POLISH MARITIME RESEARCH 4(58) 2008 Vol 15; pp. 21-29
10.2478/v10012-007-0093-y

Graphical presentation of the power of energy losses and power developed in the elements of hydrostatic drive and control system
Part II
Rotational hydraulic motor speed parallel throttling control and volumetric control systems

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ABSTRACT

Paper presents graphical interpretation of the power of energy losses in the hydrostatic drive and control system elements and also of the power developed by those elements. An individual system fed by a constant capacity pump, where rotational hydraulic motor speed control is effected by a parallel throttling control system, is analyzed and also a system with the rotational hydraulic motor speed volumetric control by a variable capacity pump, an individual system with a rotational hydraulic motor volumetric speed control by means of a simultaneous change of the pump capacity per one revolution and change of the motor capacity per one revolution, the system operating at the constant pressure in the pump discharge conduit equal to the nominal pressure of the system: $p_{P2} = p_n$ and central system (with situated in parallel and simultaneously operating motors) with volumetric speed control of each rotational hydraulic motor by a motor secondary circuit assembly, the system fed by a pump with variable capacity per one shaft revolution fitted with pressure regulator $p_{P2} = p_n$.

Keywords: hydrostatic drive and control system, power of energy losses, energy efficiency

SYSTEM OF THE MOTOR SPEED PARALLEL THROTTLING CONTROL

Fig. 14 presents the areas of power of energy losses occurring in the elements of **an individual system fed by a constant capacity pump, where rotational hydraulic motor speed control is effected by a parallel throttling control system**. The parallel throttling control assembly may have a form of a set throttling valve (Fig. 15a) or a set 2 – way regulator (Fig. 15b) placed at the pump discharge conduit branch.

The use of a constant capacity pump (Q_P = cte) in a motor speed parallel throttling control system is a necessary condition from the point of view of achieving a relatively precise change of the flow intensity Q_M towards the motor and the change of motor speed ω_M (n_M) by means of the change of flow intensity Q_0 of the stream controlled by a throttling valve (2 – way flow regulator) and directed to the tank ($Q_P - Q_0 = Q_M$).

The current useful power $P_{Mu} = M_M\ \omega_M$ of the hydraulic motor (independent of the used motor speed control structure), required by the motor driven device (and the same as in the systems shown in Fig. 2, 5, 8 and 11 - PMR 03/2008 Part. I), influences in a different way (than in the series throttling control systems) the structural losses generated in the system.

The current small hydraulic motor loading torque M_M has a direct impact on the lowered level of the pump discharge pressure p_{P2}, the pressure resulting from the decrease Δp_M of pressure in the motor and losses Δp_C of pressure in the system conduits ($p_{P2} = \Delta p_M + \Delta p_C$). In effect, the power ΔP_{stp} of structural pressure losses is reduced to zero ($\Delta P_{stp} = 0$) and the power ΔP_{stv} of structural volumetric losses is clearly smaller than in a system with constant capacity pump feeding the motor speed series throttling control assembly ($\Delta P_{stv} = p_{P2}\ (Q_P - Q_M)$).

In a rotational hydraulic motor, operating in the conditions of parallel throttling speed control, occur power ΔP_{Mm} of mechanical losses, power ΔP_{Mv} of volumetric losses and power ΔP_{Mp} of pressure losses of the values close to powers ΔP_{Mm}, ΔP_{Mv} and ΔP_{Mp}, which would occur in such a motor with the series throttling speed control (with a tendency to slight decrease of the power ΔP_{Mm} of mechanical losses and the power ΔP_{Mv} of volumetric losses, due to decrease of pressure p_{M2} at the motor outlet in the parallel throttling control system).

Power $\Delta P_C = p_C\ Q_M$ of pressure losses in the conduits of the discussed system with parallel throttling control is equal to the power ΔP_C of pressure losses in the above analyzed four systems with the series throttling control.

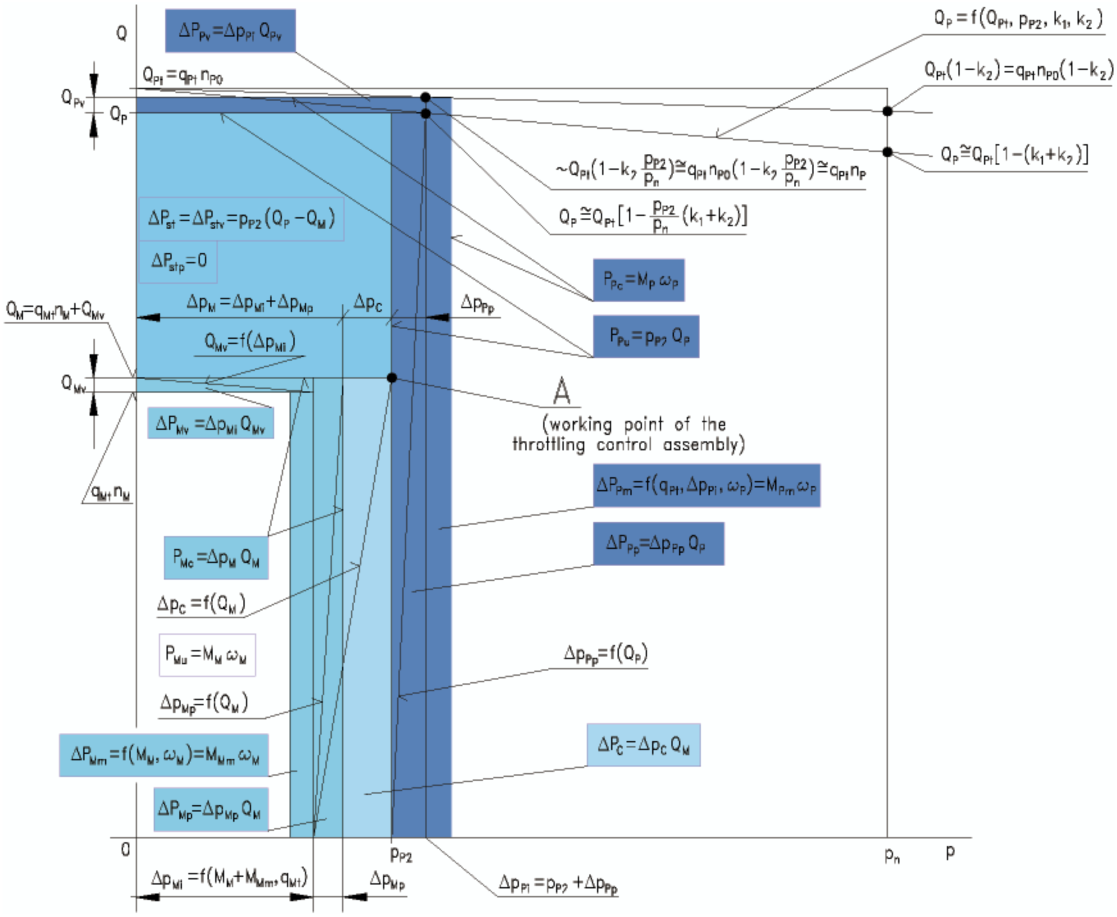

Fig. 14. *Graphical interpretation of power losses in a hydrostatic drive and control system elements. Individual system with rotational hydraulic motor speed parallel throttling control fed by a constant capacity pump; the parallel throttling control assembly in the form of: – set throttling valve, – set two-way flow regulator*

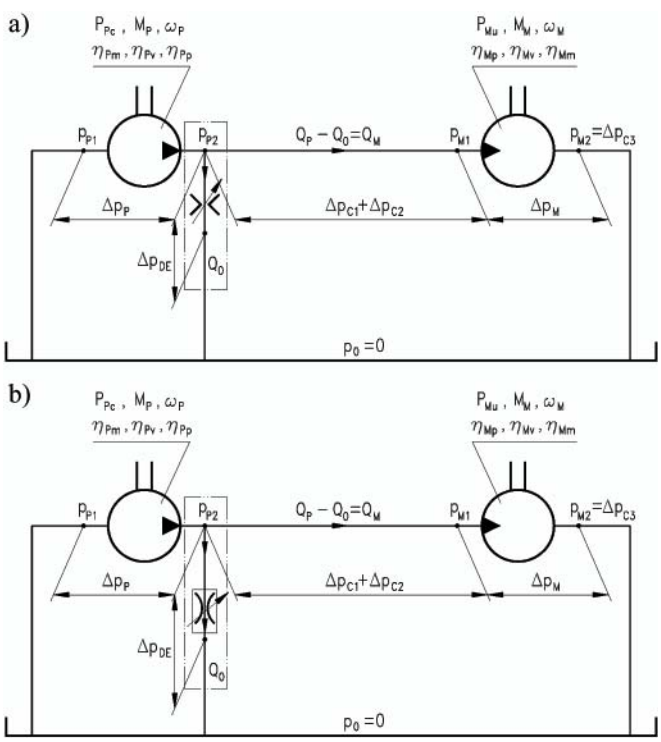

Fig. 15. *Individual system with the rotational hydraulic motor speed parallel throttling control; the throttling control assembly in the form of:* ***a)*** *set throttling valve,* ***b)*** *set two – way flow regulator*

In the constant capacity pump, due to its operation with the discharge pressure p_{P2} lower than in the series throttling control systems, a slight increase of the power $\Delta P_{Pp} = \Delta p_{Pp}\, Q_P$ of pressure losses occurs, and at the same time clear decrease of the power $\Delta P_{Pv} = \Delta p_{Pi}\, Q_{Pv}$ of volumetric losses and also decrease of the power $\Delta P_{Mp} = M_{Pm}\, \omega_P$ of mechanical losses.

When the hydraulic motor is loaded with small torque M_M, as an effect of decreased power of losses in system elements, the power $P_{Pc} = M_P\, \omega_P$ absorbed by the pump from the drive (electric or internal combustion) motor is also decreased compared with power P_{Pc} absorbed by a constant capacity pump used in the motor speed series throttling control system. With the unchanged hydraulic motor useful power $P_{Mu} = M_M\, \omega_M$, this increases the energy efficiency η of the whole system compared with the efficiency η of a series throttling control system fed by a constant capacity pump.

The discussed structure of a hydraulic system, with a constant capacity pump and with hydraulic motor parallel throttling speed control, may achieve, during the operation with maximum speed ω_{Mmax} (n_{Mmax}) of the controlled motor and in the full range $0 \le M_M \le M_{Mmax}$ of its load, energy efficiency η equal to the total efficiency η of the system with volumetric control of the hydraulic motor speed (with a variable capacity pump).

SYSTEM OF THE MOTOR SPEED VOLUMETRIC CONTROL BY A VARIABLE CAPACITY PUMP

Fig. 16 interprets the areas of power of energy losses in the elements of **a system with the rotational hydraulic motor speed volumetric control by a variable capacity pump**.

The hydraulic motor (with a constant capacity q_{Mt} per one shaft revolution) shaft speed ω_M (n_M) volumetric control by the

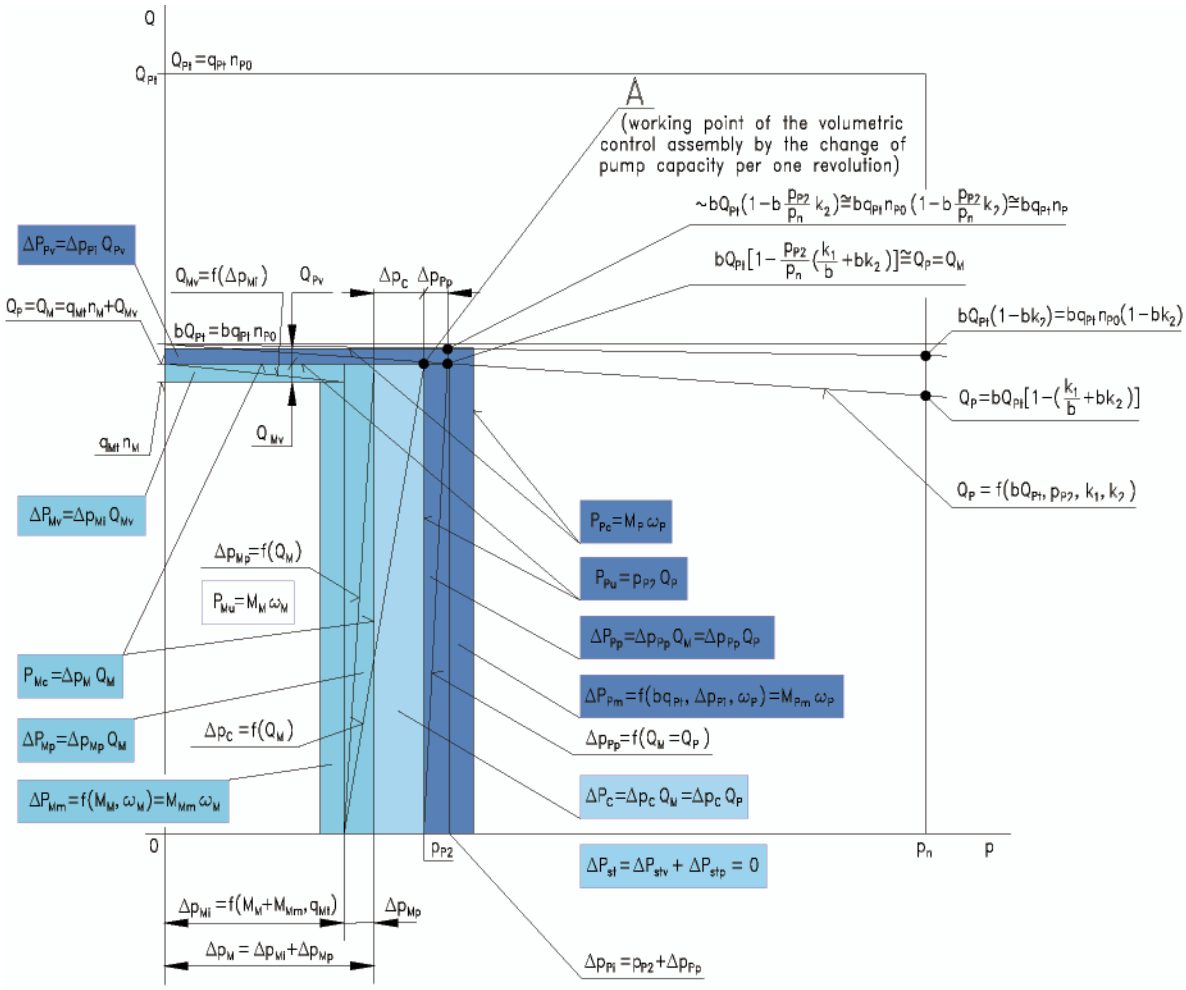

***Fig. 16.** Graphical interpretation of power losses in a hydrostatic drive and control system. Individual system with rotational hydraulic motor (with constant capacity per one revolution) speed volumetric control by a pump with variable capacity per one revolution*

change of its capacity Q_M required by the motor, allowing to achieve the speed ω_M (n_M) imposed by the motor driven device, is effected by the change of capacity of the motor feeding pump (Fig. 17).

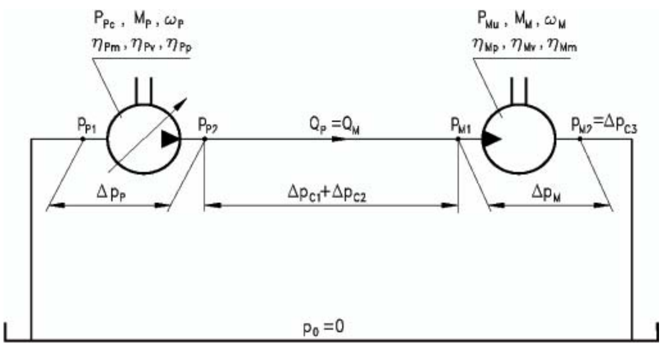

***Fig. 17.** Individual system with hydraulic motor speed volumetric control by a variable capacity pump*

The imposed motor speed ω_M (n_M) is obtained by the corresponding pump capacity Q_P ($Q_P = Q_M$). Therefore, eliminated are the structural volumetric losses ($\Delta P_{stv} = 0$) and structural pressure losses ($\Delta P_{stp} = 0$) connected with the hydraulic motor speed throttling control assembly structure.

The current pump discharge pressure p_{P2} results from the sum of current pressure decrease Δp_M required by the driven hydraulic motor and pressure losses Δp_C in the conduit between the pump and motor and in the motor outlet conduit. The maximum limit pressure value p_{P2max} in the pump discharge conduit is determined by the safety valve (closed during the normal pump operation, i.e. in the $0 < p_{P2} \le p_n$ pressure range) whose opening pressure is higher than the nominal pressure p_n of the system (maximum pressure of continuous system operation).

In a rotational hydraulic motor, operating in a speed ω_M (n_M) control system by a variable capacity pump, occur the same values of power ΔP_{Mm} of mechanical losses, power ΔP_{Mv} of volumetric losses and power ΔP_{Mp} of pressure losses as in the system where the change of speed ω_M (n_M) is controlled by series throttling control assembly and the system is fed by a constant capacity pump. Power ΔP_{Mm} of mechanical losses and ΔP_{Mv} of volumetric losses in the motor are slightly lower than power of those losses that would occur in a motor with the series throttling control system, where the flow is throttled at the motor outlet (the motor outlet pressure p_{M2} is higher).

Power $\Delta P_C = \Delta p_C\, Q_M$ of pressure losses in the conduits of the hydraulic motor speed volumetric system with a variable capacity pump is (with an unchanged motor shaft required speed ω_M (n_M)) equal to the power ΔP_C of losses in the motor speed series or parallel throttling control systems.

In a variable capacity pump feeding the hydraulic motor occurs (with the motor useful power $P_{Mu} = M_M\, \omega_M$ unchanged) decrease of power $\Delta P_{Pp} = \Delta p_{Pp}\, Q_P$ of pressure losses (due to decrease of Δp_{Pp} and Q_P), decrease of power $\Delta P_{Pv} = \Delta p_{Ppi}\, Q_{Pv}$ of volumetric losses (due to decrease of Δp_{Pi} and Q_{Pv}) and decrease of power $\Delta P_{Pm} = M_{Pm}\, \omega_P$ of mechanical losses (due to decrease of M_{Pm} and slight increase of ω_P).

In a volumetric motor (with a constant capacity per one revolution) speed control system (by a variable capacity pump), the sum of current (unchanged in relation to power P_{Mu} in the above discussed systems) value of hydraulic motor useful power $P_{Mu} = M_M \omega_M$ and of:

- ✶ power $\Delta P_{Mm} = M_{Mm} \omega_M$ of mechanical losses in the motor
- ✶ power $\Delta P_{Mv} = \Delta p_{Mi} Q_{Mv}$ of volumetric losses in the motor
- ✶ power $\Delta P_{Mp} = \Delta p_{Mp} Q_M$ of pressure losses in the motor
- ✶ power $\Delta P_C = \Delta p_C Q_M$ of pressure losses in the hydraulic system connecting conduits
- ✶ power $\Delta P_{Pp} = \Delta p_{Pp} Q_P$ of pressure losses in the pump
- ✶ power $\Delta P_{Pv} = \Delta p_{Pi} Q_{Pv}$ of volumetric losses in the pump
- ✶ power $\Delta P_{Pm} = M_{Pm} \omega_P$ of mechanical losses in the pump

decides that the power $P_{Pc} = M_P \omega_P$ required by the pump from the pump driving (electric or internal combustion) motor is smaller than power P_{Pc} absorbed by the pump in the above discussed systems with the hydraulic motor speed throttling control.

Energy efficiency $\eta = P_{Mu}/P_{Pc}$ of a system with volumetric control by a variable capacity pump is the highest efficiency η amongst the considered systems in the whole range of speed $0 \leq \omega_M \leq \omega_{Mmax}$ and hydraulic motor load $0 \leq M_M \leq M_{Mmax}$. However, it has to be noted that the energy advantage of a system with the hydraulic motor speed ω_M (n_M) control by a variable capacity pump over the systems with the motor speed throttling control decreases markedly when the motor speed ω_M approaches ω_{Mmax} and the motor load approaches M_{Mmax}.

SYSTEM OF THE MOTOR SPEED VOLUMETRIC CONTROL BY A VARIABLE CAPACITY PUMP AND A MOTOR WITH VARIABLE CAPACITY PER ONE REVOLUTION, SYSTEM OPERATING AT NOMINAL PRESSURE P_N

Fig. 18 illustrates the areas of power of energy losses in **an individual system with a rotational hydraulic motor volumetric speed control by means of a simultaneous change**

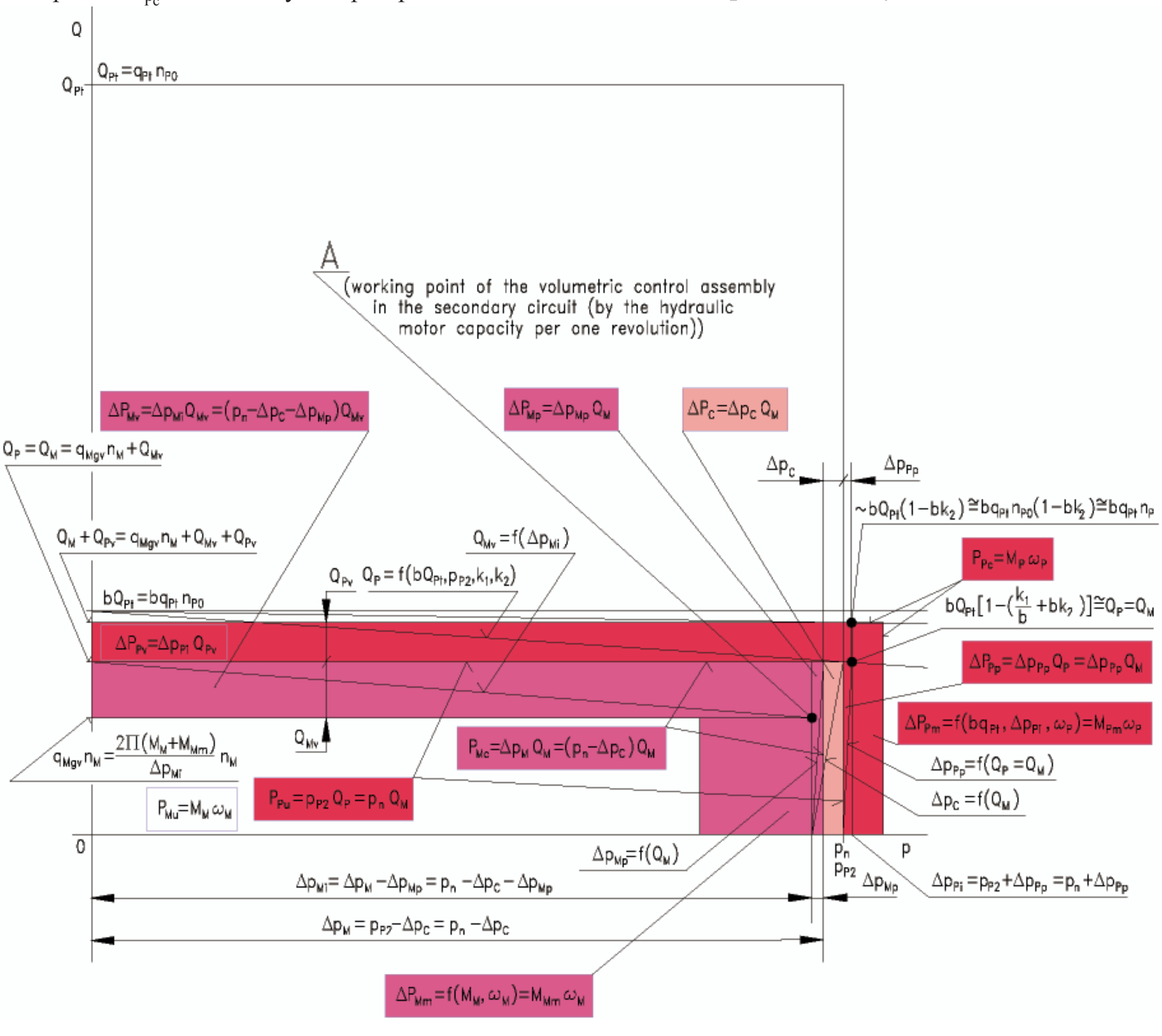

Fig. 18. *Graphical interpretation of power losses in a hydrostatic drive and control system elements:*

- *Individual system with the rotational hydraulic motor speed volumetric control by a pump with variable capacity per one revolution and a motor with variable capacity per one revolution operating at the nominal pressure $p_{P2} = cte \approx p_n$; capacity q_{Mgv} per one motor revolution determines, at a given loading M_M, the nominal pressure in the pump discharge (outlet) conduit: $p_{P2} \approx p_n$, and capacity q_{Pgv} per one pump revolution controls the motor rotational speed n_M*
- *Central system with the rotational hydraulic motor speed volumetric control in the secondary motor circuit, fed by a pump with a variable capacity per one revolution fitted with pressure regulator $p_{P2} = cte = p_n$ (Rexroth conception) – operation with one motor fed; (coefficient "a_1" of the increase of regulator controlled pump pressure: $a_1=0$)*

The Figure demonstrates the system with motor speed volumetric control by the so called secondary motor circuit (adjusting, by the change of capacity q_{Mgv} per one shaft revolution, the pressure decrease Δp_M in the motor to a value $\Delta p_M = p_n - \Delta p_C$, and at the same time setting the motor speed ω_M (n_M) required by the motor driven device); the system is fed by the pump (with variable capacity q_{Pgv} per one shaft revolution) fitted with pressure regulator $p_{P2} = p_n$.

of the pump capacity per one revolution and change of the motor capacity per one revolution, the system operating at the constant pressure in the pump discharge conduit equal to the nominal pressure of the system: $p_{P2} = p_n$.

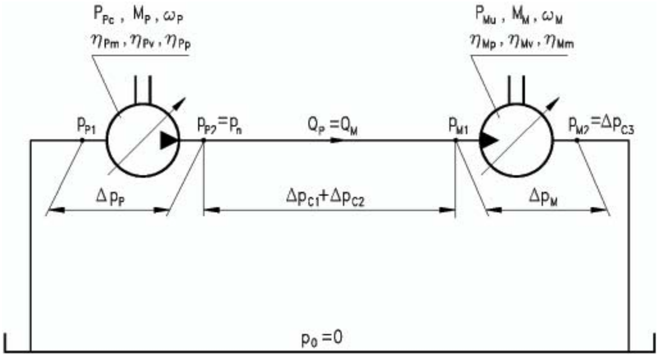

Fig. 19. *Individual system with the rotational hydraulic motor speed volumetric control by means of simul-teneous change of the pump capacity per one revolution and change of the motor capacity per one revolution; system operating at the constant pressure in the pump discharge conduit equal to the system nominal pressure: $p_{P2} = p_n$*

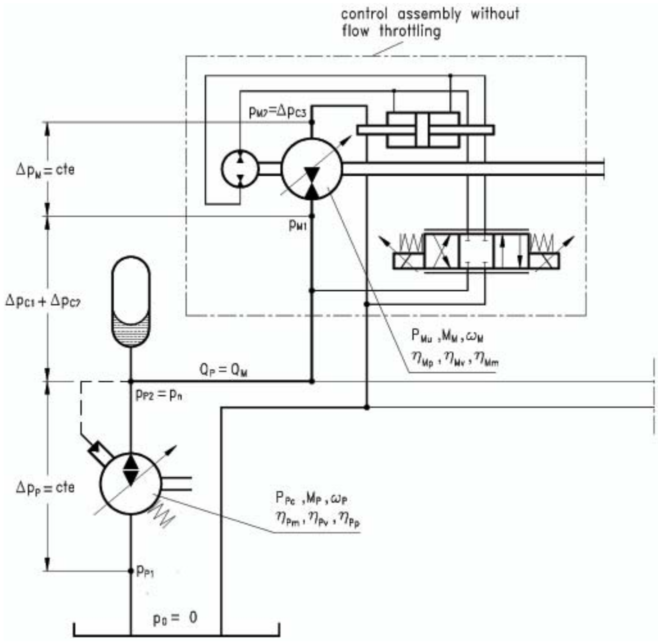

Fig. 20. *Central system, with rotational motors situated in parallel, with volumetric control of each motor speed by a secondary circuit assembly, system fed by a variable capacity pump cooperating with regulator in a constant pressure conditions: $p_{P2} = cte \approx p_n$ (conception of the Rexroth company) – during one motor operation*

This solution allows to use pump useful power P_{Pu} (resulting from a product of pump working pressure p_{P2} equal to the nominal system pressure p_n ($p_{P2} = p_n$) and maximum pump capacity Q_{Pmax} achieved at the coefficient $b = q_{Pgv}/q_{Pt} = 1$ of the change of the pump capacity per one revolution) also during the period of smaller hydraulic motor load (with the torque $M_M < M_{Mmax}$), by an increase of the motor speed ω_M above its nominal speed ω_{Mn} ($\omega_M > \omega_{Mn}$).

Nominal pressure in the pump discharge conduit: $p_{P2} = p_n$, at a given load M_M, is determined by the changing capacity q_{Mgv} per one motor shaft revolution, the motor speed ω_M (n_M) is controlled by the capacity q_{Pgv} per one pomp revolution.

The idea of feeding by one pump (of variable capacity q_{Pgv} per one revolution), at a constant pressure level $p_{P2} = p_n$ in its discharge conduit, of two or more simultaneously operating hydraulic motors (of variable capacity q_{Mgv} per one revolution) has been used by the Rexroth company in the conception of a hydraulic **central system (with situated in parallel and simultaneously operating motors) with volumetric speed control of each rotational hydraulic motor by a motor secondary circuit assembly, the system fed by a pump with variable capacity per one shaft revolution fitted with pressure regulator $p_{P2} = p_n$** (Fig. 20 – central system during one motor operation).

The speed ω_M (n_M) control of each of the simultaneously operating hydraulic motors in the central system by an own secondary circuit assembly consists in a solution where the capacity q_{Mgv} per one revolution adjusts the decrease of pressure Δp_M, required by the motor, to a value $\Delta p_M = p_n - \Delta p_C$ (i.e. to a value equal to the difference of pressure $p_{P2} = p_n$ in the pump discharge conduit and pressure drop Δp_C in the conduits of a system) and, at the same time, the motor speed ω_M (n_M), required by the motor driven device, is controlled by that assembly.

The use of the central system solution with parallelly situated rotational hydraulic motors, where the speed ω_M (n_M) of each motor (driving one of the simultaneously operating devices) is determined by its secondary circuit assembly (structurally connected with the motor), eliminates the need of setting the flow intensity Q_M, directed to each motor, by the throttling control assembly located on the branch to each motor and situated in series with the motor. Therefore, such solution eliminates the power $\Delta P_{stp} = \Delta p_{DE}\, Q_M$ of the structural pressure losses in the throttling control assembly (in the servo-valve, proportional directional valve, set throttling valve or set two - way flow regulator).

Conditions of work of such central system during feeding of one motor (from the point of view of the power of energy losses in the system) are very similar to the conditions of work of the above described individual system with volumetric control of the pump and motor (with one pump of variable capacity per one shaft revolution driving one hydraulic motor of variable capacity per one shaft revolution), where a constant pressure $p_{P2} = p_n$ in the pump discharge conduit is independent of the current hydraulic motor load M_M. These two systems differ from each other by the size of pump; in the individual system, its nominal capacity Q_{Pn} must provide the required nominal capacity Q_{Mn} of the driven motor; in the central system, the nominal capacity Q_{Pn} of the pump must be greater in order to deliver the flow of an intensity equal to the sum of the required nominal capacities Q_{Mni} of the simultaneously operating and parellelly connected motors.

However, let us assume, in order to compare the power of losses in those two systems of the hydraulic motor volumetric speed control with power of losses in the earlier discussed individual systems, that in both systems (the individual one and the central one feeding one motor) a pump is used with nominal capacity Q_{Pn} required by the nominal capacity Q_{Mn} of the controlled single motor. With such assumption, the conditions of work of both systems and power of energy losses are identical.

Let us consider the areas of power of losses in the elements of a central system according to the Rexroth company conception, with volumetric speed control of each motor by the secondary circuit assembly, the system feeding a single motor.

In a hydraulic motor with variable capacity q_{Mgv} per one shaft revolution, operating at the pressure decrease $\Delta p_M = p_n - \Delta p_C$ (i.e. only slightly lower than the nominal pressure p_n), the useful power $P_{Mu} = M_M\, \omega_M$ unchanged compared with previous system (Fig. 2, 5, 8, 11 - PMR 03/2008 Part. I, 14 and 16), resulting from the same torque M_M and motor shaft speed ω_M required by the driven device, the areas of power of losses in the motor have different shapes.

The power $\Delta P_{Mm} = M_{Mm}\ \omega_M$ of mechanical losses in the motor depends (apart from ω_M) on the torque M_{Mm} of those losses. Torque M_{Mm} of mechanical losses in the motor practically does not depend on the speed ω_M (n_M) and the current capacity q_{Mgv} per one motor shaft revolution, and depends mainly on the motor loading torque M_M. Therefore, the area of power ΔP_{Mm} of mechanical losses (Fig. 18) is practically the same as the area ΔP_{Mm} in a system with a pump with variable capacity per one revolution driving a motor with constant capacity q_{Mt} per one revolution (Fig. 16), although the shape of area ΔP_{Mm} (like the shape of area $P_{Mu} = M_M\ \omega_M$ of the unchanged useful power), presented on a plane with coordinates (p – pressure, Q – intensity), is different.

Power $\Delta P_{Mv} = \Delta p_{Mi}\ Q_{Mv}$ of volumetric losses in a hydraulic motor, with unchanged (compared with previous system) small motor useful power $P_{Mu} = M_M\ \omega_M$ (mainly at a small load M_M) increases, in the considered system, many-fold compared with power ΔP_{Mv} of volumetric losses in a motor with constant capacity q_{Mt} per one revolution (Fig. 16). This is an effect of simultaneous great increase of pressure decrease Δp_{Mi} in the motor working chambers and of accompanying great increase of intensity Q_{Mv} of the volumetric losses.

In turn, power $\Delta P_{Mp} = \Delta p_{Mp}\ Q_M$ of pressure losses in the motor channels (and in the directional control valve, if there is one), (motor with small instantaneous capacity q_{Mgv} per one shaft revolution) decreases distinctly both as an effect of the decrease of motor capacity Q_M (with unchanged n_M) and an effect of the accompanying decrease of Δp_{Mp}. Therefore, power ΔP_{Mp} of pressure losses in the motor with small capacity q_{Mgv} per one shaft revolution is lower then power ΔP_{Mp} of those losses in a motor with constant capacity q_{Mt} per one revolution (Fig. 16).

However, in an axial piston hydraulic motor with variable capacity q_{Mgv} per one revolution, operating with constant decrease Δp_M of pressure close to the system nominal pressure p_n, the energy gain connected with the smaller power ΔP_{Mp} of pressure losses will be much lower than the energy loss resulting from the increased (compared to a motor with constant capacity q_{Mt} per one revolution – Fig. 16) power ΔP_{Mv} of volumetric losses. In effect, the energy efficiency η_M of a motor operating in such conditions is lower.

Power $\Delta P_C = \Delta p_C\ Q_M$ of pressure losses in the conduits of a system operating with constant pressure $p_{P2} = p_n$ in the pump discharge is smaller than the power ΔP_C of those losses in a system with hydraulic motor with constant capacity q_{Mt} per one revolution. This is an effect of decreased flow intensity Q_M in the conduits. Also pressure losses Δp_C in the conduits are decreased.

From the point of view of power ΔP_C of losses in the connecting conduits, a system with hydraulic motor with variable capacity q_{Mgv} per one revolution, operating at constant pressure $p_{P2} = p_n$ in the pump discharge conduit, has great advantage over the systems with a motor of constant capacity q_{Mt} per one revolution, where the decreased motor loading torque M_M is accompanied by smaller decrease Δp_M of pressure required by the motor and, in effect, lower pressure p_{P2} in the pump discharge conduit ($p_{P2} < p_n$). With longer conduits and at low temperature (greater viscosity) of the working medium (hydraulic oil), the gain from energy savings in the conduits of a system with $p_{P2} = p_n$ may be considerable.

In a variable capacity pump cooperating with a $p_{P2} = p_n$ regulator, its capacity Q_P, with unchanged useful power $P_{Mu} = M_M\ \omega_M$ of the hydraulic motor (controlled by the secondary circuit assembly) driven by the pump is smaller than the capacity Q_P of the pump feeding a hydraulic motor with constant capacity q_{Mt} per one shaft revolution (i.e. in a situation when pressure p_{P2} is smaller than the nominal pressure – $p_{P2} < p_n$ (Fig. 16)). It is also accompanied by the decrease of power $\Delta P_{Pp} = \Delta p_{Pp}\ Q_P$ of pressure losses in the pump channels and the pump directional valve (due to decrease of pump capacity Q_P and decrease of pressure losses Δp_{Pp}).

Power $\Delta P_{Pv} = \Delta p_{Pi}\ Q_{Pv}$ of the pump volumetric losses, with unchanged (compared to the above described systems) small useful power $P_{Mu} = M_M\ \omega_M$ of the motor, increases many – fold, compared with power ΔP_{Pv} of volumetric losses in a variable capacity pump feeding directly a hydraulic motor with constant capacity q_{Mt} per one revolution (when $p_{P2} < p_n$; Fig. 16). Power ΔP_{Pv} of volumetric losses is practically equal to the power ΔP_{Pv} in a pump feeding a system with motor speed throttling control, the pump cooperating also with regulator in a constant pressure system $p_{P2} = p_n$ (although then the pump capacity Q_P is greater (Fig. 8)).

Analysing a pump feeding the hydraulic motor controlled in the secondary circuit (i.e. operating at constant pressure $p_{P2} = p_n$ and capacity Q_P corresponding to the current useful power $P_{Pu} = Q_P\ p_n$ of the pump related to the current useful power $P_{Mu} = M_M\ \omega_M$ of the driven motor) one can find, as the first approximation, that torque $M_{Pi} = \Delta p_{Pi}\ q_{Pgv}/2\Pi$ indicated in its working chambers is of the order of torque M_{Pi} of the variable capacity pump feeding the motor with constant capacity q_{Mt} per one revolution (Fig. 16). This is due to the fact, that with similar current useful power of the pump, for instance two-fold increase Δp_{Pi} of pressure in the pump chamber is accompanied by the nearly two-fold decrease q_{Pgv} of the capacity per one revolution. Therefore, it may be assumed with approximation that torque M_{Pm} of mechanical losses, proportional to the indicated torque M_{Pi} ($M_{Pm} \sim M_{Pi}$), will be similar in both cases. In effect, it may also be assumed, that area $\Delta P_{Pm} = M_{Pm}\ \omega_P$ of the power of mechanical losses in the pump will be similar to the area ΔP_{Pm} of mechanical losses in the pump feeding a system with hydraulic motor with constant capacity q_{Mt} per one revolution (Fig. 16), although the shape of the ΔP_{Pm} field is different .

As in a hydraulic motor controlled by own secondary circuit and operating in a continuous way with the decrease Δp_M of pressure close to the system nominal pressure p_n, the pump operation at the pressure $p_{P2} = p_n$ is connected with the energy gain resulting from the decrease of power $\Delta P_{Pp} = \Delta p_{Pp} Q_P$ of pressure losses in the pump, with a given unchanged useful power $P_{Mu} = M_M\ \omega_M$ of the motor, and with great energy loss connected with the increase of power $\Delta P_{Pv} = Q_{Pv}\ \Delta p_{Pi}$ of volumetric losses in the pump compared with the power of those losses in a variable capacity pump feeding a motor with constant capacity q_{Mt} per one shaft revolution (Fig. 16). With comparable power $\Delta P_{Pm} = M_{Pm}\ \omega_P$ of mechanical losses in the pump working in those two systems (Fig. 16 and 18), the sum of power of losses in the pump working at $p_{P2} = p_n$ is greater and its energy efficiency η_P is lower.

The sum ΔP of power of energy losses in a hydraulic motor with variable capacity q_{Mgv} per one shaft revolution, and in a pump with variable capacity q_{Pgv} per one shaft revolution working in the system with short connecting conduits, with the pump discharge conduit pressure equal to the nominal pressure ($p_{P2} = p_n$), may be, with small motor load M_M (Fig. 18), much greater than the sum ΔP of the power of losses in a system with motor with constant capacity q_{Mt} per one shaft revolution controlled volumetrically by variable capacity pump (Fig. 16).

In the operating conditions of a system with $p_{P2} = p_n$ (p_{P2} independent of the motor load M_M), a system with long connecting conduits (and greater pressure losses Δp_C in the conduits), operating with great viscosity ν of a working medium (hydraulic oil), the sum ΔP of power of losses may appear smaller than the sum ΔP of losses in a system with motor with constant capacity q_{Mt} per one shaft revolution and with variable capacity pump (Fig. 16). With great oil viscosity, the energy gains connected with decrease of pressure losses in the system elements (mainly in the connecting conduits), achieved in effect of decreasing the intensity $Q_P = Q_M$, may appear great, but the power of volumetric losses ΔP_{Pv} in the pump and ΔP_{Mv} in the motor will not increase as distinctly as with the smaller viscosity.

CONCLUSIONS

- A diagram of the direction of increase of power stream from the shaft or piston rod of a hydraulic motor to the pump shaft, power increasing as an effect of the imposed power of energy losses in the hydrostatic drive and control elements, is proposed and justified.
- Graphical interpretation of the power of energy losses in the hydrostatic drive and control system elements and also of the power developed by those elements is presented.
- Chapters 2 – 5 (PMR 03/2008, Part. I) illustrate the fields of power ΔP of energy losses in the individual system elements, where the hydraulic motor (with constant capacity per one revolution) speed control is effected by series throttling of the working medium flow in order to obtain the intensity Q_M corresponding to the angular ω_M (rotational n_M) speed required by the motor driven device. The use of a throttling directional valve (servo – valve or proportional directional valve) or else set throttling valve or set two – way flow regulator allows to change the motor speed precisely. A cheaper constant capacity pump may be used as a feeding device in the system with series throttling control, the pump cooperating with the overflow valve or controlled overflow valve, or else a variable capacity pump cooperating with a constant pressure regulator or the *Load Sensing* variable pressure regulator.
  - Chapter 2 presents an individual system of motor speed series throttling control fed by a constant capacity pump cooperating with an overflow valve in constant pressure conditions.
    The required level of nominal pressure p_n of pump operation and the required level of pump theoretical capacity Q_{Pt} during the system operation, as well as the current small loading torque M_M and current small hydraulic motor shaft angular speed ω_M are decisive in that motor speed throttling control structure of temporary power ΔP_{stp} of structural pressure losses and power ΔP_{stv} of structural volumetric losses. This is then accompanied by a very low value of the overall energy efficiency η of the system.
    The power ΔP_{stp} of the structural pressure losses in the throttling control assembly may be reduced almost to zero during the hydraulic motor operation at the maximum shaft load M_{Mmax}.
    The power ΔP_{stv} of the structural volumetric losses in the throttling control assembly may be reduced almost to zero in a situation when the hydraulic motor operates with maximum angular speed ω_{Mmax} (rotational speed n_{Mmax}).
    The hydraulic motor operation with maximum load M_{Mmax} and simultaneous maximum speed $\omega_{Mmax}(n_{Mmax})$ may cause minimization of the power of losses connected with the motor speed throttling control and the sum of energy losses in the system consists of the hydraulic motor losses, the conduit losses and pump losses. The overall system efficiency η reaches then a high value η_{max}, close to the value of energy efficiency η_{max} of a system with the motor speed volumetric control (by a variable capacity pump).
    However, in order to be able, in a system with series throttling control, to load the hydraulic motor with a maximum torque M_{Mmax} close to the maximum load M_{Mmax} of the motor in a system with volumetric speed control, the throttling slot of the throttling proportional control valve (or of the throttling valve) has to be increased to the size requiring a small decrease $\Delta p_{DEmin} \approx 0$ of pressure at the maximum flow intensity $Q_{Mmax} \approx Q_P$.
    On the other hand, in order to be able, in a system with series throttling control, to set, with a throttling proportional control valve or a throttling valve, the maximum intensity $Q_{Mmax} \approx Q_P$, i.e. close to the pump capacity, an overflow valve has to be used in the system to stabilize the pressure level $p_{SP} \approx p_n$ of the pump operation at the flow intensity $Q_P – Q_M \approx 0$ (i.e. close to zero).
  - Chapter 3 presents an individual system of motor speed series throttling control fed by a constant capacity pump cooperating with a controlled overflow valve in variable pressure conditions.
    In effect, when the hydraulic motor is loaded with a small torque M_M, the power $\Delta P_{Pc} = M_P\ \omega_P$ absorbed by the pump from the drive (electric or internal combustion) motor is also significantly reduced, which, with an unchanged hydraulic motor useful power $P_{Mm} = M_M\ \omega_M$, increases the overall system energy efficiency η compared with the constant pressure feeding system efficiency η.
    Both structures (p = cte and p = var) of the hydraulic motor speed series throttling control, fed by a constant capacity pump, may achieve, during maximum motor load M_{Mmax} and simultaneous maximum speed ω_{Mmax} (n_{Mmax}), the same maximum overall system efficiency η_{max}. It is close to the maximum energy efficiency η_{max} of a system with volumetric control (by a variable capacity pump) of hydraulic motor speed. The p = var system becomes then a p = cte system, therefore the operating conditions of both systems are the same and structural losses ΔP_{stp} and ΔP_{stv} in the throttling control assembly may be practically eliminated. However, similarly as in the constant capacity pump system p = cte, it requires increased area of the f_{DEmax} slot in the throttling directional control valve (throttling valve) to a size requiring slight pressure decrease $\Delta p_{DEmin} \approx 0$ at the maximum flow intensity $Q_{Mmax} = Q_P$. It requires also the use of a controlled overflow valve stabilizing the value $\Delta p_{SPS} = p_{P2} – p_2 =$ cte also at the flow intensity $Q_P – Q_M \approx 0$ (close to zero) and an overflow valve stabilizing the pressure level $p_{SP} \approx p_n$ at the flow intensity $Q_P – Q_M \approx 0$.
  - Chapter 4 presents an individual system of motor speed series throttling control fed by a variable capacity pump cooperating with regulator in constant pressure conditions.
    The use, as a hydraulic motor series throttling speed control system feeding source, a variable capacity pump with pressure regulator, operating at pressure $p_{P2} = \text{cte} \approx p_n$, allows, during the motor run with small speed ω_M (n_M), to reduce significantly the power $P_{Pc} = M_P\ \omega_P$ absorbed by the pump from the drive electric or internal combustion motor. With the unchanged useful power $P_{Mu} = M_M\ \omega_M$ of the hydraulic motor, the entire system energy efficiency η is significantly higher compared with efficiency η of

a constant pressure ($p = cte$) throttling assembly constant capacity pump feeding system.

The considered system may achieve, during the maximum hydraulic motor load M_{Mmax} and in the whole range of the motor speed change $0 \le \omega_M \le \omega_{Mmax}$, the overall efficiency η close to the value of energy efficiency η of a system with the motor speed volumetric control (by a variable capacity pump). The power $\Delta P_{stp} = \Delta p_{DE} Q_M$ of the structural pressure losses is then minimized. It requires, in a system with the hydraulic motor speed series throttling control, an increased area of the f_{DEmax} slot in the throttling directional control valve (or the throttling valve) to a size requiring slight pressure decrease $\Delta p_{DEmin} \approx 0$ at the maximum flow intensity $Q_{Mmax} = Q_{Pmax}$, i.e. equal to the full pump capacity. It requires also correct operation of the pump pressure regulator stabilizing the pump discharge pressure p_{P2} at the level $p_{P2} = cte \approx p_n$ in the whole range $0 \le Q_P \le Q_{Pmax}$ of the pump capacity variation.

In a situation of simultaneous maximum load M_{Mmax} and maximum speed ω_{Mmax} of a hydraulic motor controlled by series throttling, the maximum achievable energy efficiency η_{max} of a system is close to the value η_{max} of a system with hydraulic motor speed volumetric control i.e. directly by a variable capacity pump.

The greatest energy savings in the considered series throttling control system, compared with a series control system fed by a constant pressure constant capacity pump, are obtained during the hydraulic motor operation at small speed ω_M (n_M).

- Chapter 5 presents an individual system of motor speed series throttling control fed by a variable capacity pump cooperating with the *Load Sensing* regulator in variable pressure conditions.

  The use, as a feeding source of the hydraulic motor series throttling speed control system, of a variable capacity pump with the *Load Sensing* regulator operating at a pressure $p_{P2} = \Delta p_{LS} + p_2 \approx \Delta p_{LS} + p_{M1}$, i.e. slightly higher then the current pressure p_{M1} required by the hydraulic motor at its inlet (which is accompanied by decrease to a small value of the power ΔP_{st} of structural energy losses in the throttling control assembly) reduces the sum of power of energy losses in the system to a value only slightly higher then the sum of power of losses in the elements of a system with volumetric control of the motor speed (directly by a variable pump capacity). Power $P_{Pc} = M_P \omega_P$ absorbed by the pump from the electric or internal combustion drive motor is only slightly higher here then the power P_{Pc} of a variable capacity pump directly driving the hydraulic motor.

  The considered LS system operates in the whole range $0 \le M_M \le M_{Mmax}$ of the hydraulic motor load and in the whole range $0 \le \omega_M \le \omega_{Mmax}$ of its speed with the energy efficiency η only slightly lower then the efficiency η of a volumetric control system (directly by a variable capacity pump). The difference between overall efficiencies η of both systems will be inversely dependent on the capability of increase of the area of throttling proportional valve (or throttling valve) slot f_{DEmax}. The increase of f_{DEmax} allows to decrease $\Delta p_{DEmin} \approx 0$ at a maximum flow intensity $Q_{Mmax} = Q_{Pmax}$ (i.e. equal to a full pump capacity). It also requires correct operation of the pump LS regulator adjusting, in the whole range $0 \le Q_P \le Q_{Mmax}$ of the pump capacity, the discharge pressure p_{P2} at the level higher by a value $\Delta p_{LS} = p_{P2} - p_2 = cte$ then the p_2 pressure in the discharge conduit from the throttling proportional control valve (throttling valve) to the hydraulic motor.

- Chapter 6 presents an individual system with motor speed parallel throttling control. The parallel throttling control assembly may have a form of set throttling valve or set two – way flow regulator installed in the pump discharge conduit branch.

  The current useful power $P_{Mu} = M_M \omega_M$ of the hydraulic motor (independent of the used motor speed control structure), required by the motor driven device (and the same as in the systems shown in chapters 2 – 5), influences in a different way (than in the series throttling control systems) the structural losses generated in the system.

  When the hydraulic motor is loaded with small torque M_M, as an effect of decreased power of losses in system elements, the power $P_{Pc} = M_P \omega_P$ absorbed by the pump from the drive (electric or internal combustion) motor is also decreased compared with power P_{Pc} absorbed by a constant capacity pump used in the motor speed series throttling control system. With the unchanged hydraulic motor useful power $P_{Mu} = M_M \omega_M$, this increases the energy efficiency η of the whole system compared with the efficiency η of a series throttling control system fed by a constant capacity pump.

  The discussed structure of a hydraulic system, with a constant capacity pump and with hydraulic motor parallel throttling speed control, may achieve, during the operation with maximum speed ω_{Mmax} (n_{Mmax}) of the controlled motor and in the full range $0 \le M_M \le M_{Mmax}$ of its load, energy efficiency η equal to the total efficiency η of the system with volumetric control of the hydraulic motor speed (with a variable capacity pump).

- Chapter 7 presents an individual system with motor speed volumetric control by a variable capacity pump.

  In a volumetric motor (with a constant capacity per one revolution) speed control system (by a variable capacity pump), the sum of current (unchanged in relation to power P_{Mu} in the above discussed systems) values of: hydraulic motor useful power $P_{Mu} = M_M \omega_M$, power $\Delta P_{Mm} = M_{Mm} \omega_M$ of mechanical losses in the motor, power $\Delta P_{Mv} = \Delta p_{Mi} Q_{Mv}$ of volumetric losses in the motor, power $\Delta P_{Mp} = \Delta p_{Mp} Q_M$ of pressure losses in the motor, power $\Delta P_C = \Delta p_C Q_M$ of pressure losses in the hydraulic system connecting conduits, power $\Delta P_{Pp} = \Delta p_{Pp} Q_P$ of pressure losses in the pump, power $\Delta P_{Pv} = \Delta p_{Pi} Q_{Pv}$ of volumetric losses in the pump, power $\Delta P_{Pm} = M_{Pm} \omega_P$ of mechanical losses in the pump decides that the power $P_{Pc} = M_P \omega_P$ required by the pump from the pump driving (electric or internal combustion) motor is smaller than power P_{Pc} absorbed by the pump in the above discussed systems with the hydraulic motor speed throttling control.

  Energy efficiency $\eta = P_{Mu}/P_{Pc}$ of a system with volumetric control by a variable capacity pump is the highest efficiency η amongst the considered systems in the whole range of speed $0 \le \omega_M \le \omega_{Mmax}$ and hydraulic motor load $0 < M_M \le M_{Mmax}$. However, it has to be noted that the energy advantage of a system with the hydraulic motor speed ω_M (n_M) control by a variable capacity pump over the systems with the motor speed throttling control decreases markedly when the motor speed ω_M approaches ω_{Mmax} and the motor load approaches M_{Mmax}.

- Chapter 8 presents an individual system with motor speed volumetric control by means of a simultaneous change of the pump capacity per one revolution and change of the motor capacity per one revolution, system operating at a constant pressure in the pump discharge conduit equal to the nominal pressure: $p_{P2} = p_n$.

This solution allows to use pump useful power P_{Pu} (resulting from a product of pump working pressure p_{P2} equal to the nominal system pressure p_n ($p_{P2} = p_n$) and maximum pump capacity Q_{Pmax} achieved at the coefficient $b = q_{Pgv}/q_{Pt} = 1$ of the change of the pump capacity per one revolution) also during the period of smaller hydraulic motor load (with the torque $M_M < M_{Mmax}$), by an increase of the motor speed ω_M above its nominal speed ω_{Mn} ($\omega_M > \omega_{Mn}$).

In a hydraulic motor with variable capacity q_{Mgv} per one shaft revolution, operating at the pressure decrease $\Delta p_M = p_n - \Delta p_C$ (i.e. only slightly lower than the nominal pressure p_n), the useful power $P_{Mu} = M_M\,\omega_M$ unchanged compared with previous system (chapters 2 – 7), resulting from the same torque M_M and motor shaft speed ω_M required by the driven device, the areas of power of losses in the motor have different shapes.

From the point of view of power ΔP_C of losses in the connecting conduits, a system with hydraulic motor with variable capacity q_{Mgv} per one revolution, operating at constant pressure $p_{P2} = p_n$ in the pump discharge conduit, has great advantage over the systems with a motor of constant capacity q_{Mt} per one revolution, where the decreased motor loading torque M_M is accompanied by smaller decrease Δp_M of pressure required by the motor and, in effect, lower pressure p_{P2} in the pump discharge conduit ($p_{P2} < p_n$). With longer conduits and at low temperature (greater viscosity) of the working medium (hydraulic oil), the gain from energy savings in the conduits of a system with $p_{P2} = p_n$ may be considerable.

The sum ΔP of power of energy losses in a hydraulic motor with variable capacity q_{Mgv} per one shaft revolution and in a pump with variable capacity q_{Pgv} per one shaft revolution working in the system with short connecting conduits, with the pump discharge conduit pressure equal to the nominal pressure ($p_{P2} = p_n$), may be, with small motor load M_M, much greater than the sum ΔP of the power of losses in a system with motor with constant capacity q_{Mt} per one shaft revolution controlled volumetrically by variable capacity pump (chapter 7).

In the operating conditions of a system with $p_{P2} = p_n$ (p_{P2} independent of the motor load M_M), a system with long connecting conduits (and greater pressure losses Δp_C in the conduits), operating with great viscosity ν of a working medium (hydraulic oil), the sum ΔP of power of losses may appear smaller than the sum ΔP of losses in a system with motor with constant capacity q_{Mt} per one shaft revolution and with variable capacity pump. With great oil viscosity, the energy gains connected with decrease of pressure losses in the system elements (mainly in the connecting conduits), achieved in effect of decreasing the intensity $Q_P = Q_M$, may appear great, but the power of volumetric losses ΔP_{Pv} in the pump and ΔP_{Mv} in the motor will not increase as distinctly as with the smaller viscosity.

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