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PUMPING UNIT EQUIPPED WITH OSCILLATING HYDRAULIC PRESSURE INTENSIFIER FOR CYLINDERS WITH SMALL OVERALL SIZES AND LARGE LOADS THROUGHOUT THE STROKE

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ABSTRACT

Conveyance of goods and travelling of persons through the tunnels of mines or subway is done on railways of narrow or normal gauge. In interventions in case of accidents, hydraulically operated installations with small dimensions are used for lifting and rerailing the derailed rolling stock. They include hydraulic jacks (double or single acting hydraulic cylinders), supplied with low flow rates at high pressures. Feeding of hydraulic cylinders for lifting and relocating the derailed rolling stock is usually done by means of high-pressure pumps, manually operated or driven by electric / thermal motors.

The authors propose the use of a more cost-effective pumping unit with low-pressure pump and oscillating hydraulic pressure intensifier for this kind of applications - and for similar ones that involve interventions in narrow workspaces.

Such pumping units are usually used in applications designed to achieve and maintain high pressures, either in the volumes of enclosed spaces (endurance tests for pipes and tanks) or at the active stroke end of hydraulic cylinders (hydraulic presses).

The authors demonstrate, on an experimental test bench, that the displacement of a high-pressure hydraulic cylinder, with constant load over the entire stroke, can take place safely even if a low-pressure pumping unit, equipped with oscillating hydraulic pressure intensifier powers it.

Keywords: Pumping unit, low pressure, oscillating hydraulic pressure intensifier, hydraulic cylinder, displacement under load.

INTRODUCTION

An oscillating hydraulic pressure intensifier (minibooster), [1,2] figure 1, is fed through the inlet connection **IN** by a low-pressure pumping unit; it amplifies the pressure in order to supply it, through the outlet connection **H**, to a hydraulic cylinder on the advance stroke. Its structure is based on an assembly of two pistons: one of large diameter and low pressure **LP**, the other of small diameter and high pressure **HP**. These pistons move alternately in both directions, generating three variable volumes: volume 1, generated by the piston **HP**; volume 2, generated by the piston **LP**; volume 3, generated by both

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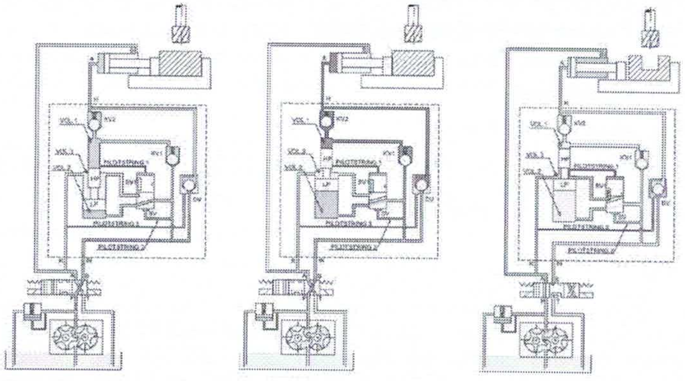
pistons. The reciprocating movement of the two pistons is controlled by a flip-flop distribution slide valve **BV/SV**, hydraulically driven, by the circuits 1 and 2.

The minibooster also includes: two check valves, **KV1** and **KV2**, for the intake / exhaust of hydraulic oil in / from volume 1; a pilot-operated check valve, **DV**, hydraulically driven by the circuit 3, to open in the restrictive direction when the hydraulic cylinder retracts.



a) b) c)

Fig.1: Structure and operation of an oscillating hydraulic pressure intensifier

Figure 1 above shows a classic example of the use of an oscillating hydraulic pressure intensifier: *hydraulic device for clamping a workpiece in order to carry out mechanical machining by cutting*. Inside the device, the minibooster is connected hydraulically in this way: low-pressure connecting fittings **R** and **IN** - to consumers **A** and **B** of the 4/3 electrically operated hydraulic directional control valve; high-pressure connecting fitting **H** – to connecting fitting **A** of the hydraulic cylinder piston chamber; connecting fitting **B** of the hydraulic cylinder rod chamber – to fitting **A** of the directional control valve and fitting **R** of the minibooster.

The operation of the hydraulic pressure intensifier (minibooster), integrated in the clamping device, is as follows:

Step I (fig.1-a), ***advance of the cylinder rod towards the device***. Connections of the 4/3 directional control valve: **P-B, A-T**. The hydraulic oil is sucked out of the tank by the low-pressure gear pump and pumped through the two check valves and the pilot-operated check valve in the piston chamber of the hydraulic cylinder for clamping the workpiece. The hydraulic cylinder rod moves towards the workpiece at maximum speed, since the entire pump flow reaches the piston chamber of the hydraulic cylinder, and the two pistons **LP** and **HP** are stationary.

Step II (fig.1-b), ***achieving and maintaining the clamping force***. Connections of the 4/3 directional control valve: **P-B, A-T**. The cylinder rod slightly bumps against the clamping device; it no longer moves and starts clamping the workpiece in the device. The oscillating hydraulic pressure intensifier comes into operation: the set of pistons **LP+HP** moves alternately linearly up and down, the direction of movement being controlled by the flip-flop distribution slide valve **BV/SV**, depending on the hydraulic controls on the pilot channels **1** and **2**; the check valves **KV1** and **KV2** alternately open / close; the valve **DV** stays closed; the high-pressure hydraulic oil reaches the piston chamber of the hydraulic cylinder through connecting fitting **H**; the low-pressure oil in the hydraulic cylinder rod chamber and volume **3** is driven through connecting fitting **R** and the 4/3 hydraulic directional control valve to the tank. The set of pistons **LP+HP** stops moving when the preset pressure on the gear pump pressure valve is reached, this pressure corresponding to the force required to clamp the workpiece. Should the workpiece clamping force tend to decrease, due to internal flow losses of the hydraulic device, the hydraulic pressure intensifier restarts operating to restore and maintain it.

Step III (fig.1-c), ***retreat of the cylinder rod from the device***. Connections of the 4/3 directional control valve: **P-A, B-T**. Processing of the workpiece is completed. The hydraulic oil is pumped via the 4/3 directional control valve into the hydraulic cylinder rod chamber and into the pilot channel **3** of the valve **DV**. The hydraulic cylinder piston chamber is discharged to the tank, via the valve **DV** and the 4/3 directional control valve, and the hydraulic cylinder rod moves away from the workpiece clamping device.

PUMPING UNIT EQUIPPED WITH OSCILLATING HYDRAULIC PRESSURE INTENSIFIER AND ITS TEST BENCH

In order to demonstrate that low-pressure pumping units equipped with an oscillating hydraulic pressure intensifier can also be used in ***displacement of cylinders with large loads throughout stroke***, the experimental test bench [3] and the pumping unit shown in figures 2 and 3 have been built, with the following structure:

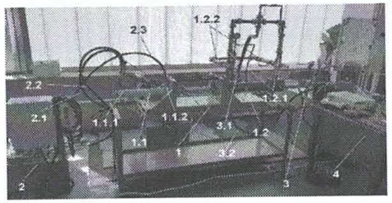


Fig.2: Experimental test bench and pumping unit with oscillating hydraulic pressure intensifier

Fig.3: Control and data acquisition module on the test bench

1: ***Module for clamping the hydraulic cylinders***; **1.1**: test cylinder with piston Ø = 38.1 mm, rod Ø = 25 mm, stroke = 257 mm, p_{max}=700 bar; **1.1.1**: fitting that connects cylinder piston chamber with secondary of the minibooster; **1.1.2**: connecting fitting and hose for cylinder rod chamber; **1.2**= load cylinder with built-in stroke transducer and: piston Ø = 80 mm, rod Ø = 45 mm, stroke = 300 mm, p_{max}=300 bar; **1.2.1**= cylinder chambers inlet check valves; **1.2.2**= cylinder chambers outlet check valves;

2: ***Low-pressure pumping unit*** equipped with: electric pump (gear pump + electric motor) of 4 kW, 200 bar, 10.5 l/min; oil tank with V= 38 l; fill and vent filter, and return filter; hydraulic block with pressure filter, pressure control valve, 4/3 hydraulic directional control valve; HC7 minibooster, 5:1 amplification ratio, outlet pressure of 0...1000 bar, outlet flow rate of 10.5...1.2 l / min; electric panel; **2.1, 2.2**= minibooster primary connection hoses;

3: ***Pumping station for filling the load cylinder*** equipped with: electric pump of 2 kW, 10 bar, 90l/min; oil tank with V= 180 l; filling pressure control valve; proportional load

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control valve; fill and vent filter; return filter; **3.1**= cylinder chambers fill connecting fitting and hose; **3.2**= cylinder chambers drain connecting fitting and hose;

4: ***Control and data acquisition module***, figure 3, acquiring data from: *1 stroke transducer* (built into the load cylinder), figure 4; *3 pressure transducers* for p_1= discharge pressure of the pump within the minibooster primary, figure 5, p_2= load pressure and p_3= discharge pressure of the filling pump, figure 6;

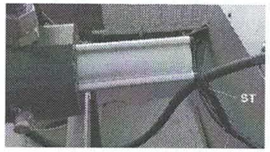

Fig.4: Stroke transducer (ST), built into the load cylinder

Fig.5: Pressure (p1) and flow (Q1) transducers

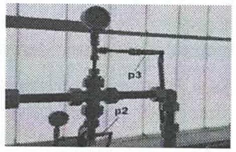

Fig.6: Pressure transducers (p2 and p3)

2 flow transducers for Q_1= test cylinder output flow rate, figure 5, Q_2= proportional valve transit flow rate, figure 7; *2 acceleration transducers* for Acc_1= acceleration measured in the direction of travel of the minibooster pistons, figure 8, Acc_2= acceleration measured in the direction of travel of the hydraulic cylinders, figure 9.

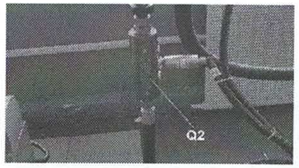

Fig.7: Flow transducer (Q2)

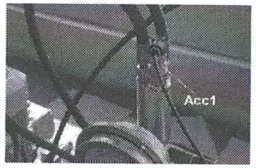

Fig.8: Acceleration transducer (Acc1)

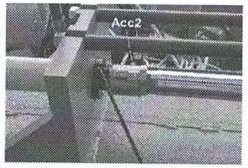

Fig.9: Acceleration transducer (Acc2)

EXPERIMENTAL TESTS ON A HYDRAULIC CYLINDER WITH CONSTANT LOAD THROUGHOUT THE STROKE, WHICH IS DRIVEN BY A PUMPING UNIT EQUIPPED WITH A MINIBOOSTER

The experimental tests, [4-9], have been conducted under the following conditions:

a) On the test bench shown in fig.2 and fig.3, the opening pressure of the normally closed proportional valve has been adjusted to an average value of ***181 bar***, which allows the load cylinder to develop an average resistance force of

$$(181x10^5)x(\pi/4)x(80x10^{-3})^2=\mathbf{90.9x10^3\ N} \quad (1)$$

b) The opening pressure of the normally closed valve that the low-pressure pumping unit is equipped with has been set to ***160 bar***. For this value, in the piston chamber of the test cylinder, fed from the high-pressure connecting fitting of the minibooster, a pressure of ***160x5 = 800 bar*** has been installed. Throughout the advance stroke, the test cylinder developed an average active force (as opposed to the resistance force of the load cylinder) of

$$(160x5x10^5)x(\pi/4)x(38.1x10^{-3})^2=\mathbf{91.2x10^3\ N} \quad (2)$$

c) The opening pressure of the safety valve of the pump filling the load cylinder has been adjusted to ***19 bar***.

d) The experimental tests have been performed on the advance stroke of the test cylinder.

e) Data acquisition, performed at a speed of ***200 samples / s*** along ***44 s***, has been made on a segment of ***27.6 mm*** (within the range ***35.1...62.7 mm***) of the test cylinder total stroke of ***257 mm***.

Under the conditions a)...e), the following characteristics have been determined experimentally: ***acceleration analysis curves***, in the time domain, figure 10, and in the frequency domain, figure 11, specific to the minibooster and hydraulic cylinders; ***time variation curves of pressures***, figure 12, from the minibooster primary, load cylinder piston chamber, filling pump discharge line; ***time variation curves of flow rates***, figure 13, from the minibooster primary and load cylinder piston chamber; ***time variation curve of displacement*** of the rods of the two (test and load) hydraulic cylinders, figure 13.

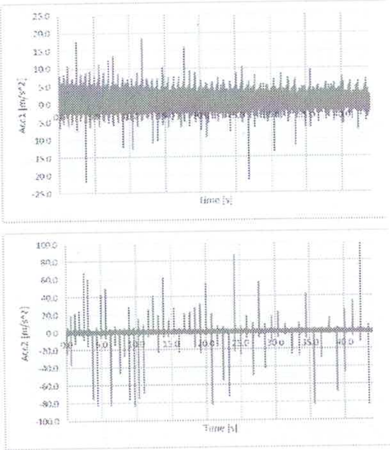

Fig.10: Time variation of the acceleration for: minibooster (top); rods of hydraulic cylinders (bottom)

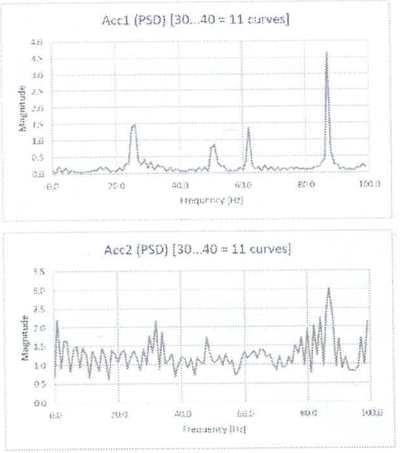

Fig.11: Power spectral density (PSD): minibooster (top); rods of hydraulic cylinders (bottom)

The reciprocating movement of the minibooster set of pistons **LP+HP** leads to a variation of the acceleration amplitudes in the time domain, fig.10, measured in: ***the direction of movement of the minibooster pistons***, with peak value=22 m/s^2 and peak-to-peak value=40 m/s^2; ***the direction of movement of the hydraulic cylinders***, with peak value=100 m/s^2 and peak-to-peak value=180 m/s^2.

The power spectral density of vibration (PSD), figure 11, highlights the magnitude variation (the ratio between the maximum and minimum amplitude of the acceleration) in the frequency domain, measured in: ***the direction of movement of the minibooster pistons***, with peak value=3.7 (for 87 Hz); ***the direction of movement of the hydraulic cylinders***, with peak value=3 (for 87 Hz). The magnitude of vibrations across the entire frequency spectrum is relatively small, at both points where it has been measured. On the graphs in figure 11 one can notice that a damping phenomenon occurs for the frequency of 87 Hz, where magnitude of vibrations decreases from 3.7 to 3.

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Due to time variations in cylinder speed, gear pump flow and pressure intensifier flow in hydraulic cylinder rods, vibrations occur, whose power must be evaluated because the amplitude of the accelerations is relatively large (± 80 m/s^2). In some applications of hydraulic drives, based on pumping units equipped with minibooster, depending on the oscillation frequency of the driven structure, this evaluation of the vibration power can prevent the occurrence of mechanical resonance.

In fig.12 one can notice that the pressure pulsations in the minibooster primary, around the average value of *160 bar*, due to the operating mode of the minibooster, cause pressure pulsations in the piston chamber of the load cylinder, around a linear value, with a decreasing slope, from *183 bar* to *180 bar*, over the time range of *0...44 s*, while the pressure of the filling pump of the load cylinder shows pulsations around an average value of 19 bar.

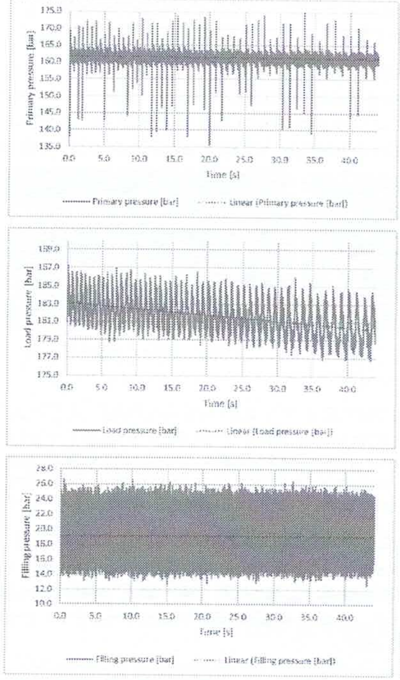

Fig.12: Time variation of pressures for: primary (top); load (middle); filling (bottom)

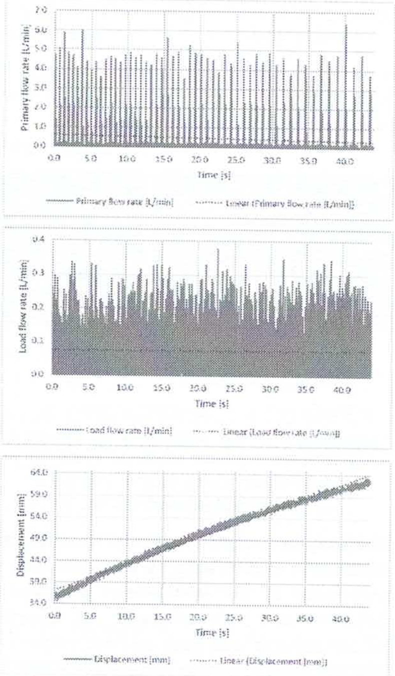

Fig.13: Time variation for: primary flow rate (top); load cylinder flow rate (middle); displacement of hydraulic cylinders (bottom)

Figure 13 shows another aspect caused by the operating mode of the minibooster: the occurrence of flow pulses. Thus, in the primary of the minibooster, flow pulses occur, with maximum values of 6 l/min and a linear value of 0.05 l/min., while in the load cylinder the flow pulses have maximum values of 0.35 l/min and a linear value of 0.08 l/min. These flow pulses cause small deviations from the continuous and linear movement of the hydraulic test cylinder, that is small decelerations and accelerations during the movement. Deviations from the uniform displacement of the cylinder are highlighted in the detail shown in the figure 14; the velocity varies cyclically, due to the operating mode of the minibooster.

Integrating the acceleration values measured with the accelerometer ***Acc2*** in relation to time, the displacement velocity of the cylinders, shown in the figure 15, has been obtained; it varies due to the pulsating nature of the minibooster flow, the compressibility of the hydraulic system and its elasticity. The mean velocity has a value of ***0.6 mm/s***; this results from the calculation below and the average value shown on the graph.

$$Velocity\ mean\ value = \frac{(0.0627-0.0355)\ m}{43.995\ s} = 6.183 \times 10^{-4}\ \frac{m}{s} = 0.62\ \frac{mm}{s} \quad (3)$$

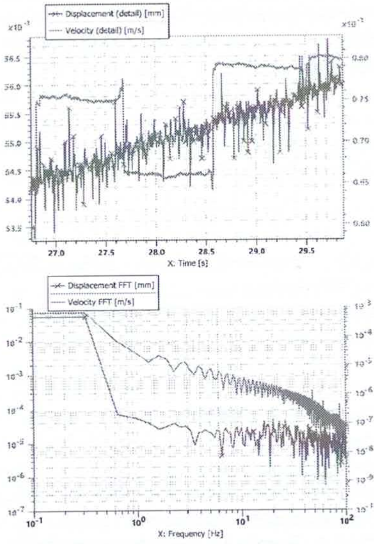

Fig.14: Detail-Time variation of *velocity* and *displacement* of hydraulic cylinders, with a load of 800 bar on the entire stroke and FFT

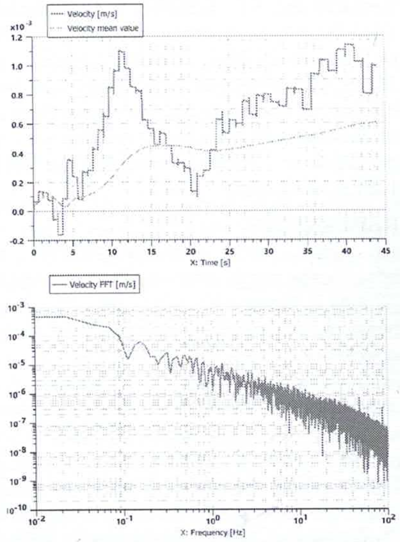

Fig.15: Time variation of *velocity* of hydraulic cylinders and FFT (Fast Fourier Transform)

CONCLUSION

In the article, it has been shown experimentally that the movement of a high-pressure hydraulic cylinder with a small size and constant load throughout the stroke can be done safely by means of a cheaper solution: *supplying the cylinder by a low-pressure pumping unit equipped with an oscillating hydraulic pressure intensifier (minibooster).*

The pulsing mode of operation of the minibooster, with a frequency of 0.01..20 Hz, caused by the alternating symmetrical movement (*up:* discharge of hydraulic oil from the high-pressure chamber; *down:* suction of hydraulic oil in the high-pressure chamber) of the set of the two pistons **LP+HP** and the hydraulically operated bistable distribution valve **BV/SV**, does not induce dangerous shocks that could affect the mechanical strength of the hydraulic cylinder or its drive system *(maximum measured acceleration =17 g)*.

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Pressure and flow pulses occurring on the supply circuit of the test cylinder piston chamber while its rod moves with a load of ***800 bar*** cause deviations from the linearity of the displacement of ± ***1.5 mm*** at most.

For moving large loads at low speeds in narrow spaces (rerailing the derailed rolling stock in tunnels of mines or subway, extrication tools used in car accidents, etc.), the solution of using low-pressure pumping units equipped with minibooster for driving hydraulic cylinders with high working pressures and small dimensions has been experimentally proven to be possible.

ACKNOWLEDGEMENTS

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