THEORETICAL DETERMINATION OF CHARACTERISTIC CURVES OF FIRST CLASS TORQUE CONVERTERS OPERATING WITH TWO - PHASE FLOW

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The theoretical determination of working characteristics, in torque converters case, has the advantages of efficiency, (inclusively through the utilization of similitude relations) and of the extension, offered trough mathematical models established.

On the other hand, the theoretical determination presents the drawback of confined validity of the results, only for degrees of filling near at $\chi_f = 100\%$ *,(absolutely filling / liquid).*

In this paper is presented a mathematical model for the theoretical determination of working characteristics of first class torque converters, operating with two- phase flow type mineral oil- air.

The theoretical working curves are compared with the experimental results. It comes out that, exists a good correlation between the theoretical results and experimental curves, for degrees of filling $\chi_f \in (95...100)\%$ *, especially.*

Keywords: torque converter, two- phase flow, analytical model, hydrodynamic transmission, operating degree of filling.

1. INTRODUCTION

The first class torque converters with the passing order pump-turbine-stator-pump, $(P \to T \to S \to P)$, or the second one, $(P \to S \to T \to P)$, [1], [2], [5], depend on the sequence very much. The working fluid passes through these elements.

In this article, is studied, especially, the problem of determination of the theoretical working curves of the first class torque converters, operating with two-phase flow (mineral oil- air). The assessment method is similar for the torque converters at the second class, [1], [2], [4], [5].

The mathematical model, established for the first class torque converter and presented in this paper, is valid and for the second class torque converter, if are substituted the geometrical and kinematic magnitudes, in accordance with the specific design and operating sequence, [1], [2], [5].

The numerical computation, with the mathematical model established in this article, was made for a special CHC- 350 torque converter, Lysholm-Smith type.

The torque converter Lysholm- Smith type is composed from a pump impeller, three stages of turbine runners and between them two stages of stators, [1], [2], [4].

Through the experimental work of the torque converter Lysholm-Smith type, are obtained overall operating performances (torque, input power rating, speed, etc.). Therefore, are not established specific operating performances of each stage of stator and of turbine runner. So, to simplify the mathematical model and to facilitate the comparative analysis of the experimental and theoretical results, in this paper, we considered that, the turbine runner and the stator have only one equivalent stage. Thus, in the numerical computation, were used the adequate extreme geometrical sizes, [5].

For the numerical computation with the established mathematical model, it is necessary to be known: the physical properties of the fluid; the geometry of hydraulic circuit (pump impeller, turbine runner, stator); the speeds, n_1 and n_2.

2. GEOMETRY OF THE TORQUE CONVERTER

We have underlined upward, that a first class torque converter is composed from a pump impeller, P, a turbine runner, T, and a stator, S, (Fig.1).

The basic geometrical sizes of first class torque converter, ($P \rightarrow T \rightarrow S \rightarrow P$), with one stage of turbine runner and one stage of stator, are presented in the Figure nr.1, [4], [5]. Also, in the Fig. 1, is presented the specific velocity triangle, [1], [2].

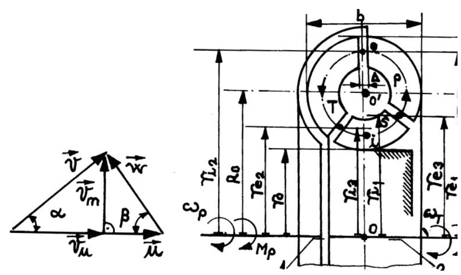


Figure. nr. 1. The basic geometrical sizes of first class torque converter. The velocity triangle.

3. ESTABLISHMENT OF THE MATHEMATICAL MODEL

3.1. Basic relations

It may be to demonstrate, [1], [2], that, the primary shaft torque of pump impeller, M_P, the secondary shaft torque of turbine runner, M_T, and the torque of stator, M_S, from a first class torque converter, operating with liquid (mono-phase flow), are:

$$M_1 \equiv M_P = \rho \cdot A \cdot r_{e_2}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_1^2 \cdot i\right] \cdot \varphi_{e_1}; \qquad (1)$$

$$M_2 \equiv M_T = \rho \cdot A \cdot r_{e_1}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_2^2 \cdot i\right] \cdot \varphi_{e_2}; \qquad (2)$$

$$M_3 \equiv M_S = \rho \cdot A \cdot r_{e_3}^3 \cdot \omega_P^2 \cdot \left[\delta_3^2 \cdot i - 1\right] \cdot \varphi_{e_3}, \qquad (3)$$

where, are used the notations established in literature, [1], [2], [4], [5], respectively:

$$i = \frac{\omega_T}{\omega_P} = \frac{n_T}{n_P} = \frac{n_2}{n_1}; \textit{speed ratio.} \qquad (4)$$

$$s = 1 - i = \frac{\omega_P - \omega_T}{\omega_P} = \frac{\omega_1 - \omega_2}{\omega_1} = \frac{n_1 - n_2}{n_1}; \quad \textit{rotational slip.} \qquad (5)$$

while, *ratio of radius, δ,* (Figure nr. 1), is:

$$\delta_1 = \frac{r_{i_1}}{r_{e_1}};$$

$$\delta_2 = \frac{r_{i_2}}{r_{e_2}} = \frac{r_{e_1}}{r_{e_2}} = \frac{r_{i_1}}{r_{e_2}} \cdot \frac{1}{\delta_1}; \qquad (6)$$

$$\delta_3 = \frac{r_{i_3}}{r_{e_3}} = \frac{r_{e_2}}{r_{e_3}} = \frac{r_{e_2}}{r_{i_1}} = \frac{1}{\delta_1 \cdot \delta_2};$$

$$\varphi_{e_i} = \frac{v_m}{u_{e_i}} = \frac{v_m}{r_{e_i} \cdot \omega_P}; \textit{ dimensionless coefficient of correlation of speeds.} (7)$$

$$A \cong \pi \cdot R^2 \cdot \left[1 - \left(\frac{r_o}{R}\right)^2\right]; \quad \textit{plan area of fluid flow.} \qquad (8)$$

When the first class torque converter works with a homogeneous two-phase flow, (for example, homogeneous mixture mineral oil - air), the relations (1), (2) and (3) become:

$$M_1 = \rho_{mix} \cdot A \cdot r_{e_2}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_1^2 \cdot i\right] \cdot \varphi_{e_1}; \qquad (9)$$

$$M_2 = \rho_{mix} \cdot A \cdot r_{e_1}^3 \cdot \omega_P^2 \cdot \left[1 - \delta_2^2 \cdot i\right] \cdot \varphi_{e_2}; \qquad (10)$$

$$M_3 = \rho_{mix} \cdot A \cdot r_{e_3}^3 \cdot \omega_P^2 \cdot \left[\delta_3^2 \cdot i - 1\right] \cdot \varphi_{e_3}; \qquad (11)$$

where, ρ_{mix} is the mean density of homogeneous two- phase flow mineral oil-air, which will be given by, [6], [7]:

$$\rho_{mix} \cong \rho_{oil} + \frac{V_{air}}{(V_{oil})_{total}} \cdot (\rho_{air} - \rho_{oil}), \left[\frac{kg}{m^3}\right] \qquad (12)$$

where, ρ_{air} is the density of air, ρ_{oil} is the density of mineral oil, while the air-mineral oil ratio is $\Pi = \frac{V_{air}}{(V_{oil})_{total}} \in [0,.......,1,0]$.

3.2. Semi - empirical relation for the mean density, ρ_{mix}

The experimental investigations, realized and presented in literature, [3], have showed that, once with the increase of operation period / time of the torque converter and with the increase of the degree of filling, χ_f, increases the fluid temperature, the secondary shaft torque, M_2, the secondary shaft speed, n_2, and the pressure from the hydraulic transmission.

Thus, we have observed that, through the alteration of the degree of filling from the value $\chi_f \cong 100\%$ at the value $\chi_f \cong 70\%$, the manometer pressure from the hydraulic transmission has decreased from the value $p \cong 5$ bar to the value $p \cong 0$! This pressure variation, evidently, alters the physical parameters of the two- phase mineral oil- air, especially, the physical properties of the air.

Thus, at these pressure variations, the density of air, ρ_{air}, alters from the value $\rho_{air} \cong 5{,}9\ \frac{kg}{m^3}$ at the value $\rho_{air} \cong 1{,}19\ \frac{kg}{m^3}$, what influences the mean density, ρ_{mix}, evidently, (relation (12)).

Therefore, it results that, the relation (12), - (that is valid for normal temperature, pressure and air humidity), - is necessary to be corrected!

It may be to consider that, at these values of pressure, the density of mineral oil, ρ_{oil}, alters very little, therefore, quite possibly to be admitted constant, $\left(\rho_{oil} \cong const. \cong 920\frac{Kg}{m^3}\right)$,[1], [2]. The variation of the density of air with the pressure quite possibly to be calculated with the following relation:

$$\rho_{air} \cong 1{,}1271333 \cdot 10^{-5} \cdot p - 0{,}047001182;[\frac{kg}{m^3}] \qquad (13)$$

where: the pressure $p[Pa]$; the relative humidity with respect to water, $\chi \cong 70\%$; the absolute temperature, $T \cong 309{,}15^\circ K$; the vapor pressure, $p_{vas} \cong 5940\ Pa$; the air constant, $R_a \cong 287\frac{KJ}{kg \cdot^\circ K}$;

On the other hand, the experimental measurement, realized and presented in literature, [3], have showed that, in the homogeneous two- phase flow mineral oil- air, the pressure increases, evidently, with the increase of the degree of filling,

χ_f. Thus, the great and fast increase of pressure was achieved at absolutely filling, $(\chi_f = 100\%)$, and in its neighborhood.

It is thought that, the variation of the pressure, p, of the homogeneous mixture mineral oil- air, and, implicitly, of the air, - with the degree of filling, $\chi_f, \chi_f \in (65,......,100)\%$, quite possibly to be expressed through the following relation:

$$p \cong \frac{\left(\chi_f - \chi_{f_o}\right)^{2,2}}{0,04}; \ [bar] \tag{14}$$

where: $\chi_f \in (0,65,...,1,0); \chi_{f_o} \cong 0,65.$

Substituting, the relation (14) in the relation (13), it results a possible dependence of the density of air, ρ_{air}, from the degree of filling, χ_f, thus:

$$\rho_{air} \cong 28,1783325 \cdot \left(\chi_f - \chi_{f_0}\right)^{2,2} - 0,047001182; \left[\frac{kg}{m^3}\right] \tag{15}$$

where: $\chi_f \in (0,65,....,1,0); \chi_{f_0} \cong 0,65.$

Then, with the relation (15), the relation (12) becomes:

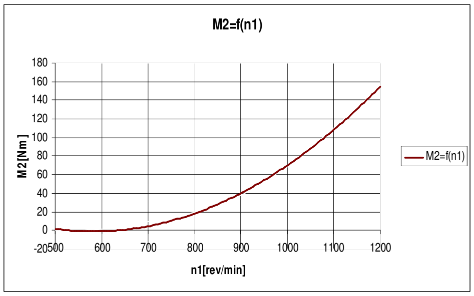
$$\rho_{mix} \cong \rho_{oil} + \frac{V_{air}}{(V_{oil})_{total}}\left[28,1783325\left(\chi_f - \chi_{f_0}\right)^{2,2} - 0,047001182 - \rho_{oil}\right]; [\frac{kg}{m^3}] \tag{16}$$

where: $\rho_{oil} \cong const.; \left(\rho_{oil} \cong 920\frac{kg}{m^3}\right); \frac{V_{air}}{(V_{oil})_{total}} \in [0,.......,1,0].$

3.3. Mathematical model of calculus

The mathematical model of calculus is composed of the relations (9), (10), (11) and (12), respectively (16), and a possible statistical relation between the primary shaft speed n_1, and the degree of filling, χ_f.

Thus, the systematic experimental investigations, realized on the torque converter Lysholm – Smith type CHC – 350, considering the speed, $n_1 \cong const. \cong 975\frac{rot}{\min}$, and altering the degree of filling, $\chi_f = 100\%,97,5\%,95\%,87,5\%,85\%,82,5\%,80\%,.....,70\%$, respectively, considering the degree of filling, $\chi_f \cong const. \cong 97,5\%$, and altering the speed of primary shaft, n_1, $n_1 = 1200,1100,1000,900,800,700,600\frac{rot}{\min}$, have allowed the determination of characteristics curves $M_2 = f(n_1)$ and $M_2 = f(\chi_f)$, Figure nr. 2.



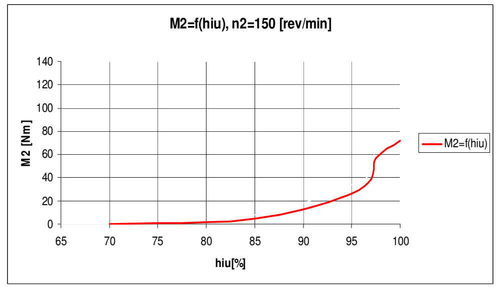


Figure nr. 2. The variation of secondary shaft torque as function of entrance speed n_1 and degree of filling, χ_f .

The experimental curve $\chi_f = f(n_1)$ quite possibly to be approximated through a mathematical relation, for example, of the following shape:

$$n_1 \cong 500 + 7{,}25 \cdot 10^{-2} \cdot (\chi_f)^{2,2} - 5{,}1 \cdot (\chi_f)^{1,2}; \left[\frac{rot}{\min}\right]. \qquad (17)$$

From the Figure nr. 2, results the mathematical dependence $\chi_f = f(n_1)$ presented in the Figure nr.3. The relation (17) associates to each degree of filling $\chi_f = const.$ a single value of speed $n_1 = const.$ and conversely.

The theoretical calculus of the torques M_P, M_T and M_S quite possibly to be effected with the relations (9), (10) and (11), together with the relations (12), (16) and (17).

Thus, the relations (9), (10), and (11) can to be written and under the following shape:

$$M_1 = \rho_{mix} \cdot A \cdot r_{e_2}^3 \cdot \frac{\pi^2}{900} \cdot \left[n_1^2 - \delta_1^2 \cdot n_1 \cdot n_2\right] \cdot \varphi_{e_1}; \qquad (18)$$

$$M_2 = \rho_{mix} \cdot A \cdot r_{e_1}^3 \frac{\pi^2}{900} \cdot \left[n_1^2 - \delta_2^2 \cdot n_1 \cdot n_2\right] \cdot \varphi_{e_2}; \qquad (19)$$

$$M_3 = \rho_{mix} \cdot A \cdot r_{e_3}^3 \cdot \frac{\pi^2}{900} \cdot \left[\delta_3^2 \cdot n_1 \cdot n_2 - n_1^2\right] \cdot \varphi_{e_3}; \qquad (20)$$

Also, we must take care about the balance equation of the mechanical torques M_ρ, M_T and M_S, which in the first class torque converter's case, $(P \to T \to S \to P)$, has the expression: [1], [2], [5]:

$$M_2 = M_1 + M_3 \qquad (21)$$

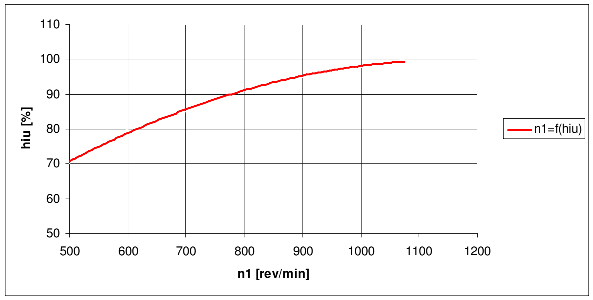


Figure nr. 3. Degree of filling in function of entrance speed of the torque converter $\chi_f = f(n_1)$.

3.4 Numerical results

A part of the theoretical results obtained are presented in the figure nr. 4, where for comparison, were superposed the experimental results, [3].We must mention the fact that in formulas (18), (19) and (20), the entrance speed, n_1 has the value actuated from the relation (17); respectively from the figure nr. 3, for the adequate filling degree, $\chi_f = const.$ and outlet speed $n_2 \in [0, \ldots, n_{2\max}]$, where the speed $n_{2\max} = n_2 |_{M_2 = 0}$

With those elements established, we may pass to the theoretical calculus of the characteristic curves of the CHC-350 torque converter, $M_2 = f(n_2)$, $M_1 = f(n_2)$ and $\eta_{CHC} = f(n_2)$, where $\eta_{CHC} = \dfrac{M_2 n_2}{M_1 n_1} = k \cdot i$, $k = i = \dfrac{n_2}{n_1}$ for different filling degrees of the torque converter. $\chi_f = const.$

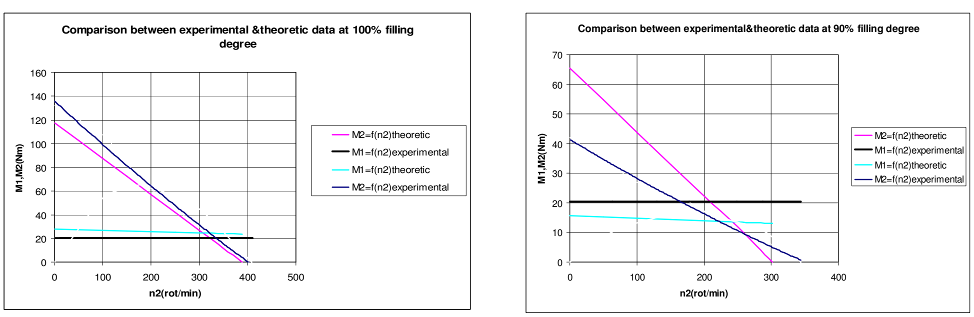


Figure nr. 4 Characteristic comparative curves for different degrees of filling a) $\chi_f = 100\%; n_1 \cong 975 rev/\min$;b) $\chi_f = 90\%; n_1 \cong 975 rev/\min$

4. CONCLUSIONS

In this paper, is squared up to the problem of theoretical determination of the characteristic curves of the hydrodynamic torque converter class I, running with two-phase flow. The concrete numerical application is referring to a hydrodynamic torque converter Lisholm-Smith CHC-350 type.

The results obtained in this paper, follows us to the next conclusions:

a) A mathematical model was establish analytically for theoretical actuate of the characteristic curves of the hydrodynamic first class torque converters, which are working with two-phase medium.

b) The established mathematical model is valid and for other hydrodynamic torque converters, class II, watching the modifying of the hydraulic circuit $(P \to S \to T \to P)$ and of it's geometry.

c) Statistical relations are established for air density ρ_{air}; mixture density, mineral oil-air ρ_{am} and for entrance speed n_1 in function of filling degree of the hydrodynamic transmission, χ_f, χ_{f0} .

d) From the comparative analysis of theoretical curves, results that, the mathematical model established in this paper approximates well the experimental results for filling degrees, $\chi_f \in (95...100)\%$.

For smaller filling degrees $\chi_f < 95\%$ the differences between theoretical and experimental curves are growing with the decreasing of the filling degree.

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