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

Part I. Model of volumetric 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 volumetric losses in the displacement machines with the rotational movement is proposed.

INTRODUCTION

With the progress achieved in recent years in the development of hydraulic systems it is becoming necessary to develop methods for precise energy efficiency calculation of such systems. It is difficult to imagine that more and more, better and better machines and control elements could be used without the possibility of mathematical tool at our disposal to enable an accurate analysis and assessment of behaviour of the system in which such machines and control elements have been applied [1÷5].

The possibility of development of methods for calculation of the actual total efficiency of a hydraulic system as a function of many parameters determining the efficiency may become a tool for total assessment of the quality of the system. The availability of such an assessment is also essential due to wide application of hydrostatic control and regulation systems and the trend to use increasingly powerful hydrostatic drives in the age of more expensive energy.

The behaviour of elements in hydraulic systems with different structures is also not researched area of fluid power technology. The proportion of different kinds of losses taking place within system elements is often not fully realised. For example, researchers concentrate on the problem of leak-tightness of elements (their volumetric efficiency) without full awareness of the losses in pressure, and especially mechanical losses that come as a price to be paid for the leak-tightness at various levels of working pressure.

A typical example of too great simplification of the problem of losses in a displacement machine is the use of the concept of η_{mh} i.e. mechanical-hydraulic efficiency of a hydraulic pump or motor [6÷10]. In fact this is the mathematical product of the mechanical η_m and pressure η_p efficiency of the hydraulic pump or motor, that represent the influence of mechanical and pressure losses due to the increase of torque M_P on the pump shaft or pressure differential Δp_M in a hydraulic motor. These losses depend, however, upon different factors. Mechanical losses result, among other things, from the load, while pressure losses do not depend on load. Mechanical losses do not practically depend on flow in the machine, while pressure losses depend exactly on the flow rate. Therefore, mechanical efficiency η_m and pressure efficiency η_p of displacement machine should be considered independently of each other so that their dependence on working conditions can be assessed, and their efficiencies possibly increased. In [11,12], the mechanical and pressure losses in the pumps and hydraulic motors are separated.

Optimisation of hydraulic systems offers, among other things, the possibility to predict energy characteristics of the system in various working conditions, in function of the 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 structure itself.

The implementation and utilisation of 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 whole system, would explain many misunderstandings in this area of knowledge.

CONNECTION OF THE DESCRIPTION OF PUMP AND HYDRAULIC MOTOR WITH THE MODEL OF ENERGY EFFICIENCY OF THE HYDRAULIC SYSTEM

Utilisation in the process of the hydraulic system design of the method of calculating its energy efficiency [2÷5] must enable the assessment of influence on efficiency of the following factors :

- system structure
- energy losses in the system elements
- decrease of rotational speed of motor which drives the pump
- characteristics of control elements
- load and speed of hydraulic motor
- viscosity of hydraulic oil which change during the system exploitation.

In order to achieve wide and easy application of this method, it is necessary to elaborate :

1. description of the mathematical model of the energy efficiency of the systems with the determinate structure, which has the principal influence on the efficiency
2. description, on the base of models, of computer programmes which enable an easy analysis of hydraulic system efficiency in function of various decisive parameters
3. determination of the approximate values of the coefficients of the energy losses in pumps, hydraulic rotational motors and hydraulic cylinders. These coefficients must be defined unequivocally, determined exactly and independently for a concrete displacement machine. A considerable advantage would be the possession and possibility of comparison of its indicatory values for a given type and size of machine, at the nominal pressure of the system in which it was applied.

The designer could also have at his disposal the possibility of evaluation of coefficients of losses on the ground of the information delivered by the producers of pumps or motors which concern the pressure losses (flow resistance) in internal pump or motor ducts at the specified viscosity of working medium (hydraulic oil).

The realization of intentions presented in point 1÷3 demands the look at the description of the energy losses and efficiency of displacement machines, with take into account :

- simplicity, which is the factor decisive in the further application, with, at the same time, precision of description of global energy efficiency, specially in the zone of nominal parameters of system work and oil viscosity recommended by producers during exploitation
- description of volumetric losses giving the possibility of evaluation of influence of the oil viscosity, dynamic μ or kinematic υ (with supposition of the mass density ρ = cte), on the whole of volumetric losses in displacement machine; this description must take in consideration an existence of two principal components of volumetric losses : the leakage of a laminar character–proportional to μ^{-1} and leakage of a turbulent character which practically do not depend on viscosity (proportional to μ^{0}).
Description of the volumetric losses can be only based on the formula, in which no dimensional coefficients of losses are used. (So, the numerical formula, related to chosen system of units [11] can not be used, even if it enables the direct determination of character of flow – on the basis of real value of the exponent α in relation $Q = C \cdot \Delta p^{\alpha}$. When $\alpha \neq 0.5$ or $\alpha \neq 1$, this relation cannot be used.) Therefore the forms with $Q \sim \Delta p$ or with $Q \sim \Delta p^{0.5}$ must be chosen even in the real situation, when the flow of volumetric losses is the sum of laminar flow and (not fully developed) turbulent flow
- separation of pressure losses and mechanical losses. To be sure, both kinds of losses increase (in the case of pump) the shaft torque or (in the case of rotational motor) the pressure drop, but they have a different character :
  - the mechanical losses, between the shaft and the working chamber are, first of all, the friction losses being principally function of work pressure. They practically do not depend on rotation speed of machine (above same critical speed) and on the viscosity of oil
  - the pressure losses in internal duct and distributor (if it exists) of machine, which are not the function of work pressure, but depend principally on volumetric flow (so – on the rotational speed) and, very rarely, on the oil viscosity.

As mentioned above, the principal influence on the hydraulic system efficiency has its structure. Its influence is generally considered with the assumption of ideal pump and motor and supposition that the real energy losses in pump and motor cause the proportional decrease of global efficiency of the system.

The image of reciprocal influence of losses in element of hydraulic system appears, however, more complicated. Temporary value of the pump energy efficiency is, apart from other factors, first of all the result of the used structure of hydraulic motor control.

The parameters connecting the pump, hydraulic motor and other elements with description of the system are :

→ theoretical working cubic capacity q_{Pt} of fixed capacity pump
→ theoretical working cubic capacity q_{Mt} of fixed capacity hydraulic motor
→ shaft rotational speed n_{Po} of not charged pump
→ nominal working pressure p_n of hydrostatic transmission (hydraulic system), which can be different from the nominal pressures of the pump and hydraulic motor.

These parameters enable the description of efficiency of the pump, hydraulic motor and other elements of the system as a function of the torque M_M and speed n_M of the motor, that is as a function of the parameters with which we present the efficiency of all hydrostatic drive.

Perception of the influence of the kinematic viscosity υ of working medium on the energy behaviour of the pump and hydraulic motor (and system) demands then the acceptance of the reference viscosity; let us assume that it is the recommended kinematic viscosity $\upsilon_n = 35$ mm²s⁻¹ (from a point of view of minimum global energy losses), proposed by producers with the characteristics of their products.

Parameters : q_{Pt}, q_{Mt}, n_{Po}, p_n and $\upsilon_n = 35$ mm²s⁻¹ could be also the reference points in the description of the three kinds of energy losses and efficiency of the pump and hydraulic motor.

It is not possible to take in the consideration, in description of particular kind of losses, the influence of every decisive factor. Therefore these descriptions have always the approximate (simplified) forms.

EXAMPLE OF THE INVESTIGATED DISPLACEMENT MACHINE

The widest energy analysis of the displacement rotational machines was made by A. Balawender [13]. Hence during the present consideration reference was made, first of all, to [13] , especially to the results of examination of hydraulic motors of SWSB type (Fig.1).

The hydraulic motor of the SWSB type is the slow-speed rotary piston engine with high output torque.

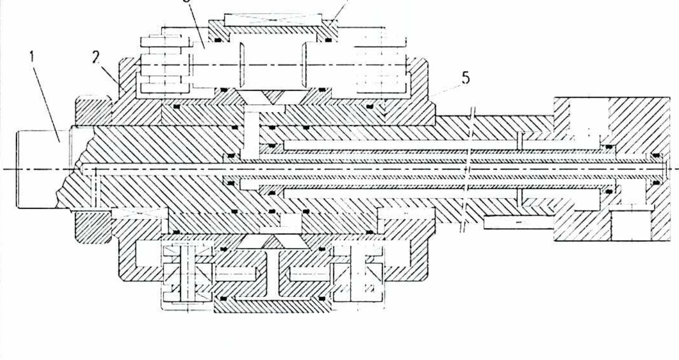

Fig. 1.
Section of the slow-speed, high-output torque rotary SWSB type motor with axial pistons in the rotational block-cylinders with annular fixed cams and cylindrical distributor

1 – axle, 2 – fixed annular cam with lobes
3 – piston plunger with roller
4 – rotational block-cylinders (rotor) with key
5 – tube (sleeve) of distributor

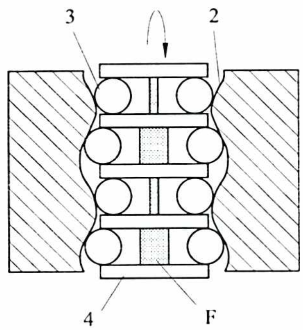

Fig.2. Principle of operation of SWSB motor

The principle of operation of such motor is presented in Fig.2. It is a slow-speed motor with axial piston-plungers with roller in rotational block-cylinders and two fixed annular cams. One obtains the great cubic capacity as a result of the great number of strokes of piston-plungers. The principal elements are : 2 – fixed annular cams, 3 – piston-plungers with roller supported on cam surface, 4 – rotational block-cylinders, F – fluid under pressure.

The SWSB-40 and SWSB-63 motors, which are the object of loss study of the next paragraphs, have the following technical parameters :

		SWSB-40	SWSB-63
theoretical working cubic capacity	q_{Mt} [cm³] :	410	630
nominal working pressure	p_{Mn} [MPa] :	6.18	6.18
maximum working pressure	p_{Mmax} [MPa] :	10	10
nominal rotational speed	n_{Mn} [min⁻¹] :	160	160
maximum rotational speed	n_{Mmax} [min⁻¹] :	250	250
input power at p_{Mmax} and n_{Mn}	P_{Mc} [kW] :	10.9	16.8

Models are also presented in this study of pumps and displacement rotational motors with propositions of the mathematical forms appropriate to the model of the global energy efficiency of hydraulic system, in which simplicity is an essential factor.

MODEL OF VOLUMETRIC LOSSES

In mathematical models of hydraulic displacement rotational machines, the authors neglect the influence of the compressibility of liquid and the change of cubic capacity on the volumetric losses Q_f (Q_{Pf} or Q_{Mf}).

Wilson [14] and Thoma [15] take into account only flow intensity of leakage of the laminar character, introducing for its presentation the function (Fig.3) :

$$Q_f = \frac{C_{vl}}{\mu}\frac{q_t}{2\Pi}\Delta p \tag{1}$$

with

- C_{vl} - non-dimensional coefficient received by experiment on real machine
- μ - dynamic viscosity of the liquid [kg m⁻¹ s⁻¹]
- q_t - theoretical cubic capacity
- Δp - decrease or increase of pressure in the machine.

Prokofiev [16] compares the sum of volumetric losses in the clearances of a piston machine with the losses in an equivalent clearance h.

He presents the laminar flow in the equivalent clearance by equation :

$$Q_f = k_v h^3 \frac{\Delta p}{\mu} \tag{2}$$

where :

- k_v - non-dimensional coefficient, a constant value for a series of hydraulic displacement machines of the same design.

The model proposed by Prokofiev is nearest to the classical relation, normally used for evaluation of losses in the laminar flow in the clearance between, for instance, two concentric cylindrical elements (Fig.4) :

$$Q_f = \frac{\Pi d}{12l} h^3 \frac{\Delta p}{\mu} \tag{3}$$

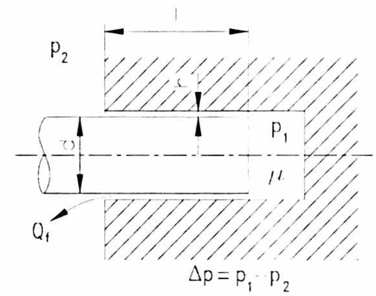

Fig.4. Parameters in the relation (3) expressing the losses of flow between two concentric cylindrical elements

Schlösser and Hilbrands [18] using the results of the experimental studies performed on different types of displacement pump, present the volumetric losses (treating them only as leakage) as a sum of two components: Q_{fl} – leakage of a laminar flow character and Q_{ft} – leakage of a turbulent flow character :

$$Q_f = Q_{fl} + Q_{ft} \tag{4}$$

The Q_{ft} component leads, for instance, for the gear pumps, to a value exceeding 50% of the total leakage.

The volumetric losses Q_{ft} of the turbulent flow are presented in the form:

$$Q_{ft} = C_{vt}\sqrt{\frac{2\Delta p}{\rho}}\sqrt[3]{\left(\frac{q_t}{2\Pi}\right)^2} \tag{5}$$

where :

- C_{vt}- non-dimenssional coefficient of proportionality.

The presence of the term Q_{ft} is justifiable by existence of clearance :

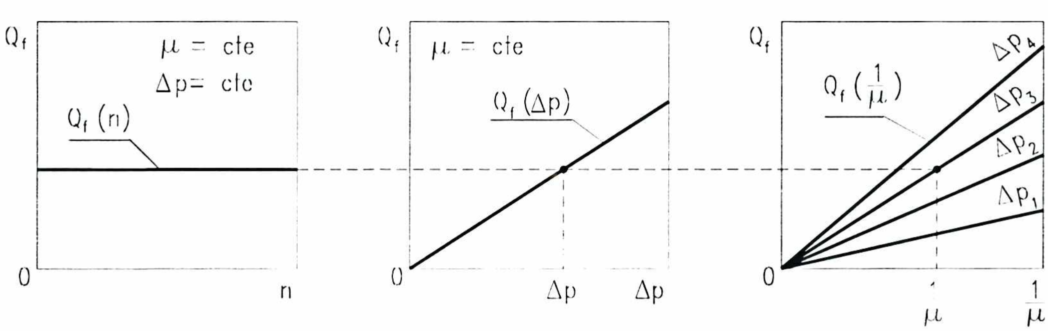

Fig.3. Model of volumetric losses according to Wilson's formula [14]

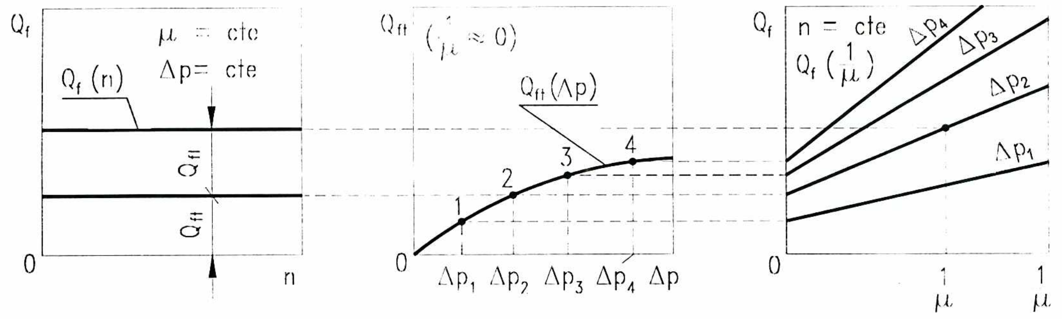

Fig.5. Model of volumetric losses according to Schlösser-Hilbrand's formula [18]

- formed by elements in linear contact (example : contact between teeth of gear machine)
- with quick section change (increase or decrease)
- with periodical perpendicular partition vibration.

The term Q_{ft} of losses of turbulent character of flow was founded on the basis of characteristics $Q_f = f\left(\frac{1}{\mu}\right)$ of volumetric global losses in function of the liquid dynamic viscosity μ, as a limit (Fig.5).

$$Q_{ft}(\Delta p) = \lim_{\frac{1}{\mu}\to 0} Q_f\left(\frac{1}{\mu}, \Delta p, n\right)\Bigg|_{n = cte}^{\Delta p = cte} \tag{6}$$

Opinions of researchers concerning the Schlösser and Hilbrands model are divided [15], [8], [13].

In [13] Balawender underlines that the volumetric loss descriptions in the displacement rotational machines do not show :

- the influence of the rotational speed of the rotor and the influence of the pressure at discharge conduit of the hydraulic motor
- the correlation with the factors, which decide of the amount of the volumetric losses : fluid compressibility, part deformation, modification of the clearance height, change of working medium viscosity during the passage through clearance, as a result of pressure and temperature influence.

In consequence, the coefficients C_{vl} and C_{vt} of laminar and turbulent character of flow change in function of machine working parameters and in a manner difficult for interpretation.

Considering the characteristics of volumetric losses in low-speed hydraulic motors (example shown in Fig.6), Balawender proposes the model of losses which is based on the losses for the speed n = 0 and which shows also the influence of pressure p_2 in motor outlet conduit.

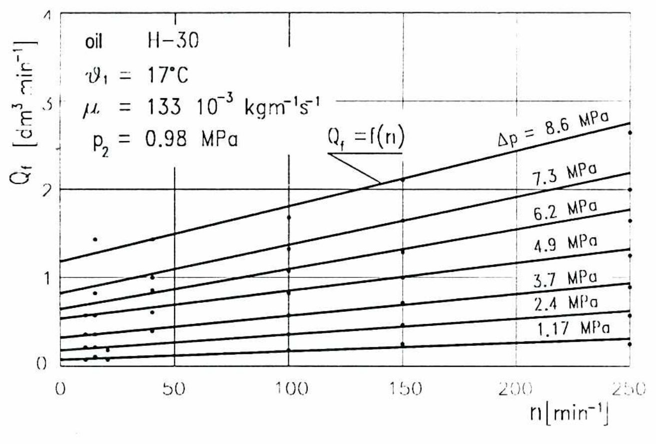

Fig.6. Characteristics Q_f of volumetric losses flow in low-speed SWSB-40 hydraulic motor in function of rotational speed n

So, the volumetric losses present according to the equation (Fig.7a) :

$$Q_f = \Delta q\,(\Delta p)\,n + Q_{fo}\,(\Delta p, p_2) \tag{7}$$

The first term of equation (7) shows the losses in function of pressure decrease Δp in motor and of rotation motor speed n.

The second one – Q_{fo} (Fig.7b) represents the volumetric losses in motor at the speed n = 0, – function of the motor pressure decrease Δp and pressure p_2 in motor outlet conduit.

Experience shows the existence of a component of volumetric losses of turbulent character even at zero speed.

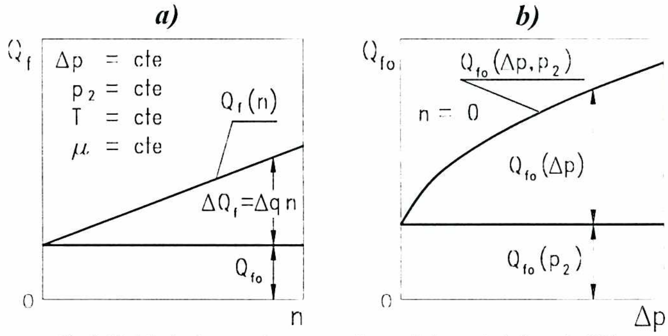

Fig.7. Model of volumetric losses according to Balawender's formula [13]

As a result of his analysis, Balawender presents [13] an extended model of the volumetric losses in the low-speed hydraulic motor :

$$Q_f = C_S \frac{q_t}{2\Pi} \Delta p\, \omega + C_{ol} \frac{q_t}{2\Pi} \frac{\Delta p}{\mu_0} + C_{0zl} \frac{q_t}{2\Pi} \frac{\Delta p}{\mu_0} + C_{0t} \sqrt{\frac{2\Delta p}{\rho}} \sqrt[3]{\left(\frac{q_t}{2\Pi}\right)^2} \tag{8}$$

Without coming into details of this model, it may be only indicated that the first term shows the influence of the speed and of the pressure drop in the motor, the second and the third–the influence of viscosity, pressure drop and the pressure in the motor outlet on the losses of laminar character. The fourth shows the influence of the mass density of liquid and the pressure drop on the losses of turbulent character.

Such a model is very complicated for modelling of the volumetric losses in the rotational machines, particularly when they constitute only the element of hydrostatic transmission, whose energy efficiency is to be evaluated. Practical requirement is that a model should be at the same time simple and as close to the reality as possible.

Within this study, intended first of all for evaluation of the global energy efficiency of a hydraulic system, the author proposes the following formula for modelling the volumetric losses of rotational displacement machines :

$$Q_f = k_{v35} \frac{q_t}{\rho v_n} \Delta p_i \left(\frac{v_n}{v}\right)^{0.5} \tag{9}$$

where :

k_{v35} - non-dimensional constant of volumetric losses, obtained experimentally on the real machine for the kinematic viscosity $\upsilon_n = 35 mm^2s^{-1}$

q_t - theoretical capacity of the displacement machine (q_{Pt} or q_{Mt})

ρ - mass density of liquid

Δp_i - indicated increase Δp_{Pi} (pump) or decrease Δp_{Mi} (motor) of pressure in the working chamber of the machine

$\left(\frac{\upsilon_n}{\upsilon}\right)^{0.5}$ - approximate form of the influence of the liquid viscosity on the combined volumetric losses. The value 0.5 of the exponent takes into account first of all two components : leakage of laminar character proportional to υ^{-1} and leakage of turbulent character independent of the viscosity (proportional to υ^0)

υ_n - recommended kinematic viscosity of the liquid, equal to $35 mm^2s^{-1}$, treated as reference level, for which producers present most often the energy efficiency of manufactured elements

υ - liquid viscosity, for which Q_f is calculated.

The k_{v35} constant in the (9) formula, which will be k_{Pv35} for pump and k_{Mv35} for hydraulic motor, is obtained for :

★ the kinematic viscosity $\upsilon_n = 35 mm^2s^{-1}$ of the liquid
★ the Δp_i value equal to the nominal pressure p_n of the hydraulic system
★ the working speed n of the machine in the hydraulic system, that is :
  - in the case of a pump, the speed n_p of its driving motor
  - in the case of a hydraulic motor, the speed n_M resulting from the nominal delivery Q_{Pn} of the pump.

The conditions for which the k_{v35} constant is determined allow to evaluate exactly the level of volumetric losses at the nominal work parameters of the system (load and speed of the motor).

The (9) formula combines the simplicity of the Wilson and Thoma model (1) and the Prokofiev model (2) with the reality of the machines shown by Schlösser model (4) and Balawender model (8).

The use of a formula $\left(\frac{\upsilon_n}{\upsilon}\right)^{0.5}$ with the exponent of 0.5 showing the influence of viscosity, allows to approximate to the real volumetric losses corresponding to a given viscosity υ of the liquid.

To increase the precision of a mathematical model (9) of volumetric losses Q_f in the case of a given displacement machine, the exponent of 0.5 in the $\left(\frac{\upsilon_n}{\upsilon}\right)^{0.5}$ formula may be replaced by another value from the (0,1) range [20], see also [21].

Let us assume that the volumetric efficiency η_{Pv} of the pump used in the hydraulic system, at a given temperature and oil viscosity, decreases with the indicated pressure increase Δp_{Pi} in the working chamber of the pump, in accordance with

$$\eta_{Pv} = 1 - \frac{k_1}{p_n}\Delta p_{Pi} \tag{10}$$

where k_1 is coefficient of volumetric losses during one pump shaft revolution, with $\Delta p_{Pi} = p_n$, related to the theoretical working capacity q_{Pt} of the pump :

$$k_1 = \frac{Q_{Pf|\Delta p_{Pi}=p_n}}{n_{P|\Delta p_{Pi}=p_n}} \frac{1}{q_{Pt}} \tag{11}$$

The k_1 coefficient and also the other k coefficients describing the transmission elements are non-dimensional values enabling further generalisation of loss and energy efficiency models after introducing reduced variables.

The following relation is valid between the k_1 coefficient and the volumetric loss constant k_{Pv35} in the pump :

$$k_1 = k_{Pv35} \frac{p_n}{\rho \upsilon_n n_{p|\Delta p_{Pi}=p_n}} \left(\frac{\upsilon_n}{\upsilon}\right)^{0.5} \tag{12}$$

So the k_1 coefficient (and same other k coefficients describing the transmission elements) is a function of oil viscosity υ.

Let us then assume that the intensity of volumetric losses Q_{Mf} in the motor used in the hydraulic system is proportional to the indicated pressure Δp_{Mi} decrease and independent of the rotational motor speed n_M :

$$Q_{Mf} = k_9 Q_{Pt} \frac{\Delta p_{Mi}}{p_n} \tag{13}$$

The k_9 coefficient is a ratio of the volumetric losses in the hydraulic motor, which would take place at the pressure decrease $\Delta p_{Mi} = p_n$, to the theoretical delivery Q_{Pt} of the pump :

$$k_9 = \frac{Q_{Mf|\Delta p_{Mi}=p_n}}{Q_{Pt}} \tag{14}$$

The following relation is valid between the k_9 coefficient and the volumetric loss constant k_{Mv35} in the motor :

$$k_9 = k_{Mv35} \frac{q_{Mt} p_n}{\rho \upsilon_n Q_{Pt}} \left(\frac{\upsilon_n}{\upsilon}\right)^{0.5} \tag{15}$$

The volumetric efficiency η_{Mv} of the motor is expressed by the equation :

$$\eta_{Mv} = 1 - k_9 \frac{Q_{Pt}}{Q_M} \frac{\Delta p_{Mi}}{p_n} \tag{16}$$

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

To be continued

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
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
f - leakage
g - geometric
i - indicated
m - mechanical
M - hydraulic motor
n - nominal
o - idle run
p - pressure
P - pump
t - theoretical
u - output
v - volumetric

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Conference

Safety at sea and marine environment protection

On 8÷10 June this year Kołobrzeg was the venue of 4th Symposium devoted to problems of safety at sea and activities for marine environment protection.

A honourary patron of the Conference was Prof. Andrzej S. Grzelakowski, Undersecretary of Ministry of Transport and Maritime Economy. Polish Parliament activities in this area were presented by Mrs. Dorota Arciszewska – Mielewczyk , Member of Parliament.

On safety at sea

11 papers were presented dealing with the following topics :

- realization of STCW –75/95 Convention provisions
- passenger safety and dangerous goods transport problems
- human factor in occupational safety at sea
- ship traffic supervision and control systems
- collision hazard situation modelling.

On search and rescue at sea

9 papers were read dealing with :

- operational effectiveness of radar reflectors
- modern rescue equipment and means
- exploitation of SAR-1500 ships and transformations of Polish Maritime Salvage Company
- evacuation system of persons within the port of Kołobrzeg
- rescue problems during exploitation of sea-bed oil resources.

On marine environment protection

11 papers were presented on the following topics :

- techniques of fighting against oil pollution and of chemical pollution detection
- dangers to marine environment due to transport of chemicals, water reservoirs and facilities and operation of naval ships
- assessment methods of factors influencing marine environment
- sea level prediction
- risk management in shipping industry.

SZKWAL - SAR rescue vessel of Polish Maritime Salvage Company