THEORETICAL INVESTIGATIONS OF TEMPERATURE VARIATION IN TORQUE CONVERTERS OPERATING WITH TWO-PHASE FLOW

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Torque converters are usually provided with a cooling circuit which assures an optimum (a convenient) oil temperature. In this article are studied the influence of closing the cooling circuit on the operation of CHC-350 torque converter by partial degree of filling with oil, namely with two-phase flow. Theoretically are deduced the shape of the characteristics curves of the torque converter operating with different oil temperatures. The numerical simulation of the flow in the first stator cascade stage, of the torque converter's torus, by different operating regimes is accomplished.

Keywords: torque converter, two-phase flow, analytical model, numerical simulation.

1. Introduction

In this article is investigated a special torque converter, Lysholm-Smith type, composed from a pump impeller, three stages of turbine runners and between them two stages of stators. The main advantages of this constructive solution are a greater speed reduction and an extended range of high efficiencies. To control the torque converter operation there was developed different methods and means. Between them, two-phase flow, namely air mixed with oil, was studied here. The maximum temperature rise of the machine is one limiting parameter for the safe operation of the machine, if it is equipped with hydrodynamic transmissions.

2. Theoretic characteristics of the torque converters

Theoretical model allowed us to analyze the behavior of a hydrodynamic torque converter in complex regimes: in normal working regime – obtaining the

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characteristic curves, or in transient regimes – generated during starting up or stopping process. Main parameters of the model are: rotational speed of the pump shaft, rotational speed of the turbine shaft and the physic parameters of the fluid.

The main steps in modeling process are:

⇨ The calculus of kinematic elements of machines cascades, i.e. speed triangle elements. In this purpose, we will consider the elements geometry of the hydrodynamic torque converter as a constant and as variable the rotational speed at the primary shaft and the load to the secondary shaft. The specific energy transferred to the fluid by the pump is given by:

$$H_P = \frac{1}{g}\left(u_2 v_{u2} - u_1 v_{u1}\right) = \frac{r_2 \omega}{g}\left(r_2 \omega - \frac{Q}{\rho_2 S_2 \,\mathrm{tg}\left(\beta_2\right)} - \frac{r_1}{r_2}\frac{Q}{\rho_1 S_1 \,\mathrm{tg}\left(\alpha_1\right)}\right). \tag{1}$$

The specific energy transferred in the turbines, are given by:

$$H_{T\,j} = \frac{1}{g}\left(u_{1j} v_{u1j} - u_{2j} v_{u2j}\right) = \frac{r_{1j} \omega}{g}\left(r_{1j} \omega - \frac{Q}{\rho_{1j} S_{1j} \,\mathrm{tg}\left(\beta_{1j}\right)} - \frac{r_{2j}}{r_{1j}}\frac{Q}{\rho_{2j} S_{2j} \,\mathrm{tg}\left(\alpha_{2j}\right)}\right). \tag{2}$$

with „*j*” the number of the turbine stages: $H_T = \sum_j H_{T\,j}$;

⇨ Calculation of hydraulic losses: hydraulic losses through shock at the entrance in blade cascades, losses dues to sudden modification of cross sections, losses dues to interblades channels curvature, friction losses between working fluid and solid walls, friction losses between adjacent fluid layers. The hydraulic losses are estimated using the Carnot – Borda relation:

$$h_p = \zeta \frac{v_0^2}{2g} \tag{3}$$

with the characteristic speed, v_0, given with the volumic flow through the machine, and the loses coefficient, ζ, calculated with the Reynolds number in which the modification of the fluid with temperature and the degree of filling is taking into account;

⇨ Calculation of mechanical losses, dues to solid – to – solid friction.

Modeling the normal working regime, at standard asinchronous rotational speed of 975 rev/min shows a variation of torques at primary and secondary shaft machines, in function of speed ratio $i = n_T / n_P$ as is presented in Fig. 1.

2.1. Modeling different degree of filling

Different degree of filling can be modeled by introducing the definition of a two-phase fluid, composed by the main liquid and air. The volume of the domain occupied by the multiphase flow represents the sum of partial volumes of components phases:

$$V = \sum_{i=1}^{N} V_i \,. \tag{4}$$

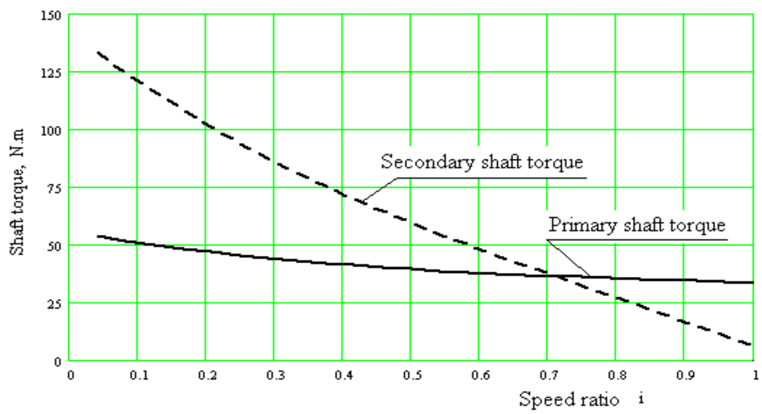


Fig. 1. Variation of shafts torques as function of speed ratio.

Volumetric concentration of component "*i*" will be given by:

$$C_i = \frac{V_i}{V} \,. \tag{5}$$

Taking into account the connection between specific masses:

$$\rho = \rho_{mix} = \rho_1 \cdot C_1 + ... + \rho_i \cdot C_i + ... + \rho_N \cdot C_N = \sum_{i=1}^{N} \left(\rho_i \cdot C_i \right). \tag{6}$$

Here, ρ_{mix} define the mean density of two-phase flow fluid, considered as a homogeneous fluid, and having this specific mass.

Reconsidering the way which has defined the degree of filling, mean density of the working fluid – considered as a two - phase flow one - is:

$$\rho_{mix} = C \cdot \rho_{air} + (1 + C) \cdot \rho_{oil} = \rho_{oil} + C \cdot (\rho_{am} - \rho_{oil}) \,, \tag{(7)}$$

where C is the main liquid (oil) concentration.

Judging in a similar way the two-phase fluid viscosity, the constitutive dynamic viscosity coefficient will be:

$$\eta_{mix} = C \cdot \eta_{air} + (1 - C) \cdot \eta_{oil} \,. \tag{8}$$

Different degrees of filling of hydraulic circuit were considered as 100 %, 95 %, 90 %, 85 %, 80 % and 70 %, as ratio between liquid (mineral oil) and total volume (liquid + gas, respectively oil + air volumes).

In Figures 2 and 3 are presented the shafts torques as function of degree of filling.

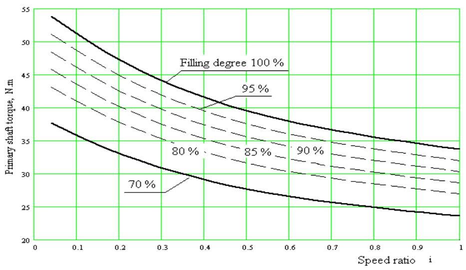


Fig. 2. Variation of primary shaft torque with speeds ratio in function of degree of filling χ_u

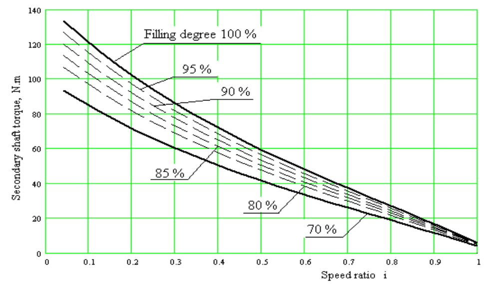


Fig. 3.Variation of secondary shaft torque with speeds ratio as function of degree of filling χ_u

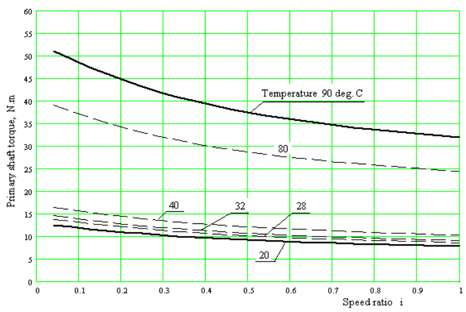


Fig. 4. Variation of primary shaft torque with the speeds ratio as function of the temperature at degree of filling χ_u = 100 %

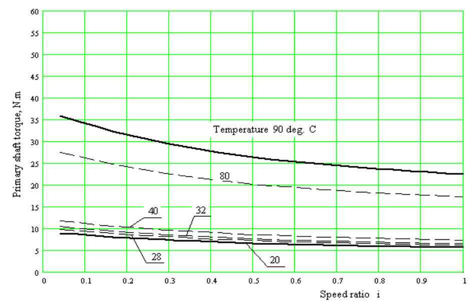


Fig. 5. Variation of primary shaft torque with speeds ratio as function of the temperature at degree of filling χ_u = 70 %

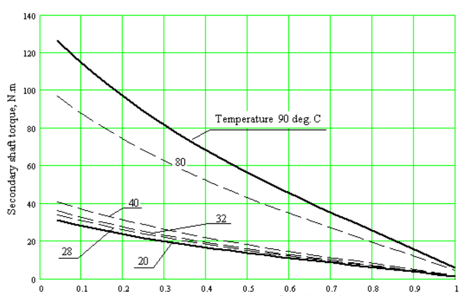


Fig. 6. Variation of secondary shaft torque with speeds ratio as function of the temperature at degree of filling χ_u = 100 %

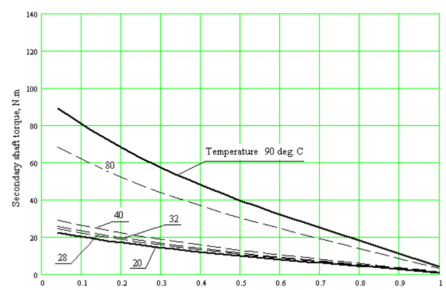


Fig. 7. Variation of secondary shaft torque with speeds ratio as function of the temperature at degree of filling χ_u = 70 %

2.2. Numerical results

Using the presented analytical model, was simulated several working regimes for a torque converter of a special type Lysholm – Smith, which has a primary machine (pump), three secondary machines (turbines), and two reactors (composed of fix blade cascade). Figures 4-7 show some of the obtained results, regarding the behavior of this complex machine in special working regimes, with partial degree of filling and considering, also, a rise of the temperature, conformal to certain experimental data.

3. Numerical simulation of the flow in the torque converter

The torque converter operating with two-phase flow, oil-air, in quite all regimes is homogeneous disperse flow as it was shown in a previous publication [2] and in our optical experiments.

This is especially true in zones with little pressure gradient along the flow so it was chosen for the analyse of the fluid movement the stator of the torque converter.

3.1 Computational domain and the streamlines

The mix of oil with air bubbles for small amount of air inside the torque converter as a homogenous disperse flow is acknowledged trough visualization of the torque converter flow.

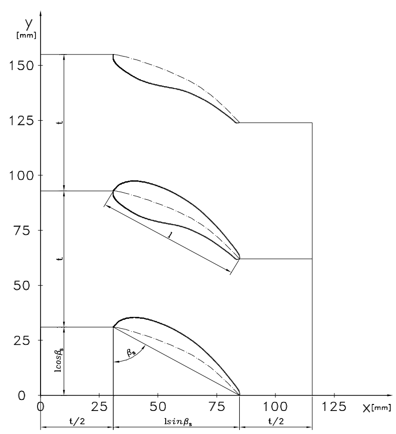


Fig. 8. Domain of analysis with FEM.

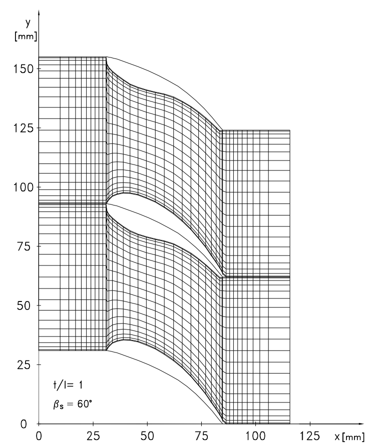


Fig. 9. Computational domain discretisation.

The blades cascade of the stator is located in the exterior axial zone of hydraulic circuit. The flow is axial and by consequence are applied all usual methods for this case. The domain of analysis is realized by tree consecutive profiles disposed in cascade at the stagger angle β_s (Fig. 8). For domain meshing are used linear, isoparametrical and quadrilateral finite elements. The size of finite elements is different (Fig. 9). The smaller elements are distributed around the profile for increasing accuracy of velocity calculation and pressure distribution.

The inviscid, incompressible and irrotational flow model employed by our software is based on the streamfunction formulation [3]. For incompressible flows the divergence of the velocity vanishes everywhere. The difference in value of streamfunction between any two streamlines is equal to the volumetric flow rate passing between the two streamlines per unit width normal to the plane of motion.

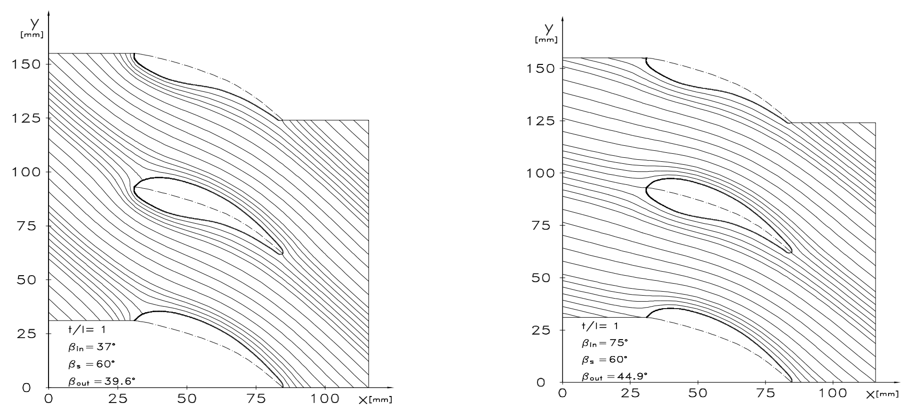


Fig. 10. Streamlines for two variants.

It is useful to recall here that in an irrotational flow of an incompressible fluid in a bounded region, the maximum velocity magnitude and the minimum pressure must occur on the boundary of this region.

In our case, the maximum velocity will always occur on the hydrofoil surface. When analyzing the flow in a cascade, one can take into account the geometrical periodicity. As a result, the computational domain will correspond to a periodic strip as shown in Fig. 8. Integrating the Laplace equation, the spatial surface of the streamfunction is obtained, which, intersected with the plane of movement, offers the curves $\psi =$ constant, representing the streamlines.

We have tested seven regimes for cascade of hydrofoils, corresponding to the same functioning regimes of torque converter. For these regimes angles β_{in} of flow was 37° and 75°. The streamlines resulted and corresponding parameters are presented in Fig. 10.

3.2. Calculation of the field of velocity and pressure

Taking into consideration the velocity relations and that which are specific to the integration on finite element [3], we can calculate the components of the velocity on each finite element, knowing the values of the streamfunction ψ in the four knots from the corners of the finite element (quadrilateral). It is obvious that any point from the domain area belongs to a line of stream, (including the borders of the profiles from the cascade) even if they are not on the calculated and graphic represented streamlines. On the entrance border the velocity V_{in} and the pressure p_{in} are the same in any point, so we can apply the equation of Bernoulli between any point from inside the area of domain and the adequate point on the entrance border.

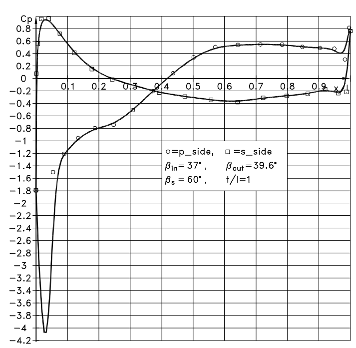
$$p = p_{in} + \frac{\rho}{2}\left(V_{in}^2 - V^2\right). \tag{9}$$

With the specific relation of FEM we obtain the field of velocities and pressures inside the area of domain and around the hydrofoil boundary. To be able to compare the results in any conditions of flow, it prefers their representation under dimensionless shape. The dimensionless values of pressure " $\overline{p}$", and velocity " $\overline{V}$", will be calculated with the relations:

$$\overline{p} = \frac{p - p_{in}}{\frac{\rho}{2}V_{in}^2}, \qquad \overline{V} = \frac{V}{V_{in}}. \tag{10}$$

Taking into consideration (10), between the dimensionless velocity and pressure we have the relation:

$$\overline{p} = 1 - \left(\overline{V}\right)^2. \tag{11}$$



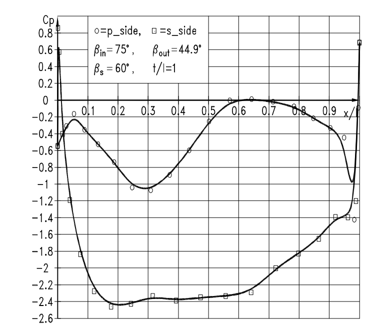


Fig. 11. Pressure coefficient distribution on the foil computed with FEM.

Into the engineering practice, the dimension "$\overline{p}$" is known in addition under the designation of "pressure coefficient" $C_p = \overline{p}$. We are interested to obtain the variation of velocities and pressures around the profile, in a clear way on pressure-side (p_side) and suction-side (s_side), and to have on the abscissa dimensionless values representing the instant abscissa related to the length of the profile's chord.

In Fig 11 is presented the variation of the pressure coefficient C_p around the profile. Important variations can be observed inside the area of the leading and trailing edge, which are explained by the behavior of the profile as an obstacle in the way of the stream.

From this analysis we can see the sensibilities zones to cavitation on the boundary of foil. It is necessary to avoid functioning of torque converter at the regimes heaving β_{in} extremes (37°, 49°, 100°, 120°,133°).

4. Conclusions

On the base of the theoretical results obtained, it can be told that the mathematical model allowed us to analyze the behavior of a complex machine as a torque converter in dynamic regimes generated in exploitation through:

- modification of the degree of filling – modeled as a modification of the volumetric concentration of working fluid;
- changing of physical proprieties of working fluid as consequence of rise of the temperature in the hydraulic circuit.

Numerical simulation of the torque converter stator in the hypothesis of homogeneous dispersed two-phase flow permitted us to determine the desired regimes of flow from energetic point of view.

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