NOTES ON THE OPTIM DESIGN OF TURBOTRANSMITIONS, REGARDING A HYDRODYNAMIC TORQUE CONVERTER OF LYSHOLM – SMITH TYPE

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Present paper analyses the conditions to provide an optimum design of a hydrodynamic torque converters. To realize this, a pseudo - dynamic model is used. The model is based on the analyze of the losses which accors in the machine. As an example of the method, is take into account a Lysholm - Smith type torque converter.

Keywords: optimization, torque converter, dynamics, model.

1. Introduction

The aim of the paper will follow:
– the determination of designing point according to certain optima criteria,
– the analyse of hydraulic losses in torque converters,
– the influence of fluid proprieties on the hydraulic performances.

The last two problems are integrated in a complex numeric model which allowed predicting the (static and dynamic) behaviour of the torque converters in complex regimes.

2. Determination of optimum design point

The determination of optimum design point suppose two steps:

– determination of the correlation head – flow for the primary machine (the pump),

– determination of the rotational speed of the secondary machine (the turbine).

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The correlation head – flow (H – Q) is realized starting from the expression of the nominal power

$$\text{Pabs} = \frac{\rho \cdot g \cdot Q \cdot H}{\eta_P} \quad , \qquad (1)$$

where the global efficiency of the pump (η_P) is supposed to be a constant [1], [2]. Having the power at the primary shaft, Pabs, the correlation H – Q results as a parabolic shape. The optimum point will be obtained plotting a tangent to this curve, having the starting point of coordinates Q = 0, $\left(H_{tP}\right)_{Qt=0} = \frac{u_{2P}^2}{g}$, according to [1].

The rotational speed of the secondary machine is related to the global transfer of energy in the torque converter and its geometric main parameters. The most important geometric parameters are the inlet (D_{1T}) and the outlet (D_{2T}) diameters of the secondary machine (the turbine).

In the case of a torque converter of Ist or IInd class, (having one secondary stage – one turbine), the characteristic diameters are approximatively equals to the characteristic diameters of the primary machine: $D_{2P} \approx D_{1T}$ and $D_{2T} \approx D_{1P}$. As consequence, the rotational speed of the turbine could be close to the rotational speed of the pump, i.e. the speed ratio is close to 1.

In the case of a complex machine, as it is the case of the Lysholm – Smith torque converter which has a pump and three turbine stages, the situation is totally different. The inlet in the first stage of the turbine has the diameter very close to the outlet diameter of the pump ($D_{1T} \approx D_{2P}$ + (1 … 3) mm) ([1], [2]). And the turbine stages are separated by stationary vanes, formatting the reactor.

In that case, the rotational speed of the turbine can be related to the outlet diameter of the first turbine stage. This relation can be deduced from the equality between the specific energies $H_{t\,(pump)} = k_T \cdot H_{T\,(turbine)}$:

$$\frac{1}{g} \cdot \frac{1}{1+p} \cdot \left(u_{2P} \cdot v_{u2P} - u_{1P} \cdot v_{u1P}\right) = k_T \cdot \frac{1}{g} \cdot \left(u_{1T} \cdot v_{u1T} - u_{2T} \cdot v_{u2T}\right) \quad , \qquad (2)$$

where k_T – îs coeficient wich consider the contribution of each turbine stage to the total specific energy of the secondary machine (in our case, k_T = 1/3). From this, a relation is obtained, wich offers the correlation between the rotational speed of the pump and of the turbine, the flow and the geometric parameters of the pump and the turbine:

$$A_P \cdot \omega_P^2 - B_P \cdot \omega_P = A_T \cdot \omega_T^2 - B_T \cdot \omega_T \quad . \tag{3}$$

Finaly, this equation offer a relation between the speed ratio ($i = \omega_T / \omega_P$) as a function of flow and geomety of the machine.

3. Hydraulic losses in a torque converter

The model we discuss here, allowed to evaluate the behaviour and performances of torque converters. The proposed model is a quasistatic ones, but makes possible the analyse of dynamic behaviour of such complex machine as a Lysholm – Smith turboconverter.

As initial data we suppose to have the geometry of the turboconverter; the parameters we consider are the rotational speeds of the shafts and the physical parameters of the working liquid.

The steps in building the model are:

– calculation of the kinemaic values of the speed triangles, and the specific energy for each machine,

– calculation of the hydraulic losses, using the Borda – Carnot formula:

$$h_p = \zeta \cdot \frac{v^2}{2 \cdot g} \quad , \tag{4}$$

where ζ = the losses coefficient and v = the characteristic speed. The types of loses taked into account were:

* shoc losses at the entrance of the pump, turbines and reactors cascade,
* losses dues to sudden variation of cros section at the entrance of cascades,
* friction losses in the chanels between the blades,
* losses dues to the changes of the direction of the flow, caused by the blades curvature,
* losses caused by sudden variation of the cross section at the outlet of the cascades,
* losses in the bends of hydraulic circuit;

with this losses, the balance equations are:

$$Q_P = Q_T = Q \quad , \tag{5}$$

$$H_{tP} - \sum H_{Tj} - \sum h_{pi} = 0 \quad , \tag{6}$$

where: * Q_P – flow in the pump,
* Q_T – flow in the turbines,
* $H_{t\,P}$ – theoretic specific energy of the pump,
* $\Sigma\ H_{T\,j}$ – specific energies of the turbines,
* $\Sigma\ h_{p\,i}$ – hydraulic losses;

The considerations presented above are integrated into a n analitical model of the torque converter, which permit to evaluate the characteristic curves of this machine.

4. Numerical results

The theory presented were illustrated on a 35 kW Lysholm – Smith torque converter having the pump rotational speed fo 975 rev/min. The hydraulic circuit is presented in figure 1.

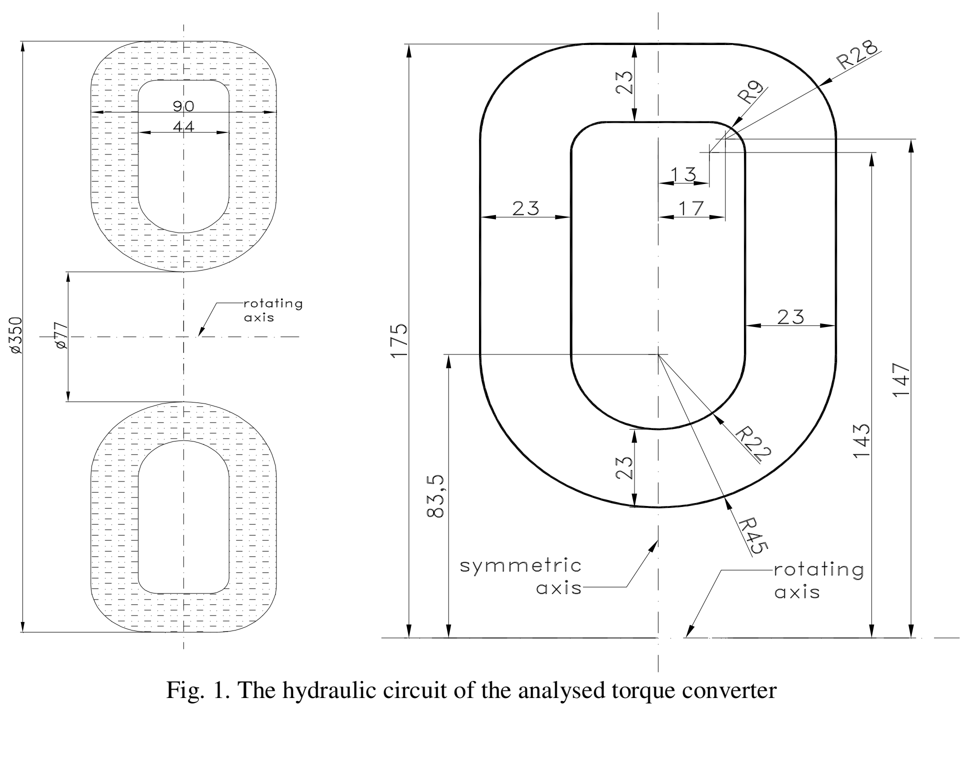

Fig. 1. The hydraulic circuit of the analysed torque converter

The variation of the rotational speed of the secondary machine, according to equation (3), is presented in figure 2.

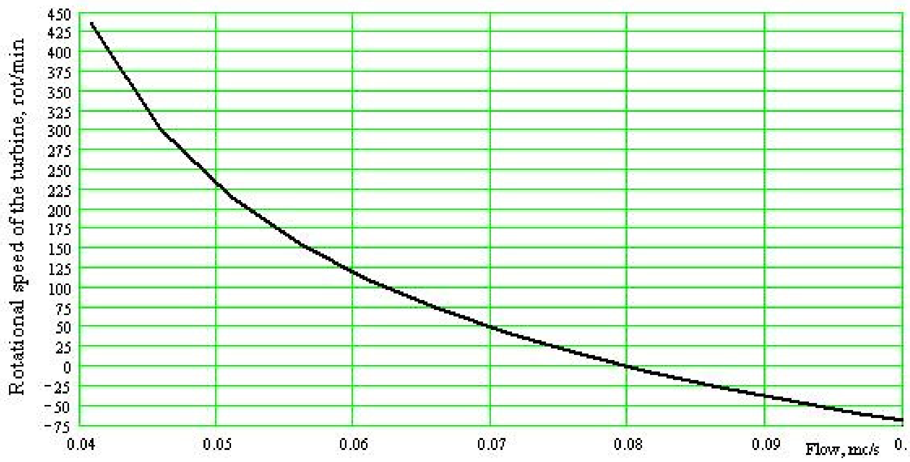

Fig. 2. Turbine rotational speed as function of the flow

In figure 3 is presented the variation of the specific energy of the pump as function of the flow.

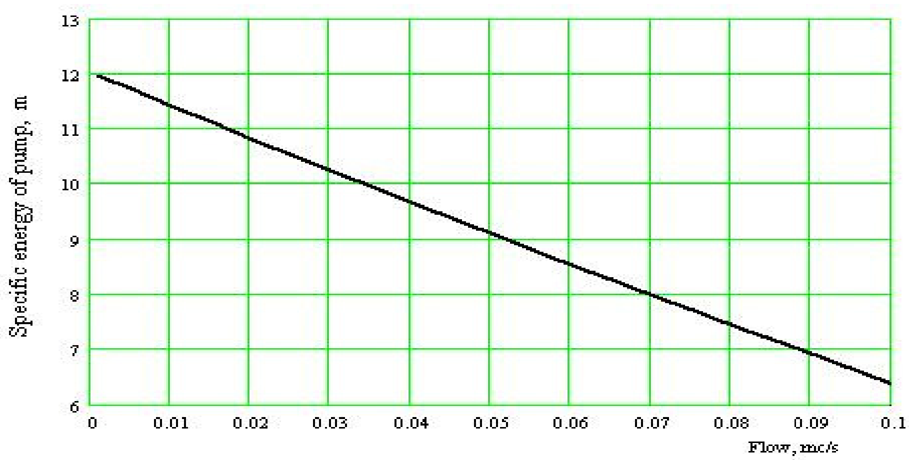

Fig. 3. The variation of the specific energy of the pump, as function of flow

In figures 4 and 5 are presented the variations of the hydraulic losses in the turbine stages and the total losses in the machine.

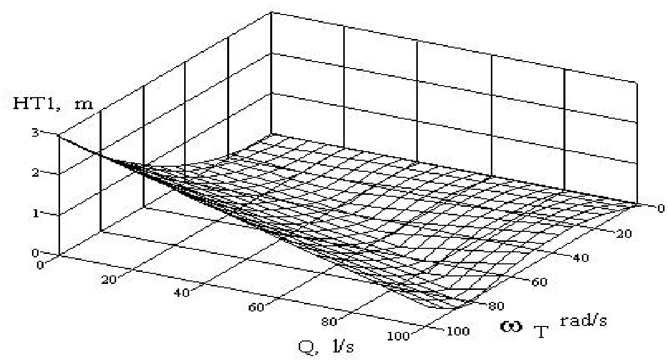

Fig. 4. The variation hydraulic losses in the first turbine stage

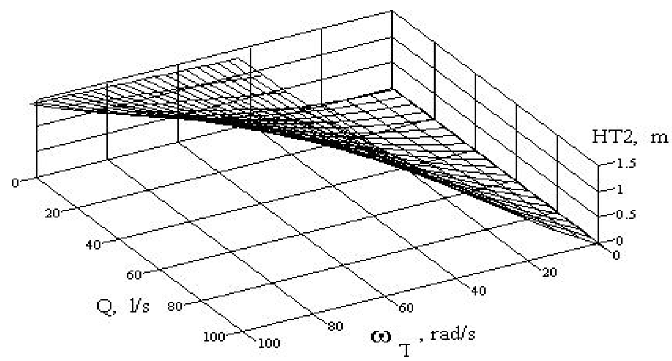

Fig. 4. The variation hydraulic losses in second turbine stage

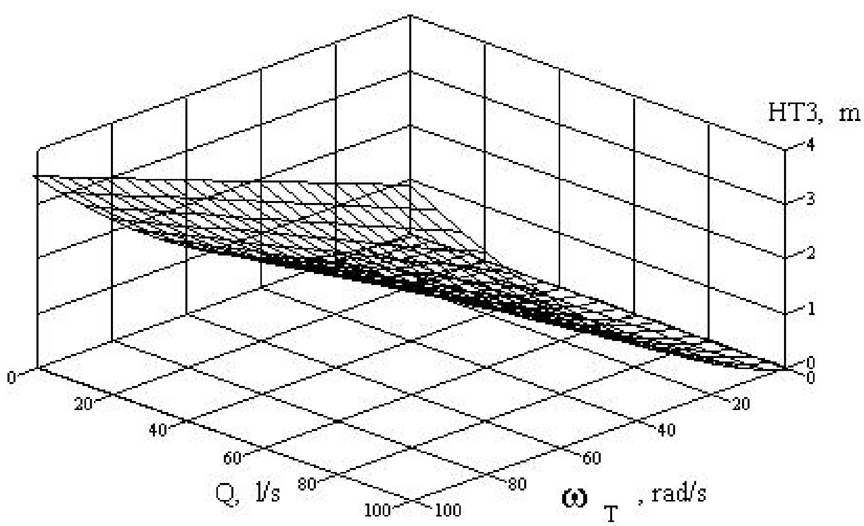

Fig. 5. The variation hydraulic losses in the third turbine stage

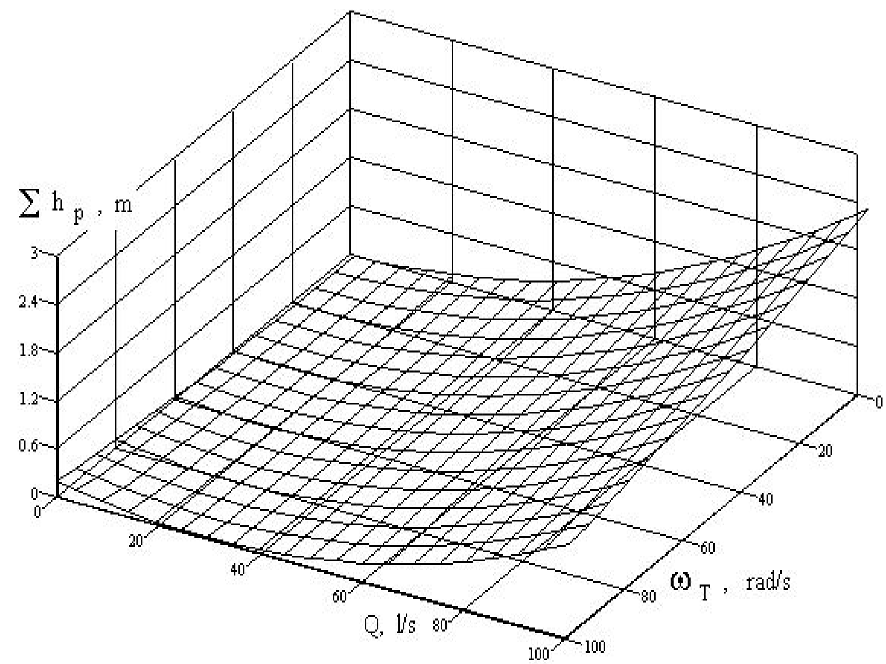

Fig. 6. The variation hydraulic losses in torque converter

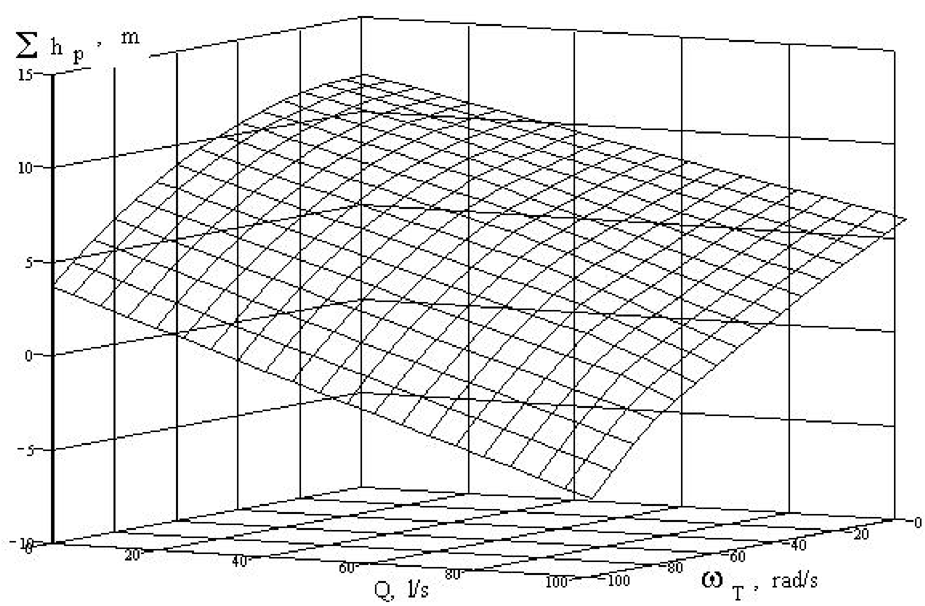

Fig. 7. The variation the energies, according to relation (6)

6. Conclusions

The obtained results allowed to estimate the behaviour of a torque converter in the design process and to evaluate the influence of several parameters on the characteristic curves, i.e. to anticipate the behaviour of this machine.

The presented model can be used in several conditions.

As an example, the characteristics of the fluid propreties, the behaviour of the machine functioning with two-phases fluid.

In the same time, imposing a time behaviour of certains parameters, the behaviour of the machine in time can be anticipated.

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