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Energy Saving in a Hydraulic Servomechanism System – Theory and Examples of Laboratory Verification

Original scientific paper

A control system with a directional control servo valve or a proportional directional throttling control valve, controlling a linear hydraulic motor (cylinder) is used in the ship steering gear drive, in the controllable pitch propeller control, in the variable capacity pump control system for hydraulic deck equipment motors or fixed pitch propellers in small ships (e.g. ferries).

The most popular solution is a system where a throttling control valve is fed by a constant capacity pump cooperating with an overflow valve. This system, working with a constant pressure, achieves a high energy efficiency value η only at the point of maximum motor load coefficient $\bar{M}_M$ and maximum motor speed coefficient $\bar{\omega}_M$.

Energy savings in a constant capacity pump operation may be achieved by means of an overflow valve controlled by the oil outlet point pressure between the directional control valve and the cylinder. Although structural volumetric losses cannot be eliminated in such a system, it is possible to reduce considerably structural pressure losses, mechanical losses and volumetric losses in the pump, and also mechanical losses in the cylinder.

The paper discusses these energy savings using an earlier developed mathematical model of losses in elements, the energy efficiency of the system and the operating range of the cylinder [9].

Key words: *energy efficiency, hydrostatic servo systems, hydrostatic transmissions, ship control systems*

Uštede energije u hidrauličnom servo sustavu – teorija i primjer laboratorijske potvrde

Izvorni znanstveni rad

Upravljački sustav s proporcionalnim razvodnikom ili proporcionalnim prigušnim ventilom, koji upravlja linearnim hidrauličkim motorom (cilindrom) upotrebljava se u: brodskom kormilarskom mehanizmu, upravljanju zakretnih lopatica propelera, upravljačkom sustavu s pumpom promjenjive dobave za pogon hidrauličkih motora na palubi i za pogon propelera na malim brodovima (npr. trajektima).

Najpopularnije rješenje je sustav u kojemu je prigušni regulacijski ventil napajan pumpom konstantne dobave spregnute s rasteretnim ventilom. Ovaj sustav, radeći pri konstantnom tlaku, postiže visoku energetsku učinkovitost η samo u točki maksimalnog koeficijenta opterećenja $\bar{M}_M$ i maksimalnog koeficijenta brzine $\bar{\omega}_M$ motora.

Ušteda energije u radu pumpe konstantne dobave može biti ostvarena pomoću rasteretnog ventila upravljanog na temelju tlaka u ulju u točki između razvodnika i cilindra. Iako strukturni volumentrički gubitci ne mogu biti eliminirani u takvom sustavu, moguće je značajno reducirati strukturne gubitke tlaka, mehaničke i volumetričke gubitke u pumpi, te mehaničke gubitke u cilindru. Ovaj članak prezentira takve energetske "uštede" bazirajući se na ranije razvijenom matematičkom modelu gubitaka u elementima, energetskoj učinkovitosti sustava i području rada cilindra [9].

Ključne riječi: *brodski upravljački sustavi, energetska učinkovitost, hidraulički servo sustavi, hidrostatički prijenos*

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Nomenclature

cte - constant

f - throttling slot section

f_{DEmax} - maximal throttling slot section of directional control valve (servo valve, proportional valve)

f_0 - reference throttling slot section which gives the intensity of flow to hydraulic motor equal to theoretical pump delivery – $Q_M = Q_{Pt}$ at pressure decrease in directional control valve equal to nominal pressure of hydrostatic transmission – $\Delta p_{DE} = p_n$

$\bar{f}$ - relative throttling slot section $\bar{f} = f/f_0$

$\bar{f}_{DE\,max}$ - maximal relative throttling slot section of directional control valve (servo valve, proportional valve) $\bar{f}_{DE\,max} = f_{DEmax}/f_0$

F - load, force

F_M - hydraulic linear motor (cylinder) load

F_{Mi} - force indicated on the piston of the hydraulic linear motor (cylinder)

F_{Mm} - hydraulic linear motor mechanical losses

F_{Mn} - hydraulic linear motor nominal load (force)

k_1 - coefficient of relative volumetric losses per one shaft revolution of fixed capacity pump

k_2 - coefficient of relative decrease in pump rotational speed

k_3 - coefficient of relative pressure losses (flow resistance) in internal pump ducts, at theoretical pump delivery Q_{Pt}

$k_{4.1}$ - 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 in pressure in pump working chambers

k_5 - coefficient of relative pressure losses (flow resistances) in the line joining the pump with the throttle control unit, at theoretical pump delivery Q_{Pt}

$k_{6.1}$ - coefficient of relative pressure losses (flow resistances) in the line joining the throttle control unit with the hydraulic motor, at theoretical pump delivery Q_{Pt}

$k_{6.2}$ - coefficient of relative pressure losses (flow resistances) in hydraulic motor outlet line, at theoretical pump delivery Q_{Pt}

$k_{7.1}$ - coefficient of relative mechanical losses in hydraulic motor – cylinder, at a force $F_M = 0$

$k_{7.2}$ - coefficient of relative increase of mechanical losses in motor – cylinder, at increase of force F_M

k_8 - coefficient of relative pressure losses (flow resistances) in internal ducts of hydraulic motor, at theoretical pump delivery Q_{Pt}

k_9 - coefficient of relative volumetric losses in hydraulic motor

k_{10} - coefficient of relative minimum pressure decrease in 2-way flow control valve, which still ensures the flow regulation, or coefficient of relative pressure decrease in 3-way flow control valve

k_{11} - coefficient of relative pressure decrease Δp_{DE} in directional control valve (servo valve, proportional valve) demanded by a maximal throttling section f_{DEmax} for receiving flow intensity equal to theoretical pump delivery Q_{Pt}

M - torque

$\bar{M}_M$ - hydraulic motor relative load coefficient $\bar{M}_M = F_M/F_{Mn}$

M_P - pump shaft load (torque)

M_{Pm} - torque of pump mechanical losses

p - relative pressure (overpressure or under-pressure)

p_n - nominal (rated) working pressure of hydrostatic transmission (hydraulic system)

p_{P2} - pump supplying pressure

$\bar{p}_{P2}$ - relative value of the pump supplying pressure

Δp - change of pressure, flow resistance

Δp_{DE} - decrease of pressure in directional control valve (servo valve, proportional valve)

Δp_{Mi} - decrease of pressure (pressure drop) in hydraulic motor working chambers

Δp_{Pi} - increase of pressure in pump working chambers

P - power

P_{Mu} - hydraulic motor output power

P_{Pc} - pump shaft input power

q - cubic capacity

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

$\bar{Q}_M$ - flow coefficient Q_M/Q_{Pt}

Q_P - pump delivery

Q_{Pt} - theoretical pump delivery

η - energy efficiency

η_M - hydraulic motor total efficiency

η_{Mm} - hydraulic motor mechanical efficiency

η_{Mp} - hydraulic motor pressure efficiency

η_{Mv} - hydraulic motor volumetric efficiency

η_P - pump total efficiency

η_{Pm} - pump mechanical efficiency

η_{Pp} - pump pressure efficiency

η_{Pv} - pump volumetric efficiency

η_{st} - circuit structural efficiency

ϑ - temperature

ν - viscosity

v_M - hydraulic motor linear speed

v_{Mn} - hydraulic motor nominal linear speed

ω - angular speed

$\bar{\omega}_M$ - hydraulic motor speed coefficient – ratio of instantaneous speed to the nominal one of a hydraulic motor – $\bar{\omega}_M = v_M/v_{Mn}$

ω_P - pump shaft angular speed

Indices

c - input

C - conduit

g - geometric

i - internal

m - mechanical

M - hydraulic motor (cylinder)

n - nominal

o - idle run

p - pressure

P - pump, power

t - theoretical

u - output

v - volumetric

1 Introduction

The most often used hydraulic servomechanism system (Figure 1) or hydraulic rotational or linear motor (cylinder) proportional control system, in the case of proportional directional valve with $\Delta p_{DE2} = \Delta p_{DE1}$ (Figure 2), is a system where the directional control valve is fed by a constant capacity pump cooperating with an overflow valve stabilizing the feed pressure level (p = cte).

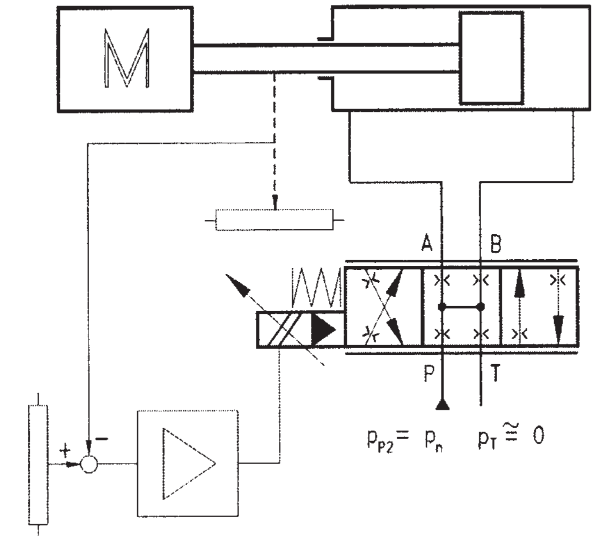
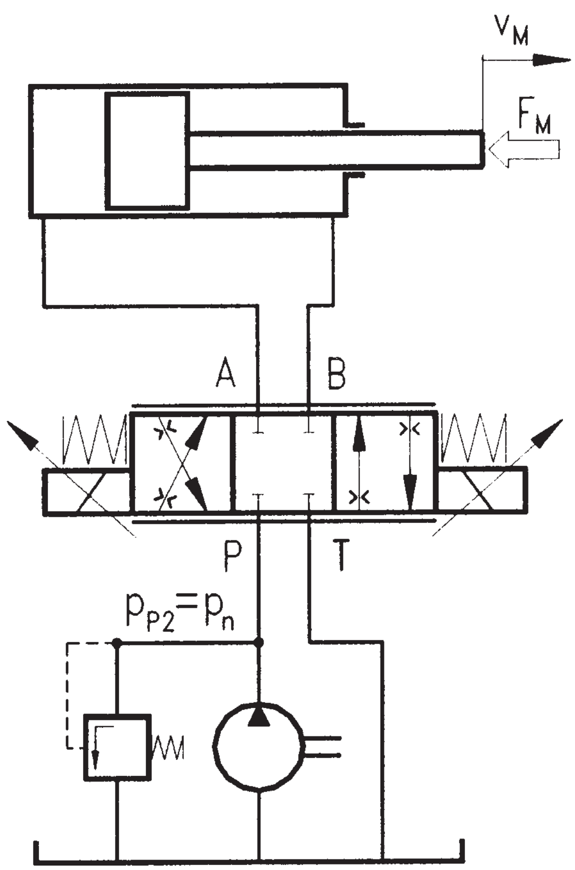


Figure 1 **Control system with servovalve representing two throttling slots – at the inlet and outlet of the cylinder**
Slika 1 **Kontrolni sustav sa servoventilom s dva prigušna otvora – na ulazu i izlazu iz cilindra**

Figure 2 **System with proportional directional control valve fed by a constant capacity pump in a constant pressure system**
Slika 2 **Sustav s proporcionalnim razvodnikom napajan pumpom konstantnog protoka u sustavu konstantnog tlaka (p = konst.)**



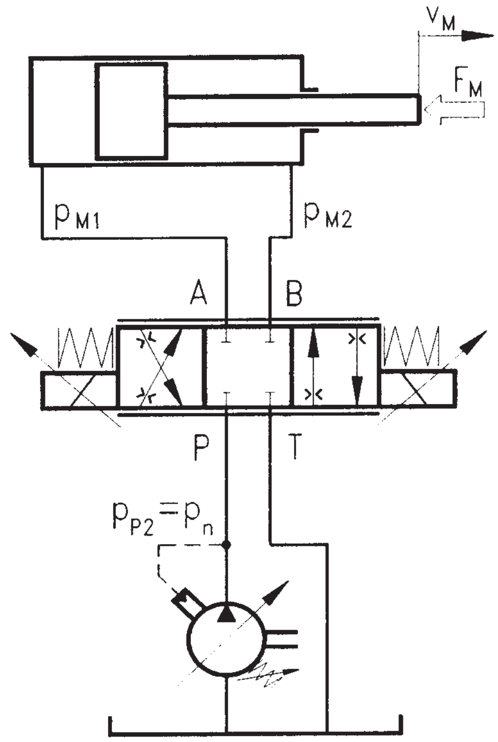
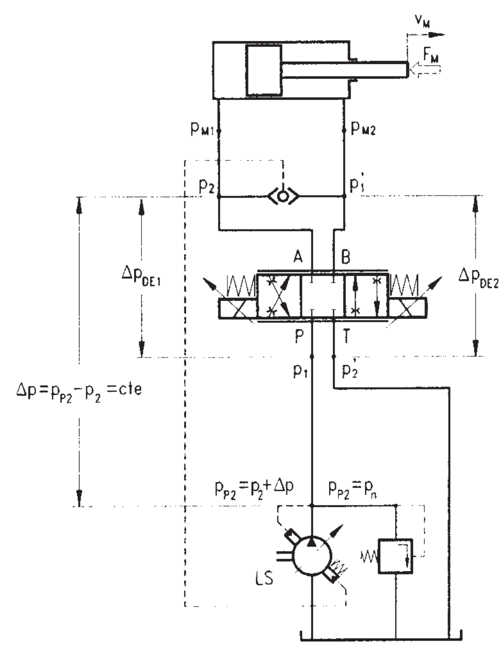


Figure 3 **System with proportional directional control valve fed by a variable capacity pump with pressure regulator (p = cte)**
Slika 3 **Sustav s proporcionalnim razvodnikom napajan pumpom promjenjivog protoka s regulatorom tlaka (p = konst.)**

Figure 4 **System with proportional directional control valve fed by a variable capacity pump with Load Sensing regulator (p = var)**
Slika 4 **Sustav s proporcionalnim razvodnikom napajan pumpom promjenjivog protoka s regulatorom upravljanim opterećenjem (p = var.)**



The mathematical description of the energy behaviour of such a system is presented in [1,2,3].

The laboratory verification of the simulation description of energy efficiency of elements and the system as a whole is presented in the research report [4]. The work was carried out in the laboratory of the Chair of Hydraulics and Pneumatics, Faculty of Mechanics, Gdansk University of Technology, and the results are presented in [5,6,7].

The system with constant feed pressure achieves high energy efficiency, equal to the efficiency of the system without the throttling control, only in the points of the maximum $\bar{M}_M$ coefficient and $\bar{\omega}_M$ coefficient of the controlled motor. The system efficiency η decreases rapidly with decreasing motor load and particularly with the simultaneously decreasing motor speed.

There are possibilities of reducing energy losses in the elements of a proportional control system (in the pump, in the throttling assembly and in the hydraulic linear motor – cylinder), therefore there are possibilities of increasing the energy efficiency of a directional control valve system.

They are connected, for instance, with the elimination of structural volumetric losses in the throttling assembly by using a variable capacity pump with a pressure regulator p=cte as a directional control valve feeding system (Figure 3).

The mathematical description of losses and energy efficiency of such a system is given in [1,3,8].

The use of a variable capacity pump with a *Load Sensing* regulator in the proportional control system (Figure 4) gives a possibility of elimination of structural volumetric losses, significant reduction of the structural pressure losses, reduction of mechanical losses in the linear hydraulic motor – cylinder and also reduction of mechanical and pressure losses in the pump.

The mathematical description of losses and energy efficiency of the *Load Sensing* feeding system (p = var) is also given in [1,3,8].

The use of a variable capacity pump with a p = cte or p = var regulator is connected with the high cost of the pump and regulator and should be decided on after an economic analysis, i.e. a comparison of additional investment cost with energy gains from the operation of such a system.

2 Energy–saving system with a constant capacity pump fed in a variable pressure system p = var

Energy savings may also be achieved in a system with a servo valve or with a proportional directional valve fed by a cheaper pump, i.e. a constant capacity pump cooperating with an overflow valve controlled by the discharge pressure from the servo valve to the hydraulic motor, a linear motor in particular (Figure 5,6).

In such a system, with p = var, the p_{P2} pressure in the discharge end of the pump is adapted to the p_2 pressure in the discharge conduit from the directional control valve to the motor (Figure 7).

The structural volumetric losses in such a system cannot be eliminated, but the structural pressure losses, mechanical and volumetric losses in the pump and also the mechanical losses in the cylinder can be considerably reduced. The reduction of mechanical losses in the cylinder is a result of significant decrease in pressure in the cylinder discharge conduit (Figure7).

A mathematical description of losses and the efficiency of elements and of the whole system with proportional control (or a hydraulic servomechanism system) with a hydraulic linear motor, fed by a constant capacity pump in a variable pressure system p = var, has been developed in [9, 10].

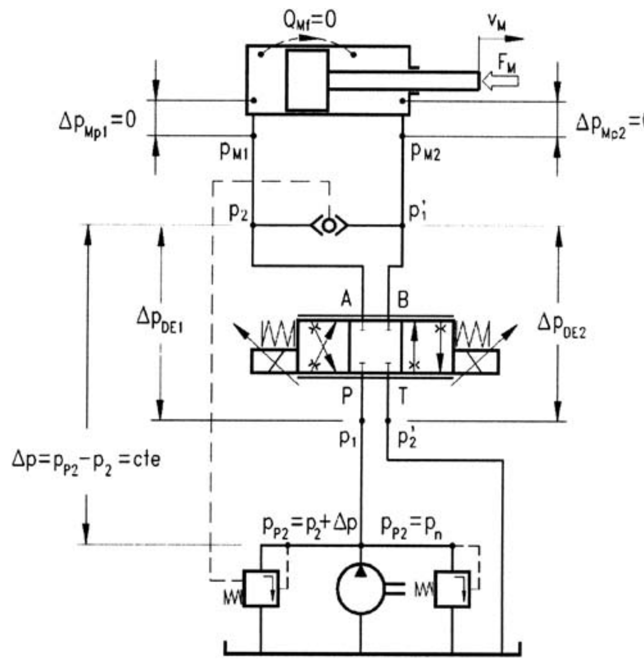


Figure 5 **System with proportional directional control valve fed by a constant capacity pump in a variable pressure system (p = var)**

Slika 5 **Sustav s proporcionalnim razvodnikom napajan pumpom konstantnog protoka s promjenjivim tlakom u sustavu (p = var.)**

An important problem is the laboratory verification of such a mathematical description. It is a part of the doctor thesis of Grzegorz Skorek [6], now at an advanced stage of elaboration.

3 Pressures and pressure drops in a system fed in the variable pressure system p = var

The system requires a directional control valve with maximum throttling section f_{DEmax} defined by the coefficient k_{11} fulfilling the condition: $\frac{k_{11}}{2} \langle k_{10}$.

Fulfilling the condition guarantees obtaining maximum intensity Q_{Mmax} in the proportional valve equal to the pump capacity Q_P, when the pump discharge conduit pressure p_{P2} increases to $p_{P2} = p_n$, i.e. when $p_{P2} < p_n$.

The system working parameters are presented in Figure 6. Figure 7 presents the relations of pressures, pressure drops in the proportional valve throttling slots and the flow resistances in the system connecting conduits to the pressure drop Δp_M in the cylinder.

Figure 8 presents a possibility of reducing the friction force F_{Mm} in the cylinder as an effect of a considerable decrease in throttling (reduction of the pressure drop Δp_{DE2} in the proportional valve slot f_{DE2}) in the cylinder discharge. This decrease in throttling is associated with a decrease in pressure p_{M2i} in the cylinder discharge chamber.

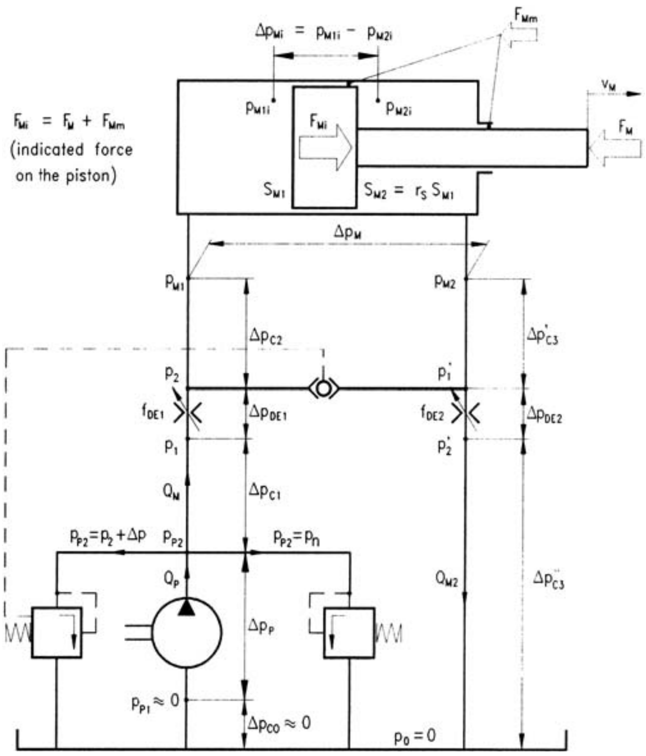


Figure 6 **Working parameters of the system with proportional valve fed by a constant capacity pump cooperating with an overflow valve controlled in a variable pressure system (p = var)**

Slika 6 **Radni parametri sustava s proporcionalnim ventilom napajanim pumpom konstantnog protoka spregnutom s rasteretnim ventilom upravljanim u sustavu s promjenjivim tlakom**

The p_{P2} pressure in the pump discharge conduit (Figure 5) is controlled by means of an overflow valve, by the p_2 pressure at the outlet from the proportional valve to the cylinder, as it is in the capacity pump system with the *Load Sensing*:

$$p_{P2} = p_2 + \left(k_5 + k_{10}\right) p_n . \qquad (1)$$

Relations of: p_{P2}, p_1, p_2, p_{M1}, p_{M2}, $p_1^{'}$, $p_2^{'}$ pressures, Δp_{DE1} and Δp_{DE2} pressure drops in the f_{DE1} and f_{DE2} proportional valve slots, Δp_{C1}, Δp_{C2}, $\Delta p_{C3}^{'}$, $\Delta p_{C3}^{''}$ flow resistances in the system connecting conduits to the pressure decrease Δp_M in the cylinder at a given proportional valve controlled Q_M = cte flow intensity. The proportional valve pressure decrease Δp_{DE} coefficient k_{11} was adopted as required by the maximum throttling slot section f_{DEmax}, in order to obtain the flow intensity equal to the theoretical pump capacity Q_{Pt} determined by the $k_{11}/2 = k_{10}$ equation.

The pressure difference $\Delta p = p_{P2} - p_2 = \left(k_5 + k_{10}\right) p_n$, between the pressure level p_{P2} in the pump discharge conduit and the pressure level p_2 in the outlet conduit from the proportional valve to the cylinder, is caused by the necessity of overcoming the maximum flow resistance Δp_{C1max} in the conduit between the pump and the proportional valve. The maximum flow resistance $\Delta p_{C1max} = k_5 p_n$ occurs in the case of the flow intensity Q_M equal to the theoretical pump capacity Q_{Pt} ($Q_M = Q_{Pt}$).

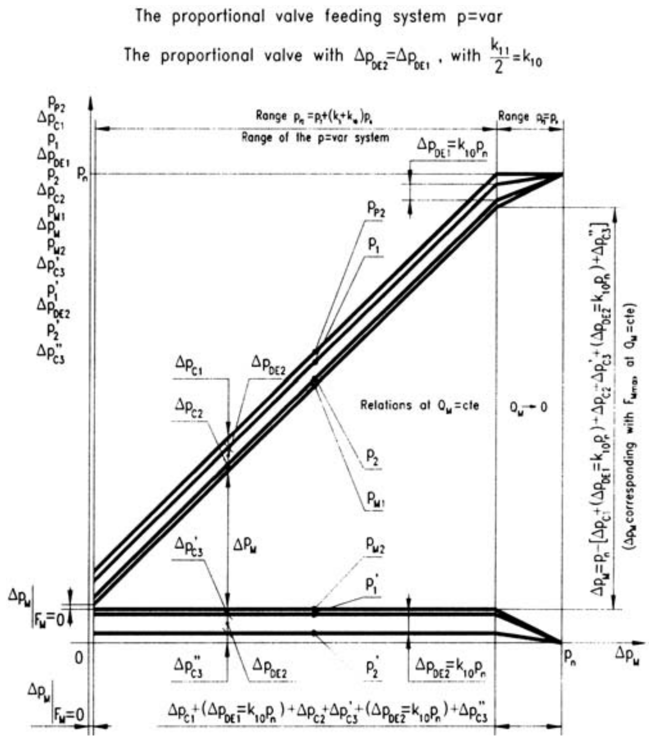


Figure 7 **The proportional valve variable pressure feeding system p = var with $\Delta p_{DE2} = \Delta p_{DE1}$, fed with:**
- variable capacity pump cooperating with the Load Sensing regulator controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder i.e. $p_{P2} = p_2 + (k_5 + k_{10})\, p_n$,
- constant capacity pump cooperating with an overflow valve controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder i.e. $p_{P2} = p_2 + (k_5 + k_{10})\, p_n$.

Slika 7 **Sustav napajanja varijabilnog tlaka s proporcionalnim ventilom p = var s $\Delta p_{DE2} = \Delta p_{DE1}$, napajan:-**
- pumpom promjenjivog protoka spregnutom s opteretno osjetljivim regulatorom tlaka upravljanim s p_2 tlakom na izlazu iz proporcionalnog ventila na cilindar $p_{P2} = p_2 + (k_5 + k_{10})\, p_n$
- pumpom konstantnog protoka spregnutom s rasteretnim ventilom upravljanim s p_2 tlakom na izlazu iz proporcionalnog ventila na cilindar $p_{P2} = p_2 + (k_5 + k_{10})\, p_n$

At the same time, in the proportional valve throttling slot, with a maximum area f_{DE1max}, a pressure drop of $\Delta p_{DE1}\big|_{f_{DE1max},\, Q_{Pt}} = k_{10}\ p_n$ must be ensured to guarantee Q_M intensity in the slot equal to the theoretical pump capacity Q_{Pt}.

In effect, the pressure drop $\Delta p_{DE1} = p_1 - p_2$ in the proportional directional valve throttling slot f_{DE1} (at the cylinder inlet) is, with a given $\bar{Q}_M$ value of the flow intensity coefficient adjusted by the proportional valve, equal to:

$$\Delta p_{DE1} = p_1 - p_2 = \left(k_5 + k_{10}\right) p_n - k_5 p_n \bar{Q}_M . \qquad (2)$$

The $\Delta p_{DE2} = p_1^{'} - p_2^{'}$ pressure drop in the f_{DE2} proportional valve throttling slot (at the cylinder outlet) will be equal to:

a. in the case of proportional valve with $\Delta p_{DE2} = \Delta p_{DE1}$:

$$\Delta p_{DE2} = \Delta p_{DE1} = \left(k_5 + k_{10}\right) p_n - k_5 p_n \bar{Q}_M , \qquad (3)$$

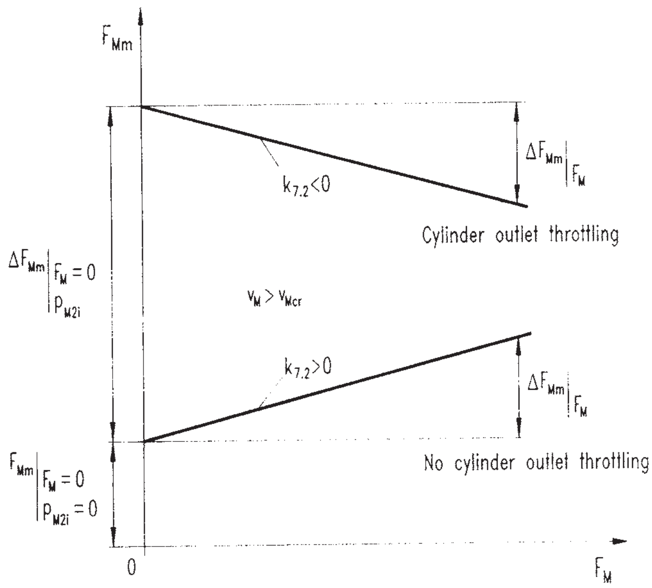


Figure 8 **Friction force F_{Mm} in cylinder as a function of external load F_M and the cylinder mechanical loss coefficient $k_{7.2}$. The system fed with a constant capacity pump cooperating with an overflow valve stabilizing the pressure level in the pump discharge conduit (system p = cte).**

Slika 8 **Sila trenja F_{Mm} u cilindru kao funkcija vanjskog tereta F_M i koeficijenta mehaničkih gubitaka cilindra $k_{7.2}$. Sustav je napajan pumpom konstantnog protoka spregnutom s rasteretnim ventilom stabilizirajući tlak u pumpnom povratnom vodu (p = konst.).**

b. in the case of proportional valve with $f_{DE2} = f_{DE1}$ or a servo valve:
   – with a single piston rod cylinder:

$$\Delta p_{DE2} = r_S^2 \ \Delta p_{DE1} = r_S^2 \left[\left(k_5 + k_{10}\right) p_n - k_5 p_n \bar{Q}_M \right], \quad (4)$$

   – with a double piston rod cylinder:

$$\Delta p_{DE2} = \Delta p_{DE1} = \left(k_5 + k_{10}\right) p_n - k_5 p_n \bar{Q}_M . \quad (5)$$

The p_2 pressure at the cylinder feed proportional valve outlet (cylinder inlet) has the following form:

$$p_2 = k_{6.1} \ p_n \bar{Q}_M + p_{M1i} = k_{6.1} \ p_n \bar{Q}_M + \frac{F_{Mi}}{S_{M1}} + r_S \ p_{M2i}$$
$$= k_{6.1} \ p_n \bar{Q}_M + \frac{F_{Mi}}{S_{M1}} + r_S \left(r_S \ k_{6.2} \ p_n \bar{Q}_M + \Delta p_{DE2} \right). \quad (6)$$

Following (1) and (6), the p_{P2} pump discharge pressure is given by the formula:

$$p_{P2} = p_2 + \left(k_5 + k_{10}\right) p_n = \frac{F_{Mi}}{S_{M1}} + k_{6.1} \ p_n \bar{Q}_M$$
$$+ r_S \left(r_S \ k_{6.2} \ p_n \bar{Q}_M + \Delta p_{DE2} \right) + \left(k_5 + k_{10}\right) p_n , \quad (7)$$

where the Δp_{DE2} pressure drop in the f_{DE2} proportional valve slot (cylinder outlet) will be determined, depending on the cylinder and proportional valve type used, in accordance with one of the following formulae (3), (4) or (5).

When F_{Mi} in equation (7) is replaced by:

$$F_{Mi} = k_{7.1} F_{Mn} + \left(1 + k_{7.2}\right) F_M , \quad (8)$$

the cylinder rated (nominal) force F_{Mn} by:

$$F_{Mn} = F_{M \left| \Delta p_M = p_n ; \ p_{M2} = 0; \ \eta_{Mm} = 1; \ \eta_{Mp} = 1 \right.} = S_{M1} p_n , \quad (9)$$

the F_M/F_{Mn} ratio by:

$$\bar{M}_M = \frac{F_M}{F_{Mn}} = \frac{F_M}{S_{M1} p_n} , \quad (10)$$

and F_M by:

$$F_M = \bar{M}_M F_{Mn} , \quad (11)$$

then the dimensionless expression $\bar{p}_{P2} = p_{P2}/p_n$, describing the ratio of p_{P2} pressure in the pump discharge conduit to the nominal system pressure p_n, will be presented as:

$$\bar{p}_{P2} = \frac{p_{P2}}{p_n} = k_{7.1} + (1 + k_{7.2}) \ \bar{M}_M + k_{6.1} \ \bar{Q}_M +$$
$$+ r_S \left(r_S \ k_{6.2} \bar{Q}_M + \frac{\Delta p_{DE2}}{p_n} \right) + k_5 + k_{10} \quad (12)$$

Similarly, the dimensionless ratio $\dfrac{\Delta p_{DE2}}{p_n}$ in formula (12) will be written as follows:

a. in the case of proportional directional valve with $\Delta p_{DE2} = \Delta p_{DE1}$ – in accordance with:

$$\frac{\Delta p_{DE2}}{p_n} = k_5 \left(1 - \bar{Q}_M \right) + k_{10} , \quad (13)$$

b. in the case of proportional directional valve with $f_{DE2}=f_{DE1}$ or servo valve:
   – with single piston rod cylinder – in accordance with:

$$\frac{\Delta p_{DE2}}{p_n} = r_S^{\ 2} \left[\ k_5 \left(1 - \bar{Q}_M \right) + k_{10} \right], \quad (14)$$

   – with double piston rod cylinder – in accordance with:

$$\frac{\Delta p_{DE2}}{p_n} = k_5 \left(1 - \bar{Q}_M \right) + k_{10} . \quad (15)$$

4 Possibilities of energy savings in a system with a constant capacity pump driven in a variable pressure system p = var

The basic energy gain resulting from feeding the system with directional proportional valve by a constant capacity pump in a variable pressure system is the reduction of structural pressure losses.

The adopted p_{P2} pressure setting system in the pump discharge conduit makes the pressure losses in the directional proportional valve (pressure structural efficiency η_{stp}) dependent on the pressure losses in the system conduits (conduit efficiency η_C). The product of η_{stp} and η_C allows to make conclusions about the sum of pressure losses.

The product of pressure structural efficiency η_{stp} and the conduit efficiency η_C has the form:

$$\eta_{stp}\,\eta_C = \frac{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M + \left(k_{6.1} + r_S\;k_{6.2}\right)\bar{Q}_M}{\bar{p}_{P2} - k_5\bar{Q}_M} \times$$

$$\times \frac{\bar{p}_{P2} - k_5\bar{Q}_M}{\bar{p}_{P2}}\;\frac{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M}{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M + \left(k_{6.1} + r_S k_{6.2}\right)\bar{Q}_M} = \quad (16)$$

$$= \frac{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M}{\bar{p}_{P2}}.$$

Replacing $\bar{p}_{P2}$ in formula (16) with expression (12) gives the $\eta_{stp}\,\eta_C$ product in the form:

$$\eta_{stp}\eta_C \frac{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M}{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M + k_{6.1}\,\bar{Q}_M + r_S\left(r_S\,k_{6.2}\bar{Q}_M + \frac{\Delta p_{DE2}}{p_n}\right) + k_5 + k_{10}}, \quad (17)$$

with the $\frac{\Delta p_{DE2}}{p_n}$ ratio according to (13), (14) or (15).

For example, using the directional proportional valve with $f_{DE2} = f_{DE1}$ or servo valve and double piston rod cylinder (expression (15)), gives the $\eta_{stp}\,\eta_C$ product in the form:

$$\eta_{stp}\eta_C = \frac{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M}{k_{7.1} + \left(1+k_{7.2}\right)\bar{M}_M + \left(k_6 - k_5\right)\bar{Q}_M + 2\left(k_5 + k_{10}\right)}\;. \quad (18)$$

This indicates that the sum of pressure structural losses and conduit pressure losses, given by the expression:

$$\left[\,\left(k_6 - k_5\right)\bar{Q}_M + 2\left(k_5 + k_{10}\right)\,\right]p_n\,, \quad (19)$$

is independent of a momentary value of the cylinder load coefficient $\bar{M}_M$ and is practically also independent of the momentary value of the $\bar{Q}_M$ coefficient of flow intensity set by the directional proportional valve (i.e. of the cylinder speed coefficient $\bar{\omega}_M$). The sum reaches a value of

$$2\,(k_5 + k_{10})\;p_n\,. \quad (20)$$

Assuming the k_5 coefficient equal to 0.02 and the k_{10} coefficient equal to 0.04, the sum of pressure structural losses and conduit losses may be evaluated as a constant approximately equal to 0.12 p_n of the system. That value of pressure losses in the directional proportional valve and system conduits, equal to 0.12 p_n, may be seen in Figure 7 as a difference between a momentary pump discharge pressure p_{P2} and a momentary value of the cylinder pressure drop Δp_M.

Therefore, the sum of structural pressure losses and conduit losses (Figure 6) is independent of a momentary pressure drop value Δp_M in the cylinder. It is relatively small. This gives a basic energy saving resulting from using the proportional valve or servo valve variable pressure feeding system p = var, in comparison with a constant pressure feeding system p = cte, i.e. with the nominal pressure p_n in the discharge conduit regardless of a momentary pressure drop value Δp_M in the cylinder (Figure 9).

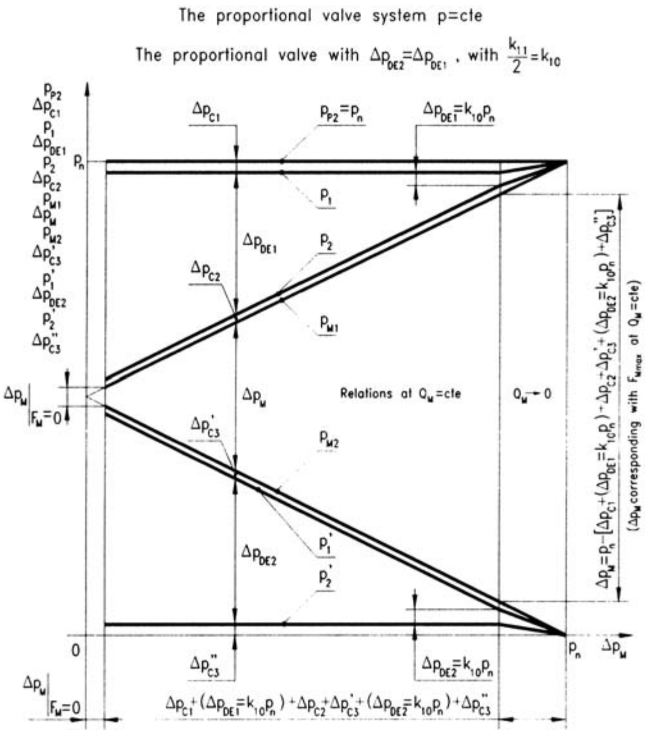


Figure 9 **The directional proportional throttling valve feeding system p =cte with $\Delta p_{DE2} = \Delta p_{DE1}$, fed with:**
– constant capacity pump cooperating with an overflow valve – coefficient a = 0 ,
- variable capacity pump cooperating with a pressure controller.

Slika 9 **Proporcionalni prigušni ventil sustav napajanja p = konst. s $\Delta p_{DE2} = \Delta p_{DE1}$, napajan:**
- pumpom konstantnog kapaciteta spregnutom s rasteretim ventilom – koeficijent a = 0,
- pumpom promjenjivog protoka spregnutm s regulatorom tlaka.

Relations of: p_{P2}, p_1, p_2, p_{M1}, p_{M2}, p'_1, p'_2 pressures, Δp_{DE1} and Δp_{DE2} pressure drops in the f_{DE1} and f_{DE2} proportional valve slots, flow resistances: Δp_{C1}, Δp_{C2}, $\Delta p'_{C3}$, $\Delta p''_{C3}$ in the system connecting conduits to the pressure decrease Δp_M in the cylinder at a given valve controlled Q_M = cte flow intensity. The proportional valve pressure decrease Δp_{DE} coefficient k_{11} was adopted as required by the maximum throttling slot section f_{DEmax} in order to obtain the flow intensity equal to the theoretical pump capacity Q_{Pt} determined by the $k_{11}/2 = k_{10}$ equation.

In a p = cte system (Figure 9), the sum of structural pressure losses and conduit losses reached a value of 0.96 p_n in a tested system (in a situation when a maximum pressure drop Δp_M, required by an unloaded cylinder, was $\Delta p_M\big|_{F_M=0} = 0.04 p_n$ – Figure 9).

It should be noted that in a p = cte system, in a situation of unloaded cylinder ($F_M = 0$), the p_{M2} pressure at the cylinder outlet is very high and generates higher mechanical losses in the cylinder. These higher losses required a higher pressure drop Δp_M in the tested unloaded cylinder of a value $\Delta p_M\big|_{F_M=0} = 0.04 p_n$.

In a p = var system (Figure 6), the p_{M2} pressure in the cylinder discharge conduit is relatively low in the whole range of cylinder load F_M (and cylinder pressure drop Δp_M). In the tested system it was approximately $p_{M2} = 0.06\, p_n$. The low p_{M2} pressure makes the cylinder mechanical losses decrease. The tested unloaded cylinder ($F_M = 0$) required a lower pressure drop of a value $\Delta p_M|_{F_M=0} = 0.01 p_n$ (Figure 6), i.e. several times lower than the situation in the p = cte system.

The energy gains connected with the use of a variable pressure p = var directional throttling valve feeding system, in accordance with the expression $p_{P2} = p_2 + (k_5 + k_{10})\, p_n$ (Figure 5, 6, 7, 8,), obtained in comparison with a p = cte nominal pressure $p_{P2} = p_n$ feeding system, occur only when the pump discharge pressure p_{P2} is lower than the nominal pressure p_n. As soon as the pump discharge conduit pressure reaches the nominal pressure value $p_{P2} = p_n$, the further increase in the cylinder load F_M and associated increase in the pressure drop Δp_M in the cylinder is accompanied by a constant value $p_{P2} = p_n$. In consequence, the directional valve pressure drop Δp_{DE1} (and Δp_{DE2}) decreases to zero.

The pressure drop Δp_{DE1} and the Δp_{DE2} reduction cause in turn the reduction to zero of the flow intensity controlled by the directional throttling valve ($Q_M \to 0$) and the conduit flow resistance Δp_C. The cylinder pressure drop Δp_M may reach a value of the system nominal pressure $p_{P2} = p_n$ with the simultaneous reduction to zero of the cylinder piston rod speed v_M ($v_M = 0$).

In the $p_{P2} = p_n$ range of p = var system operation, the energy behaviour of the system is identical with the behaviour of a p = cte system.

Apart from the energy saving connected with a considerable reduction of structural pressure losses and also the saving from the reduction of mechanical losses in the cylinder, the use of a variable pressure p = var feeding of the proportional throttling valve in a constant capacity pump system allows the reduction of volumetric and mechanical losses in the pump and also the unloading of the pump driving motor by reducing the torque M_P required by the pump.

The pump volumetric efficiency η_{Pv} is given by the expression:

$$\eta_{Pv} = 1 - k_1\, \bar{p}_{P2}\,, \tag{21}$$

with $\bar{p}_{P2}$ in accordance with expression (12):

$$\bar{p}_{P2} = \frac{p_{P2}}{p_n} = k_{7.1} + (1 + k_{7.2})\, \bar{M}_M + k_{6.1}\, \bar{Q}_M + \\ + r_S\left(r_S\, k_{6.2}\bar{Q}_M + \frac{\Delta p_{DE2}}{p_n}\right) + k_5 + k_{10}\,,$$

where the dimensionless relation $\frac{\Delta p_{DE2}}{p_n}$ has a form given in (13), (14) or (15).

The $k_1\bar{p}_{P2}$ component in formula (21) presents a relative value of volumetric losses, which decreases with reducing the pump discharge pressure p_{P2} and simultaneously gives an increase in the η_{Pv} efficiency.

The pump mechanical efficiency η_{Pm} is described by the formula:

$$\eta_{Pm} = \frac{\bar{p}_{P2} + k_3\, \bar{Q}_P^2}{k_{4.1} + (1 + k_{4.2})\left(\bar{p}_{P2} + k_3\, \bar{Q}_P^2\right)}, \tag{22}$$

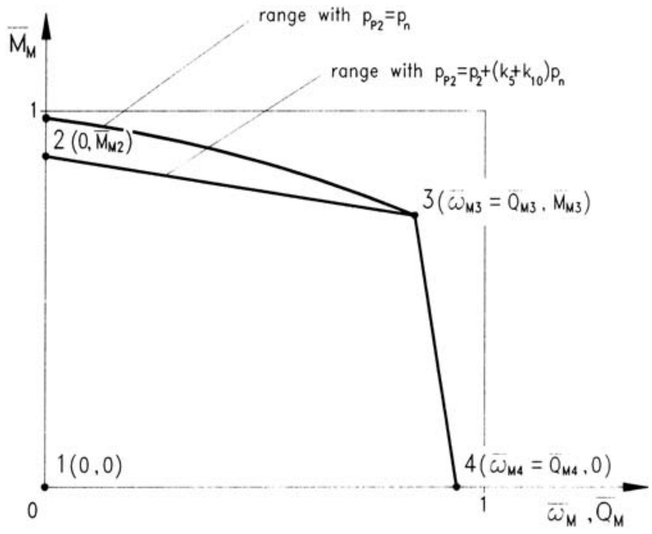


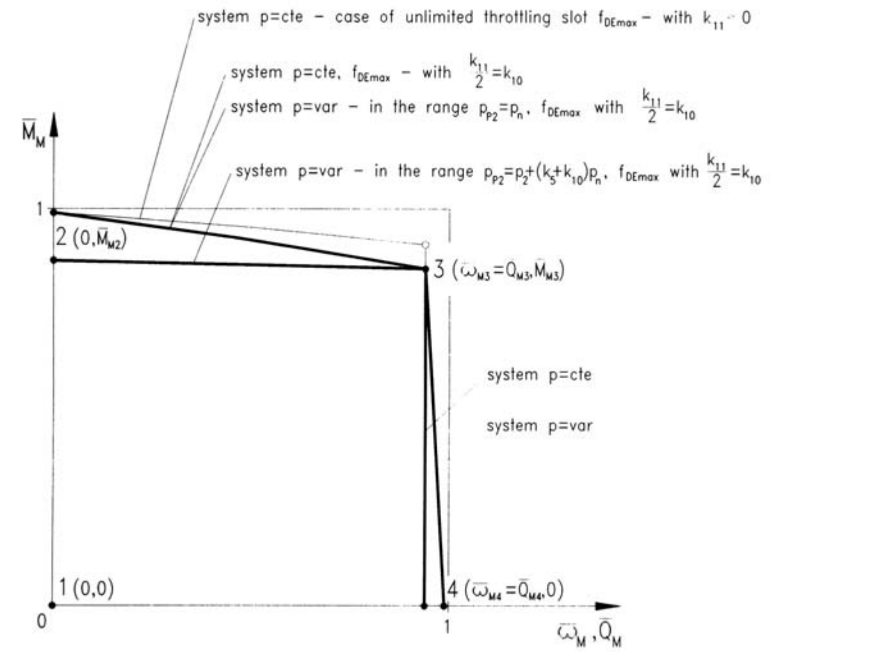
Figure 10 **The range of change of the $\bar{\omega}_M = \bar{Q}_M$ speed coefficient and $\bar{M}_M$ load coefficient in a system with proportional control of a cylinder fed by variable pressure system p = var; $\bar{\omega}_M = \bar{Q}_M$ as the intensity of internal leaks Q_{Mf} in the cylinder is considered as negligible – $Q_M = 0$ ($k_9 = 0$). Possibilities of feeding the directional proportional throttling valve:**
– variable capacity pump cooperating with the Load Sensing regulator controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder, i.e. $p_{P2} = p_2 + (k_5 + k_{10})p_n$,
– constant capacity pump cooperating with an overflow valve controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder, i.e. $p_{P2} = p_2 + (k_5 + k_{10})p_n$.

Slika 10 **Područje promjene $\bar{\omega}_M = \bar{Q}_M$ koeficijenta brzine i $\bar{M}_M$ koeficijenta opterećenja u sustavu sa razmjernom kontrolom punjenja cilindra sa promjenjivim tlakom sustava p = var; $\bar{\omega}_M = \bar{Q}_M$ pri čemu se intenzitet unutarnjeg propuštanja Q_{Mf} u cilindru smatra zanemarivim – $Q_M = 0$ ($k_9 = 0$). Mogućnost napajanja izravno proporcionalnog prigušnog ventila:**
– pumpa promjenjivog kapaciteta sa *Load Sensing* regulatorom kontroliranim s tlakom p_2 na izlazu proporcionalnog ventila u cilindar, odnosno $p_{P2} = p_2 + (k_5 + k_{10})p_n$,
– pumpa stalnog kapaciteta u sudjelovanju s jednim preljevnim ventilom kontroliranim s tlakom p_2 na izlazu proporcionalnog ventila u cilindar, odnosno $p_{P2} = p_2 + (k_5 + k_{10})p_n$.

with $\bar{p}_{P2}$ in accordance with expression (12) and $\bar{Q}_P$ given by:

$$\bar{Q}_P = \frac{Q_P}{Q_{Pt}} = 1 - \left[\begin{array}{l} k_{7.1} + (1 + k_{7.2})\, \bar{M}_M + k_{6.1}\bar{Q}_M + \\ + r_S\left(r_S\, k_{6.2}\bar{Q}_M + \frac{\Delta p_{DE2}}{p_n}\right) + k_5 + k_{10} \end{array}\right](k_1 + k_2). \tag{23}$$

The $k_{4.2}\left(\bar{p}_{P2} + k_3\, \bar{Q}_P^2\right)$ component of the formula (22) denominator presents an increase of mechanical losses in the pump connected with the pressure increase Δp_{Pi} in the pump working chambers and the increase in p_{P2}. This component decreases with the reduction in the p_{P2} pressure. The reduction in the p_{P2} pressure and the $k_{4.2}\left(\bar{p}_{P2} + k_3 \bar{Q}_{P2}^2\right) p_n$ component reduces the torque



system p=cte – case of unlimited throttling slot f_{DEmax} with $k_{11} \approx 0$:

$$\bar{M}_{M2} = \frac{1-k_{7.1}}{1+k_{7.2}};\ \bar{\omega}_{M3} = (1-k_1)(1-k_2);\ \bar{M}_{M3} = \frac{1-k_{7.1}-k_5-k_6}{1+k_{7.2}};\ \bar{\omega}_{M4} = (1-k_1)(1-k_2)$$

system p=cte: $\bar{M}_{M2} = \frac{1-k_{7.1}}{1+k_{7.2}};\ \bar{\omega}_{M3} = (1-k_1)(1-k_2);\ \bar{M}_{M3} = \frac{1-k_{7.1}-k_5-k_6-2k_{10}}{1+k_{7.2}};\ \bar{\omega}_{M4} = (1-k_1)(1-k_2)$

system p=var – in the range $p_{P2}=p_n$:

$$\bar{M}_{M2} = \frac{1-k_{7.1}}{1+k_{7.2}};\ \bar{\omega}_{M3} = 1-(k_1+k_2);\ \bar{M}_{M3} = \frac{1-k_{7.1}-k_5-k_6-2k_{10}}{1+k_{7.2}};\ \bar{\omega}_{M4} = \frac{1-(k_1+k_2)[k_{7.1}+2(k_5+k_{10})]}{1+(k_1+k_2)(k_6-k_5)}$$

system p=var – in the range $p_{P2}=p_2+(k_5+k_{10})p_n$:

$$\bar{M}_{M2} = \frac{1-k_{7.1}-2(k_5+k_{10})}{1+k_{7.2}};\ \bar{\omega}_{M3} = 1-(k_1+k_2);\ \bar{M}_{M3} = \frac{1-k_{7.1}-k_5-k_6-2k_{10}}{1+k_{7.2}};\ \bar{\omega}_{M4} = \frac{1-(k_1+k_2)[k_{7.1}+2(k_5+k_{10})]}{1+(k_1+k_2)(k_6-k_5)}$$

Figure 11 **The range of $\bar{Q}_M$ flow intensity coefficient, $\bar{\omega}_M$ speed coefficient and $\bar{M}_M$ load coefficient of the cylinder in an individual system with directional proportional valve of a maximum throttling slot cross section f_{DEmax} determined by the pressure drop Δp_{DE} coefficient $k_{11}/2 = k_{10}$ in order to obtain the flow intensity equal to the theoretical pump capacity Q_{Pt}. System p =cte: feeding of the directional proportional throttling valve with $\Delta p_{DE2} = \Delta p_{DE1}$ with:**
– constant capacity pump cooperating with an overflow valve of a = 0 coefficient,
– variable capacity pump cooperating with a pressure controller.
System p = var: feeding of the directional proportional throttling valve with $\Delta p_{DE2} = \Delta p_{DE1}$ by:
– variable capacity pump cooperating with a Load Sensing regulator controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder, i.e. $p_{P2} = p_2 + (k_5 + k_{10})p_n$,
– constant capacity pump co-operating with an overflow valve controlled by the p_2 pressure at the outlet from the proportional valve to the cylinder, i.e. $p_{P2} = p_2 + (k_5 + k_{10})p_n$.

Slika 11 **Područje $\bar{Q}_M$ koeficijenta intenziteta protoka, $\bar{\omega}_M$ koeficijenta brzine i $\bar{M}_M$ koeficijenta opterećenja cilindra u pojedinačnom sustavu s izravno proporcionalnim ventilom s maksimalnom površinom poprečnog presjeka prigušnog otvora f_{DEmax} određenog prema koeficijentu $k_{11}/2 = k_{10}$ pada tlaka Δp_{DE} da bi se dobio intenzitet protoka jednak teoretskom kapacitetu pumpe Q_{Pt}. Sistem p =cte: napajanje izravno proporcionalnog prigušnog ventila sa $\Delta p_{DE2} = \Delta p_{DE1}$ gdje su:**
– pumpa stalnog kapaciteta u sudjelovanju s jednim preljevnim ventilom uz koeficijent a = 0,
– pumpa promjenjivog kapaciteta u sudjelovanju s regulatorom tlaka.
Sistem p = var: napajanje izravno proporcionalnog prigušnog ventila sa $\Delta p_{DE2} = \Delta p_{DE1}$ gdje su:
– pumpa promjenjivog kapaciteta u sudjelovanju s *Load Sensing* regulatorom kontroliranim s tlakom p_2 na izlazu proporcionalnog ventila u cilindar, odnosno $p_{P2} = p_2 + (k_5 + k_{10})p_n$,
– pumpa stalnog kapaciteta u sudjelovanju s jednim preljevnim ventilom kontroliranim s tlakom p_2 na izlazu proporcionalnog ventila u cilindar, odnosno $p_{P2} = p_2 + (k_5 + k_{10})p_n$.

required by the pump. However, the mechanical efficiency of the pump decreases.

The reduction in the p_{P2} pressure in the pump discharge conduit, in the reduced cylinder load F_M zone, as an effect of the reduction of mechanical losses F_{Mm} in the cylinder, the reduction of structural pressure losses (the sum $\Delta p_{DE} = \Delta p_{DE1} + \Delta p_{DE2}$ of the pressure drop in the directional proportional throttling valve) allows the reduction of mechanical losses in the pump, the reduction of the shaft moment M_P and the increase of the rotational speed n_P of the pump.

The increase of n_P and the decrease of volumetric losses in the pump allow the increase in its capacity in accordance with the formula:

$$Q_P = Q_{Pt}\left\{1-\left[\begin{array}{l} k_{7.1}+\left(1+k_{7.2}\right)\bar{M}_M + k_{6.1}\bar{Q}_M + \\ +r_S\left(r_S\ k_{6.2}\bar{Q}_M + \dfrac{\Delta p_{DE2}}{p_n}\right)+k_5+k_{10} \end{array}\right]\left(k_1+k_2\right)\right\}, \qquad (24)$$

where $\frac{\Delta p_{DE2}}{p_n}$ ratio is described by expressions (13), (14) or (15).

The increase in Q_P allows the increase in the range of $\bar{\omega}_M = \bar{Q}_M$ cylinder speed coefficient, i.e. the system operation range (Figure 10) compared with the p = cte feeding system.

Figure 11 presents summary information on the ranges ($\bar{\omega}_M = \bar{Q}_M$, $\bar{M}_M$) of change of work parameters of the system with directional proportional throttling valve fed in a constant pressure system p = cte or a variable pressure system p = var.

5 Examples of laboratory verification of energy saving in the system with a constant capacity pump in a variable pressure system – p = var, in comparison with a constant pressure system p = cte

Grzegorz Skorek in his currently prepared doctor thesis [6] verifies experimentally the energy loss [1, 3, 8, 9, 10] and energy efficiency descriptions of the linear hydraulic motor (cylinder) system controlled by a proportional directional control valve and fed by a constant capacity pump in:

– constant pressure (c = cte) system – Figure 2,
– variable pressure (p = var) system – Figure 5.

The following components were used in the tested systems:

– axial piston pump with displaceable rotor HYDROMATIC type A7.VSO.58DR, operating with fixed theoretical capacity $Q_{Pt} = 0.000805\ m^3s^{-1}$ ($48.30\ dm^3min^{-1}$),
– directional proportional control valve, REXROTH type 4WRA10E60-21/G24N9K4, with identical throttling slots $f_{DE1} = f_{DE2}$,
– double piston cylinder HYDROSTER type CD-63/36x500, piston diameter D = 63 mm and piston rod diameter d = 36 mm,
– indirect operation overflow valve REXROTH type DBW10A3-52/315XU GE 62 4N9K4,
– controlled overflow valve REXROTH type ZDC10PT-23/XM (only in the variable pressure – p = var system).

The nominal pressure of the tested systems was $p_n = 16$ MPa, the hydraulic oil Total Azola 46 was used with kinematics viscosity of $\upsilon = 35\ mm^2s^{-1}$ (at the temperature $\vartheta = 43°C$) and volumetric mass of $\rho = 873.3\ kgm^{-3}$.

6 Conclusions

The test results, shown in Figures 12-18, allow a comparison of the energy loss power values, expressed in Watts [W], in the p = cte and p = var system elements. The following conclusions may be drawn:

1. The most significant reduction of energy losses, when a p = cte system is replaced by a p = var system, is obtained in the case of the structural pressure loss power ΔP_{stp} (Figure 12) in the proportional directional control valve. With the cylinder load coefficient $\overline{M}_M = 0$ and speed coefficient $\overline{\omega}_M = 0.875$, the loss power is reduced from approx. 9800 W to approx. 1800 W, i.e. by 7.5 times. The pressure loss power ΔP_{stp} in both systems equalizes in the maximum cylinder load area (maximum $\overline{M}_M$ values), i.e. in the area where the p = var system begins to operate as a p = cte system. The pressure structural loss power ΔP_{stp} in both systems is then relatively small – below 2300 W.
2. The volumetric structural loss power ΔP_{stv} (Figure 13), occurring in the overflow valve (p = cte system) or in the controlled overflow valve and the overflow valve (p = var system), decreases also when the p = cte system is replaced by the p = var system. But the power reduction is not as significant as in the case of pressure structural loss power ΔP_{stp}.
With the cylinder coefficient $\overline{M}_M = 0$ and speed coefficient $\overline{\omega}_M = 0.063$, the volumetric loss power ΔP_{stv} is reduced from ca. 12000 W to approx. 2400 W, ie. 5 fold. The volumetric loss power ΔP_{stv} in both systems equalizes in the cylinder maximum load area (maximum $\overline{M}_M$ values, i.e. in the area of the p = var system operating as a p = cte system). However, the same volumetric loss power ΔP_{stv} in both systems is at its maximum – it reaches 12000 W at $\overline{\omega}_M = 0.063$.

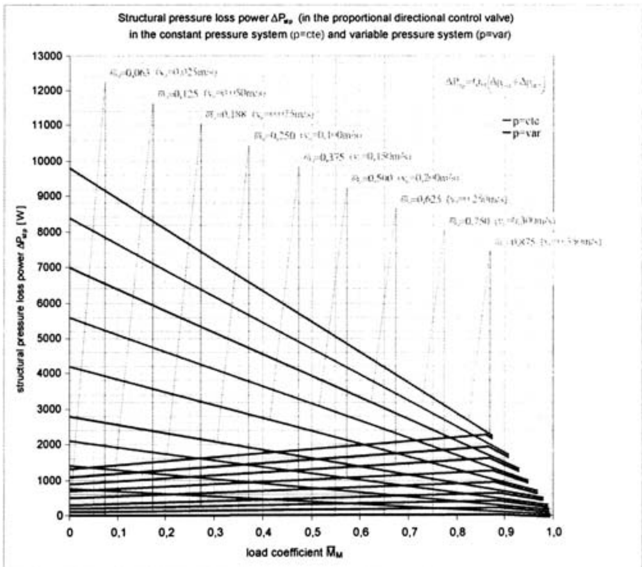


Figure 12 **Relation between the structural pressure loss power ΔP_{stp} (in the directional proportional control valve), in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\overline{M}_M$ with different speed coefficients $\overline{\omega}_M$ of the cylinder [6]**

Slika 12 **Odnos snage ΔP_{stp} strukturnih gubitaka tlaka (u izravno proporcionalnom regulacionom ventilu), u sustavu konstantnog tlaka (p = cte) i sustavu promjenjivog tlaka (p = var), i koeficijenta opterećenja $\overline{M}_M$ pri različitim koeficijentima brzine $\overline{\omega}_M$ cilindra [6]**

3. The mechanical loss power ΔP_{Mm} in the cylinder (Figure 14) decreases when the p = cte system is replaced by a p = var system. With $\overline{M}_M = 0$ and $\overline{\omega}_M = 0.875$, the loss power is reduced from aprox. 350 W to approx. 84 W, i.e. by 4.2 times. Mechanical loss power ΔP_{Mm} in the cylinder equalizes in both systems in the maximum cylinder load area (maximum $\overline{M}_M$ values) i.e. where the p = var system begins to operate as a p = cte system. The mechanical loss value ΔP_{Mm} in the cylinder is then relatively small – below 100 W.
4. The volumetric loss power ΔP_{Pv} in the pump (Figure 15) decreases when the p = cte system is replaced by a p = var system. The loss power ΔP_{Pv} does not depend on the cylinder speed coefficient $\overline{\omega}_M$ value, but it depends on the cylinder load coefficient $\overline{M}_M$. With the $\overline{M}_M = 0$, the volumetric loss power ΔP_{Pv} in the pump decreases from approx. 80 W (p = cte system) to approx. 5 W (p = var system), i.e. by 16 times. Volumetric loss power ΔP_{Pv} in both systems equalizes in the maximum cylinder load area (maximum $\overline{M}_M$ values) i.e. where the p = var system begins to operate as a p = cte system. The volumetric loss power ΔP_{Pv} in pump is then maximal – 80 W.

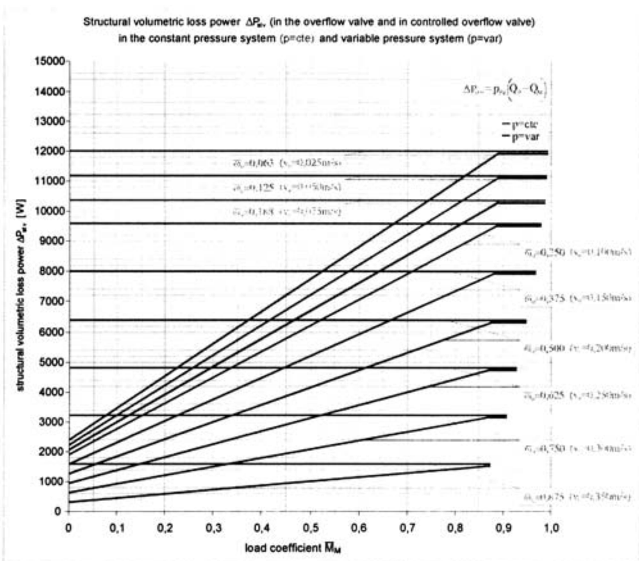


Figure 13 **Relation between the structural volumetric loss power ΔP_{stv} (in the overflow valve and in controlled overflow valve), in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\bar{M}_M$ with different speed coefficients $\bar{\omega}_M$ of the cylinder [6]**

Slika 13 **Odnos između strukturnog volumetrijskog gubitka snage ΔP_{stv} (u rasteretnom ventilu i reguliranom rasteretnom ventilu), u sustavu konstantnog tlaka (p = cte) i sustavu promjenjivog tlaka (p = var), i koeficijenta opterećenja $\bar{M}_M$ pri različitim koeficijentima brzine $\bar{\omega}_M$ cilindra [6]**

5. The mechanical loss power ΔP_{Pm} in the pump (Figure 16) decreases when the p = cte system is replaced by a p = var system. The loss power ΔP_{Pm} does not depend on the cylinder speed coefficient $\bar{\omega}_M$ value, but it depends on the cylinder load coefficient $\bar{M}_M$. With the $\bar{M}_M = 0$, the mechanical loss power ΔP_{Pm} in the pump is reduced from approx. 193 W (in p = cte system) to approx. 157 W (in p = var system) i.e. by 1.2 times. The mechanical loss power ΔP_{Pm} in both systems equalizes in the maximum cylinder load area (maximum $\bar{M}_M$ values). i.e. where the p = var system begins to operate as a p = cte system. The mechanical loss power ΔP_{Pm} in the pump is then maximal – 193 W.
6. The pressure loss power ΔP_{Pp} in the pump (Figure 17) increases slightly after replacing the p = cte system with the p = var system (because of reducing the volumetric losses in the pump, i.e. increasing the pump capacity). The loss power ΔP_{Pp} does not depend on the cylinder speed coefficient $\bar{\omega}_M$ value. In the p = cte system, the ΔP_{Pp} power is also independent of the cylinder load coefficient $\bar{M}_M$, and is constant at approx. 27 W. In the p = var system it decreases from the ΔP_{Pp} value of approx. 32 W ($\bar{M}_M = 0$) to $\Delta P_{Pp} = 27$ W (in the cylinder maximum load area – maximum $\bar{M}_M$ values) i.e. where the p = var system begins to operate as a p = cte system.
7. The energy loss power ΔP_C in the p = cte system conduits and that in the p = var system conduits (Figure 18) are identical. They do not depend on the cylinder load coefficient $\bar{M}_M$ values, but on the cylinder speed coefficient $\bar{\omega}_M$ values. The

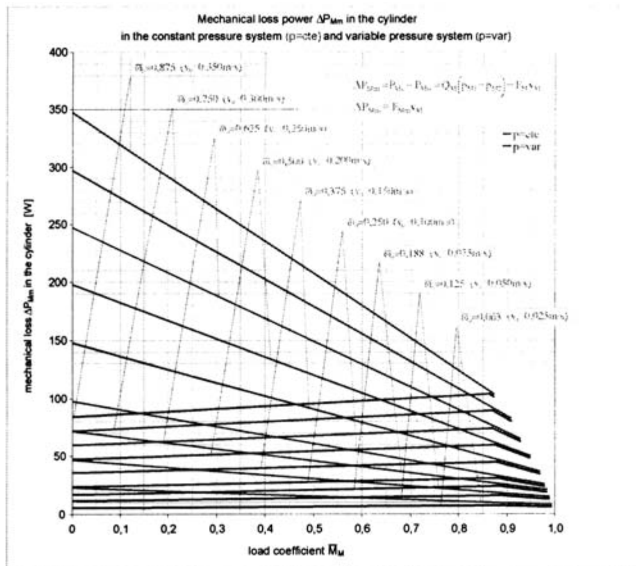
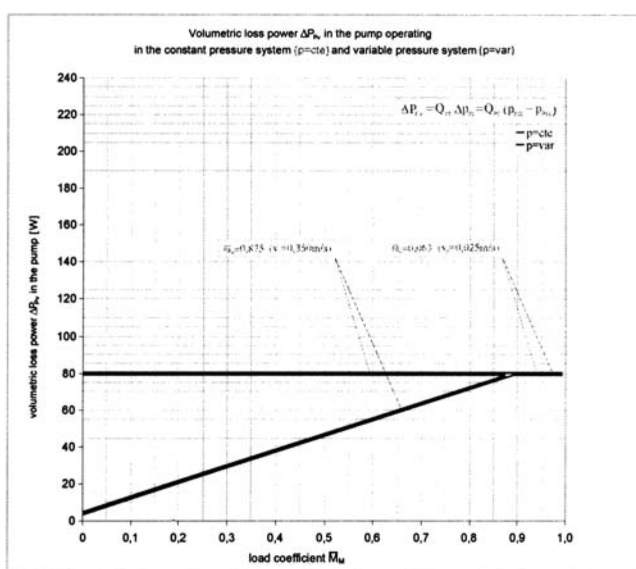


Figure 14 **Relation between the mechanical loss power ΔP_{Mm} in the cylinder, in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\bar{M}_M$ with different speed coefficients $\bar{\omega}_M$ of the cylinder [6]**

Slika 14 **Odnos snage ΔP_{Mm} mehaničkih gubitaka u cilindru, u sustavu konstantnog tlaka (p = cte) i sustavu promjenjivog tlaka (p = var), i koeficijenta opterećenja $\bar{M}_M$ pri različitim koeficijentima brzine $\bar{\omega}_M$ cilindra [6]**

Figure 15. **Relation between the volumetric loss power ΔP_{Pv} in the pump, operating in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\bar{M}_M$ with different speed $\bar{\omega}_M$ coefficients of the cylinder [6]**

Slika 15 **Odnos snage ΔP_{Pv} volumetričkih gubitaka u pumpi, u režimu rada sustava konstantnog tlaka (p = cte) i sustava promjenjivog tlaka (p = var), i koeficijenta opterećenja $\bar{M}_M$ pri različitim koeficijentima brzine $\bar{\omega}_M$ cilindra [6]**



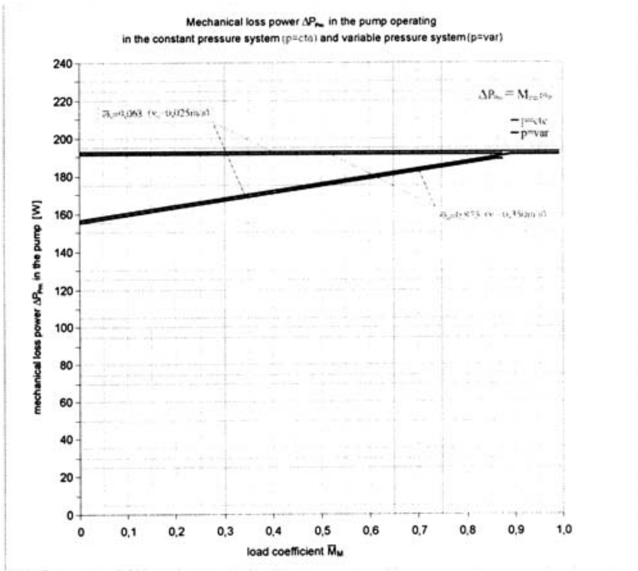
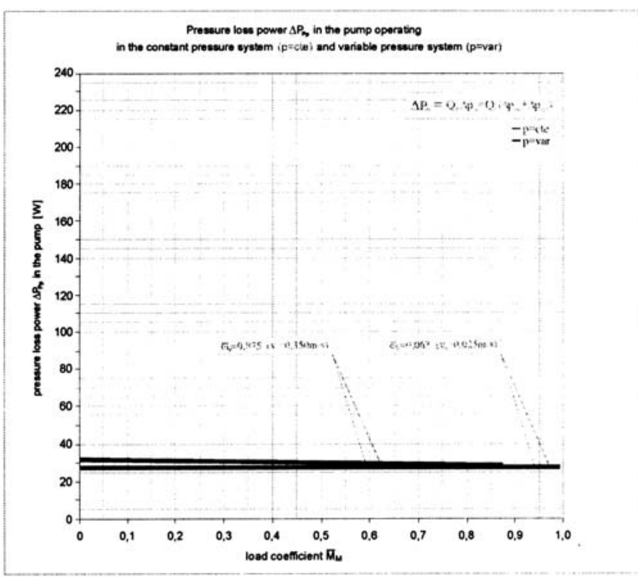


Figure 16 **Relation between the mechanical loss power ΔP_{Pm} in the pump, operating in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\overline{M}_M$ with different speed coefficients $\overline{\omega}_M$ of the cylinder [6]**

Slika 16 **Odnos snage ΔP_{Pm} mehaničkih gubitaka u pumpi, u režimu rada sustava konstantnog tlaka (p = cte) i sustava promjenjivog tlaka (p = var), i koeficijenta opterećenja $\overline{M}_M$ pri različitim koeficijentima brzine $\overline{\omega}_M$ cilindra [6]**

Figure 17 **Relation between the pressure loss power ΔP_{Pp} in the pump, operating in the constant pressure system (p = cte) and variable pressure system (p = var), and the load coefficient $\overline{M}_M$ with different speed coefficients $\overline{\omega}_M$ of the cylinder [6]**

Slika 17 **Odnos snage ΔP_{Pp} mehaničkih gubitaka tlaka u pumpi, u režimu rada sustava konstantnog tlaka (p = cte) i sustava promjenjivog tlaka (p = var), i koeficijenta opterećenja $\overline{M}_M$ pri različitim koeficijentima brzine $\overline{\omega}_M$ cilindra [6]**



energy loss power ΔP_C in the system conduits increases from $\Delta P_C = 0$ (with $\overline{\omega}_M = 0$) to $\Delta P_C = 498$ W (with $\overline{\omega}_M = 0.94$ in the constant pressure p = cte system) and to $\Delta P_C = 550$ W (with $\overline{\omega}_M = 0.98$ in the variable pressure p = var system).

8. The laboratory verification results [6] confirm with great accuracy the theoretical considerations and mathematical simulation descriptions of the energy losses in the elements of a hydraulic servo mechanism or the proportional directional control valve systems operating in the p = cte and p = var feed systems [1, 3, 8, 9, 10].

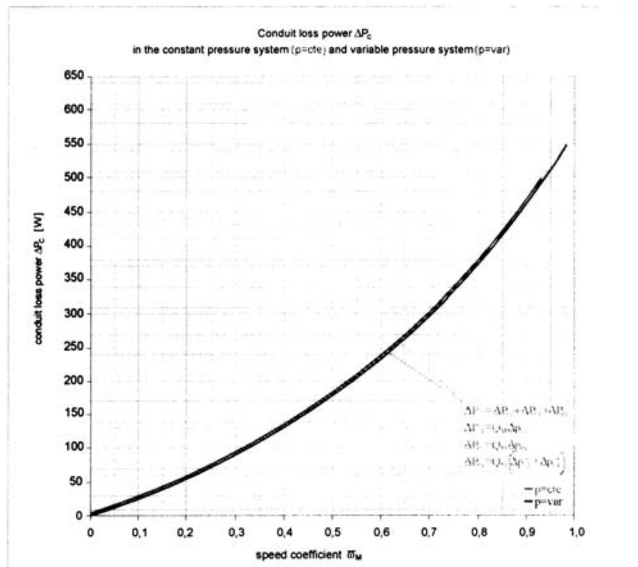


Figure 18 **Relation between the conduit loss power ΔP_C, in the constant pressure system (p = cte) and variable pressure system (p = var), and the cylinder speed coefficient $\overline{\omega}_M$ [6]**

Slika 18 **Odnos snage ΔP_C gubitaka u vodovima, u sustavu konstantnog tlaka (p = cte) i sustavu promjenjivog tlaka (p = var), i koeficijenta opterećenja $\overline{M}_M$ te koeficijenta brzine $\overline{\omega}_M$ cilindra [6]**

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