Zygmunt PASZOTA, Assoc.Prof.,D.Sc.,M.E.
Technical University of Gdańsk
Faculty of Ocean Engineering
and Ship Technology

Model of the energy losses in the displacement rotational machines for description of efficiency of the hydrostatic drive

Part II. Model of mechanical and pressure losses

SUMMARY

This paper presents an analysis of the work concerning the mathematical descriptions of the energy losses in pumps and hydraulic motors, which are applied in the hydrostatic transmissions. These losses are estimated from the point of view of the possibility to introduce the whole of the hydrostatic drive into the simulation model of energy efficiency. Model of the mechanical and pressure losses in the displacement machines with the rotational movement is proposed.

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MODEL OF MECHANICAL LOSSES

Models found in technical literature, concerning the moment losses in the displacement rotational machine, in the majority of cases combined :

⇨ ***mechanical losses*** resulting from Coulomb dry friction (between solids) and Newton viscous friction (between solids with a film of oil between surfaces)

⇨ ***pressure losses*** resulting from liquid flow in the machine channels.

The M_p loss of the moment is defined in these models as a difference between the theoretical moment M_t and effective moment M_e on the motor shaft for a given pressure decrease Δp in the machine :

$$M_p = M_t - M_e \tag{17}$$

where:

$$M_t = \frac{q_t}{2\Pi}\Delta p \tag{18}$$

The balance of the losses M_p is most often presented in the following form :

$$M_p = M_{Tk} + M_{Tp} + M_{T\mu} + M_\rho \tag{19}$$

where:

- M_{Tk} - constant moment loss resulting from mechanical friction, independent of pressure
- M_{Tp} - moment loss resulting from mechanical friction, proportional to pressure and independent of viscosity
- $M_{T\mu}$ - moment loss resulting from viscous friction in the clearances between mechanical parts, proportional to speed
- M_ρ - moment loss resulting from hydraulic losses, basically the pressure losses in the machine channels, but also from friction between rotor and liquid inside the casing.

The (19) model was proposed by Schlösser [17,18] and completed the earlier published Wilson model [14] with the M_ρ component.

Fig.8 shows the components of the model (19) as a function of the rotational speed and pressure, for a constant viscosity and mass density of the liquid.

The moment loss M_{Tk} is a constant value for different operational conditions, which means that it does not depend on the rotational speed, pressure, viscosity or the mass density of the liquid. These losses result from the mechanical friction in the sealing rings of shafts, in the slide or roll bearings. Their magnitude is a function of the adapted technology and the permitted manufacturing tolerances.

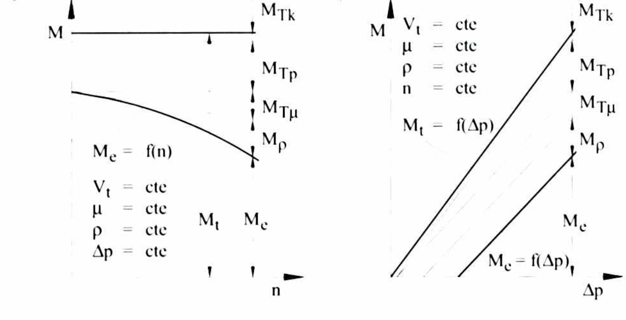

Fig.8. *Components of the moment losses M_p in the rotational hydraulic motor, according to Schlösser model [17,18], as a function of **a)** - rotational speed, **b)** - pressure*

The moment loss M_{Tp} is due to dry mechanical friction proportional to stresses between mechanical parts in contact; these stresses result from the pressure difference Δp. The M_{Tp} moment may be presented as proportional to theoretical moment M_t :

$$M_{Tp} = C_{Tp} M_t = C_{Tp}\,\Delta p \frac{q_t}{2\Pi} \tag{20}$$

where :

C_{Tp} - proportionality factor.

As the M_{Tp} moment does not depend on the rotational speed, the C_{Tp} factor may be considered constant.

The moment loss $M_{T\mu}$ result from the viscous friction of the liquid. The authors of the model place these losses first of all between the surface of parts where an oil film is created. The model of this moment may be the following :

$$M_{T\mu} = C_{T\mu}\,\mu\,\omega \frac{q_t}{2\Pi} \tag{21}$$

where :

$C_{T\mu}$ - proportionality factor

ω - angular speed of the motor shaft.

The moment loss M_p, the basic cause of which are the pressure losses of the turbulent flow of liquid in the machine channels and distributor, is proportional to the square of the liquid speed in the channels, that means proportional to the square of the angular speed of shaft, according to the model :

$$M_p = C_p\,\rho\,\omega^2 \sqrt[3]{\left(\frac{q_t}{2\Pi}\right)^5} \tag{22}$$

where :

C_p - proportionality factor

ρ - mass density of the liquid.

In the C_p factor value, Schlösser included not only the influence of pressure losses in the channels and distributor, but also internal friction in bearings and that existing between the rotor and liquid inside the casing, friction proportional to the mass density ρ of liquid and square of angular speed.

The total loss M_p of moment (19) may then be presented in the form :

$$M_p = M_{Tk} + C_{Tp}\,\Delta p \frac{q_t}{2\Pi} + C_{T\mu}\,\mu\,\omega \frac{q_t}{2\Pi} + C_p\,\rho\,\omega^2 \sqrt[3]{\left(\frac{q_t}{2\Pi}\right)^5} \tag{23}$$

The experimental studies confirm the usefulness of this model first of all in the high-speed displacement machines.

Thoma [15], taking the (19) model as a basis, obtained a valid model of variable capacity displacement machines and presented the M_p expression in the form :

$$M_p = C_p\, b^3 \rho\, \omega^2 \sqrt[3]{\left(\frac{q_t}{2\Pi}\right)^5} \tag{24}$$

where $0 < b < 1$ is a variation coefficient of the momentary capacity q_{gv} of the machine in relation to its maximum theoretical capacity q_t.

Prokofiev [16], using a model close to the Wilson model, takes into account only the loss of moment resulting from the Coulomb dry friction and Newton viscous friction. He neglects in the balance the loss of moment M_p due to pressure losses during the flow of liquid in the channels and distributor of the machine. From this hypothesis results a linear model of moment losses, very useful in the theory of similarity of the hydraulic displacement machines.

In order to verify the use of model (23) in the low-speed hydraulic motors, Balawender [13] analysed, on the experimental basis, characteristics of six different low-speed motors among them SWSB-40 and SWSB-63. Fig.9. shows the results.

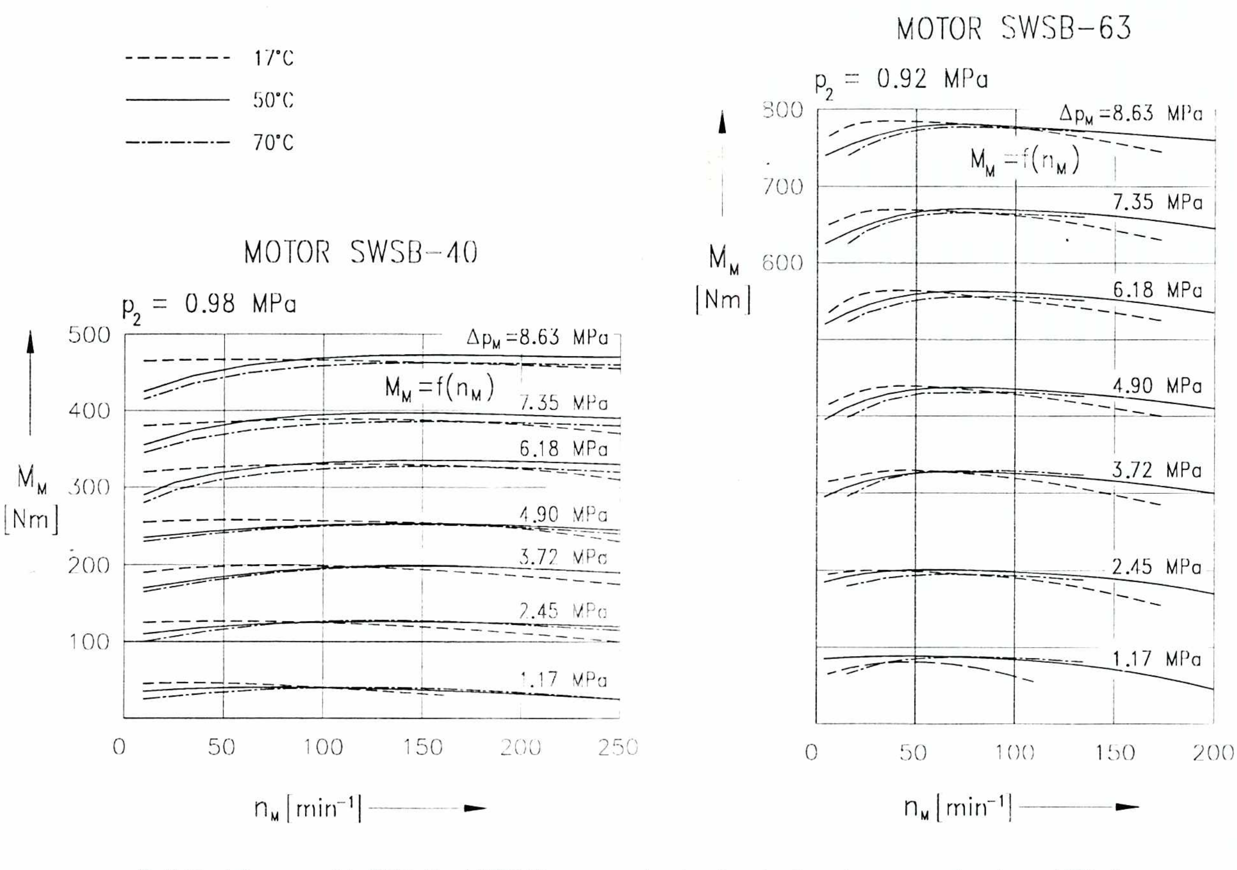

***Fig.9.** M_M shaft moment of the SWSB-40 and SWSB-63 motors as a function of rotational speed n_M, pressure drop Δp_M and H30 oil temperature*

It may be stated that real characteristics differ distinctly, in the zone of low speed, in comparison with the model (23), (Fig.8a). When the speed n_M tends to zero, the shaft moment of the motor decreases, at the pressure drop Δp_M = cte. This is a consequence of the increase of dry mechanical friction forces as the sliding speed between mechanical parts decreases.

To illustrate this phenomenon, an experimental study [13] was carried out of the friction moment between the sleeve 5 and rotor 4 of the cylindrical distributor in the SWSB-63 motor (see Fig.1). The pistons 3 were removed and replaced by mandrels. Pressure supply of the motor was provided and, driving the rotor, the M_R moment, corresponding to the mechanical losses in the distributor, was directly measured.

The source of mechanical losses is here the viscous friction in the clearance and dry or mixed friction of the sealing rings under the pressure.

The results of these measurements are presented in Fig.10 a,b,c,d. It shows a non-linear decrease and then slight linear increase of the mechanical loss moment as a function of speed (Fig.10 a). It shows also a linear change of this moment as a function of pressure (Fig.10 b,c).

The M_R moment of mechanical losses in distributor may be presented as a sum of the following components :

$$M_R = M_{Tk} + M_{T\mu} + M_{Tp} + \Delta M_{Tm} + \Delta M_{Tr} \qquad (25)$$

where:

M_{Tk} = cte - constant component of mechanical friction, independent of the speed and pressure, resulting from precompression of the sealing system and the manufacturing tolerances of various parts

$M_{T\mu} = C_{R\mu}\mu$ - component of the viscous friction in the clearance, proportional to the rotational speed n with the

$$C_{R\mu} = \frac{dM}{dn} = \text{cte}$$

coefficient independent of the pressure (Fig.10 a)

$M_{Tm} = M_{Tp} + \Delta M_{Tm}$ - component of the mixed mechanical friction, a function of pressure and speed, where :

$M_{Tp} = C_{Rp}p$ - component of the mixed mechanical friction, proportional to the pressure, with the

$$C_{Rp} = \frac{dM}{dp} = \text{cte}$$

coefficient in the zone above the critical speed $n > n_{cr}$ (Fig.10 c)

$\Delta M_{Tm} = \Delta C_{Rm}p = f(n)$ - component allowing for the change of mixed friction as a function of speed in the zone below the critical speed $0 < n < n_{cr}$, with

$$\frac{dM_R}{dp} = \Delta C_{Rm} + C_{Rp} = f(n) \quad \text{(Fig.10 d)}$$

$\Delta M_{Tr} = M_{Rrmax} - M_{Rrmin}$ - component allowing for the modification of the mixed friction during starting (Fig.10 a). The ΔM_{Tr} value depends, among others, on the pressure and the time of immobility of the motor before starting (Fig.10 b).

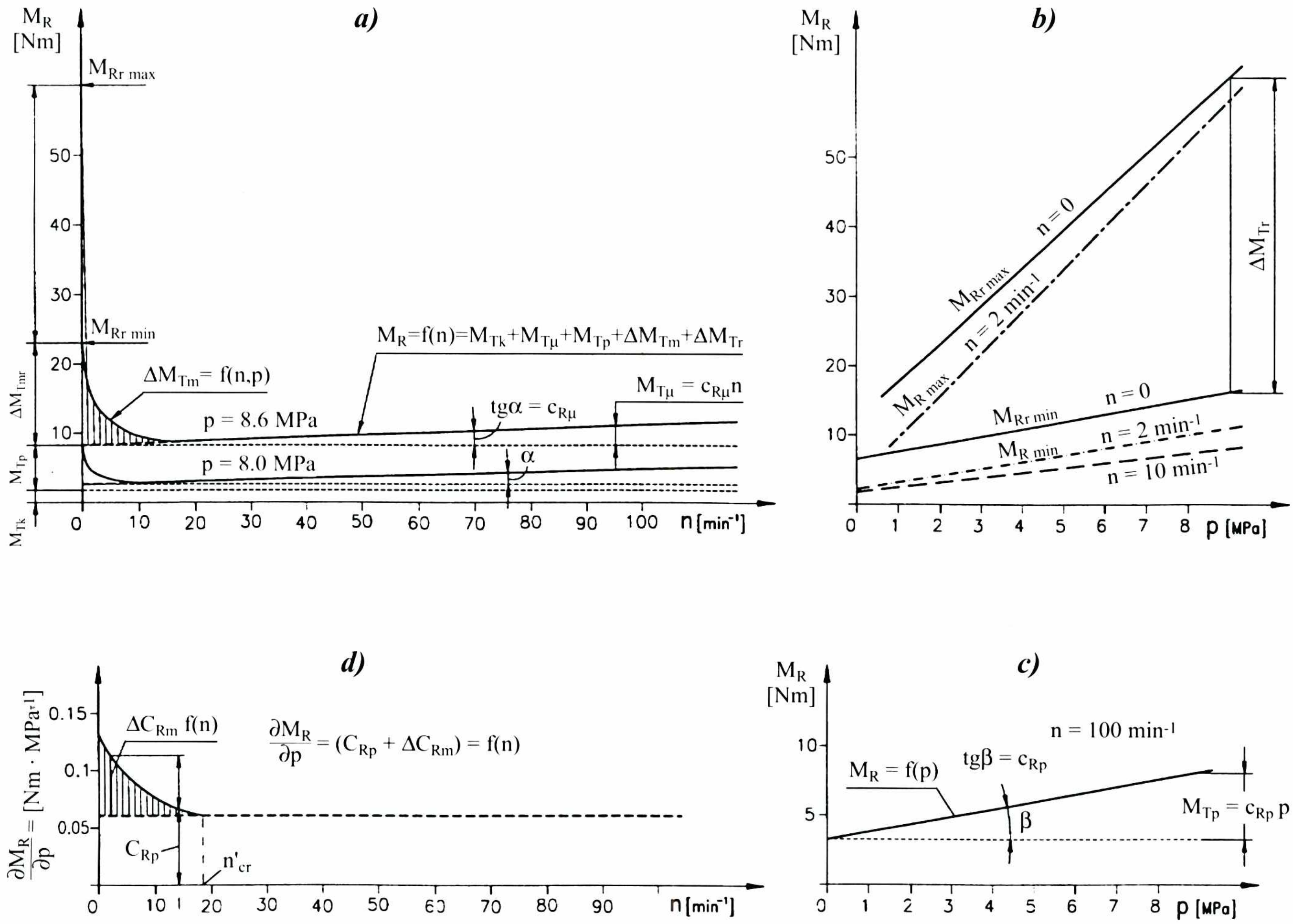

Fig.10. Friction moment between the slave and rotor of the cylindrical distributor in the SWSB-63 hydraulic motor

The example shown above demonstrates that the moment losses proposed by Schlösser (23), should be completed, in the case of low--speed motors, with a ΔM_{Tm} component allowing for increase of friction moment for very low speeds (below the critical speed n_{cr}) and with a ΔM_{Tm} component allowing for the sudden increase of friction moment as a difference between the static and kinetic friction.

The remarks presented by Balawender are very important particularly for the analysis of the behaviour of the low-speed hydraulic motors operating at very low speed. It is in this configuration that the phenomenon of the irregularity of motor speed can be observed, which is connected with the total increase of friction moment. On the other hand, the movement irregularity problem is much more serious with the SWSB type motors than with other motors.

In this study it is proposed to separate the mechanical friction losses and the pressure losses in the channels of the machine. Both these losses reduce in the case of hydraulic motor, for a given pressure drop Δp, effective shaft moment M_e. But, on the other hand, the losses are of different character :

- mechanical losses between moving parts, corresponding first of all to dry friction, are mainly a function of working pressure and do not depend practically on the rotational speed (in the zone above the critical speed, very low in the considered solutions) or the viscosity of the liquid
- pressure losses in the channels and in the distributor of the machine are not a function of pressure and they depend mainly on the flow intensity, in other words on the rotational speed, and, in certain cases (mainly for low temperatures and small intensity flows) on viscosity of the liquid.

Disregarding a narrow zone where the rotational speed is irregular in the majority of low speed motors and, on the other hand, taking into account the fact that, in the area where the speed is stable, the losses and mechanical efficiency depend little on the rotational speed and the viscosity of the liquid, one can assume that the **model of mechanical losses in the hydraulic motor** has the form (Fig.11) :

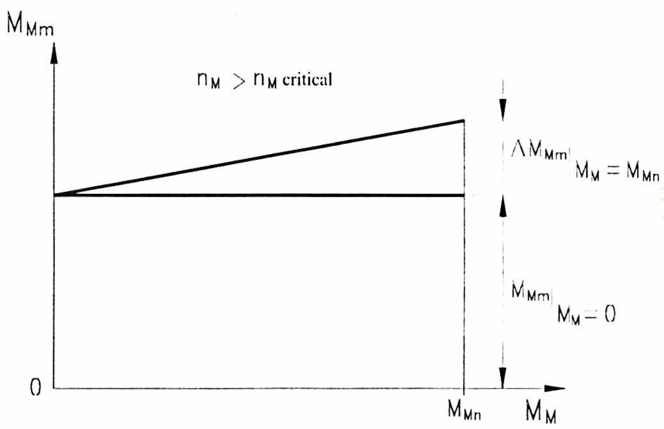

Fig.11. Approximate relation of the M_{Mm} moment of mechanical friction losses in a hydraulic motor of a constant capacity to the shaft moment M_M

$$M_{Mm} = M_{Mm|M_M=0} + \Delta M_{Mm|M_M} \tag{26}$$

where :

$$M_{Mm|M_M=0} = k_{7.1} M_{Mn} = k_{7.1} \frac{q_{Mt} p_n}{2\Pi} \tag{27}$$

and

$$\Delta M_{Mm|M_M} = k_{7.2} M_M \tag{28}$$

with the nominal moment M_{Mn} of the motor :

$$M_{Mn} = M_M \left| \begin{array}{l} \Delta p_M = p_n \\ \eta_{Mm} = 1 \\ \eta_{Mp} = 1 \end{array} \right. = \frac{q_{Mt} p_n}{2\Pi} \tag{29}$$

where p_n is the nominal pressure of the hydraulic system in which the motor is used.

The coefficient :

$$k_{7.1} = \frac{M_{Mm|M_M=0}}{M_{Mn}} = \frac{2\Pi\, M_{Mm|M_M=0}}{q_{Mt} p_n} \tag{30}$$

will receive the value obtained for the kinematic viscosity of the liquid $\upsilon_n = 35$ mm²s⁻¹ recommended by the manufactures of elements, whose energy efficiency is often presented.

Elsewhere the coefficient :

$$k_{7.2} = \frac{\Delta M_{Mm|M_M=M_{Mn}}}{M_{Mn}} = \frac{2\Pi\, \Delta M_{Mm|M_M=M_{Mn}}}{q_{Mt} p_n} \tag{31}$$

will receive the value obtained for an M_{Mn} moment corresponding to the nominal pressure p_n and $\upsilon_n = 35$ mm²s⁻¹.

The (26) model demonstrates at the same time that in the case of hydraulic motor the moment of mechanical losses is not a direct function of the pressure drop Δp_M but the M_M shaft moment of the motor.

In the case of a displacement machine working as a pump the speed n_P does not influence the M_{Pm} moment of mechanical losses (because it is always high) and the mechanical losses are a function of the indicated pressure increase Δp_{Pi} in the working chamber (the pump shaft moment M_P depends on the pressure increase Δp_{Pi} and on the M_{Pm} moment of mechanical losses between the working chamber and the driving shaft). The model of the M_{Pm} moment of mechanical losses may be presented in the form (Fig.12) :

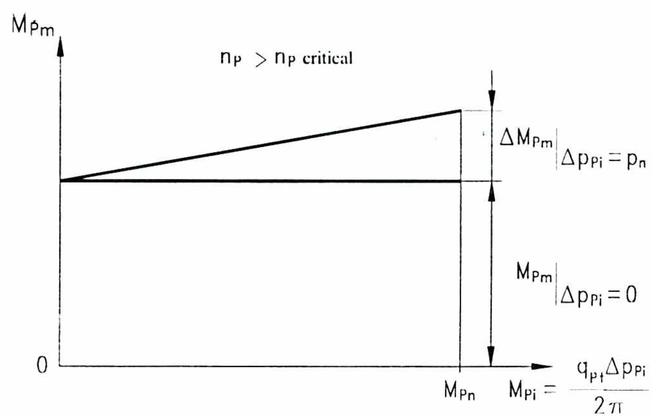

Fig.12. Approximate relation of the M_{Pm} moment of mechanical losses in a constant capacity pump to the M_{Pi} moment resulting from the pressure increase Δp_{Pi} in the pump working chamber and from theoretical working capacity q_{Pt}

$$M_{Pm} = M_{Pm|\Delta p_{Pi}=0} + \Delta M_{Pm|\Delta p_{Pi}} \tag{32}$$

where :

$$M_{Pm|\Delta p_{Pi}=0} = k_{4.1} M_{Pn} = k_{4.1} \frac{q_{Pt} p_n}{2\Pi} \tag{33}$$

and

$$\Delta M_{Pm|\Delta p_{Pi}} = k_{4.2}\frac{q_{Pt}\,\Delta p_{Pi}}{2\Pi} = k_{4.2}M_{Pi} \tag{34}$$

with the nominal moment M_{Pn} of the pump :

$$M_{Pn} = M_P \left| \begin{array}{l} \Delta p_P = p_n \\ \eta_{Pm} = 1 \\ \eta_{Pp} = 1 \end{array} \right. = \frac{q_{Pt}p_n}{2\Pi} \tag{35}$$

In this expression, p_n is the nominal pressure of the hydraulic system in which the pump is used and the coefficients :

$$k_{4.1} = \frac{M_{Pm|\Delta p_{Pi}=0}}{M_{Pn}} = \frac{2\Pi\, M_{Pm|\Delta p_{Pi}=0}}{q_{Pt}p_n} \tag{36}$$

$$k_{4.2} = \frac{\Delta M_{Pm|\Delta p_{Pi}=p_n}}{M_{Pn}} = \frac{2\Pi\,\Delta M_{Pm|\Delta p_{Pi}=p_n}}{q_{Pt}p_n} \tag{37}$$

take the value obtained for Δp_{Pi} equal to the nominal pressure p_n.

MODEL OF PRESSURE LOSSES

Schlösser [17] presents the moment loss (22) as a result particularly of the local pressure losses in the channels of the machine :

$$\Delta p_p = \xi \rho \frac{v^2}{2} \tag{38}$$

where :
- ξ - coefficient of the local pressure losses (determined experimentally)
- ρ - mass density of the liquid
- v - average speed of oil flow in the channels.

It is assumed that ξ coefficient remains constant.

According to studies [22] of the pressure losses in hydraulic elements, the ξ coefficient may be presented as a function of the Reynold's number Re of liquid flow in the channel :

$$\xi = \frac{K_1}{Re} + K_2 \tag{39}$$

where :

$$Re = \frac{d_h v}{\upsilon} \tag{40}$$

with d_h - hydraulic diameter of the channel in the machine :

$$d_h = \frac{4S_h}{U} \tag{41}$$

where :
- S_h - cross-section area of the channel
- U - perimeter of the wet section
- $Q = \frac{q_t}{2\Pi}\omega = D_t^3\omega$ - flow intensity in the channel
- υ - kinematic viscosity of the liquid [m^2s^{-1}].

In most cases, the increase of ξ coefficient occurs for Re < 1000.

Introducing (39) to (38), one obtains :

$$\Delta p_p = \frac{K_1 \upsilon \rho}{2d_h}\frac{Q}{S_h} + K_2\frac{\rho}{2}\left(\frac{Q}{S_h}\right)^2 \tag{42}$$

After introducing the non-dimensional constants $C_{p\mu}$ and $C_{p\rho}$ satisfying the relations :

$$\frac{K_1}{2d_hS_h} = \frac{C_{p\mu}}{D_t^3} \tag{43}$$

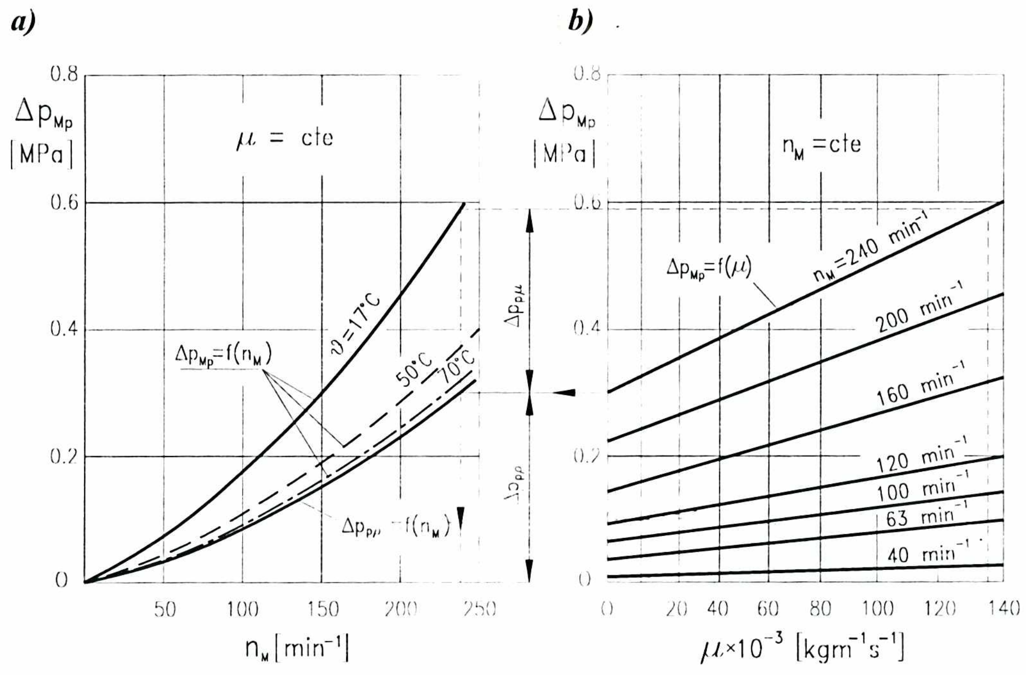

Fig.13. Characteristics of the pressure losses Δp_p in the channels of low-speed SWSB-63 hydraulic motor as a function of : a) - rotational speed n_M, b) - viscosity μ of the fluid

$$\frac{K_2}{2S_h^2} = \frac{C_{p\rho}}{D_t^4} \tag{44}$$

one obtains the equation giving the pressure losses Δp_p in the displacement machines in the form :

$$\Delta p_p = C_{p\mu}\mu\,\omega + C_{p\rho}\rho\,D_t^2\omega^2 \tag{45}$$

So one can describe the total pressure loss Δp_p as a sum of the $\Delta p_{p\mu}$ component depending on the viscosity μ of the liquid and $\Delta p_{p\rho}$ component depending on the mass density ρ of the liquid :

$$\Delta p_p = \Delta p_{p\mu} + \Delta p_{p\rho} \tag{46}$$

To illustrate this phenomenon the examples can be presented of the characteristics of pressure losses in the channels of a low-speed SWSB-63 motor elaborated by Balawender [13]. The results shown in Fig.13 confirm the appearance of two components $\Delta p_{p\rho}$ and $\Delta p_{p\mu}$. The $\Delta p_{p\rho}$ component is defined in the Fig.13 b by extrapolation of a function $\Delta p_p = f(\mu)$ for $\mu \to 0$.

The $\Delta p_p = f(n)$ characteristics found for the viscosity μ of the liquid corresponding to the 50° and 70° temperatures (kinematic viscosity $\upsilon = 28\ mm^2s^{-1}$ and $13\ mm^2s^{-1}$) are practically the same as those of $\Delta p_{p\rho} = f(n_M)$. For a high viscosity (oil temperature 17°C with kinematic viscosity $\upsilon = 150\ mm^2\,s^{-1}$) the difference between the characteristics becomes much greater.

Fig.14 shows characteristics of the ξ coefficient of pressure losses as a function of Re in the channels of SWSB-63 motor.

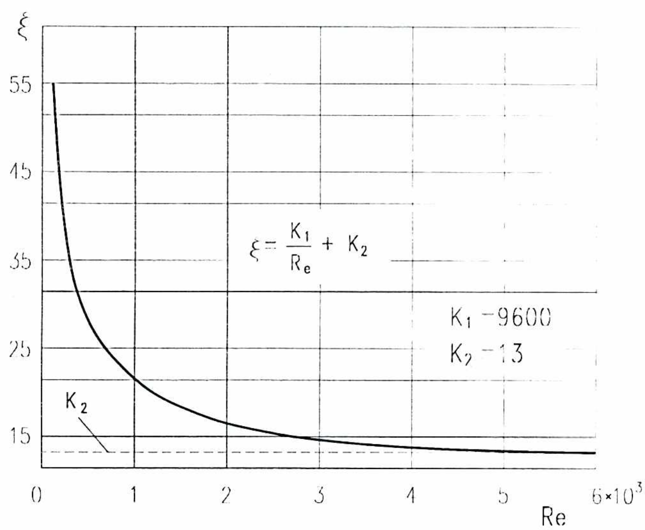

Fig.14. Characteristics of the ξ coefficient of local pressure losses as a function of Re number in the channels of SWSB motor [13]

The above analysis concerning the results of the investigation of pressure losses Δp_p in the SWSB-63 motor shows that in the zone of recommended viscosity ($\upsilon = 35\ mm^2\,s^{-1}$) one can neglect, in the first approximation, the component $\Delta p_{p\mu}$ of the pressure losses depending on the viscosity of the liquid.

One can choose the model (38) in which the flow in the channels and in the distributor of the machine is considered turbulent with pressure losses proportional to the square of flow intensity through the machine.

In the study of efficiency of the elements and hydrostatic transmission the following **models of pressure losses Δp_P** are proposed in Fig.15 and Fig.16.

For the pump (Fig.15) :

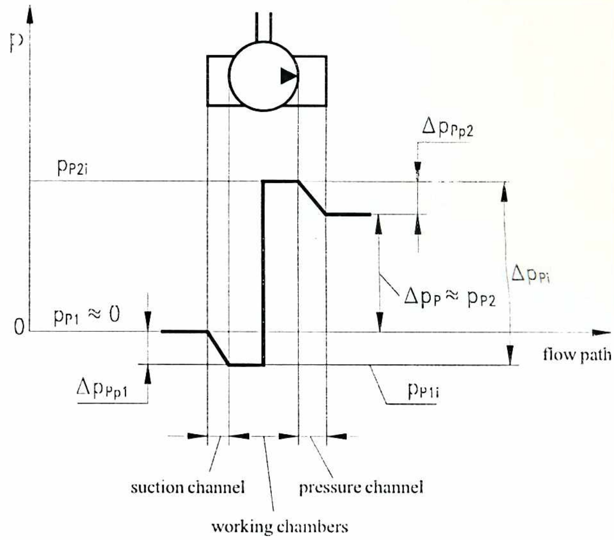

Fig.15. Approximate changes of working liquid pressure along the flow path in the pump

$$\Delta p_{Pp} = k_3 p_n \left(\frac{Q_P}{Q_{Pt}}\right)^2 \tag{47}$$

with :

- p_n - nominal pressure of the hydraulic system in which the pump is used
- Q_P - pump delivery
- Q_{Pt} - theoretical pump delivery

and

$$k_3 = \frac{\Delta p_{Pp|Q_P=Q_{Pt}}}{p_n} \tag{48}$$

representing the pressure losses Δp_{Pp} coefficient in the internal channels of the pump, pressure losses which would exist with the Q_P delivery equal to theoretical Q_{Pt} delivery, in comparison with the nominal pressure p_n. It is proposed to adopt the k_3 value corresponding to the fluid viscosity $\upsilon = 35\ mm^2\,s^{-1}$, recommended by the producers of elements, for which their energy efficiency is most often presented.

For the hydraulic motor (Fig.16) :

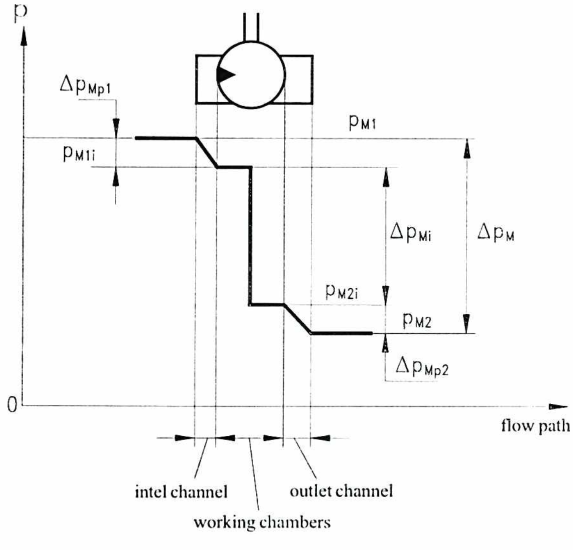

Fig.16. Approximate changes of working liquid pressure along the flow path in the hydraulic motor

$$\Delta p_{Mp} = k_8 p_n \left(\frac{Q_M}{Q_{Pt}} \right)^2 \tag{49}$$

with :

p_n - nominal pressure of the hydraulic system in which the motor is used

Q_M - absorbing flow intensity of the motor

Q_{Pt} - theoretical pump delivery

and

$$k_8 = \frac{\Delta p_{Mp|Q_M=Q_{Pt}}}{p_n} \tag{50}$$

representing the pressure loss Δp_{Mp} coefficient in the internal channels of the hydraulic motor, pressure losses which would exist with the Q_M absorbing flow equal to theoretical Q_{Pt} delivery of the pump, in comparison with the nominal pressure p_n.

It is proposed to adopt the k_8 value corresponding to the fluid viscosity $\upsilon = 35\ mm^2 s^{-1}$.

CONCLUSIONS

- Optimisation of hydraulic system is, among other things, a possibility to predict the energy characteristics of the system in various working conditions, in function of speed and load of the hydraulic motor, viscosity of the working medium, losses in elements, and especially as an effect of the influence of the system's structure itself.
- The implementation and utilisation of the objective, laboratory tested, methods of determining the energy efficiency of these systems, and a look at the efficiency of elements from the point of view of the efficiency of the whole system which they constitute, would explain many misunderstandings in this area, such as the problem of maximum efficiency of certain structures.
- The principal factor decisive of temporary value of energy efficiency of the pump, hydraulic motors and the whole hydrostatic drive is the used structure of hydraulic motor speed control.
- Elaboration of the model of hydraulic system energy efficiency with definite structure, which enables a simulation of energy behaviour of this system, demands an application of a simple mathematical description of losses in displacement machines. Description must show its dependence on system working parameters especially in the zone of nominal parameters. The aim of this description is estimation of the sum of the influences of all phenomena on the mechanical, pressure and volumetric losses and also the global estimation of the size of losses and their proportions.
- The known descriptions of energy losses in pump and hydraulic motor serve mainly for explanation of physical phenomenon during the work of displacement machine. They have simplified forms because it is not possible to consider all factors influencing the losses. Simultaneously, these forms do not allow to describe the global image of dependence of particular kinds of losses on the working system parameters.
- Model of system efficiency, considering the influence of kinematic viscosity υ of working medium, requires an application of the description of the volumetric losses which shows the proportion between the principal elements of these losses – the leakages of laminar character and that of turbulent character. This description demands acceptance of the reference viscosity υ_n.
  It is proposed that the recommended viscosity is the kinematic viscosity $\upsilon_n = 35\ mm^2 s^{-1}$ given by producers with characteristics of their products.
  The proposed description of volumetric losses in pumps and hydraulic motors combines the simplicity of Wilson [14] and Thoma [15] models and Prokofiew [16] models with Schlösser and Hilbrands models [18] and Balawender models [13] which consider the phenomenon of simultaneous occurrence of leakages of laminar and turbulent character. This model is at the same time precise at the reference parameters: p_n, n_{Po}, $\upsilon_n = 35\ mm^2 s^{-1}$.
- For the description of hydraulic system efficiency as a function of the load and speed of the hydraulic motor, it is necessary to separate, in displacement machines, the mechanical losses and pressure losses, which depend differently on the level of pressure and flow intensity of the working medium.
- The proposed descriptions of losses and efficiency of pump and hydraulic motor, with coefficients k_i of energy losses, make possible the generalization of efficiency models of elements and system after introduction of reduced variables.
- Coefficients k_i of energy losses inform about the level and proportion of particular kinds of losses in pump and hydraulic motor at the nominal parameters of the system (especially at the nominal pressure p_n) in which that machine is utilized.
- Elaboration of the tables with the approximate values of coefficients k_i in pumps and hydraulic motors of different types and sizes (also in linear motors – cylinders) could be an important information for the designer of the system during the selection of these machines.

Appraised by Józef Szala, Prof.,D.Sc.,M.E.

NOMENCLATURE

cte	-	stands for constant
k_1	-	coefficient of volumetric losses per one shaft revolution of fixed capacity pump, defined by formula (11)
k_2	-	coefficient of decrease in pump rotational speed
k_3	-	coefficient of pressure losses (flow resistance) in internal pump ducts, at theoretical pump delivery
$k_{4.1}$	-	coefficient of mechanical losses in pump, at $\Delta p_{Pi} = 0$, defined by formula (36)
$k_{4.2}$	-	coefficient of increase of mechanical pump losses, at increase in pressure in pump cylinder, defined by formula (37)
$k_{7.1}$	-	coefficient of mechanical losses in hydraulic motor, at a torque $M_M = 0$, defined by formula (30)
$k_{7.2}$	-	coefficient of increase in mechanical losses in motor, occurring at a torque M_{Mn}, defined by formula (31)
k_8	-	coefficient of pressure losses (flow resistance) in internal ducts of hydraulic motor, at the flow intensity equal to theoretical pump delivery
k_9	-	coefficient of volumetric losses in hydraulic motor, defined by formula (14)
M	-	torque, moment
M_{Mn}	-	hydraulic motor shaft nominal torque resulting from motor pressure decrease equal to nominal pressure $\Delta p_M = p_n$ of a system and from theoretical motor working cubic capacity q_{Mt}
n	-	rotational speed
p	-	relative pressure - overpressure or underpressure
p_n	-	nominal working pressure of hydrostatic transmission (hydraulic system)
Δp	-	pressure increase or decrease, flow resistances
q	-	cubic capacity
q_{Mt}	-	theoretical working cubic capacity of fixed capacity hydraulic motor
q_{Pt}	-	theoretical working cubic capacity of fixed capacity pump
Q	-	flow intensity, delivery, absorbing flow
Q_M	-	hydraulic motor absorbing flow, intensity of flow directed to hydraulic motor
Q_P	-	pump delivery
Q_{Pt}	-	theoretical pump delivery resulting from theoretical pump working cubic capacity q_{Pt} and speed n_{Po}
η	-	energy efficiency
η_{Mm}	-	hydraulic motor mechanical efficiency
η_{Mp}	-	hydraulic motor pressure efficiency
η_{Mv}	-	hydraulic motor volumetric efficiency
η_{Pm}	-	pump mechanical efficiency
η_{Pp}	-	pump pressure efficiency
η_{Pv}	-	pump volumetric efficiency
ϑ	-	temperature
μ	-	dynamic viscosity
υ	-	kinematic viscosity
ρ	-	mass density
ω	-	angular speed

Indices

c	- input		n	- nominal
e	- effective		o	- idle run
f	- leakage		p	- pressure
g	- geometric		P	- pump
i	- indicated		r	- during starting
m	- mechanical		t	- theoretical
M	- hydraulic motor		u	- output

v - volumetric

Notice:
Wherever in this paper formulas of other authors are cited , their original nomenclature is maintained.

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Conference

Transport machines and systems

Since 1992 Faculty of Maritime Technology, Technical University of Szczecin has organized cyclic scientific conferences under the general heading of „Shipbuilding and Ocean Technology". However their main area of interest has been design, operational and testing problems of shipboard machines in correlation with port transport systems and various reloading techniques.

1st Conference held in Szczecin in 1992 and two successive ones held in Międzyzdroje in 1995 and 1997 were named „Development of shipboard machines and reloading techniques". In their scope more and more attention was devoted to problems of quality, reliability and safety. The same tendency was also observed in the scope of two successive conferences held in Międzyzdroje, that under heading „Port technology and shipboard equipment" organized in 1998 and that under heading „Transport machines and systems" organized in 2000.

The profile of 5th Conference (held on 6÷7 June this year) demonstrated some common area of interest of scientists and engineers from many scientfic centres, research institutes and industrial enterprises, not always connected with maritime technology and economy. The conference proceedings confirmed that transport machines and systems are a broad research area even if limited only to maritime technology problems.

Topics of the presented papers were diversified: beginning from general problems, through quality, reliability and safety, and ending at detail ones dealing with specific design solutions.

During the Conference 40 papers were presented: 6 of them at the plenary session and the remaining during 8 topic sessions.

The following papers were presented during the plenary session :

- *Searching for methods for machine quality evaluation* by Prof. Stanisław Oziemski
- *Navigation risk analysis as an assessment method of waterway modernization effectiveness* – by Prof. Stanisław Gucma
- *Simulation of influence of operational parameters on reliability of ocean technology units* – by Prof. Mieczysław Hann
- *Classification of cranes complying with EU standards and guidelines* – by Prof. Sylwester Markusik
- *Analysis of safety indices by means of interval method* by Prof. Jerzy Lewitowicz
- *Loads of containers due to ship motion in waves* – by Prof. Tadeusz Szelangiewicz

Papers read and discussed during topic sessions dealt with the following areas :

- Ship propulsion and power systems
- Hydrostatic drive power engineering
- Ship vibroacoustics
- Novel ship design solutions
- Exploitation loads and operation of loading equipment
- Failure frequency and degradation processes of work machines, floating and ocean technology objects
- Work machine ergonomics
- Modelling and analysis of exploitation processes
- Simulation and service investigations of transport objects and facilities
- Shipping models and methods
- Safety assessment and its improvement methods
- Management of maintenance processes
- Application of fuzzy sets to transport problems.

The papers were prepared by authors from 14 universities, research centres and institutes. Authors from Technical University of Szczecin (8 papers), Technical University of Gdańsk (7 papers), Technical University of Łódź and Maritime University of Szczecin (6 papers each) contributed most as regards the number of papers.

In the Conference 100 persons took part from 28 universities, research centres and institutes as well as enterprises connected with shipbuilding, shipping and maritime economy. Apart from establishing mutual contacts and exchanging experience the participants had an opportunity to take part in a voyage to Sassnitz.