DISPLACEMENT CONTROLLED LINEAR ACTUATOR WITH DIFFERENTIAL CYLINDER - A WAY TO SAVE PRIMARY ENERGY IN MOBILE MACHINES

Robert Rahmfeld and Monika Ivantysynova
Technical University of Hamburg-Harburg, Aircraft Systems Engineering, Nesspriel 5, D-21129 Hamburg, Germany
R.Rahmfeld@tu-harburg.de

ABSTRACT

The paper introduces a new hydraulic circuit for a pump controlled actuator with differential cylinder. The actuator uses a constant low pressure source for the compensation of the difference between the in- and outgoing flow of the cylinder chambers. The pump controlled actuator allows an improved utilization of primary energy due to the omitted valves especially when taking into account recovery of potential load and brake energy and its use for other drives. The possible amount of saved primary energy compared to today's valve controlled systems for general mobile machines is to be estimated in this paper with the help of a typical working cycle of a reference system.

KEYWORDS

Linear Actuator, Displacement Control, Energy Saving

INTRODUCTION

Considering the developments in the field of construction, agricultural, mining and earth-moving machines, a strong trend towards more automation of working cycles and support of the user can be obtained. Taking the state of the art into account then it is obvious that there is a high demand for new electro hydraulic servo drives which are suitable for mobile machines. This technology demands intelligent actuators which work in a closed control loop. Not only continuously rising fuel costs but also increasing pollution of the environment claim for a higher energy efficiency of the actuators used in these machines. Today's mobile machines mostly contain hydraulic valve controlled drives using an open loop control. For energy saving purpose the constant pressure controlled pumps are very often replaced by load-sensing controlled pumps, and consequently load-sensing valve technology is used today. The use of load-sensing technology for hydraulic actuators in close loop motion control, as needed for machines with automatic motion control, requires very complex multi-variable control concepts. Additionally, a greater expenditure according to sensors and signals is necessary. An alternative concept is shown in this paper.

DISPLACEMENT CONTROLLED ACTUATOR WITH DIFFERENTIAL CYLINDER

Because of space reasons in the boom structures, today mostly a differential cylinder is used as linear motor. To operate a differential cylinder in closed hydraulic circuit in four quadrant operating mode with a servo pump as final control element, new circuit solutions in order to balance the unequal volume flows are necessary. From literature some previous concepts mainly for stationary applications ([1], [4], [10]) are known, for example the implementation of a hydraulic transformer with an additional pump [10], the use of two servo pumps [4] and the use of the INNAS hydraulic transformer with an additional high pressure source [1]. Due to the relatively high number of components, and in some cases very complicated multi-variable control concepts, these circuit concepts are not appropriate for mobile machines. The replacement of a valve controlled actuator by a displacement controlled actuator, with two or more displacement machines and perhaps even further components, is too extensive for mobile machines.

Figure 1 shows a new developed circuit solution with low component expenditure for a displacement controlled linear drive with differential cylinder. The proof of function in four quadrant operation was successfully done by experiment on a laboratory test rig [11] and additionally on a mobile machine [12].

Here, differential volume is balanced on the low pressure side. In Fig. 1 a charge pump (4) together with an accumulator (5) are used for this task. Main advantage of this concept is that in case of more than one actuator in a machine, the low pressure lines can be coupled. Two pilot operated check valves (3) make sure that the low pressure side of the cylinder (2) is always connected to the accumulator (5), which depends on the operating quadrant. A servo pump (1) works as final control element of the linear drive. In case of positive loads and forward movements of the cylinder ($\dot{x} > 0$) the servo pump works in pumping mode and sucks oil additionally from the accumulator. When the cylinder is to move backwards under the same load condition the servo pump works in motoring mode under an aiding load, and the accumulator is filled from the low pressure side. The adjustment system of the servo pump is also supplied by the charge pump (4) which is, as mentioned above, also used for balancing of volumetric losses of the servo pump. Of course, this actuator needs two high pressure and one low pressure relief valves.

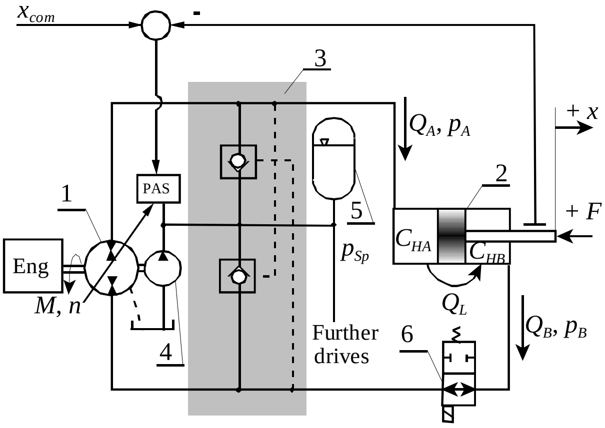

Fig. 1 Displacement controlled drive with differential cylinder

The system shown in Fig. 1 uses a servo pump with a low pressure adjustment system, which is sufficient for many applications. If higher bandwidths are requested for the final control element, a third pressure level can be installed by separating low pressure line pressurization and adjustment system supply. The grey box in Fig. 1 indicates that all elements inside the box can be usefully integrated into the servo pump housing. Today's servo pumps for closed circuit mostly contain several components like for example relief valves, check valves and a charge pump, i.e., for one actuator there are only three individual components required (servo pump, accumulator and cylinder). Thus, for a total machine with i actuators with a coupled low pressure line with one common accumulator the resulting number of elements in the whole hydraulic circuit is: $2i + 1$.

The coupling of actuators on the low pressure side and the mechanical coupling of servo pumps by a common distributing gear allows here the use of potential load and brake energy for other drives implemented in the machine. For example, if loads are lowered and the boom structure of the machine also rotates simultaneously, potential load energy can be used for the rotational drive. An alternative is using this energy for the hydrostatic transmission, which for instance is very useful in wheel loaders. By the use of a shut-off valve (6) loads can be hold also in case of an engine or pump failure. Further, a simple open loop operation of the actuator can be realized by the help of this valve.

Following advantages are combined with this type of hydraulic linear actuator:

- less energy dissipation due to omitting throttling losses, which allows less fuel consumption and means less pollution,
- use of load and brake energy for other drives in the machine – especially for propel or steering drive,
- simplification of hydraulic system (less components) which leads to less installation and maintenance costs,
- reduction of total system mass,
- less cooling power,
- nearly load independent system behaviour and
- less filtration ratio needed in main circuit compared to control valve technology.

Due to energy saving aspects a pressure compensated pump should be used in combination with an accumulator as low pressure source. The size of the low pressure source (pump (4) and accumulator (5)) should be optimized according to typical working cycles.

ACTUATOR LOSS MODEL

In order to evaluate the potential of energy saving effects of this new actuator concept a precise mathematical model describing the losses of the actuator was developed. Hereby, loss models for the following loss types were developed:

- servo pump losses,
- cylinder losses,
- pressure losses in hydraulic lines,
- pressure losses of pilot operated check valve and
- charge pump losses.

Figure 2 demonstrates the energy flow for this type of actuator. All in the loss model considered sources of energy dissipation are shown in the figure.

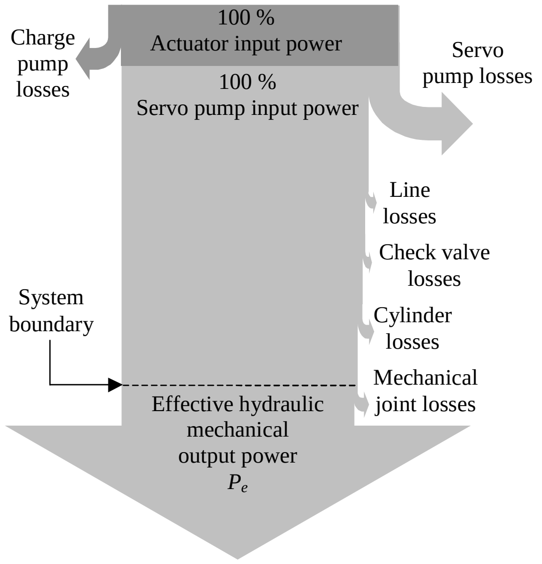

Fig. 2 Sankey-Diagramme of displacement controlled linear actuator with differential cylinder in pumping mode

Servo Pump Losses

The main source of losses in this type of actuator usually is represented by the servo pump. To achieve an adequate dynamic performance of the actuator the use of swash plate axial piston pumps is recommended. It should be mentioned here, that the bandwidth of the servo pump is mainly determined by the bandwidth of the applied control valve of the pump control system in case of swash plate pumps. Investigations have shown that using simple one stage servo valves (direct drive) bandwidths of servo pump control system in the range of 80 Hz are achievable [2].

For the proof of function and further experimental investigations of this actuator type a swash plate pump was used here. Both volumetric and torque losses are a function of operating pressure difference between high and low pressure line of the actuator, pump speed and the adjusted displacement volume. Additionally, both type of losses depend on the viscosity of the oil. For a precise mathematical description of the very complex loss behaviour of a variable displacement machine only a measurement based modelling method can be used [7]. The main reason for this is the very complex behaviour of frictional losses in the individual gaps of the swash plate machine which are at the moment not describable by general valid mathematical analytical models. For this purpose servo pump was measured in a circuit according to ISO4409 with about 700 measurement points considering 6 adjustment positions, 5 pressure levels, 3 pump speeds and 2 temperatures in pumping and motoring mode.

Servo pump losses are modelled with a polynomial in contrast to previous models [3] and [6] which are not precise in modelling loss dependency of adjusted displacement volume. The polynomial is gained by interpolation of measurements as a function of displacement volume, pump speed and pressure difference for a specific temperature with real coefficient matrix **K** and natural exponents:

$$Q_S(V_i, n, \Delta p)_{T=const.} = \sum_{i=0}^{q}\sum_{j=0}^{r}\sum_{k=0}^{s} \mathbf{K}(i,j,k)\cdot V_i^i \cdot n^j \cdot \Delta p^k \quad (1)$$

The real coefficients are found by minimizing sum of error squares. All together we receive two equations each with $(q+1)(r+1)(s+1)$ elements for the description of volumetric and hydraulic-mechanical losses.

For ingenious results it is important to create a two-quadrant model based on measurements for pumping and motoring mode which is extended for the missing two quadrants symmetrically whereby a symmetric behaviour of the servo pump is assumed. Figure 3 contains measured and modelled volumetric and torque losses of a typical axial piston swash plate servo pump. Now no unsteadiness points result by the model in areas of small adjusted displacement volume which is coherent to physical behaviour. Especially for a servo pump, which works very often in this small adjustment volume area, an improved loss model is of importance. Of course accuracy of polynomial approximation is lower compared to one quadrant models. One has to find a compromise between accuracy and numerical optimization for simulation. Servo pump power losses follow as:

$$P_{SP} = Q_S \ \Delta p + M_S \ 2\pi \ n \quad (2)$$

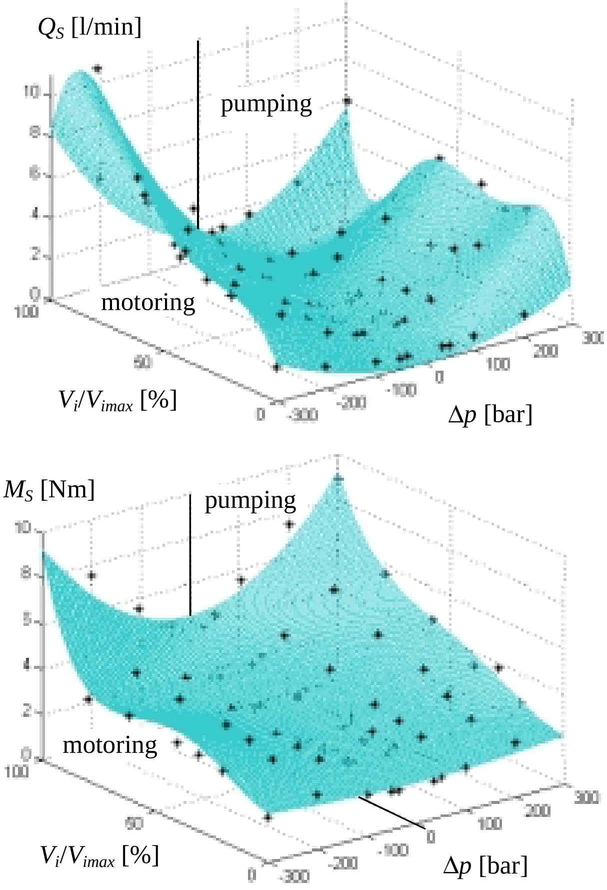

Fig. 3 Two quadrant loss model and measurement points of volumetric and torque losses for a servo pump with V_{imax} = 10,5 cm^3/U (pump speed 5000 rpm, oil viscosity 20 cSt)

Cylinder Losses

Cylinder friction losses occur on both sealings (rod and piston) and similar to the servo pump model an analytical description of this physical behaviour is very complex. So again a measurement based modelling is applied. For instance, Fig. 4 illustrates a measured friction force for a typical differential cylinder (stroke of 0,5 m, and maximum cylinder force of 100 kN) which was determined by using Newton equation:

$$F_R = -F - m\,\ddot{x} + p_A \, A_K - p_B \, A_K \ \alpha \quad (3)$$

where m stands for all moved masses at the cylinder. In Fig. 4 clearly a Stribeck curve can be seen, friction force consists then of the following essential parts:

- viscous friction as cylinder velocity proportional,
- coulomb friction and

- static friction.

Friction in form of Stribeck curve containing all three elements mentioned above can be described as follows:

$$F_R = f_v\,\dot{x} + \mathrm{sign}(\dot{x})\left(F_C + F_H\, \mathrm{e}^{-c_H|\dot{x}|}\right) \qquad (4)$$

In order to prevent numerical problems by solving the differential equations in points of discontinuity friction force is rewritten by replacing the expressions:

$$\mathrm{sign}(\dot{x}) \rightarrow \tanh(\dot{x}\,\gamma) \;\text{ and }\; |\dot{x}| \rightarrow \dot{x}\tanh(\dot{x}\,\gamma) \qquad (5)$$

whereby γ can be varied in order to optimize the consistency of approximation. Figure 5 includes result of approximation according to Eq. 4 and the modified approximation according to Eq. 5. It is obvious that with high values of γ a high approximation without discontinuity can be achieved.

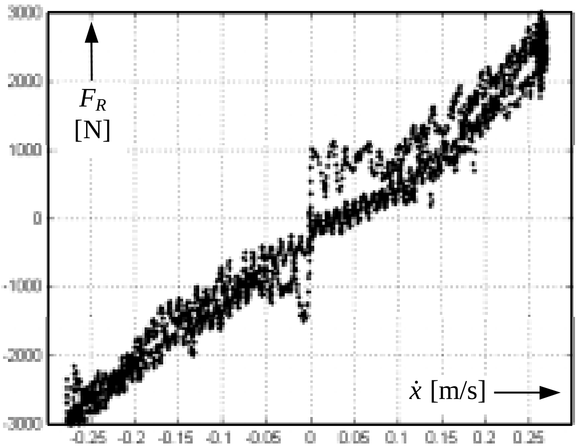

Fig. 4 Measured friction force of a typical linear actuator with differential cylinder (oil viscosity 20 cSt, HLP32)

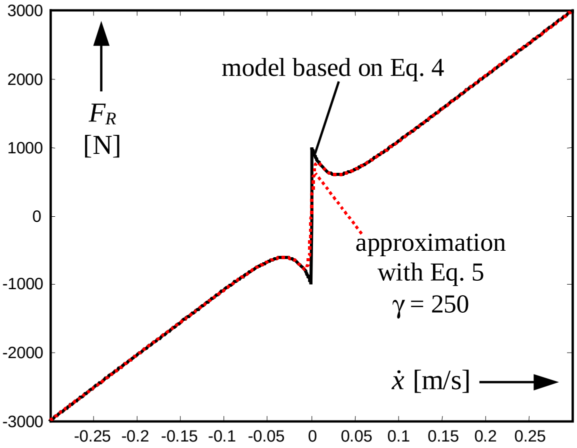

Fig. 5 Friction force models of measured friction force in Fig. 4

The measured friction force shows a clear linear dependence on cylinder velocity. This fact allows us to calculate volumetric losses in the gaps under the cylinder sealings with the help of a simplified model. This simplified model consists of a total gap length l_G, a gap height h and a gap width assumed to $\pi \cdot d_K$. Then the viscous friction yields:

$$f_v\,\dot{x} = \frac{\pi\, d_K\, l_G\, \mu}{h}\,\dot{x} \qquad (6)$$

With the help of this assumption it is now possible to estimate cylinder volumetric losses using the following expression:

$$Q_L = \frac{\pi\, d_K\, h}{2}\left(\dot{x} + \frac{h^2}{6\, l_G\, \mu}\,(p_A - p_B)\right) \qquad (7)$$

where the gap height h is calculated using Eq. 4 for the measured friction force. Cylinder power losses then become in case of symmetric line losses Δp_L (see below):

$$P_{SC} = F_R\,\dot{x} + Q_L\,(\Delta p - \mathrm{sign}(\dot{x})\,2\,\Delta p_L) \qquad (8)$$

Pressure Losses in Lines

Losses occurring in hydraulic lines of the actuator can be generally modelled by the use of the following equation, e.g. cylinder annulus chamber supply line losses:

$$\Delta p_L = \left[\lambda(\mathrm{Re})\,\frac{l_L}{d_L} + \sum \zeta\right]\frac{\rho}{2}\left(\frac{\alpha\, A_K}{A_L}\,\dot{x}\right)^2 \qquad (9)$$

Whereby the first term represents losses due to the flow of a viscous fluid and the second term represents losses due to the impulse change of the fluid appearing on individual hydraulic resistances, for instance pipe quarter bend. Note, that Δp_L has to be added or subtracted from cylinder chamber pressure in order to calculate pressure at servo pump port depending upon cylinder movement direction. Corresponding power losses are:

$$P_{SL} = \Delta p_L\;\alpha\, A_K\;\dot{x} / A_L \qquad (10)$$

Pilot Operated Check Valve Losses

Pilot operated check valve loss description is also based on measurement results. Both directions pressure-flow characteristics are described by polynomials as a function of volume flow. Measurements were done on a separate test rig according to ISO4411 again for 2 temperatures and several flow rates. Figure 6 shows measured valve pressure drop in both flow directions for a pilot operated check valve of nominal size 10. Main difference of both flow directions behaviour is that in normal flow direction from A_E to B_E a spring force has to be overcome which corresponds to 1.5 bar opening pressure. Power losses of this valve are in case all volumetric losses are balanced by one valve:

$$P_{SCV} = \Delta p_{CV}\left[(1-\alpha)\,A_K\dot{x} + Q_S + Q_L\right] \qquad (11)$$

Charge Pump Losses

For the fixed displacement charge pump constant volumetric and torque losses for particular temperatures are taken into account because pressure difference is nearly constant. In case using a pressure compensated pump a similar model as for the servo pump which depend on pressure level and adjusted displacement volume is utilized. Charge pump losses are written therefore as:

$$P_{SCP} = Q_{SCP}\, p_{Sp} + M_{SCP}\, 2\pi\, n \qquad (12)$$

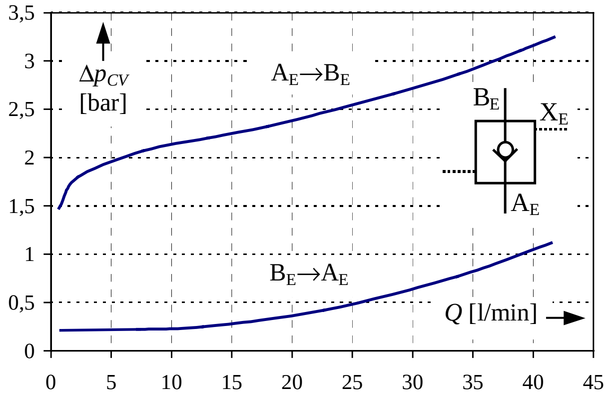

Fig. 6 Pressure-flow characteristics of typical pilot operated check valve SL10 in both flow directions (20 cSt, HLP32)

Total Loss Model

All particular losses are implemented in a dynamic Matlab/Simulink model of typical actuators which allows the simulation of typical working cycles. Total power losses of one actuator, which are transformed into heat, are the following sum:

$$P_S = P_{SP} + P_{SC} + P_{SL} + P_{SCV} + P_{SCP} \qquad (13)$$

ADDITIONAL SAVING EFFECTS

For comparison with today's hydraulic systems for mobile machines in addition to power losses, regained load and brake energy must be considered. Total required power is the sum of mechanical input power of servo pumps and additional pumps, whereby the system consists of N_P pumps. At first only linear drives are regarded. In case of negative torque servo pump works in motoring mode and transfers energy to other pumps via shaft or distributing gear whereby excess regained energy is dissipated in the diesel engine.

Setting system boundary between pump shafts and mechanical output of cylinders delivers difference between mechanical input and output power of one total machine, whereby the system contains N_A linear actuators:

$$P_S^* = P - P_e = 2\pi \sum_{j=1}^{N_P} M_j n_j - \sum_{i=1}^{N_A} F_i\, \dot{x}_i ,\quad P_S^* \geq 0 \qquad (14)$$

The amount of wasted primary energy due to hydraulic energy losses is then specified by the diesel engine efficiency (maximum around 40 %) and the efficiency of mechanical devices like gear boxes (about 98 %).

This energy can be transferred to a quantity of physical work by using the integral with the working cycle time t. Saved volumetric amount of fuel V_S follows with calorific value (H_{Diesel} = 42,5 MJ/kg) and density (ρ_{Diesel} = 835 kg/m^3) of Diesel fuel.

$$V_S = \frac{\int_0^t P_S^*(\tau)\, \mathrm{d}\tau}{\eta_{Diesel}\ \eta_{mech}\ H_{Diesel}\ \rho_{Diesel}} \qquad (15)$$

I.e., V_S is therefore the interesting comparison aspect in order to find out how much primary fuel energy can be saved with this new drive technology.

RESULTS AND CONCLUSION

For proving energy saving possibilities in typical mobile machines a reference machine with a typical hydraulic LS system was defined. Figure 7 includes the demolition excavator with 160 kW maximum Diesel engine output power and seven linear drives with differential cylinder. A mechanical model of this machine was created in multibody simulation tool Pro/Mechanica in order to receive forces and masses for hydraulic simulation. Further, simulation results for a typical working cycle of the conventional open loop LS system and a displacement controlled system, where each actuator owns its own servo pump, can be seen. Whereby the shovel with a weight of 1300 kg is lifted vertically over a distance of 7 m in a time of t = 10 s.

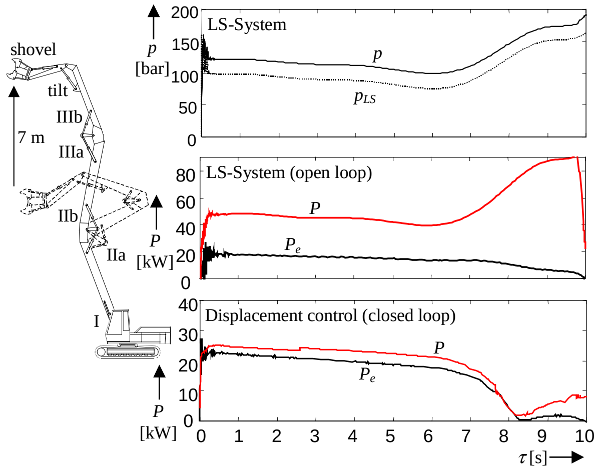

Fig. 7 Simulation results of reference system for displacement and LS controlled hydraulic system: lifting shovel 7 m

The first plot in Fig. 7 shows system and LS pressure of LS system. LS pressure difference is set to 20 bar and the simulation results verify this demand. Second plot depicts effective mechanical output power P_e and required Diesel engine power P for LS system. It is obvious that efficiency (= P_e/P) of LS system is always lower than about 40 %. Compared to that, third plot

describes the results for the displacement controlled system which is close loop controlled. It is obvious that effective mechanical output power P_e is somewhat different for both systems due to open and closed loop and flow behaviour differences. In case of displacement control maximum total machine efficiency is 70 %. Resulting V_S values of LS and displacement controlled (DC) mobile machine are for this working cycle:
$V_{S,LS}$ = 21.46 ml (=21.46 l for 1000 cycles in ~1 week),
$V_{S,DC}$ = 6.8 ml (=6.8 l for 1000 cycles in ~1 week),
resulting in following ratio: $V_{S,LS}/V_{S,DC} = 3.15$.
Other results of further independent working cycles confirmed this trend. The presented results proof the possibilities of energy savings which can be achieved with this new displacement controlled drive technology. Further, it is clear that LS systems are not energy efficient in case of different actuator load pressures in a machine. Further more, it can be concluded that cooling power in a machine can be reduced enormously with displacement control.

NOMENCLATURE

A_K	Differential cylinder piston area	m^2
A_L	Supply line area	m^2
c_H	Friction Force parameter	s/m
C_H	Hydraulic capacity	m^5/N
d_K	Cylinder piston diameter	m
d_L	Supply line inner diameter	m
f_v	Viscous friction parameter	kg/s
F	Cylinder load	N
F_C	Coulomb friction force	N
F_H	Static friction force	N
F_R	Friction force	N
h	Gap height	m
H	Calorific value	J/kg
l_L	Supply line length	m
m	Mass	kg
M	Torque	Nm
n	Pump speed	rpm
p	Pressure	N/m^2
Q	Flow rate	m^3/s
Q_L	Volumetric losses of the cylinder	m^3/s
t	Working cycle duration	s
V	Fuel volume	m^3
V_i	Displacement volume	cm^3/rev
x	Cylinder linear position	m
α	Differential cylinder area ratio	-
γ	Friction force parameter	-
η	Efficiency	-
μ	Dynamic oil viscosity	kg/s/m
λ	Tube resistance number	-
ρ	Density	kg/m^3
ζ	Pressure loss factor	-

Subscript:

A, B	Cylinder chamber A, B
CP	Charge pump
CV	Pilot operated check valves
DC	Displacement control
Diesel	Diesel engine
LS	Load-sensing
L	Hydraulic lines
P	Servo pump
S	Losses

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