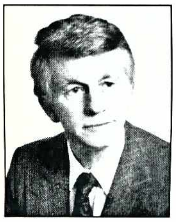

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Energy-saving structures of hydrostatic drives for ship deck machines

SUMMARY

This paper presents a comparison of the energy behaviour of two widespread structures (of Load Sensing and Adaptive Secondary Control) of the central feeding systems of hydraulic motors in the case of supplying oil to one motor. The research was made possible due to elaboration of the computer simulation method of energy efficiency of hydrostatic transmissions.

Both system solutions are shortly described together with presentation of equations of the total efficiency η of the system, graphs of changes of the speed coefficient $\overline{\omega}_M$ and charge coefficient $\overline{M}_M$ of the motor.

Finally, results and analyses of computer simulations of efficiency of the systems are also described.

INTRODUCTION

Development of the hydrostatic drive for ship deck machines and machines used in other industries is connected with the search for energy-efficient solutions, e.g. central systems with parallel situated motors.

The possibility of the simultaneous supply of many receivers by one delivery pump is connected with the assumption that a central system should more effectively economize the energy wasted in the assembly of the motor speed throttle control.

One solution is a central system with throttle control, supplied by a variable delivery pump with a variable pressure regulator [1,2], called *Load Sensing* system. In the system, energy loss minimization in the throttle control assembly is effected by decreasing the pressure drop within the assembly as a result of adapting the pump pressure to the pressure demanded by the currently most charged receiver.

The system with the *Adaptive Secondary Control* [3,4] applied in the industry by the firm Rexroth as an energy-saving system is still an interesting idea from the cognitive point of view. The analysed energy savings occurred as a consequence of applying the hydraulic motor control structure. The structure eliminates the throttle control unit being the source of losses, and at the same time introduces the variable capacity motor and work at the maximum constant pressure.

The above-mentioned solutions were possible to be compared due to elaboration and improvement of the computer simulation methods of energy efficiency of hydraulic systems, presented in [5,6].

LOAD SENSING SYSTEM WORKING WITH ONE MOTOR FED THROUGH A TWO–WAY FLOW CONTROLLER

This system is equivalent to the central system considered in the case of one-motor feeding, shown in Fig.1.

In the system in question the pressure p_{P2} of a supply pump is continuously adapted to the working pressure demanded by the most charged motor. The pressure p_{P2} is stabilized at a little higher level than the pressure p_2 prevalent in the central pipe of the control. Non-return valves act as pressure selectors with the effect that the temporary pressure p_2 required by the most charged motor, occurs in the pipe of the control.

The system shown in Fig.1 works with one-motor feeding. The throttle control unit serves as a two-way flow controller. Characteristics of the system elements are presented in Tab.1.

The pressure difference $(p_{P2} - p_2)$ defined by the force R of the pump spring, should guarantee the minimum pressure drop Δp_{EEmin} still corresponding with the proper work of the flow controller. This is the case of the largest pressure drop Δp_{C1max} in the supply conduit of the pump. Therefore the state in which the condition $\Delta p_{C1} < \Delta p_{C1max}$ is satisfied demands an increase of the pressure drop Δp_{EE} in the throttle assembly (flow controller) :

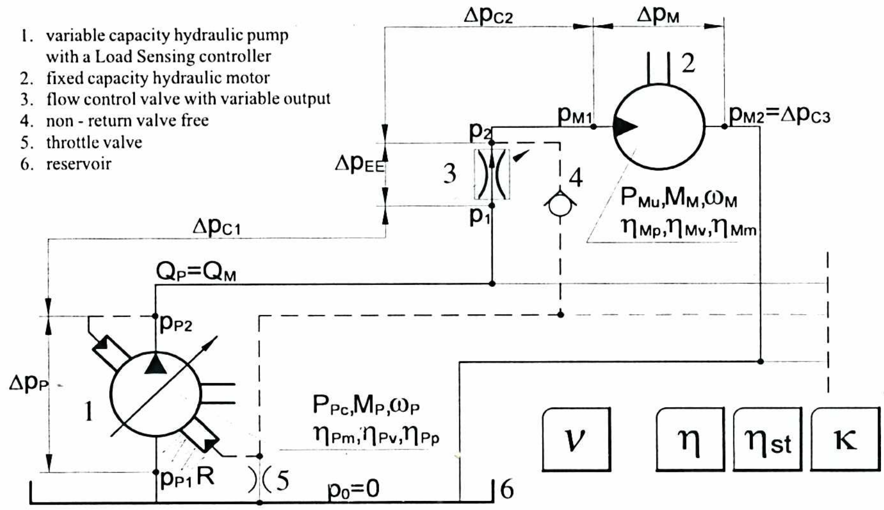
$$\Delta p_{C1} < \Delta p_{C1\max} \Rightarrow \Delta p_{EE} > \Delta p_{EE\min} \qquad (1)$$

In this case the flow in the central conduit is lower than the maximum flow or the oil viscosity is lower than the value ν assumed for the calculation of Δp_{C1max}.

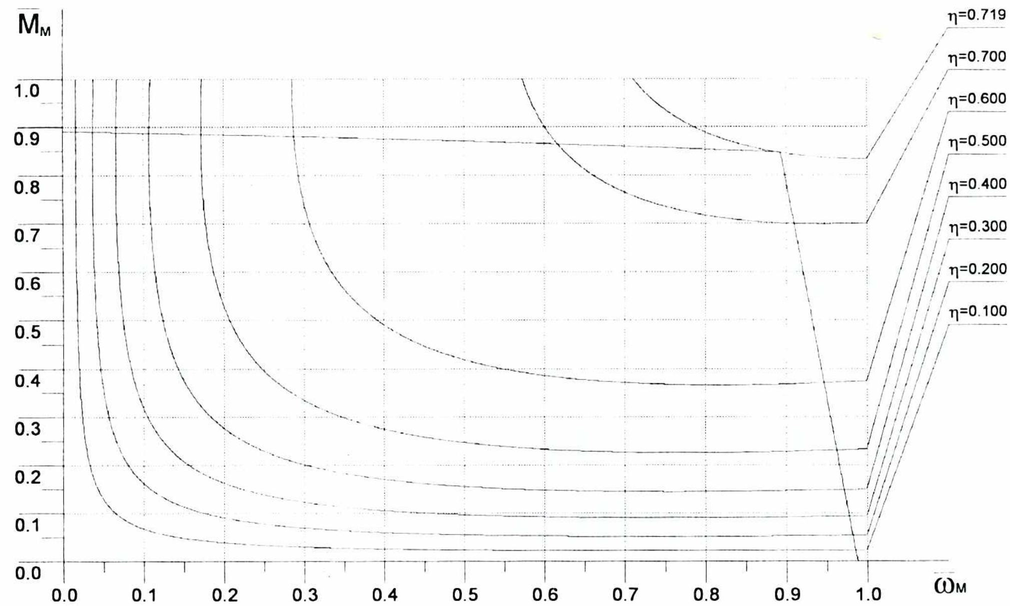
The value of the minimum pressure drop Δp_{EEmin} in the throttle assembly, which still guarantees the correct functioning of the flow regulator, is proportional to the nominal pressure p_n of the system :

$$\Delta p_{EE\min} = k_{10}\, p_n \qquad (2)$$

Fig.1. Central system with a variable delivery pump and Load Sensing controller



a) *Schematic diagram of the system*



b) *Relationship between the motor charge coefficient $\overline{M}_M$, its speed coefficient $\overline{\omega}_M$ and total efficiency η of the system*

Tab.1. Characteristics of the system elements

Pump	Hydraulic motor	Conduits
$k_1 = 0.04$ $k_2 = 0.03$ $k_3 = 0.02$ $k_{4.1} = 0.05$ $k_{4.2} = 0.02$	$k_{7.1} = 0.05$ $k_{7.2} = 0.02$ $k_8 = 0.02$ $k_9 = 0.04$	$k_5 = 0.01$ $k_{6.1} = 0.01$ $k_{6.2} = 0.02$
Flow control valve $k_{10} = 0.03$		**Viscosity** $\nu = 35$

It may be assumed that the value of the maximum pressure drop Δp_{C1max} in the conduit between the pump and the motor flow controller can be expressed as follows :

$$\Delta p_{C1\max} = k_5 p_n \tag{3}$$

The value of the pressure p_{P2} in the supply pipe of the pump should be regulated to the following level :

$$p_{P2} = p_2 + (k_5 + k_{10}) p_n \tag{4}$$

The global efficiency η of the system is expressed by the following equations in compliance with [7,8,9] :

➲ **when $k_2 = 0$:**

$$\eta = \frac{P_{Mu}}{P_{Pc}} = \frac{\{\overline{Q}_M - k_9[k_{7.1} + (1+k_{7.2})\overline{M}_M]\}\overline{M}_M}{k_{4.1} + [(1+k_{4.2})\overline{p}_{P2} + k_3\overline{Q}_M^2](\overline{Q}_M + k_1\overline{p}_{P2})} \tag{5}$$

➲ **when $k_2 \neq 0$:**

$$\eta = \frac{P_{Mu}}{P_{Pc}} = \frac{(X - k_1 \overline{p}_{P2})Y}{\{k_{4.1} + [(1+k_{4.2})\overline{p}_{P2} + k_3\overline{Q}_M^2]X\}\overline{Q}_M} \quad (6)$$

in which :

$$X = \frac{1}{2k_2}\frac{1}{\overline{p}_{P2}} - \left[\left(\frac{1}{2k_2}\frac{1}{\overline{p}_{P2}}\right)^2 - \frac{k_1}{k_2} - \frac{1}{k_2}\frac{1}{\overline{p}_{P2}}\overline{Q}_M\right]^{0.5}$$

$$Y = \{\overline{Q}_M - k_9[k_{7.1} + (1+k_{7.2})\overline{M}_M]\}\overline{M}_M$$

where the relative value of the pump supplying pressure $\overline{p}_{P2}$ is determined by the relationship :

$$\overline{p}_{P2} = \frac{p_{P2}}{p_n} = k_{7.1} + (1+k_{7.2})\overline{M}_M + k_5 + k_{10} + k_6\overline{Q}_M + k_8\overline{Q}_M^2 \quad (7)$$

In order to present η as a function of the coefficients $\overline{\omega}_M$ and $\overline{M}_M$, the coefficient $\overline{Q}_M$ should be replaced by the following expression :

$$\overline{Q}_M = \overline{\omega}_M + k_9[k_{7.1} + (1+k_{7.2})\overline{M}_M] \quad (8)$$

In the case of supplying one motor the upper limit $\overline{\omega}_{M\max}$ (of the range : $0 \leq \overline{\omega}_M \leq \overline{\omega}_{M\max}$) corresponds with the maximum flow $Q_{M\max}$ equal to the pump delivery and is calculated as follows :

$$Q_{P\max} = Q_{Pt}\left[1 - \frac{p_{P2}}{p_n}(k_1 + k_2)\right] =$$

$$= Q_{Pt}\{1 - (k_1+k_2)[k_{7.1} + (1+k_{7.2})\overline{M}_M + \quad (9)$$

$$+ k_5 + k_{10} + k_6\overline{Q}_{M\max} + k_8\overline{Q}_{M\max}^2]\}$$

The flow coefficient $\overline{Q}_{M\max}$ is expressed in this case as a function of $\overline{M}_M$:

$$\overline{Q}_{M\max} = 1 - (k_1+k_2)[k_{7.1} + (1+k_{7.2})\overline{M}_M + k_5 + k_{10} + k_6\overline{Q}_{M\max} + k_8\overline{Q}_{M\max}^2] \quad (10)$$

The motor speed coefficient $\overline{\omega}_{M\max}$ is the function of $\overline{Q}_{M\max}$ in accordance with the following relationship :

$$\overline{\omega}_{M\max} = \overline{Q}_{M\max} - k_9[k_{7.1} + (1+k_{7.2})\overline{M}_M] \quad (11)$$

When $p_{P2} = p_n$, i.e. $\overline{p}_{P2} = 1$, the upper limit $\overline{M}_{M\max}$ (of the range : $0 \leq \overline{M}_M \leq \overline{M}_{M\max}$) can be described by the equation :

$$\overline{M}_{M\max} = \frac{1 - (k_{7.1} + k_5 + k_{10} + k_6\overline{Q}_M + k_8\overline{Q}_M^2)}{1 + k_{7.2}} \quad (12)$$

On substitution of the flow coefficient $\overline{Q}_M$ in (12) by the formula :

$$\overline{Q}_M = \overline{\omega}_M + k_9[k_{7.1} + (1+k_{7.2})\overline{M}_{M\max}] \quad (13)$$

one can find $\overline{M}_{M\max}$ as a function of the motor speed coefficient $\overline{\omega}_M$.

SYSTEM WITH ADAPTIVE SECONDARY CONTROL IN ONE-MOTOR FEEDING

The system of central feeding the parallel working receivers, shown in Fig.2, is equipped with the variable delivery pump having a pressure regulator and accumulator which makes it possible to accumulate energy while using the hydraulic motor as a pump.

The system comprises motors of the geometric capacity q_{Mgv} which is automatically adapted to the pressure existing in the supplying conduit as well as to the actual driving moment M_M. The speed n_M of the motor and its rotation sense are controlled by a self-contained assembly.

In result, in the central supplying conduit as well as in the conduit branches directed to particular motors, the constant maximum working pressure will be observed equal to the nominal pressure in the supplying conduit of the pump.

Description of the total efficiency η of the system takes the following forms (acc. to [10]) :

➲ **where $k_2 = 0$:**

$$\eta = \frac{[1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2]}{k_{4.1} + (1 + k_{4.2} + k_3\overline{Q}_M^2)(\overline{Q}_M + k_1)} \times$$

$$\times \{\overline{Q}_M - k_9[1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2]\} \times \quad (14)$$

$$\times \frac{\overline{M}_M}{k_{7.1} + (1+k_{7.2})\overline{M}_M}$$

➲ **where $k_2 \neq 0$:**

$$\eta = \frac{(Z - k_1)[1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2]}{k_{4.1} + (1 + k_{4.2} + k_3\overline{Q}_M^2)Z} \times$$

$$\times \frac{\overline{Q}_M - k_9[1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2]}{\overline{Q}_M} \times \quad (15)$$

$$\times \frac{\overline{M}_M}{k_{7.1} + (1+k_{7.2})\overline{M}_M}$$

where :

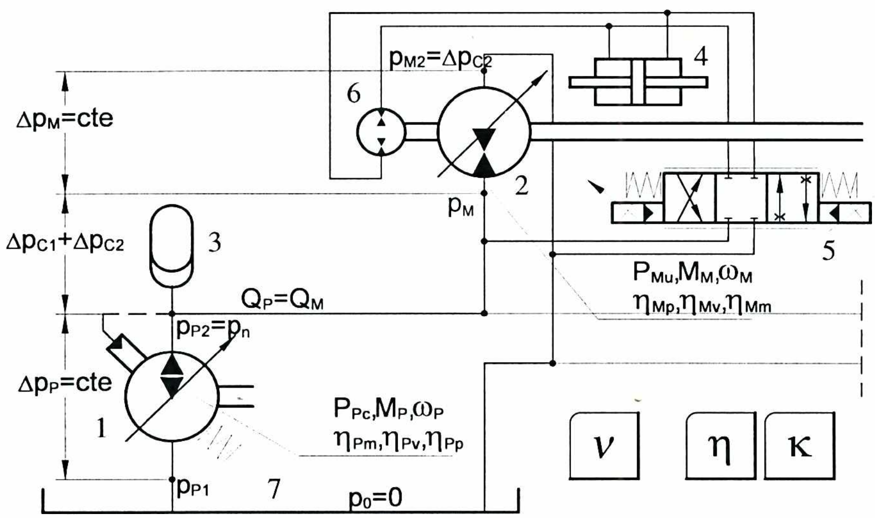
$$Z = \frac{1}{2k_2} - \left[\left(\frac{1}{2k_2}\right)^2 - \frac{k_1}{k_2} - \frac{\overline{Q}_M}{k_2}\right]^{0.5}$$

In order to express characteristics of the total transmission efficiency η as a function of the speed coefficient $\overline{\omega}_M$ and the load coefficient $\overline{M}_M$ for the controlled motor, the relationship between the flow coefficient $\overline{Q}_M$ and the coefficients $\overline{\omega}_M$ and $\overline{M}_M$ should be determined at first. The relationship in question takes the following form (acc. to [10]) :

$$\overline{Q}_M = \frac{\overline{\omega}_M[k_{7.1} + (1+k_{7.2})\overline{M}_M]}{1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2} + \quad (16)$$

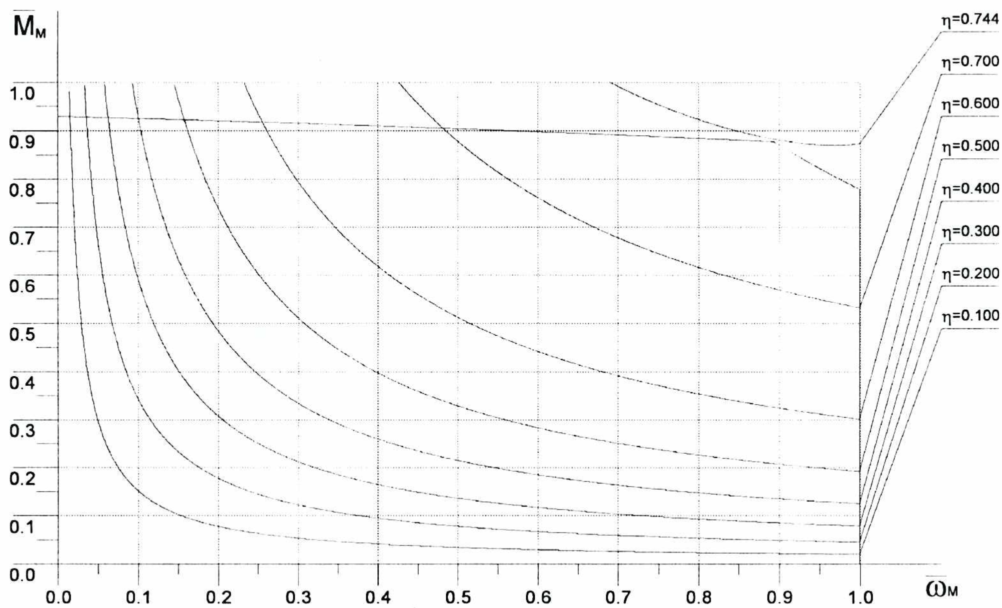
$$+ k_9[1 - (k_5+k_6)\overline{Q}_M - k_8\overline{Q}_M^2]$$

Fig.2. Central system with Adaptive Secondary Control



1. variable capacity hydraulic pump/motor unit with a constant pressure controller and with two directions of flow
2. variable capacity hydraulic motor/pump unit with an Adaptive Secondary Control and with two directions of flow
3. accumulator
4. double acting cylinder
5. throttling directional control valve
6. fixed capacity pump/motor unit with two directions of flow
7. reservoir

a) *Schematic diagram of the system*



b) *Relationship between the motor charge coefficient $\overline{M}_M$, its speed coefficient $\overline{\omega}_M$ and total efficiency η of the system*

Tab.2. *Characteristics of the system elements*

Pump	Hydraulic motor	Conduits
k_1 = 0.04 k_2 = 0.03 k_3 = 0.02 $k_{4.1}$ = 0.05 $k_{4.2}$ = 0.02	$k_{7.1}$ = 0.05 $k_{7.2}$ = 0.02 k_8 = 0.02 k_9 = 0.04	k_5 = 0.01 $k_{6.1}$ = 0.01 $k_{6.2}$ = 0.02
Viscosity ν = 35		

The system shown in Fig.2 works with one-motor feeding. Characteristics of the system elements are presented in Tab.2.

Due to possible changing the geometrical working capacity q_{Mgv}, the motor can achieve - at the lower load M_M - the speed ω_M (n_M) higher than the nominal speed ω_{Mn} (n_{Mn}). Thus in this case the speed coefficient $\overline{\omega}_M$ can exceed the value 1.

When the load coefficient $\overline{M}_M$ is given, the upper limit $\overline{\omega}_{M\max}$ (of the range: $0 \le \overline{\omega}_M \le \overline{\omega}_{M\max}$) is limited by the maximum pump delivery, i.e. the maximum value of the flow coefficient $\overline{Q}_{M\max}$.

When $\frac{p_{P2}}{p_n} = 1$, the flow $Q_{M\max} = Q_{P\max}$ is defined by the equation :

$$Q_{M\max} = Q_{P\max} = (1-k_1)(1-k_2)Q_{Pt} \tag{17}$$

therefore :

$$\overline{Q}_{M\max} = (1-k_1)(1-k_2) \tag{18}$$

The knowledge of $\overline{Q}_{M\max}$ makes it possible to express $\overline{\omega}_{M\max}$ in function of $\overline{M}_M$ according to the equation transformed from equation (16) with $\overline{Q}_{M\max}$ replaced by (18) :

$$\overline{\omega}_{M\max} = \frac{\overline{Q}_{M\max} - k_9\left[1-(k_5+k_6)\overline{Q}_{M\max} - k_8\overline{Q}^2_{M\max}\right]}{k_{7.1} + (1+k_{7.2})\overline{M}_M} \times$$

$$\times\left[1-(k_5+k_6)\overline{Q}_{M\max} - k_8\overline{Q}^2_{M\max}\right] = \qquad (19)$$

$$= \frac{(1-k_1)(1-k_2)-k_9}{k_{7.1}+(1+k_{7.2})\overline{M}_M} \times$$

$$\times\left[1-(k_5+k_6)(1-k_1)(1-k_2)-k_8(1-k_1)^2(1-k_2)^2\right]^2$$

The upper limit $\overline{M}_{M\max}$ (of the range: $0 \leq \overline{M}_M \leq \overline{M}_{M\max}$) given when $q_{Mgv} = q_{Mt}$, is equal to that of the individual system with volumetric control by the variable delivery pump. In this case the relationship between $\overline{M}_{M\max}$ and $\overline{Q}_M$, given when $\frac{p_{P2}}{p_n} = 1$, is represented by the equation (acc. to [10]) :

$$\overline{M}_{M\max} = \frac{1-k_{7.1}-(k_5+k_6)\overline{Q}_M - k_8\overline{Q}^2_M}{1+k_{7.2}} \qquad (20)$$

$\overline{M}_{M\max}$ can be presented in function of $\overline{\omega}_M$ after replacement of $\overline{Q}_M$ in (20) by the following expression :

$$\overline{Q}_M = \overline{\omega}_M + k_9\left[k_{7.1} + (1+k_{7.2})\overline{M}_{M\max}\right] \qquad (21)$$

which is valid only if $q_{Mgv} = q_{Mt}$.

CHARACTERISTICS OF THE SYSTEM ELEMENTS AND INPUT DATA USED IN THE EXEMPLARY CALCULATIONS

The examplary elements used in the compared systems are of the average characteristics of energy losses and work characteristics, namely :

1. *The axial piston pump* with swash plate, of the energy loss coefficients determined at the viscosity v_n and presented in Tab.3 below :

Table 3

Energy loss coefficient	Energy loss type	Condition
k_1 = 0.04	volumetric losses	$\Delta p_{Pi} = p_n$
k_3 = 0.02	pressure losses	$Q = Q_{Pt}$
$k_{4.1}$ = 0.05	mechanical losses	$\Delta p_{Pi} = 0$
$k_{4.2}$ = 0.02	increase of mechanical losses	Δp_{Pi} increases from 0 to p_n

2. *The electric motor* which drives the pump, of the speed drop coefficient $k_2 = 0.03$ at 50 kW nominal motor power.

3. *The axial piston hydraulic motor* with swash plate, of the energy loss coefficients determined at the viscosity v_n and presented in Tab.4 below :

Table 4

Energy loss coefficient	Energy loss type	Condition
$k_{7.1}$ = 0.05	mechanical losses	$M_M = 0$
$k_{7.2}$ = 0.02	increase of mechanical losses	M_M increases from 0 to M_{Mn}
k_8 = 0.02	pressure losses	$Q = Q_{Pt}$
k_9 = 0.04	volumetric losses	$\Delta p_{Mi} = p_n$

4. *The conduits* of the pressure loss coefficients determined at the viscosity v_n and presented in Tab.5 below :

Table 5

Energy loss coefficient	Energy loss type	Condition
k_5 = 0.01	pressure losses	$Q = Q_{Pt}$
$k_{6.1}$ = 0.01		
$k_{6.2}$ = 0.02		

5. *The throttle control assembly* of the motor speed in *Load Sensing* system, i.e. two-way flow controller of the coefficient $k_{10} = 0.03$ of the minimal pressure drop Δp_{EEmin}, which guarantees its proper work.

6. L-HM 46 hydraulic oil (acc. to PN-C-96057-5 standard, 1994) is assumed to be applied in the system, of the viscosity ν changing along with the change of the temperature ϑ as follows (Tab.6) :

Table 6

Viscosity value	At temperature	Comments
ν = 300 mm^2s^{-1}	$\vartheta \approx 10°C$	
ν = 100 mm^2s^{-1}	$\vartheta \approx 24°C$	
ν = 35 mm^2s^{-1}	$\vartheta \approx 46°C$	
ν = 10 mm^2s^{-1}	$\vartheta \approx 80°C$	
ν_{min} = 10 mm^2s^{-1}		admissible minimum viscosity
ν_{max} = 300 mm^2s^{-1}		admissible maximum viscosity
ν_n = 35 mm^2s^{-1}		value proposed by producers

7. The energy behaviour of the system was also determined at higher values of the coefficients k_1 and k_9 equal to 0.07 and 0.10, respectively, in order to make it possible to estimate the influence of volumetric losses in the pump and hydraulic motors on the total system efficiency.

RESULTS OF THE RESEARCH

Computer simulation results of the two analyzed systems: *Load Sensing* (LS) and that with *Adaptive Secondary Control* (ASC) are exemplified in Fig.1 and 2 as well as Tab.7 to 13. In the figures and tables the energy efficiency η of the analyzed systems is compared in the form of the function $\eta = f(k_1, k_9, \nu, \overline{\omega}_M, \overline{M}_M)$.

Tab.7. $k_1 = 0.04$ $k_9 = 0.04$ $v_n = 35\ mm^2s^{-1}$

$\overline{\omega}_M$	0.89	0.89	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.85	0.87	0.50	0.10	0.50	0.10
η (LS)	0.72 (η_{max})		0.62	0.30	0.35	0.14
η (ASC)		0.74 (η_{max})	0.59	0.24	0.26	0.08

Tab.8. $k_1 = 0.04$ $k_9 = 0.04$ $v_{min} = 10\ mm^2s^{-1}$

$\overline{\omega}_M$	0.83	0.83	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.88	0.90	0.50	0.10	0.50	0.10
η (LS)	0.69 (η_{max})		0.61	0.32	0.31	0.14
η (ASC)		0.71 (η_{max})	0.51	0.18	0.20	0.05

Tab.9. $k_1 = 0.04$ $k_9 = 0.04$ $v = 100\ mm^2s^{-1}$

$\overline{\omega}_M$	0.72	0.67	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.80	0.84	0.50	0.10	0.50	0.10
η (LS)	0.69 (η_{max})		0.61	0.27	0.38	0.14
η (ASC)		0.72 (η_{max})	0.63	0.29	0.32	0.09

Tab.10. $k_1 = 0.04$ $k_9 = 0.04$ $v_{max} = 300\ mm^2s^{-1}$

$\overline{\omega}_M$	0.40	0.38	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.71	0.80	0.50	0.10	0.50	0.10
η (LS)	0.59 (η_{max})		0.52	0.20	0.37	0.12
η (ASC)		0.65 (η_{max})	0.63	0.31	0.36	0.10

Tab.11. $k_1 = 0.07$ $k_9 = 0.07$ $v_n = 35\ mm^2s^{-1}$

$\overline{\omega}_M$	0.84	0.84	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.85	0.88	0.50	0.10	0.50	0.10
η (LS)	0.68 (η_{max})		0.59	0.30	0.31	0.13
η (ASC)		0.70 (η_{max})	0.52	0.19	0.20	0.05

Tab.12. $k_1 = 0.10$ $k_9 = 0.10$ $v_n = 35\ mm^2s^{-1}$

$\overline{\omega}_M$	0.78	0.78	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.85	0.88	0.50	0.10	0.50	0.10
η (LS)	0.63 (η_{max})		0.56	0.30	0.27	0.12
η (ASC)		0.65 (η_{max})	0.46	0.15	0.16	0.04

Tab.13. $k_1 = 0.10$ $k_9 = 0.10$ $v_{min} = 10\ mm^2s^{-1}$

$\overline{\omega}_M$	0.76	0.60	0.50	0.50	0.10	0.10
$\overline{M}_M$	0.51	0.91	0.50	0.10	0.50	0.10
η (LS)	0.56 (η_{max})		0.50	0.30	0.20	0.12
η (ASC)		0.53 (η_{max})	0.35	0.10	0.10	0.02

ANALYSIS OF THE RESULTS

Work of the systems at the recommended viscosity $v_n = 35\ mm^2s^{-1}$ (Tab.7)

- The maximum efficiency $\eta_{max} = 0.72$ of the LS system is a little lower than $\eta_{max} = 0.74$ in relation to the ASC system – as a result of the limitation of the maximum value of $\overline{M}_M$ in the LS system, which is the consequence of using the flow controller. This assembly demands a minimum pressure drop described by the coefficient $k_{10} = 0.03$.
- Lowering the coefficient $\overline{\omega}_M$ or $\overline{M}_M$ causes a faster drop of the efficiency η in the ASC system than in the LS system. In effect, the efficiency η of the LS system becomes higher than that of the ASC system. The relative difference of these two efficiencies is higher when values of $\overline{\omega}_M$ and $\overline{M}_M$ are lower.

Work of the systems at the minimum viscosity $v_{min} = 10\ mm^2s^{-1}$ (Tab.8)

- One can observe a decrease (in comparison with work at $v_n = 35\ mm^2s^{-1}$) in the maximum efficiency of both systems: to $\eta_{max} = 0.69$ of the LS system and $\eta_{max} = 0.71$ of the ASC system (as the result of an increase in volumetric losses of the pump and motor).
- Lowering the $\overline{\omega}_M$ and $\overline{M}_M$ causes a faster fall of the efficiency η of the ASC system than that of the LS system. In effect, the predominance of the efficiency of the LS system over that of the ASC system becomes even higher than in the case of work at $v_n = 35\ mm^2s^{-1}$.

Work of the systems at the large viscosity $v = 100\ mm^2s^{-1}$ (Tab.9)

- One can observe a decrease (in comparison with work at $v_n = 35\ mm^2s^{-1}$) in the maximum efficiency of both systems: to $\eta_{max} = 0.69$ of the LS system and $\eta_{max} = 0.72$ of the ASC system (as the result of higher pressure losses in conduits of the system).
- Decreasing the $\overline{\omega}_M$ and $\overline{M}_M$ causes a slower drop (in comparison with work at $v_n = 35\ mm^2s^{-1}$ and $v_{min} = 10\ mm^2s^{-1}$) of efficiency of the ASC system. In effect, the efficiency of the ASC system at $\overline{\omega}_M = 0.50$ is higher than that of the LS system. Instead, at $\overline{\omega}_M = 0.10$, the efficiency of the ASC system is still lower than that of the LS system.

Work of the systems at the maximum viscosity $v_{max} = 300\ mm^2s^{-1}$ (Tab.10)

- One can observe (in comparison with work at $v = 100\ mm^2s^{-1}$) a further decrease in the maximum efficiency of both systems: to $\eta_{max} = 0.59$ of the LS system and $\eta_{max} = 0.65$ of the ASC system. So the drop of η_{max} of the LS system is more distinct. The reason is a great increase of the pressure losses Δp_{C1} (Fig.1) in the conduit between the pump and flow controller, resulting from the increase of the viscosity v.
- Decreasing the $\overline{\omega}_M$ and $\overline{M}_M$ causes a slower (in comparison with work at $v_n = 100\ mm^2s^{-1}$) drop of efficiency η of the ASC system. In effect, the efficiency of the ASC system, at $\overline{\omega}_M = 0.50$, is higher than that of the LS system. Instead, at $\overline{\omega}_M = 0.10$, the efficiency of the ASC system is a little lower than that of the LS system.

Work of the systems at the increased value of the volumetric loss coefficient $k_1 = 0.07$ of the pump and $k_9 = 0.07$ of the hydraulic motor, at the recommended viscosity $v_n = 35\ mm^2s^{-1}$ (Tab.11)

- One can observe a decrease (in comparison with the case $k_1 = 0.04$ and $k_9 = 0.04$) in the maximum efficiency of both systems: to $\eta_{max} = 0.68$ of the LS system and $\eta_{max} = 0.70$ of the ASC system (as a result of the increase of volumetric losses in the pump and motor).
- Decreasing the $\overline{\omega}_M$ and $\overline{M}_M$ causes a faster drop of the efficiency η of the ASC system. In effect, the predominance of the efficiency η of the LS system over that of the ASC system becomes higher than in the case of work of the system with the pump of $k_1 = 0.04$ and motor of $k_9 = 0.04$.

Work of the systems at the large value of the volumetric loss coefficient $k_1 = 0.10$ of the pump and $k_9 = 0.10$ of the hydraulic motor, at the recommended viscosity $v_n = 35\ mm^2s^{-1}$ (Tab.12)

- One can observe a further decrease (in comparison with the case $k_1 = 0.07$ and $k_9 = 0.07$) of the maximum efficiency of both systems: to $\eta_{max} = 0.63$ of the LS system and $\eta_{max} = 0.65$ of the ASC system (as the result of the further increase of volumetric losses in the pump and motor).
- Decreasing the $\overline{\omega}_M$ and $\overline{M}_M$ causes a still faster drop of the efficiency η of the ASC system than that of the LS system. In effect, the predominance of the efficiency η of the LS system over that of the ASC system becomes still higher than in the case of work of the system with the pump of $k_1 = 0.07$ and motor of $k_9 = 0.07$.

Work of the systems at the large value of the volumetric loss coefficient $k_1 = 0.10$ of the pump and $k_9 = 0.10$ of the motor , at the minimum viscosity $v_n = 10\ mm^2s^{-1}$ (Tab.13)

- One can observe the greatest drop of the maximum efficiency of both systems : to $\eta_{max} = 0.56$ of the LS system and still lower $\eta_{max} = 0.52$ of the ASC system.
- Decreasing the $\overline{\omega}_M$ and $\overline{M}_M$ causes the quickest fall of the efficiency η of the ASC system. In effect, the predominance of the efficiency η of the LS system over that of the ASC system becomes very large.

CONCLUSIONS

By using the method described in [6], the computer simulation results of the energy efficiency η were compared for two of the most widespread energy-saving structures of hydrostatic drive in central system : the *Load Sensing* system (LS) with the throttle control by a two-way flow regulator and the system with *Adaptive Secondary Control*. The comparison is made for the case of supplying one hydraulic motor. In result, the following general conclusions can be offered :

- ❑ The energy efficiency of hydraulic systems of various structures, but composed of the same pumps and motors working in the recommended range of oil viscosity, is similar at conditions of the nominal working parameters of the motor.The structure of the motor speed control influences the energy efficiency first of all in the lower range of values of the speed coefficient $\overline{\omega}_M$ and load coefficient $\overline{M}_M$.
- ❑ The η_{max} value of the ASC system is a little higher than that of the LS system. This difference is influenced by the value of the minimum pressure drop coefficient $k_{10} = 0.03$ of the flow controller used in the LS system. At $k_{10} = 0.03$ and the range of viscosity :10 mm²s⁻¹≤ $\nu \leq$ 100 mm²s⁻¹, the difference of η_{max} values of the hypothetical systems is about 0.02÷0.03.
- ❑ The basic energy gain resulting from the use of the *Load Sensing* energy-saving system is achieved in the area of lower values of the coefficients $\overline{\omega}_M$ and $\overline{M}_M$.
- ❑ The decrease of the efficiency η of the LS system in the range of the lower motor speed is connected with the value of coefficient k_5 of the pressure losses in the conduit between the pump and flow controller. The coefficient k_5 must be as low as possible and the system must work at the recommended viscosity ν_n = = 35 mm²s⁻¹.
- ❑ The presented comparison made for the case of supplying one hydraulic motor shows that the ASC solution, in result of creating difficult operating conditions of the pump and hydraulic motor, is characterized by the energy efficiency lower than that of the LS solution with the throttle motor control.

NOMENCLATURE

cte - constant
F - load
k_1 - coefficient of relative volumetric losses per one shaft revolution of fixed capacity pump
k_2 - coefficient of relative decrease of pump rotational speed
k_3 - coefficient of relative pressure losses (flow resistance) in internal pump ducts, at theoretical pump delivery
$k_{4.1}$ - coefficient of relative mechanical losses in pump, at $\Delta p_{pi} = 0$
$k_{4.2}$ - coefficient of relative increase of mechanical pump losses, at increase of pressure in pump cylinder
k_5 - coefficient of relative pressure losses (flow resistances) in the line joining the pump with throttle control unit, at theoretical pump delivery
$k_{6.1}$ - coefficient of relative pressure losses (flow resistances) in the line joining the throttle control unit with hydraulic motor, at theoretical pump delivery
$k_{6.2}$ - coefficient of relative pressure losses (flow resistances) in hydraulic motor outlet line, at theoretical pump delivery
$k_{7.1}$ - coefficient of relative mechanical losses in hydraulic motor, at torque $M_M = 0$
$k_{7.2}$ - coefficient of relative increase of mechanical losses in motor, at increase of torque M_M
k_8 - coefficient of relative pressure losses (flow resistances) in internal ducts of hydraulic motor, at theoretical pump delivery
k_9 - coefficient of relative volumetric losses in hydraulic motor
k_{10} - coefficient of relative minimum pressure decrease in 2 - way flow control valve, or coefficient of relative pressure decrease in 3 - way flow control valve
M - torque
M_M - hydraulic motor shaft torque
$\overline{M}_M$ - hydraulic motor relative load coefficient
M_{Mn} - hydraulic motor shaft nominal torque
M_P - pump shaft torque
n - rotational speed
n_M - hydraulic motor shaft rotational speed
p - relative pressure (overpressure or underpressure)
p_n - nominal working pressure of hydrostatic transmission (hydraulic system)
$\overline{p}_{P2}$ - relative value of the pump supplying pressure
Δp - change of pressure, flow resistance
P - power
P_{Mu} - hydraulic motor shaft power output
P_{Pc} - pump shaft input power
q - cubic capacity
q_{Mgv} - geometrical variable working cubic capacity of hydraulic motor
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 capacity
Q_M - hydraulic motor absorbing capacity, intensity of flow to hydraulic motor
$\overline{Q}_M$ - flow coefficient Q_M / Q_{Pt}
Q_P - pump delivery
Q_{Pt} - theoretical pump delivery
R - force
v - linear speed
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
η_{st} - circuit structural efficiency
ϑ - temperature
κ - coefficient of decrease in the total energy efficiency of hydrostatic transmission with controllable speed hydraulic motor, related to the efficiency of hydrostatic transmission with variable capacity hydraulic pump
ν - viscosity
ω - angular speed
ω_M - hydraulic motor shaft angular speed
$\overline{\omega}_M$ - hydraulic motor speed coefficient - ratio of instantaneous speed to nominal one of a rotary hydraulic motor, or ratio of linear instantaneous speed to nominal one of a linear hydraulic cylinder
ω_P - pump shaft angular speed

Indices

c - input
g - geometric
i - internal
m - mechanical
M - hydraulic motor
n - nominal
o - idle run
P - pump
t - theoretical
u - output
v - volumetric

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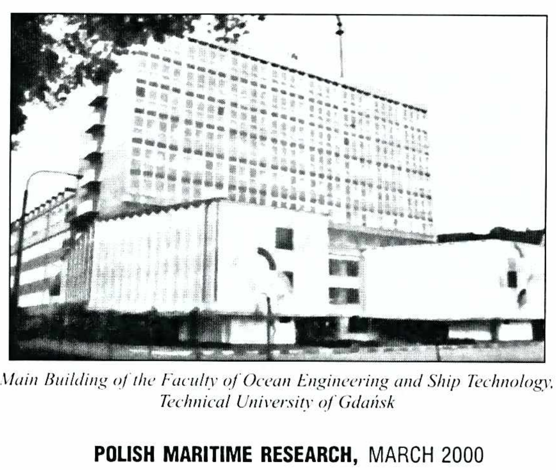

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