a very sensitive parameter. Since the volume loss in the circulatory motion is less than 0.2% of the circulatory flow rate, the volume loss can be ignored in the flow field calculation. On the other hand, the through-flow area of each converter wheel is design to be approximately identical. Theoretically, it is reasonable that each converter wheel can be considered to have the same meridional velocity. The meridional velocity v_m can be initially estimated by the pump rotational speed n_p and torque T_p. According to the pump wheel torque formula. The meridional velocity is given by:

$$v_{\mathrm{m}}=\frac{R_{\mathrm{P2}}^2\omega_{\mathrm{P}}-\sqrt{(R_{\mathrm{P2}}^2\omega_{\mathrm{P}})^2-4(R_{\mathrm{S2}}\cot\beta_{\mathrm{S2}}-R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}})T_{\mathrm{P}}/(\rho A_{\mathrm{t}})}}{2(R_{\mathrm{S2}}\cot\beta_{\mathrm{S2}}-R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}})} \tag{1}$$

2.3 Automatically Generating Parameters

The automatically generating parameters include 3 blade entrance angles and 3 blade exit angles. The blade angle of this article refers to the included angle from the circumferential velocity vector to the tangential vector of the blade camber line.

The blade angle at the pump exit has a very great influence on the permeability, stall torque ratio and economy of a torque converter. Usually it is considered that $75° \le \beta_{\mathrm{P2}} \le 135°$.

The blade angle at the stator exit will also affect the performance of the torque converter to a certain extent. In general, it is considered that the range of the stator blade exit angle is $22° \le \beta_{\mathrm{S2}} \le 40°$.

The blade angle at the turbine exit has the influence on the performance of the torque converter not as significantly as the blade angle at the pump exit. It is generally considered that $138° \le \beta_{\mathrm{S2}} \le 154°$.

III. PROGRAM DESIGN

3.1 Determination of Blade Angles

If flow deviation is neglected, each flow angle at the exit is equal to the blade exit angle. The pump wheel torque equation is:

$$T_{\mathrm{P}}=\rho A_{\mathrm{t}}v_{\mathrm{m}}\left[\omega_{\mathrm{P}}R_{\mathrm{P2}}^2+v_{\mathrm{m}}\left(R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}}-R_{\mathrm{S2}}\cot\beta_{\mathrm{S2}}\right)\right] \tag{2}$$

The turbine torque equation is:

$$T_{\mathrm{T}}=\rho A_{\mathrm{t}}v_{\mathrm{m}}\left[(\omega_{\mathrm{T}}R_{\mathrm{T2}}^2-\omega_{\mathrm{P}}R_{\mathrm{P2}}^2)+v_{\mathrm{m}}\left(R_{\mathrm{T2}}\cot\beta_{\mathrm{T2}}-R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}}\right)\right] \tag{3}$$

The expression of torque ratio at the design point is:

$$k_{\mathrm{d}}=\left|T_{\mathrm{T}}/T_{\mathrm{P}}\right|=\left|(\omega_{\mathrm{P}}R_{\mathrm{P2}}^2+v_{\mathrm{m}}R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}}-\omega_{\mathrm{P}}iR_{\mathrm{T2}}^2-v_{\mathrm{m}}R_{\mathrm{T2}}\cot\beta_{\mathrm{T2}})/(\omega_{\mathrm{P}}R_{\mathrm{P2}}^2+v_{\mathrm{m}}R_{\mathrm{P2}}\cot\beta_{\mathrm{P2}}-v_{\mathrm{m}}R_{\mathrm{S2}}\cot\beta_{\mathrm{S2}})\right| \tag{4}$$

Theoretically, the efficiency at the design point is:

$$\eta_{\mathrm{d}}=|k_{\mathrm{d}}i_{\mathrm{d}}|=\left|(\cot\beta_{\mathrm{P2}}-\mathrm{A}\cot\beta_{\mathrm{T2}}+\mathrm{B})/(\cot\beta_{\mathrm{P2}}-\mathrm{C}\cot\beta_{\mathrm{S2}}+\mathrm{D})\right|i_{\mathrm{d}} \tag{5}$$

where $\mathrm{A}=R_{\mathrm{T2}}/R_{\mathrm{P2}}$; $\mathrm{B}=\omega_{\mathrm{P}}(R_{\mathrm{P2}}-iR_{\mathrm{T2}}^2/R_{\mathrm{P2}})/v_{\mathrm{m}}$; $\mathrm{C}=R_{\mathrm{S2}}/R_{\mathrm{P2}}$; $\mathrm{D}=\omega_{\mathrm{P}}R_{\mathrm{P2}}/v_{\mathrm{m}}$.

The maximum value of the above expression can be obtained by using an optimization method, and the optimization model is as follows:

The optimization variables are β_{P2}, β_{T2} and β_{S2}, respectively. The objective function is:

$$\eta_{\mathrm{d}}=\left|(\cot\beta_{\mathrm{P2}}-\mathrm{A}\cot\beta_{\mathrm{T2}}+\mathrm{B})/(\cot\beta_{\mathrm{P2}}-\mathrm{C}\cot\beta_{\mathrm{S2}}+\mathrm{D})\right|i_{\mathrm{d}}\to\max \tag{6}$$

The constraint condition are $75° \le \beta_{\mathrm{P2}} \le 135°$, $138° \le \beta_{\mathrm{S2}} \le 154°$ and $22° \le \beta_{\mathrm{S2}} \le 40°$.

Obviously, it is very difficult to obtain an analytical solution for the above optimization mode. For convenient calculation, the above formulas are programmed. With the help of the program code, three exit blade angles can be obtained.

After blade exit angles have been determined, the velocity circulation at the entrance and exit can be obtained. The velocity circulation at the pump entrance and exit are:

$$\begin{cases}\varGamma_{\mathrm{P1}}=2\pi R_{\mathrm{P1}}(R_{\mathrm{P1}}\omega_{\mathrm{P}}+v_{\mathrm{m}}\cot\beta_{\mathrm{P1}})\\ \varGamma_{\mathrm{P2}}=2\pi R_{\mathrm{P2}}(R_{\mathrm{P2}}\omega_{\mathrm{P}}+v_{\mathrm{m}}\cot\beta_{\mathrm{P2}})\end{cases}$$

The velocity circulation at the turbine entrance and exit are:

$$\begin{cases} \Gamma_{\mathrm{T1}} = 2\pi R_{\mathrm{T1}}(R_{\mathrm{T1}}\omega_{\mathrm{P}}i_{\mathrm{d}} + v_{\mathrm{m}}\cot\beta_{\mathrm{T1}}) \\ \Gamma_{\mathrm{T2}} = 2\pi R_{\mathrm{T2}}(R_{\mathrm{T2}}\omega_{\mathrm{P}}i_{\mathrm{d}} + v_{\mathrm{m}}\cot\beta_{\mathrm{T2}}) \end{cases}$$

The velocity circulation at the stator entrance and exit are:

$$\begin{cases} \Gamma_{\mathrm{S1}} = 2\pi R_{\mathrm{S1}}(v_{\mathrm{m}}\cot\beta_{\mathrm{S1}}) \\ \Gamma_{\mathrm{S2}} = 2\pi R_{\mathrm{S2}}(v_{\mathrm{m}}\cot\beta_{\mathrm{S2}}) \end{cases}$$

Make the assumption that no torque is transferred to the core and shell in the bladeless section, there is no change in the velocity circulation. The circulation conservation law in the bladeless section are expressed as:

$$\begin{cases} \Gamma_{\mathrm{P1}} = \Gamma_{\mathrm{S2}} \\ \Gamma_{\mathrm{T1}} = \Gamma_{\mathrm{P2}} \\ \Gamma_{\mathrm{S1}} = \Gamma_{\mathrm{T2}} \end{cases} \tag{7}$$

After velocity circulation expressions are substituted into the above equations, each blade angle at the entrance can be obtained as following:

$$\begin{cases} \beta_{\mathrm{P1}} = \arctan\{v_{\mathrm{m}} / [(R_{\mathrm{S2}} / R_{\mathrm{P1}})v_{\mathrm{m}} / \tan\beta_{\mathrm{S2}} - R_{\mathrm{P1}}\omega_{\mathrm{P}}]\} \\ \beta_{\mathrm{T1}} = \arctan\{v_{\mathrm{m}} / [R_{\mathrm{P2}}\omega_{\mathrm{P}}(1-i_{\mathrm{d}}) + v_{\mathrm{m}} / \tan\beta_{\mathrm{P2}}]\} \\ \beta_{\mathrm{S1}} = \arctan\{v_{\mathrm{m}} / [(R_{\mathrm{T2}} / R_{\mathrm{S1}})(R_{\mathrm{T2}}\omega_{\mathrm{P}}i_{\mathrm{d}} + v_{\mathrm{m}} / \tan\beta_{\mathrm{T2}})]\} \end{cases} \tag{8}$$

3.2 Design of Central Stream Surface

The design of the central stream surface (or blade camber surface) is the core content of a torque converter design. In order to reduce the power losses, the curvature of a streamline should reach its minimum. A perfect streamline should be a circular arc[11]. For the pump and turbine, the three-dimensional circular arc serves as the three-dimensional streamline which can be regarded as the intersection line of a sphere and a plane[12]. Thus, the arc equation of three-dimensional central streamline can be expressed as:

$$\begin{cases} (x_0 - a)^2 + (y_0 - b)^2 + (z_0 - c)^2 = R^2 \\ Ax_0 + By_0 + Cz_0 = 1 \end{cases} \tag{9}$$

For the stator, because the stator blade angle variation from entrance to exit is very large and the stator flow passage is substantially short, it was unfeasible for arcs to serve as 3D streamlines via mathematical verification. Besides straight line segments and circular arcs, helixes possess constant or approximately constant curvature. Naturally, helixes were chosen as 3D streamlines. The generalized torus helix method was used as the design of the central stream surface[13]. The helix equation can be written as follows:

$$\begin{cases} x_0 = \rho\cos\theta \\ y_0 = r\cos\phi \\ z_0 = r\sin\phi \end{cases} \tag{10}$$

where $r = h[(1+\delta\sin\theta)^2 + 2\xi\sigma\sin\theta]^{1/2}$; $\rho = (r-h)/\sin\theta$; $\phi = A\sinh[k(\theta-\theta_1)]$.

3.3 Blade Design of pump or turbine

Blade design is actually to determine coordinates of blade surface points. Since the equation of the blade camber surface is a planar equation given by $Ax_0+By_0+Cz_0=1$, the equation of the blade surface can be expressed as $Ax+By+Cz=D$. The normal line equation through a point (x_0, y_0, z_0) is given by:

$$(x-x_0)/A = (y-y_0)/B = (z-z_0)/C \tag{11}$$

The distance from the point (x, y, z) of the blade surface to the point (x_0, y_0, z_0) is $t/2$， yeilding:

$$(x-x_0)^2 + (y-y_0)^2 + (z-z_0)^2 = t^2/4 \tag{12}$$

From Eqs. (11), the follows can be obtained:

$$\begin{cases} (y-y_0)^2 = (B/A)^2(x-x_0)^2 \\ (z-z_0)^2 = (C/A)^2(x-x_0)^2 \end{cases} \tag{13}$$

The above two expressions substituted into Eq.(11), the x-coordinate of the point on the blade pressure surface is given by:

$$x = x_0 + [A(A^2+B^2+C^2)^{-1/2}/2]t \tag{14}$$

The other two coordinates of the point on the pressure surface of the blade are:

$$\begin{cases} y = y_0 + (B/A)(x-x_0) \\ z = z_0 + (C/A)(x-x_0) \end{cases} \tag{15}$$

Similarly, the coordinates of the point on the suction surface of the blade are:

$$\begin{cases} x = x_0 - [A(A^2 + B^2 + C^2)^{-1/2} / 2]t \\ y = y_0 - (B / A)(x - x_0) \\ z = z_0 - (C / A)(x - x_0) \end{cases} \tag{16}$$

3.4 Blade Design of Stator

The blade camber surface expressed by Eq.(10) can be considered as a two-variable function with respect to blade span wise parameter ξ and polar angle θ.

The tangential vector of the blade camber surface along the blade span wise is given by:

$$\mathbf{V}_1 = (\partial x / \partial \xi, \partial y / \partial \xi, \partial z / \partial \xi) \tag{17}$$

The tangential vector of the blade camber surface along stream wise can be expressed as:

$$\mathbf{V}_2 = (\partial x / \partial \theta, \partial y / \partial \theta, \partial z / \partial \theta) \tag{18}$$

The normal vector of the stator blade camber surface $\mathbf{V}$ equals the cross product of vector $\mathbf{V}_1$ and $\mathbf{V}_2$:

$$\mathbf{V} = \mathbf{V}_1 \times \mathbf{V}_2 = \begin{vmatrix} i & j & k \\ \partial x / \partial \xi & \partial y / \partial \xi & \partial z / \partial \xi \\ \partial x / \partial \theta & \partial y / \partial \theta & \partial z / \partial \theta \end{vmatrix} = v_x i + v_y j + v_z k \tag{19}$$

If the blade thickness is *t*, the coordinates of a point on the blade surface are:

$$\begin{cases} x = x_0 \pm [v_x (v_x^2 + v_y^2 + v_z^2)^{-1/2} / 2]t \\ y = y_0 \pm [v_y (v_x^2 + v_y^2 + v_z^2)^{-1/2} / 2]t \\ z = z_0 \pm [v_z (v_x^2 + v_y^2 + v_z^2)^{-1/2} / 2]t \end{cases} \tag{20}$$

where positive sign is used to calculate the coordinates of points on the blade pressure surface; while negative is used to calculate the coordinates of points on the blade suction surface.

3.5 Interface Design of the Program

To facilitate data input, a dialog box interface was designed, as shown in the Fig. 6.

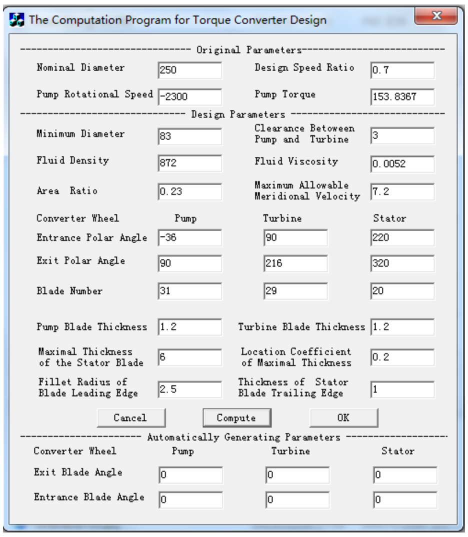

Figure 6: Parameter-imputed interface of torque converter design program

IV. SEMI-AUTOMATIC MODELING TECHNIQUE

Semi-automatic modeling refers to programming, outputting formatted data which are counterparts of models, and then rapidly generating models with the help of 3D software [14].

4.1 Type of Geometry Model

There are two types of geometric model used to calculate the flow field of torque converters [15]. The first type takes the flow passage space between two adjacent blades as the geometric model, and the flow field calculation needs to be carried out independently. The second is to cut the flow passage along the center stream surface, wrap a blade in the model, and calculate the flow field together with the pump, turbine, and stator.

The first model ignores the bladeless zone of the flow field, and the modeling and meshing are relatively simple. And the boundary conditions are relatively simple as well. Most importantly, it is necessary for mesh to be generated automatically. And time consumption can be reduced exponentially in the course of modeling and mesh generation. In addition, the requirements for computer configuration in flow field calculation are relatively low and the meshes can be finer. The second type of geometric model includes the bladeless zone of the flow field, which is relatively complex in modeling and meshing. The boundary conditions are relatively complex as well. The calculation of the flow field requires relatively high computer configuration requirements. Based on these considerations, the first geometric model was adopted. For such a geometric model, it can be regarded as a revolving solid cut with two curved surfaces. In addition, corresponding to each blade with a round leading edge, the flow passage model should possess sharp edges at the entrance.

4.2 Generation of Revolving Solid

The torus of a torque converter consists of complex curved lines. In order to model and fabricate conveniently, either core or shell contour line of each converter wheel is approximated with one arc or two or three arcs. Each approximation error is controlled within 0.3mm. The torus of each converter wheel can be generated automatically by using the program code, and then revolved into a three-dimensional revolving solid.

4.3 Determination of Cutting Surfaces

The two cutting surfaces used to model are actually the pressure surface and suction surface of the flow passage; while the point (x', y', z') on the pressure or suction surfaces of the flow passage are precisely transformed from points (x,y,z) on the pressure or suction surfaces of the blades. If z_P, z_T and z_S are used to denote the number of pump blades, the number of turbine blades and the number of stator blades, respectively. The center angle of the flow passage is given by $\psi = 2\pi / z_P$ or $\psi = 2\pi / z_T$ or $\psi = 2\pi / z_S$, respectively. Transformation formulas are:

$$\begin{cases} x' = x \\ y' = y\cos(\pm\psi/2) - z\sin(\pm\psi/2) \\ z' = y\sin(\pm\psi/2) + z\cos(\pm\psi/2) \end{cases} \tag{21}$$

where positive sign is used to transform the coordinates of points on the blade pressure surface; while negative is used transform the coordinates of points on the blade suction surface.

4.4 Generation of geometric model

First, convert point cloud data into surfaces with the help of 3d software. Second, insert the revolving solid. Next, cut the revolving solid with surfaces. Eventually, the geometric models of 3 flow passages can be obtained(Fig. 7, 8 and 9). With the program code, each converter wheel can be modeled within 8 minutes.

Figure 7: Turbine flow passage **Figure 8: Stator flow passage** **Figure 9: Pump flow passage**

V. MESH GENERATION

The Tetra/Mixed hybrid grid was used for the geometric model. In addition, the pyramid grid was used on the wall boundary CORE, SHELL, PRESSURE and SUCTION, with a maximum pyramid grid size of 0.3mm, a height of 0.15, and a height ratio of 1.3. For this kind of model, mashes of each converter wheel can be generated within 12 minutes. The main grid parameters of converter flow passage are shown in the Table I.

Table I. **Main grid parameters of converter flow passages**

Converter wheel	Blade number	Total elements	Mesh quality metric
Pump	31	1532799	MIN. 0.362398
Turbine	29	1849473	MIN. 0.345114
Stator	20	1956919	MIN. 0.231300

VI. SIMULATION OF FLOW FIELD

In the numerical simulation of a torque converter, generally, following assumptions are made [16]:

(1)Relative to each rotating reference frame, the flow field in the flow passage is steady. Therefore, the flow parameter is time independent.

(2)The flow is periodic from one passage to another (cycle symmetry). Only one flow passage is modeled for each component of torque converter.

(3) No leakage flow exists. The cooling flow is neglected since it is less than 0.2% of the total mass flow rate.

(4)The fluid flow is simulated at a constant temperature, with no heat transfer simulated.

(5)The fluid is impressible with constant physical properties (density and viscosity).

The selection of turbulence model is very important in the internal flow field simulation of torque converter. The $k-\varepsilon$ model has high stability and fast convergence performance, which is more suitable for the calculation of complex three-dimensional flow field inside torque converter.

The accuracy and runtime taken into account, it is recommended that each residual can be set to 0.0003.

The setting of relaxation factor is vitally important, and convergence often depends on this step. It can be set to (0.3, 1, 1, 0.4, 0.4, 0.4, 1).

Run the fluent program. After a certain number of iterations, the residuals of continuity, x-velocity, y-velocity, z-velocity, k, and ε are all less than presetting values, the iterative calculation stops. The torque of each converter wheel can be obtained. Design speed ratio taken into account, the peak efficiency can be obtained.

VII. CONCLUSION

The design methodology of hydrodynamic torque converter was investigated, a 31-parameter design model was proposed. The following conclusions are drawn from this investigation:

(1) The fully parameterized design of torque converter has been achieved successfully. As a result, the imagination from the input of parameters to the output of the resultant performance parameter has become a reality.

(2) This program code is suitable for the design of widely used 3-element centripetal turbine torque converters.

(3) Each converter wheel model can be accomplished within 8 minutes, mesh generation can be completed within 12 minutes, and the most vital performance parameter, peak efficiency, can be obtained within several hours. Therefore, the research and development cycle of torque converter can be shorted greatly.

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