THEORETICAL AND EXPERIMENTAL STUDIES ON TORQUE CONVERTERS

by

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Lysholm-Smith torques converters has a special importance, due to the fact that this kind of machine can realize a continuously variation of the torque and of the rotational speed to the outlet shaft as function of the resistant torque and, in the same time, an amortization of the shocks and vibrations. This type of torque converter has three turbine stages, intercalated by two stages of reactors (having stationary blades).

The present paper presents theoretical and experimental results obtained on a Lysholm-Smith torque converter CHC-380 in the laboratory of Hydraulic Machinery Department from "POLITEHNICA" University of Timişoara, ROMÂNIA. Theoretical, was studied the behavior of the torque converter, expressed by the characteristic curves. Experimentally, was studied the variation of the temperature inside the torque converter with and without the cooling system and, also, the influence of the filling degree on the characteristic curves. The paper discusses, in the same time the definition and the variation of the degree of transparency for this particular torque converter.

Key words*: torque converter, Lysholm-Smith, temperature, efficiency, degree of filling, degree of transparency*

Introduction

Hydrodynamic transmission found a large area of applications in automotive industry, heavy civil engineering machines, in mining, in ships and railway engines, *etc*. The hydrodynamic torque converter is a transmission which consists from a combination between a hydrodynamic pump and a turbine in the same carcass (envelope) [1-4]. The Lysholm-Smith CHC-380 torque converter has four rotors and three stators, placed between the turbine rotors. The main hydraulic circuit is plotted in fig. 1, with: the pump (P) – the first turbine stage (T1)

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– the first reactor (Re1) – the second turbine stage (T2) – the second reactor (Re2) – the third turbine stage (T3).

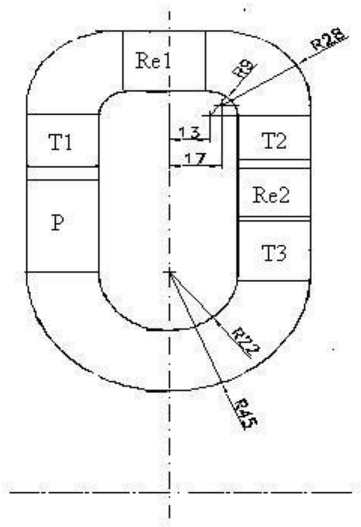

Figure 1. Hydraulic circuit of the torque converter

In the case of Lysholm-Smith torque convertor, for constant values of the rotational speed at the entrance, the entrance torque increase continuously in the same time with the resistance torque, *i. e.* in the same time that decrease the ratio n_T/n_P [2, 5]. This characteristic of the torque convertor is called "transparency". The variation of the resistance outlet torque produces proportional variations of the inlet torque.

Theoretical model

The paper presents a quasistatic mathematical model to estimate the performances of torque converters (of Lysholm-Smith type) [1, 2].

Theoretical specific energies

For the pump, by definition, the theoretic head is given by:

$$H_{\mathrm{tP}} = \frac{1}{g\cdot(1+ps_{\mathrm{P}})}\cdot\left(u_{2\mathrm{P}}\cdot v_{\mathrm{u2P}} - u_{1\mathrm{P}}\cdot v_{\mathrm{u1P}}\right) = \\ = \frac{1}{g\cdot(1+ps_{\mathrm{P}})}\cdot r_{2\mathrm{P}}\cdot\omega_{\mathrm{P}}\cdot v_{\mathrm{u2P}}\cdot\left(1-\frac{r_{1\mathrm{P}}\cdot v_{\mathrm{u1P}}}{r_{2\mathrm{P}}\cdot v_{\mathrm{u2P}}}\right) \tag{1}$$

For the first stage of the turbine the characteristic energy is considered from the characteristic equation, taking into account that the momentum at the inlet is given by the law:

$$r_{1\mathrm{T1}}\cdot v_{\mathrm{u1T1}} = r_{2\mathrm{P}}\cdot v_{\mathrm{u3P}} = r_{2\mathrm{P}}\cdot\frac{v_{\mathrm{u2P}}}{1+ps} \tag{2}$$

and the total head of the turbine is divided by the rule:

$$H_{\mathrm{tT(j)}} = k_{(\mathrm{j})}\cdot H_{\mathrm{tT}}, \quad \sum_{\mathrm{j}=1}^{3} k_{\mathrm{j}} = 1 \tag{3}$$

And so, will be obtained, for the first turbine,

$$k_1 \cdot H_{\mathrm{t\,T}} = \frac{1}{g} \cdot \frac{1}{\eta_{\mathrm{hT1}}} \cdot r_{1\mathrm{T1}} \cdot \omega_{\mathrm{T}} \cdot v_{\mathrm{u1T1}} \cdot \left(1 - \frac{r_{2\mathrm{T1}}}{r_{1\mathrm{T1}}} \cdot \frac{v_{\mathrm{u2T1}}}{v_{\mathrm{u1T1}}}\right) \tag{4}$$

For the third stage of the turbine, the kinematic and angular elements are considered as function of the pump inlet:

$$r_{2\mathrm{T3}} \cdot v_{\mathrm{u2T3}} = \frac{r_{1\mathrm{P}} \cdot v_{\mathrm{u1P}}}{1 + \dfrac{2 \cdot \pi \cdot c_{\mathrm{f}}}{Q} \cdot r_{1\mathrm{P}} \cdot v_{\mathrm{u1P}} \cdot \Delta x_{\mathrm{P\text{-}T3}}} \tag{5}$$

and

$$k_3 \cdot H_{\mathrm{t\,T}} = \frac{1}{g} \cdot \frac{1}{\eta_{\mathrm{h\,T3}}} \cdot r_{1\mathrm{T3}} \cdot \omega_{\mathrm{T}} \cdot v_{\mathrm{u1T3}} \cdot \left(1 - \frac{r_{2\mathrm{T3}}}{r_{1\mathrm{T3}}} \cdot \frac{v_{\mathrm{u2T3}}}{v_{\mathrm{u1T3}}}\right) \tag{6}$$

Between the second stage of the reactor and the third stage of the turbine, the transfer of the momentum is considered given by $r_{2\mathrm{Re2}} \cdot v_{\mathrm{t3Re2}} = r_{1\mathrm{T3}} \cdot v_{\mathrm{u1T3}}$, and

$$v_{\mathrm{t3Re2}} = \frac{r_{1\mathrm{T3}}^2 \cdot \omega_{\mathrm{T}}}{r_{2\mathrm{Re2}}} \cdot \left(1 - \frac{Q_{\mathrm{t\,T3}}}{2 \cdot \pi \cdot r_{1\mathrm{T3}}^2 \cdot \omega_{\mathrm{T}} \cdot b_{2\mathrm{Re2}} \cdot \mathrm{sd}_{1\mathrm{T3}} \cdot \tan(\beta'_{1\mathrm{T3}})}\right) \tag{7}$$

$$\alpha'_{3\mathrm{Re2}} = \mathrm{atan}\left(\frac{v'_{\mathrm{m2\,Re2}}}{v_{\mathrm{t3Re2}}}\right)$$

And the tangential speed at the entrance of the second reactor will be:

$$v_{\mathrm{t1Re2}} = \frac{v'_{\mathrm{m1\,Re2}}}{v'_{\mathrm{m2\,Re2}}} \cdot \frac{b_{1\mathrm{Re2}}}{b_{2\mathrm{Re2}}} \cdot p_{\mathrm{Re2}} \cdot \left((1 + p_{\mathrm{Re2}}) \cdot v_{\mathrm{t3Re2}} - v'_{\mathrm{m2\,Re2}} \cdot \mathrm{ctg}(\alpha'_2)\right) \tag{8}$$

Between the outlet from the second stage of the turbine and the entrance in the second stage of the reactor, the transfer of the momentum is considered given from $r_{2\mathrm{T2}} \cdot v_{\mathrm{u2T2}} = r_{1\mathrm{Re2}} \cdot v_{\mathrm{t1Re2}}$, and the specific energy on the second stage of the turbine will be:

$$\begin{aligned} k_2 \cdot H_{\mathrm{t\,T}} &= \frac{1}{g} \cdot \frac{1}{\eta_{\mathrm{h\,T2}}} \cdot \left(u_{1\mathrm{T2}} \cdot v_{\mathrm{u1T2}} - u_{2\mathrm{T2}} \cdot v_{\mathrm{u2T2}}\right) = \\ &= \frac{1}{g} \cdot \frac{1}{\eta_{\mathrm{h\,T2}}} \cdot r_{1\mathrm{T2}}^2 \cdot \omega_{\mathrm{T}}^2 \cdot \left(1 - \frac{Q_{\mathrm{t\,T2}}}{2 \cdot \pi \cdot (r'_{1\mathrm{T2}})^2 \cdot \omega_{\mathrm{T}} \cdot b_{1\mathrm{T2}} \cdot \rho_{2\mathrm{T2}} \cdot \tan(\beta'_{1\mathrm{T2}})}\right) \cdot \\ &\quad \cdot \left(1 - \frac{r_{1\mathrm{R2}}}{r_{1\mathrm{T2}}} \cdot v_{\mathrm{t1Re2}}\right) \end{aligned} \tag{9}$$

The transfer of the momentum between the outlet of the first stage of the turbine and the inlet in the first stage of the reactor given from:

$$r_{1\,\mathrm{Re1}} \cdot v_{\mathrm{t1\,Re1}} = \frac{r_{2\,\mathrm{T1}} \cdot v_{\mathrm{u2\,T1}}}{1 + \dfrac{2 \cdot \pi \cdot c_{\mathrm{f}}}{Q} \cdot r_{2\,\mathrm{T1}} \cdot v_{\mathrm{u2\,T1}} \cdot \Delta x_{\mathrm{T1-Re1}}} \tag{10}$$

with tangential speed at the outlet from the turbine stage:

$$v_{\mathrm{u2\,T1}} = u_{2\,\mathrm{T1}} - \frac{Q_{\mathrm{t\,T1}}}{2 \cdot \pi \cdot r_{2\,\mathrm{T1}} \cdot b_{2\,\mathrm{T1}} \cdot \mathrm{sd}_{2\,\mathrm{T1}} \cdot \tan(\beta'_{2\,\mathrm{T1}})} \tag{11}$$

And at the outlet form the first stage of the reactor:

$$r_{1\,\mathrm{T2}} \cdot v_{\mathrm{u1\,T2}} = \frac{r_{2\,\mathrm{R1}} \cdot v_{\mathrm{t3\,Re1}}}{1 + \dfrac{2 \cdot \pi \cdot c_{\mathrm{f}}}{Q} \cdot r_{2\,\mathrm{Re1}} \cdot v_{\mathrm{t3\,Re1}} \cdot \Delta x_{\mathrm{Re1-T2}}} \tag{12}$$

from was results:

$$v_{\mathrm{t3\,Re1}} = \frac{r_{1\,\mathrm{T2}}}{r_{2\,\mathrm{Re1}}} \cdot \frac{v_{\mathrm{u1\,T2}}}{1 - \dfrac{2 \cdot \pi \cdot c_{\mathrm{f}}}{Q} \cdot r_{1\,\mathrm{T2}} \cdot v_{\mathrm{u1\,T2}} \cdot \Delta x_{\mathrm{Re1-T2}}} \tag{13}$$

And from the equation of the momentum will be obtained the tangential speed at the outlet of the reactor:

$$v_{\mathrm{t2\,Re1}} = \frac{r_{1\,\mathrm{Re1}}}{r_{2\,\mathrm{Re1}}} \cdot v_{\mathrm{t1\,Re1}} - (1 + p_{\mathrm{Re1}}) \cdot \left(\frac{r_{1\,\mathrm{Re1}}}{r_{2\,\mathrm{Re1}}} \cdot v_{\mathrm{t1\,Re1}} - v_{\mathrm{t3\,Re1}} \right) \tag{14}$$

and the characteristic angles:

$$\alpha'_{1/3\,\mathrm{Re1}} = \mathrm{atan}\left(\frac{v'_{\mathrm{m1/2\,Re1}}}{v_{\mathrm{t1/3Re1}}} \right) = \mathrm{atan}\left(\frac{1}{v_{\mathrm{t1/3Re1}}} \cdot \frac{Q}{2 \cdot \pi \cdot r_{1/3\,\mathrm{Re1}} \cdot b_{1/2\,\mathrm{Re1}} \cdot sd_{1/2\,\mathrm{Re1}}} \right) \tag{15}$$

$$\alpha'_{2\,\mathrm{Re1}} = \mathrm{atan}\left(\frac{1}{(1 + p_{\mathrm{Re1}}) \cdot \mathrm{ctg}(\alpha'_{3\,\mathrm{Re1}}) - p_{\mathrm{Re1}} \cdot \dfrac{b_{2\,\mathrm{Re1}}}{b_{1\,\mathrm{Re1}}} \cdot \mathrm{ctg}(\alpha'_{1\,\mathrm{Re1}})} \right) \tag{16}$$

Volumetric losses

The lost flows are obtained calculating the losses in the seals:

$$Q_{\mathrm{p(i)}} = Cd_{\mathrm{et(i)}} \cdot S_{\mathrm{et(i)}} \cdot \sqrt{2 \cdot \mathrm{g} \cdot \frac{\Delta p_{\mathrm{et(i)}}}{\gamma}} \tag{17}$$

with the pressures Δp_{et} obtained from:

- for the case of the pump:

$$\frac{\Delta p_{\mathrm{et\,P}}}{\gamma} = \frac{1}{g} \cdot \left(u_{2\mathrm{P}} \cdot v_{\mathrm{u3P}} - u_{1\mathrm{P}} \cdot v_{\mathrm{u1P}} \right) - \frac{v_{3\mathrm{P}}^2 - v_{1\mathrm{P}}^2}{2 \cdot g} - \frac{\omega_{\mathrm{P}}^2}{8 \cdot g} \cdot \left(r_{2\mathrm{P}}^2 - r_{1\mathrm{P}}^2 \right) \tag{18}$$

- for the case of the turbine:

$$\frac{\Delta p_{\mathrm{et\,T(j)}}}{\gamma} = \frac{1}{g} \cdot \left(u_{1\mathrm{T(j)}} \cdot v_{\mathrm{u1T(j)}} - u_{2\mathrm{T(j)}} \cdot v_{\mathrm{u2T(j)}} \right) - \\ - \frac{v_{1\mathrm{T(j)}}^2 - v_{2\mathrm{T(j)}}^2}{2 \cdot g} - \frac{\omega_{\mathrm{P}}^2}{8 \cdot g} \cdot \left(r_{1\mathrm{T(j)}}^2 - r_{2\mathrm{T(j)}}^2 \right) \quad \left(\mathrm{j} = \overline{1,3} \right) \tag{19}$$

Hydraulic losses

Hydraulic losses are written in the form:

$$h_{\mathrm{p}} = \zeta \cdot \frac{v^2}{2 \cdot g} \tag{20}$$

where the characteristic speed, v, is considered ay the outlet of the respective blade. Hydraulic losses in the rotors can be, also, calculated with:

$$h_{\mathrm{p}} = \zeta(s) \cdot \frac{v_{\mathrm{m2}}^{'2}}{2 \cdot g} \tag{21}$$

where the slide "s" is defined by:

$$s = \frac{\omega_{\mathrm{P}} - \omega_{\mathrm{T}}}{\omega_{\mathrm{P}}} = 1 - i \tag{22}$$

(with "i" the speed ratio of the rotational speeds).

The shock losses at the inlet of the blades are considered, at the inlet of the rotors:

$$h_{\mathrm{p\,soc\,R\,(i)}} = \zeta_{\mathrm{soc}} \cdot \frac{u_1^2}{2 \cdot g} \cdot \left(1 - \frac{Q}{\mathrm{Q}_0} \right)^2 \tag{23}$$

with R (i) the considered rotor, Q – current flow, Q_0 – nominal flow, and for the reactors:

$$h_{\mathrm{p\ soc\ Re(i)}} = \zeta_{\mathrm{soc}} \cdot \frac{u_2^2}{2 \cdot g} \cdot \left(\frac{r_{2\ \mathrm{R(i)}}}{r_{1\ \mathrm{Re(i)}}} \right)^2 \cdot \left(\frac{Q}{Q_0} - 1 \right)^2 \tag{24}$$

Dividing the sum of hydraulic losses, for each stage, from specific energy (head), the real specific energy will be obtained.

The transmission ratio will give the relation between rotational speeds.

The torques momentum will result as: $M_{\mathrm{P}} = (\rho \cdot g \cdot Q \cdot H_{\mathrm{P}})/(\omega_{\mathrm{P}} \cdot \eta_{\mathrm{P}})$ and, respectively, $M_{\mathrm{T}} = (\rho \cdot g \cdot Q \cdot H_{\mathrm{T}} \cdot \eta_{\mathrm{T}})/\omega_{\mathrm{T}}$.

Having the connection between torque momentum $\mu = M_T / M_P$, the global efficiency of the torque converter is given by:

$$\eta = \frac{P_{\mathrm{arbT}}}{P_{\mathrm{arbP}}} = \frac{M_{\mathrm{T}} \cdot \omega_{\mathrm{T}}}{M_{\mathrm{P}} \cdot \omega_{\mathrm{P}}} = \mu \cdot i \tag{25}$$

Numerical results

Based on previous relations, the characteristic curves for a torque converter of Lysholm-Smith type was calculated, using a personal code.

Some of the obtained results are presented in figures 2, 3, 4, 5, 6, 7, and 8.

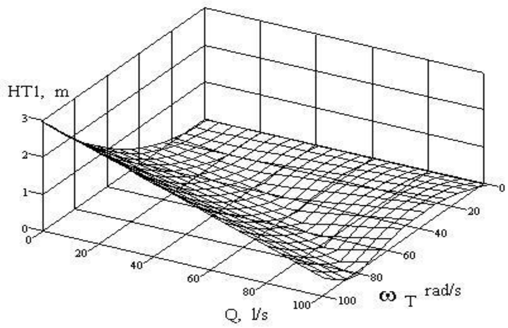

Figure 2. The variation of hydraulic losses in the first turbine stage

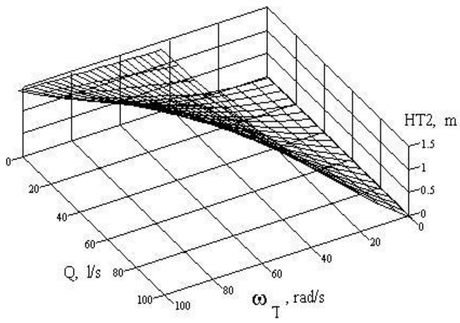

Figure 3. The variation of hydraulic losses in second turbine stage

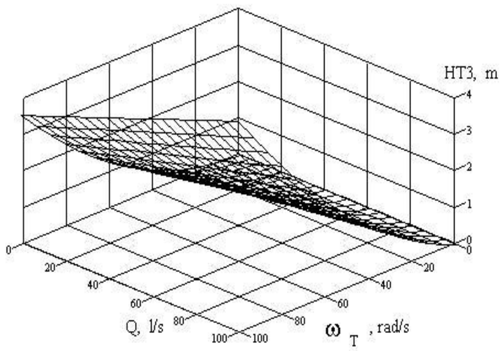

Figure 4. The variation hydraulic losses in the third turbine stage

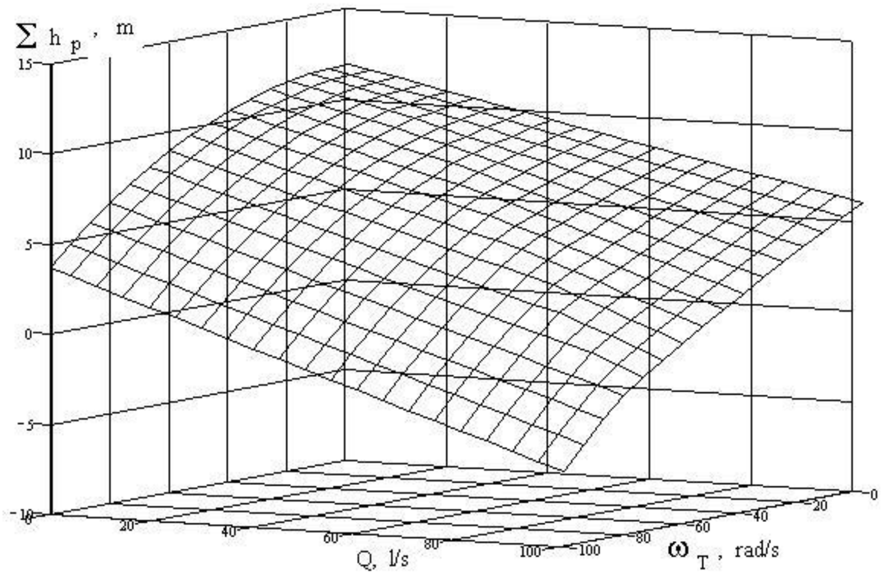

Figure 5. The variation the losses in the torque convertor

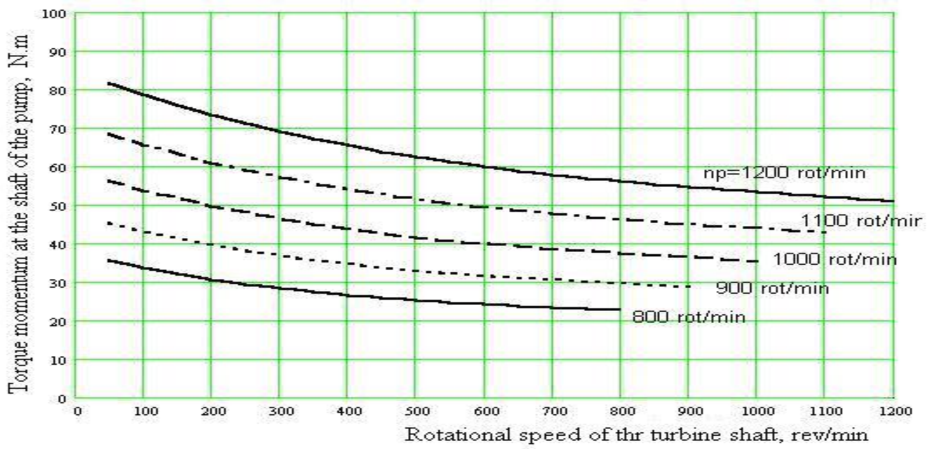

Figure 6. The variation of the torque momentum at the shaft of the pump

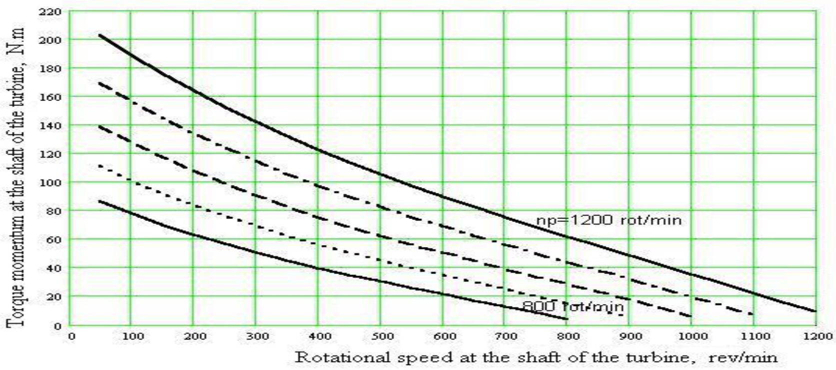

Figure 7. The variation of the torque momentum at the shaft of the turbine

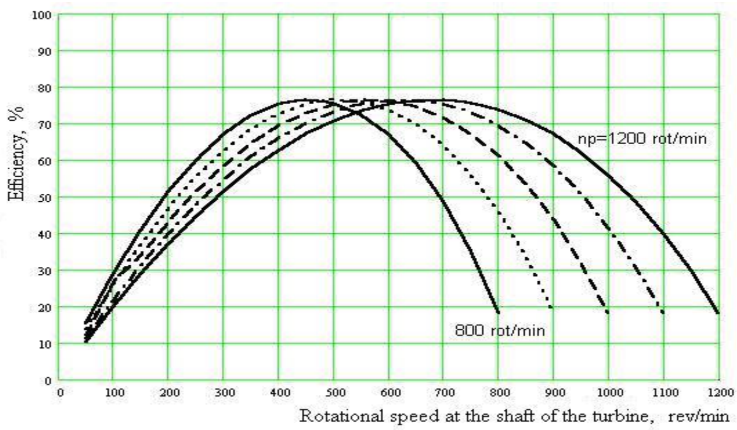

Figure 8. The dependence of the efficiency from the turbine rotational speed

Experimental results

The experimental rig

The testing rig (figure 9) consists from: 1 – torque converter Lysholm–Smith, 2 – multiple speed induction motor, 3 – dynamo for load, 4 – Ward – Leonard system for power regeneration, 5 – additional electric aggregate for excitation, 6 – closed circuit for coupling brake circuit, 7 – gear pump, 8 – radiator, 9 – starter equipment, 10 – gauge for torque, 11 – data acquisition system: FC – frequency converter, 12 – data acquisition equipment and computer [1, 4, 5].

The experiments were made in two different cases: with and without the cooling system connected.

In both cases was measured the variation of the temperature inside the CHC – 380 torque converter, for different filling degrees: 100%, 95 % and 75%. In order to measure the oil temperature, was used a type K chrome – alumel probe, having the diameter of 1.5 mm, and DS 1820 sensor. The advantages of the use of the thermocouple are: the direct contact with the oil, the possibility of its direct connection to the measurement equipments, low sensibility to shocks and vibrations, high resistance at pressure variations, resistance to corrosion. The obtained signal from the thermocouple is in mV and is measured with a voltmeter; the value of the temperature is obtained from the characteristic curve of the thermocouple. For the DS 1820 sensor, the temperature is read on an electronic display.

The characteristics of both instruments are presented in table 1.

Figure 9.1. The testing rig

Figure 9.2 The cooling system

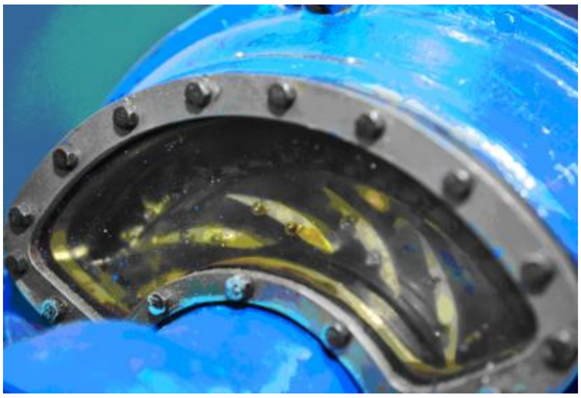

Figure 9.3 View of the torque convertor – the stator blades

Table 1.

Temperature probe	Temperature range	Accuracy
Chrome-Alumel type K	$0 \div 1000\ ^0C$	± 0,75 %.
Sensor DS 1820	$-55\ \div +125^0C$	$0{,}5^0C$

Experimental results

In figure 10 are presented the results of the measured values of the temperature for a filling degree of 100 % with the cooling system coupled, and in figure 11 the variation of the temperature for filling degrees of 95% and 75% without the cooling system. In the first case, the temperature reaches a constant value of 63°C. In the second case, the temperature has continuum rising tendency in time.

For different values of the pump rotational speed was obtained thought ABB 550 converter, position 11 in figure 9, and taken as constant by the converter, the following values was measured:

– the torque at the pump shaft, given by the ABB 550 converter;
– the torque at the turbine shaft, measured with a torque transducer of HBM T22/200 Nm type, having the nominal torque at N · m, precision class 0,5 and the output signal at -5 ... 5 V or 10 ± 8 mA, and
– the rotational speed at the turbine shaft was measured with a transducer of slot – disc type.

Data was acquired with a National Instruments NI 6212 devise, using a LabView 8.6 soft.

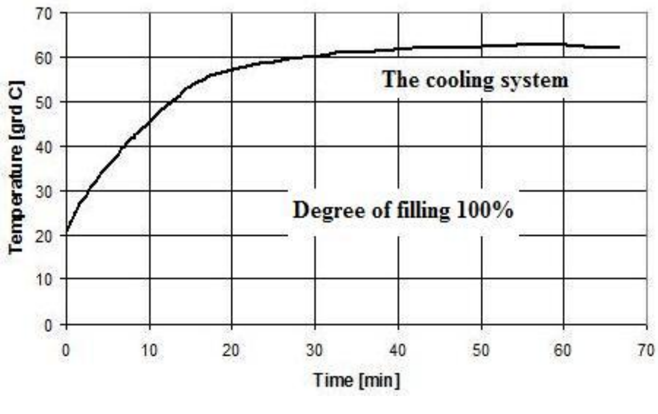

Figure 10. The variation of the temperature of the oil in torque converter with cooling system

From this data, using the relation for the mechanical power $P = M \cdot \omega$, was obtained the angular speed ω.

The efficiency of the torque converter was given by:

$$\eta = \frac{P_{\text{arbT}}}{P_{\text{arbP}}} \tag{26}$$

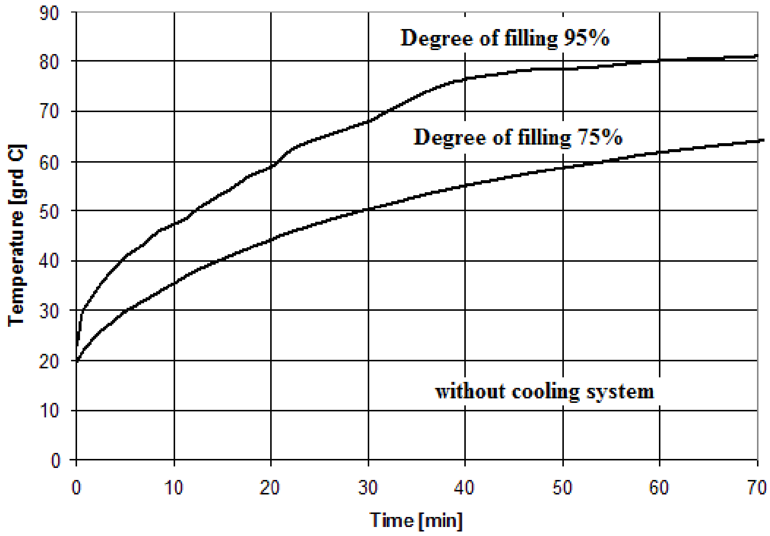

Figure 11. The variation of the temperature of the oil in torque converter without cooling system

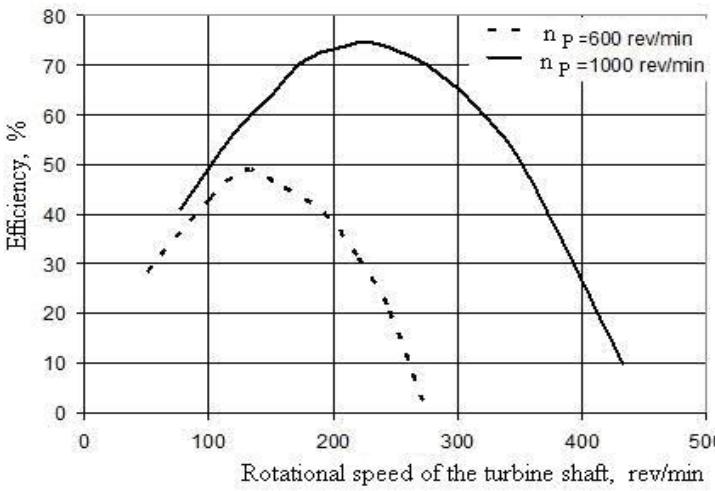

Figure 12. The variation of the torque converter efficiency, with cooling system coupled

Analyzing the curves from figure 12, results that when the cooling system was coupled, the efficiency of the torque converter increase proportionally with the pump rotational speed. In the same time, for the same rotational speed of the shaft of the pump, as for example for n_P = 1000 rot/min, as is shown in figure 13, results a decrease of the efficiency up to 20 % in the case of not coupling the cooling system.

Characteristic for the Lysholm-Smith type torque converter is the fact that, for constant values of the rotational speed at the inlet shaft, the torques increase as the same time the resistant torques at the outlet shaft increase too, *i. e.* in the same time that decrease the rotational speed ratio. This characteristic of the torque converter is called "the transparency". The variation of the torque at the outlet shaft produces variation of the rotational speed at the inlet shaft.

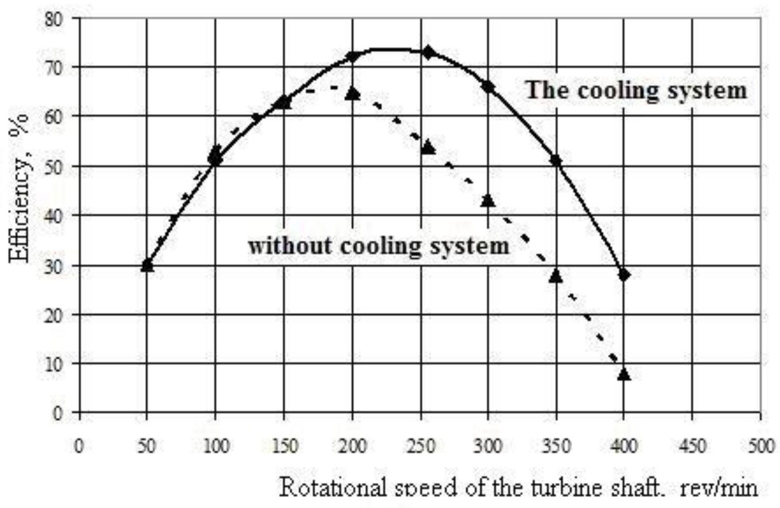

Figure 13. The variation if the torque converter efficiency at n_P = 1000 rev/min

Degree of transparency

As function of their constructive solution, "the transparency" can have larger or smaller values, this fact characterizing the Lysholm-Smith type converters.

The degree of transparency is defined as:

$$DT = \frac{M_{P\left(\frac{n_T}{n_P}=0\right)}}{M_{P(M_P=M_T)}} \quad . \tag{27}$$

For a more complete analyze of the behavior of the torque converters at different working regimes, is necessary to know the variation of the transparency degree at different working regimes.

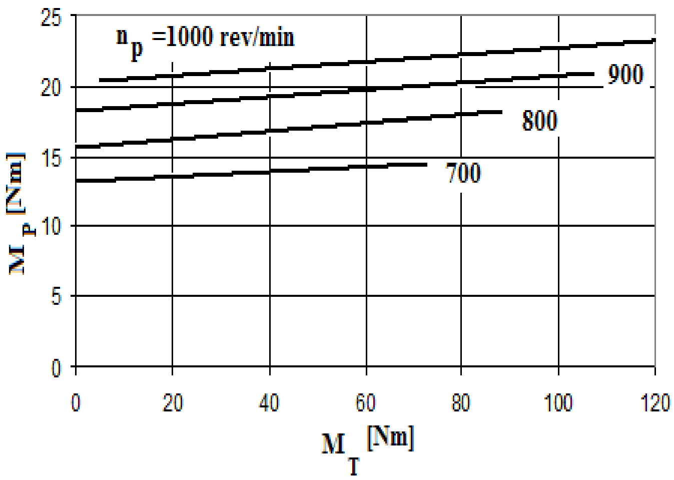

Figure 14. The variation of the torque at the shaft of the pump

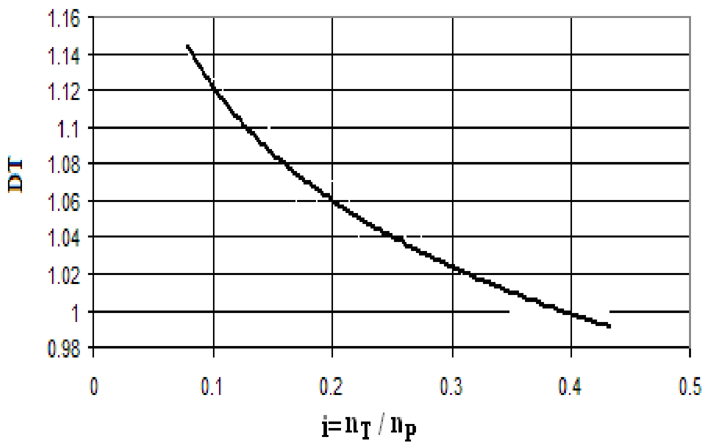

Figure 15. The variation of the transparency degree

That why the degree of transparency as function of the working regimes is defined:

$$DT = \frac{k_{M_P}\left(\frac{n_T}{n_P}=i\right)}{k_{M_P}\left(\frac{n_T}{n_P}=i_{max}\right)}, \quad i = 0;\ \ 0,1;\ \ 0,2;\ \ \ldots; i_{max} \tag{28}$$

For the analyzed torque converter, the dependency of the variation of the pump shaft as the of the variation of the turbine shaft is presented in figure 14. In figure 15 is presented the variation of the transparency degree.

Conclusions

The obtained results allowed to estimate the behavior 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 behavior of this machine.

The presented model can be used in several conditions.

As an example, the characteristics of the fluid proprieties, the behavior of the machine functioning with two-phases fluid.

In the same time, imposing a time behavior of certain parameters, the behavior of the machine in time can be anticipated.

The temperature of the oil in the converter has a continuous increase tendency in the case that the cooling system is out of function, no matter what is the value of the filling degree. But if the cooling system is coupled, the temperature stabilizes to a certain value.

The variation of the resistant torque induces variations to the inlet rotational speed, that characteristic of the torque converters of Lysholm-Smith types been called "transparency".

Acknowledgment

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Nomenclature

b – meridian width, [m]
C_d – flow coefficient, [–]
c_f – friction coefficient, [–]
DT – degree of transparency, [–]
g – gravitational acceleration, [ms^{-1}]
H – specific energy, [m]
h_p – losses, [m]
i – rotational speed ratio, [–]
k – coefficient, [–]
M – torque momentum, [Nm]
n – rotational speed, [rpm]
p – pressure, [Nm^{-2}]
p_s – sleeping factor of the pump rotor, [–]
Q – flow, [m^{3}s^{-1}]
Q_p – lost flow, [m^{3}s^{-1}]
r – radius, [m]
S – area, [m^2]
s – rotational speed slide, [–]
s_d – shuttered degree, [–]
u – transport speed, [ms^{-1}]
v – speed, [ms^{-1}]
v_t – tangential speed in reactors, [ms^{-1}]
v_u – tangential speed in rotors, [ms^{-1}]
x – coordinate, [m]

Greek letters

α – characteristic angle, [degree]
β – characteristic angle, [degree]
γ – specific weight, [Nm^{-3}]
Δ – difference, [–]
ζ – local losses coefficient [–],
η – efficiency, [–]
μ – torque momentum ratio, [–]
ρ – density, [kg]
ω – angular speed, [rad s^{-1}]

Indexis

' – shuttered values
0 – nominal
1 – inlet
2 – outlet
3 – outlet
arb – shaft
et – seal
h – hydraulic
i – current contor,
j – current contor
m – meridian

P	– pump,	soc	– shock,
R	– rotor	T	– turbine
Re	– reactor,	t	– theoretic

References

[1] Bai, L., Fiebing, M., Mitra, N. K., Numerical Analysis of Turbulent Flow in Fluid Couplings, *Journal of Fluids Engineering*, *119* (1997), 3, pp. 569-576

[2] Barglazan, A., Dobanda, V., Transmisiile Hidraulice. Construcția, Calculul, Exploatarea si Incercarea lor, Editura Tehnica, Bucharest, 1957

[3] Barglazan, M., Transmisii Hidrodinamice, Editura Politehnica, Timisoara, Romania, 2002

[4] Backstorm T. W., Lakshminarayama B., Perspective: Fluid Dynamics and Performance of Automotive Torque Converters: An Assessment, *Journal of Fluids Engineering, 118* (1996), 4, pp. 665-676

[5] Dong Y., Lakshminarayama B., Experimental Investigation of the Flow Field in a Automotive Torque Converter Stator, *Journal of Fluids Engineering, 121* (1999), 4, pp. 788-797

[6] Peligrad, N., Cuplaje Hidraulice si Convertizoare Hidraulice de Cuplu, Editura Tehnica, Bucharest, 1985

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