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Use of pumping units equipped with oscillating hydraulic pressure intensifiers for displacement of cylinders with heavy loads over the entire stroke

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Teodor-Costinel POPESCU [1*], Alexandru-Polifron CHIRIȚĂ [1], Ana-Maria Carla POPESCU [1], Alina-Iolanda POPESCU [1]

Author's affiliations and addresses:

[1] National Institute of Research & Development for Optoelectronics / INOE 2000-subsidiary Hydraulics and Pneumatics Research Institute –IHP Bucharest, Romania

*** Correspondence:**

e-mail: popescu.ihp@fluidas.ro

Abstract:

Working and moving in confined, limited and narrow spaces, specific to underground mining activities, also requires the use of hydraulically operated equipment, capable of developing large forces, with small dimensions. An example of such equipment includes pumping units comprising low-pressure electric pumps and oscillating hydraulic pressure intensifiers. They use low pressure in the primary side of the intensifier and generate high pressure in the secondary side of the intensifier. Such pumping units are usually used to achieve and maintain high pressure, either in the volumes of closed spaces (in strength tests on pipes and tanks) or at the end of the active stroke of hydraulic cylinders (in hydraulic presses). On an experimental laboratory bench, which comprises a test cylinder, powered by a pumping unit, equipped with an oscillating hydraulic pressure intensifier, and a load cylinder, powered by another pumping unit, with the possibility of load control, the authors show that: the application range of these pumping units can be extended in the third direction, too, useful for underground mining activities, namely for drive of hydraulic cylinders with low gauge / displacement speeds and constant high load (high working pressure) over the entire working stroke length; the uniformity of displacement of these cylinders, with load throughout the stroke length, which are powered and driven with such pumping units, is slightly affected by the pulsating mode of operation of the hydraulic pressure intensifier. A set of experimental measurement results is presented for a constant value of the load over the entire displacement stroke of the test cylinder.

Keywords: low-pressure pumping unit, oscillating hydraulic pressure intensifier, high pressure, hydraulic cylinder

1. Introduction

The operating regime of hydraulic drive systems includes certain work phases in which the actuated displacement engines, hydraulic cylinders and rotary hydraulic engines have to develop large static or dynamic forces and torques, and consequently have to be powered at high pressure rates.

These high pressure rates, of the order of hundreds or thousands of bars, can be generated by two types of pumping units:

a) ***pumping units*** comprising ***displacement pumps and high-pressure hydraulic equipment***; they directly supply linear or rotary displacement hydraulic engines;
b) ***pumping units*** comprising ***displacement pumps and low-pressure hydraulic equipment***; they supply pressurized closed volumes or displacement hydraulic engines by means of ***hydraulic pressure intensifiers***, which are placed between the pumping unit and the hydraulic consumers.

The first category of pumping units is more expensive and, as a rule, is suitable in dynamic applications, where linear or rotary movement of large loads with uniform speed rates is required. This category is not energy efficient for static applications, where the aim is to produce and maintain high pressure for a long time in a closed enclosure, or for dynamic applications that, for example, require high pressure rates only over a short segment of the hydraulic cylinder advance stroke, and the rest of the stroke, as well as the entire retreat stroke, must be done at high speed rates and low pressure.

The pumping units under the second category make the energy-dissipating applications of the first category more energy efficient, but they have the disadvantages of low and pulsating flow rates for high-load dynamic applications. They are developed in a wide range of construction variants, intended to offer an optimal solution, with reduced pulsations and increased flow rates for dynamic applications of short duration.

Hydraulic pressure intensifiers with oscillating pistons are known in the specialized literature under several names: *oscillating hydraulic pressure amplifiers (intensifiers)* [1,2,3], *oscillating pumping units, boosters, miniboosters* [4,5,6].

There is a diverse range of construction solutions for miniboosters, single or double acting in relation to high-pressure pumping, designed to boost the pressure of low-pressure pumping units. They have been tested experimentally [7,8,9], investigated by mathematical modeling / numerical simulations [10,11,12,13] and used in various applications of hydraulic drive systems [14,15,16,17]. They can power, for example, single-acting or double-acting hydraulic cylinders that move large loads linearly at the end of the advance stroke, or achieve and maintain high pressure in a closed enclosure. They are mounted between the pumping unit and the actuated cylinder (Fig. 1), as close to the latter as possible, and comprise: an oscillator (**OP**), consisting of two oscillating pistons of different diameters and a 3/2 bistable directional control valve; two check valves (**KV1**, **KV2**); a pilot-operated check valve (**DV**), bypassing the oscillator. The low-pressure connecting fittings **IN** (inlet) and **R** (return), in the primary side of the minibooster, are connected to consumers of the hydraulic directional control valve of the pumping unit, and the high-pressure connecting fitting **H**, in the secondary side of the minibooster, is connected to the piston chamber of the actuated cylinder.

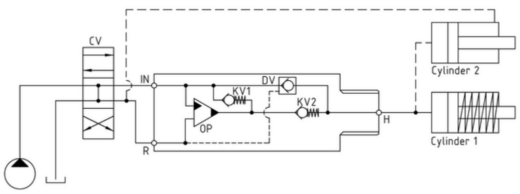

Fig. 1. Simplified hydraulic diagram of a minibooster

State of the art in technical applications related to the use of low-pressure pumping units, equipped with oscillating hydraulic pressure intensifiers (miniboosters), does not include applications involving displacements of hydraulic cylinders that have high loads over the entire working stroke length. ***The objective*** of this paper is to experimentally demonstrate the possibility of using miniboosters in such applications, too, with reasonable limits of the displacement uniformity and continuity of the actuated cylinders, under the conditions of the pulsating operation mode of these pressure intensifiers [18, 19, 20].

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2. Materials and Methods

To achieve the intended objective, the authors have used an experimental method by which they have monitored the dynamic behaviour of a hydraulic cylinder, which, being fed into the piston chamber through a minibooster, moves with a constant load, equivalent to the pressure of $700x10^5$ *Pa*, over the entire stroke length. The minibooster has been integrated into a low-pressure pumping unit, while the hydraulic cylinder under testing – into a test bench.

2.1. Low-pressure pumping unit equipped with minibooster

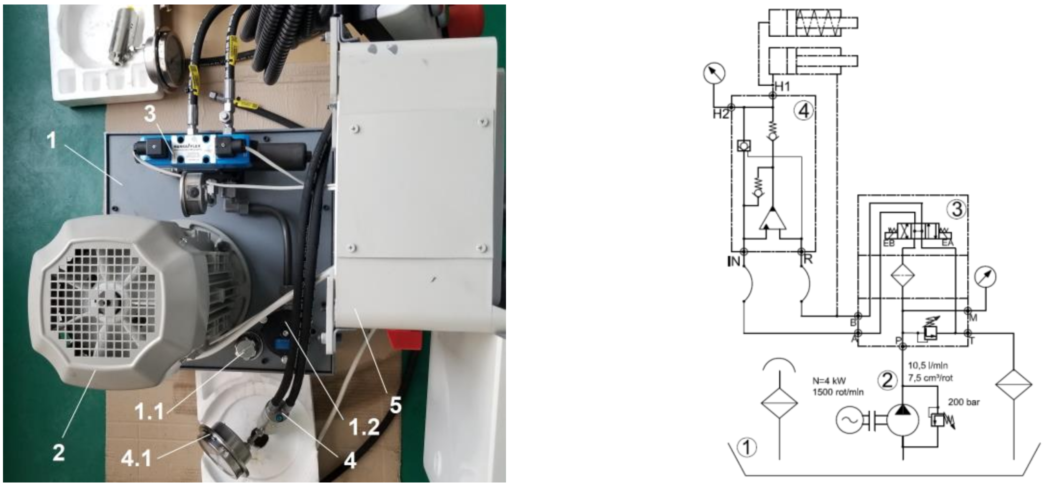

Fig. 2. Pumping unit equipped with minibooster (left side – general view; right side - hydraulic diagram)

The pumping unit in Fig. 2 comprises: **1**= oil tank ($38x10^{-3}$ m^3 volume); **1.1**= fill and vent filter; **1.2**= return filter; **2**= low-pressure electric pump (*4 kW*; *1500 rev/min*; $7{,}5x10^{-6}$ m^3/rev; $250x10^5$ *Pa*); **3**= block with hydraulic devices (pressure control valve; pressure filter; 4/3 hydraulic directional control valve, DN6, electrically actuated; $250x10^5Pa$ pressure gauge); **4**= HC7 minibooster (*5:1* amplification ratio; $0...200x10^5$ *Pa* pressure in the primary side; $0...1000x10^5$ *Pa* pressure in the secondary side; $14x1{,}667x10^{-5}$ m^3/s low-pressure inlet maximum flow rate; $1{,}6x1{,}667x10^{-5}m^3/s$ amplified pressure outlet maximum flow rate); **4.1**= $2500\ x10^5$ *Pa* pressure gauge; **5**= electric panel.

Pressure adjustment in the circuit **H1**, which supplies the hydraulic cylinder actuated by the pumping unit, is done using the pressure control valve on the hydraulic block of the unit, and the adjusted value of this pressure can be read on the high-pressure gauge, mounted in the high-pressure connecting fitting **H2.**

2.2. Test bench for low-pressure pumping unit equipped with minibooster

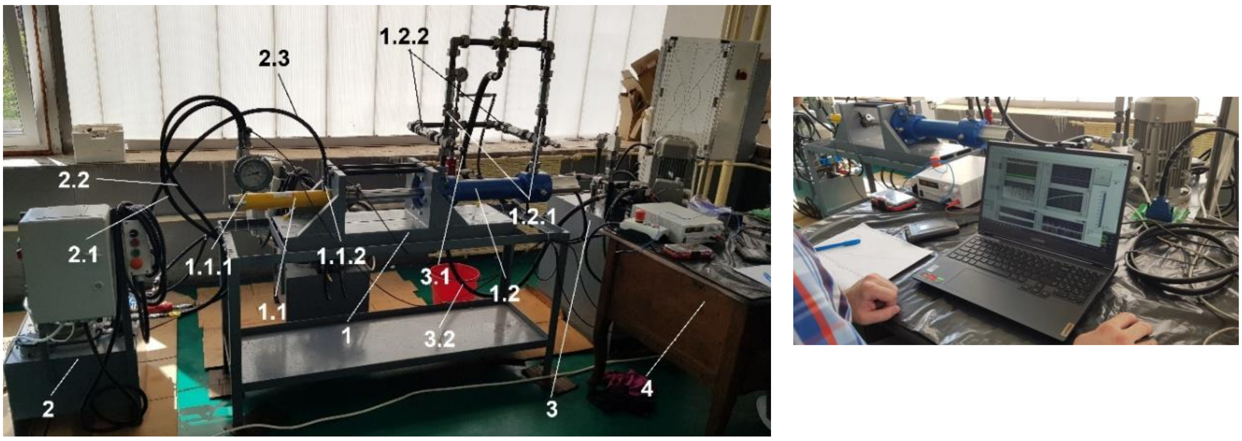

Fig. 3. Test bench for pumping unit equipped with minibooster

The structure of the test bench and the pumping unit in Fig. 3 is as follows:

1: Module for clamping the hydraulic cylinders; **1.1**= test cylinder with piston Ø= $38{,}1x\ 10^{-3}\ m$, rod Ø= $25x10^{-3}\ m$, stroke length= $257x10^{-3}\ m$, p_{max}= $700x10^{5}\ Pa$; **1.1.1**= fitting that connects cylinder piston chamber with secondary side 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= $80x10^{-3} m$, rod Ø= $45x10^{-3}\ m$, stroke length= $300x10^{-3}\ m$, p_{max}= $300x10^{5}\ Pa$; **1.2.1**= cylinder chambers inlet check valves; **1.2.2**= cylinder chambers outlet check valves;

2: Low-pressure pumping unit (previously presented); **2.1, 2.2**= hoses connecting the primary side of the minibooster to the consumers of the hydraulic directional control valve of the unit;

3: Pumping station for filling the load cylinder equipped with: electric pump of $2\ kW$, $10x10^{5}Pa$, $90x1{,}667x10^{-5}\ m^3/s$; oil tank with V= $180x10^{-3}\ m^3$; filling pressure control valve; proportional pressure control valve (acting as load); fill and vent filter; return filter; **3.1**= cylinder chambers fill fitting and hose; **3.2**= cylinder chambers drain fitting and hose;

4: Control and data acquisition module (Fig. 3 - right side) acquiring data from the transducers of: acceleration, measured in the direction of displacement of the minibooster pistons (**Acc1**); acceleration, measured in the direction of displacement of the hydraulic cylinders on the test bench (**Acc2**); minibooster primary side pump pressure (**p1**); load cylinder pressure (**p2**); filling pump pressure (**p3**); minibooster primary side input flow rate (**Q1**); load cylinder flow rate (**Q2**); **stroke** (built into the load cylinder).

2.3. Test conditions for the pumping unit equipped with a minibooster

In the following Item the results of the experimental tests for a constant value of the test cylinder load, namely $\boldsymbol{700x10^{5}\ Pa}$, equivalent to the resisting force of $\boldsymbol{80x10^{3}\ N}$, which has been maintained throughout the advance stroke, are presented. The load value of the test cylinder has been adjusted using the proportional valve on the pumping station for filling the load cylinder.

The experimental tests have been carried out under the following conditions: on the test bench shown in Fig. 3, opening pressure of the normally closed valve with which the low-pressure pumping unit is provided, has been adjusted to $\boldsymbol{160x10^{5}\ Pa}$ (equivalent to max. $\boldsymbol{800x10^{5}\ Pa}$ in the secondary side of the intensifier); pressure to open the safety valve of the load cylinder filling pump has been adjusted to $\boldsymbol{19x10^{5}\ Pa}$; the tests have been carried out only during the advance stroke of the test cylinder; data acquisition has been done with a sampling frequency of ***200 samples/s*** and duration of ***20 s***, exploring a number of ***4000 samples*** for a segment of $\boldsymbol{169{,}6x10^{-3}\ m}$ from the total stroke **(**$\boldsymbol{257x10^{-3}\ m}$**)** of the test cylinder.

3. Results

Testing the pumping unit equipped with a minibooster, presented in Item 2.1., has aimed at determining the influence of the pulsating operation mode of this type oscillating pressure intensifier on the displacement continuity and uniformity of the test cylinder which is a part of the test bench, under the conditions in which it is "loaded" with the load equivalent to the working pressure of $700x10^{5}\ Pa$.

To capture the physical phenomena that occur when the intensifier is fed with the full flow rate of the pump, when it bypasses the intensifier through the pilot-operated check valve DV (Fig. 1), first the data acquisition has been started, then the pumping unit. On the graph in Fig. 7, which shows the time-variation of the flow rate of the pumping unit, it can be seen that, for less than a second, the pump flow bypasses the intensifier and the average flow rate Q1 decreases from $9x1{,}667x10^{-5}\ m^3/s$ to $7x1{,}667x10^{-5}\ m^3/s$.

The data from eight transducers mounted on the test bench have been acquired, and the results of their graphical processing are presented further on.

3.1. Graphical processing of data acquired with the acceleration transducers

The first factor disturbing the displacement continuity and uniformity of the test cylinder includes ***mobile parts*** of the minibooster, which have a symmetrical alternating displacement with small amplitudes (vibrations) and variable frequency rates in the range 0...20 Hz, namely: the assembly of the two (low and high pressure) pistons; the 3/2 bistable directional control valve, with bistable position and hydraulic control; the check valves for the intake and exhaust of oil into / from the high-pressure piston chamber. The accelerations generated by the vibrating parts of the minibooster have been detected with the acceleration transducer **Acc1**, mounted on the minibooster housing (perpendicular to its longitudinal axis), and the accelerations generated by the displacement of the test cylinder rod have been detected with the acceleration transducer **Acc2**, mounted on the cylinder rod.

Fig. 4 shows the time-variations of the acceleration of the vibrating parts of the minibooster (top), and of acceleration of the test cylinder rod (bottom), when it operates at the working pressure of $700x10^5$ *Pa*, linearly displacing the load of $80x10^3$ *N*.

It is found that variation of acceleration of the vibrating parts of the minibooster has an average value of ±*60* m/s^2, and variation of acceleration of the hydraulic cylinder rod - an average value of ±*80* m/s^2. The negative influence of the vibrating parts of the minibooster on the displacement continuity of the cylinder rod is found in the ±*20* m/s^2 increase of its vibration amplitude.

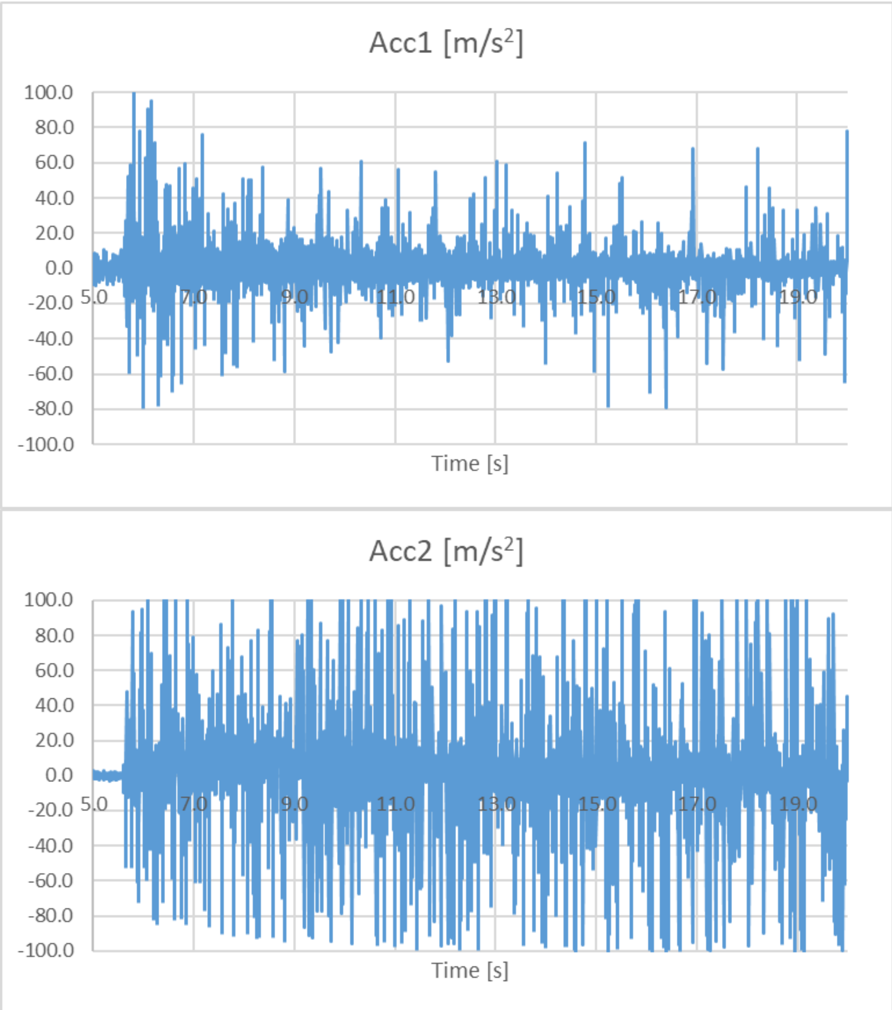

Fig. 4. Time-variation of acceleration measured with Acc1 (top) and Acc2 (bottom)

The accelerations measured with the two transducers have been processed using the power spectral density (PSD) function. The results, depicted graphically in Fig. 5, show the strength of the vibrational

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energy variations as a function of frequency, variation measured in the direction of: displacement of the minibooster pistons, Fig. 5-top; displacement of the hydraulic cylinders, Fig. 5-bottom. At the maximum operating frequency of the minibooster, namely 20 Hz, it is found that the magnitude has the value of approx. 5, resulting from the Fourier transformation of the data acquired with the transducer **Acc1**, and the magnitude of approx. 40, resulting from the processing of the data acquired with the transducer **Acc2**, corresponds to the direction of displacement of the load cylinder.

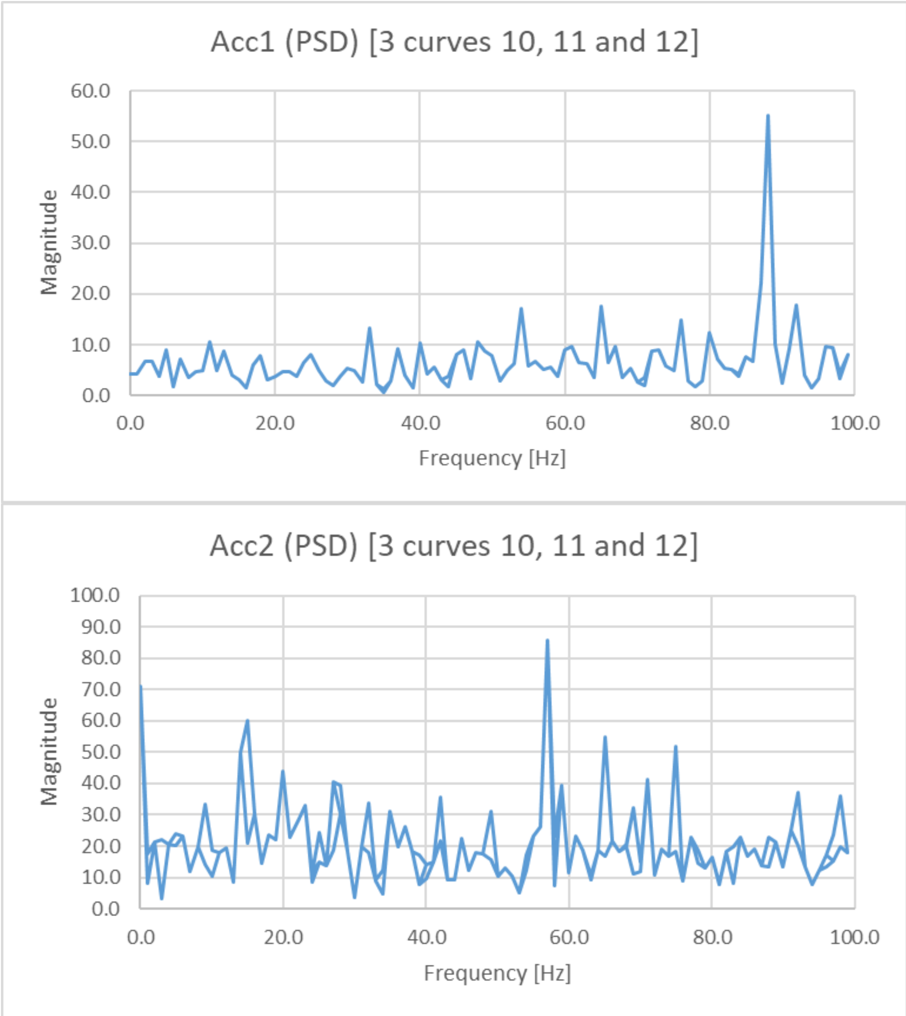

Fig. 5. Power spectral density determined with Acc1 (top) and Acc2 (bottom)

3.2. Graphical processing of data acquired with the pressure transducers

The second factor disturbing the displacement continuity and uniformity of the test cylinder is ***the time-variation of the pressure*** in the high-pressure piston chamber of the minibooster. It has a pulsating mode of operation, with symmetrical alternating cycles, of constant amplitude and variable frequency (*0...20 Hz*), for oil suction and discharge into/from its chamber.

Fig. 6 shows the time-variation of the pressure in: the primary side of the minibooster, measured with transducer **p1** (top); the load cylinder piston chamber (secondary side of the minibooster), measured with transducer **p2** (centre); the discharge circuit of the filling pump, measured with transducer **p3** (bottom). Pressure values measured with transducer **p2** are multiplied by **4.41**, which is the value of the ratio of the piston areas of the two (load and test) cylinders on the bench.

From the graphical representation, it can be seen that pressure variation in the primary side of the minibooster is around the average value of $150x10^5$ *Pa*, pressure variation in the load cylinder is around the average value of $700x10^5$ *Pa* (4,41x159), and pressure variation in the load cylinder filling circuit is around the average value of *19x10⁵ Pa*. ***It is found that the pressure pulsations of the minibooster are lower than those of the gear pump in the structure of the tested pumping unit.***

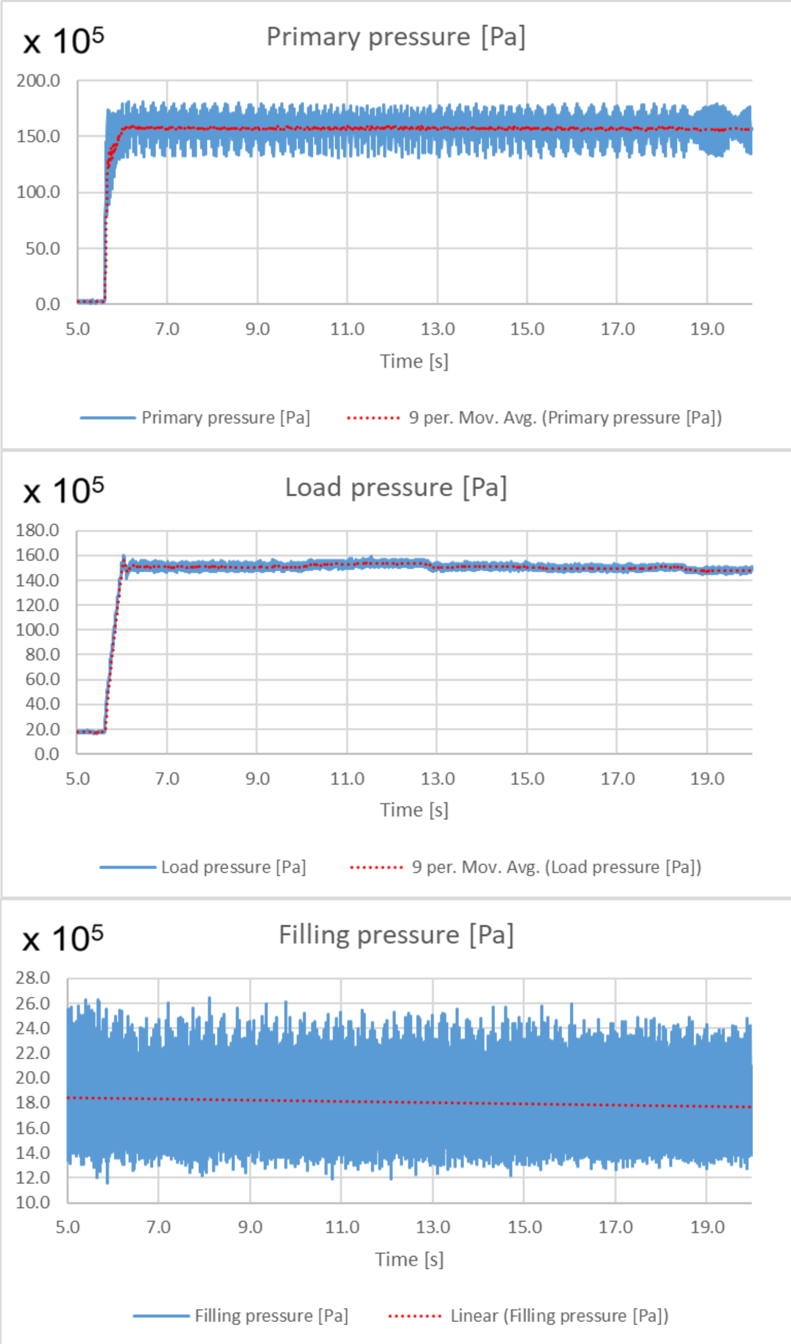

Fig. 6. Time-variation of pressure measured with p1 (top), p2 (centre) and p3 (bottom)

3.3. Graphical processing of data acquired with the flow transducers

The third factor disturbing the displacement continuity and uniformity of the test cylinder is ***the time-variation of the flow rate*** in the high-pressure piston chamber of the minibooster.

Fig. 7 shows the time-variation of the flow entering, at low pressure, the primary side of the minibooster, measured with the transducer **Q1** (top), and the variation of the flow rate in the piston chamber of the load cylinder, identical to the flow rate that comes out, at high pressure, from the secondary side of the minibooster, measured with the transducer **Q2** (bottom).

Variation of the flow rate in the primary side of the minibooster is around the average value of *7x1,667x10^{-5} m^3/s*, and variation of the flow rate in the load cylinder is around the average value of *2,7x1,667x10^{-5} m^3/s.*

During the displacement of the test cylinder under the *700x10^5 Pa* load ***it is found that variation of the flow rate coming out from the secondary side of the minibooster at high pressure is less than variation of flow rate entering the primary side of the minibooster at low pressure***.

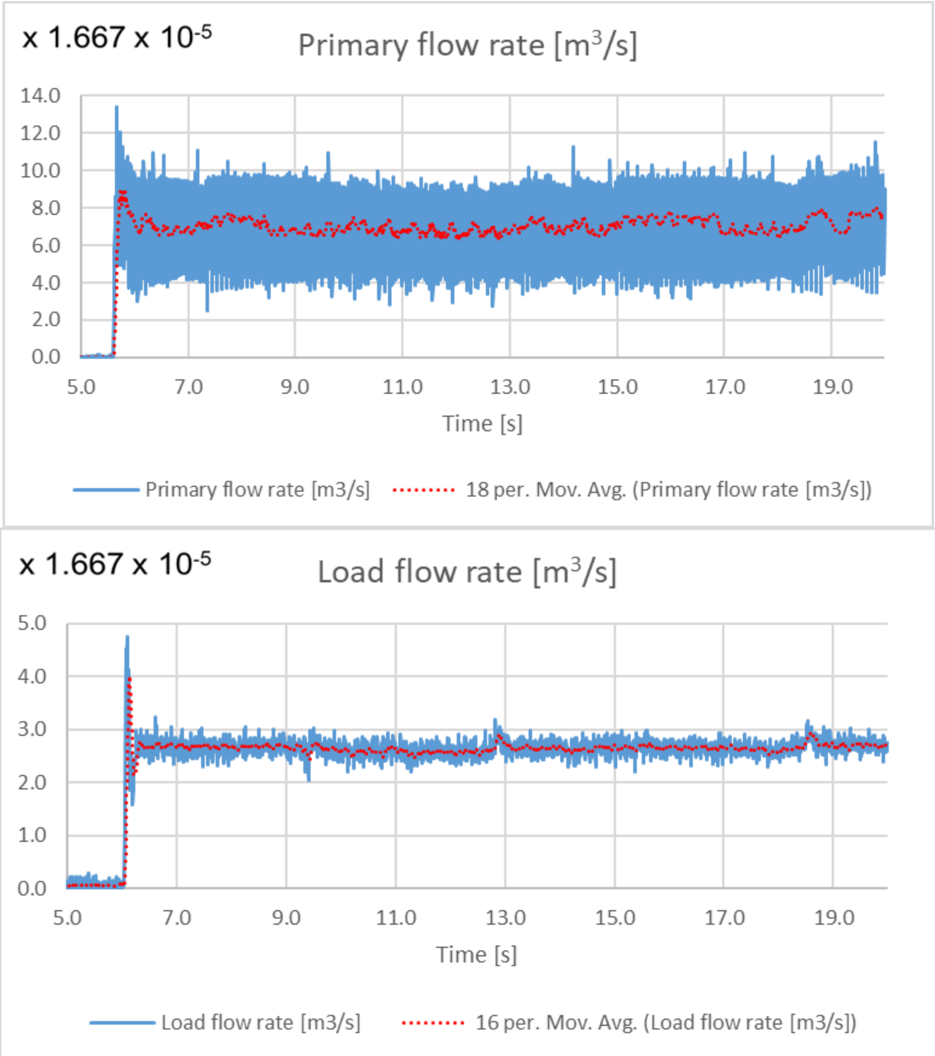

Fig. 7. Time-variation of flow rate measured with Q1 (top), and Q2 (bottom)

3.4. Graphical processing of data acquired with the displacement transducer

The cumulative effect of the three factors disturbing the displacement continuity and uniformity of the test cylinder results from the graphical processing of the data acquired with the displacement transducer. Fig. 8 shows the time-variation of the displacement of the test cylinder, and Fig. 9 – a representative detail of the deviations from the linearity of this displacement. It can be noticed that, over the entire stroke of displacement under load of the cylinder (*180x10^{-3} m*), the maximum value of these deviations is approximately *1x10^{-3} m*.

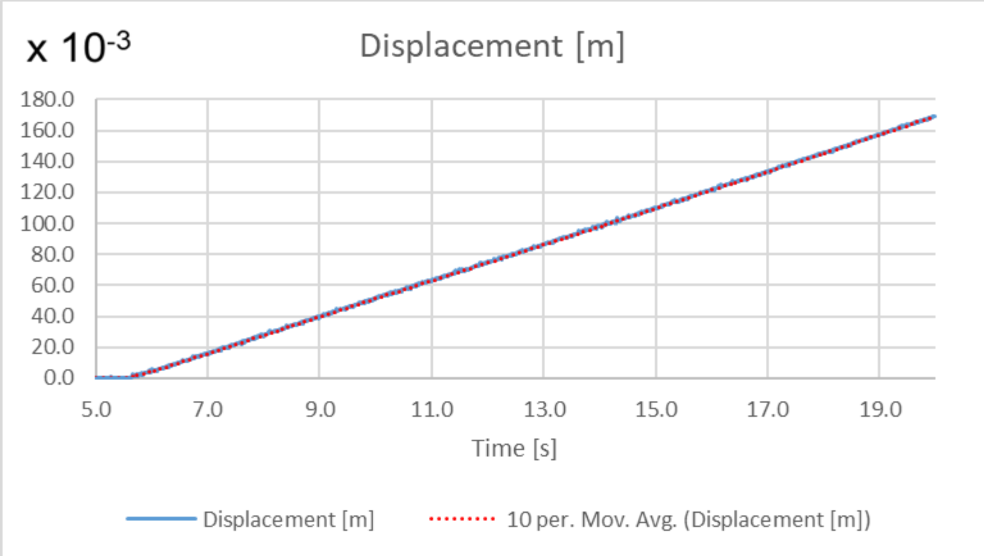

Fig. 8. Time-variation of the displacement of the test cylinder at 700 x 10^5 Pa load

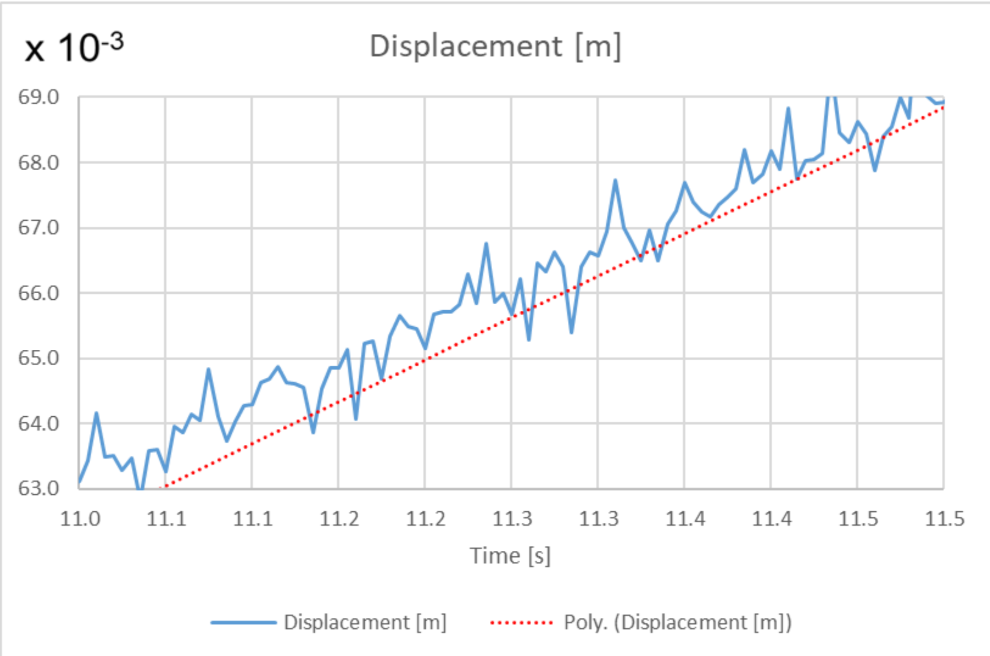

Fig. 9. Deviations from the linearity of the displacement of the test cylinder under load

4. Discussion

It is experimentally demonstrated that the compact and small size pumping units comprising low-pressure electric pump, oscillating hydraulic pressure intensifier, hydraulic directional control valve, pressure control valve and filters, can be used for safe displacement of hydraulic cylinders, with a constant load throughout the stroke, being suitable for activities in confined spaces (e.g., mining activities).

The pulsating mode of operation of the minibooster, caused by the alternating symmetrical displacement of its mobile parts, with the frequency of 0,01...20 Hz, as well as flow and pressure pulsations do not induce dangerous shocks that could affect the mechanical strength of the hydraulic cylinder and its drive system.

Deviations from the linearity of the displacement are small, and the displacement speed at high loads increases if the normally closed valve of the pumping unit is adjusted to a higher opening pressure than that corresponding to the load.

The vibrating parts of the minibooster, which move linearly, bidirectionally, over short distances and with variable frequencies in the range of 0,01...20 Hz, are the main cause of deviations from the linearity of the displacement of the test cylinder under load.

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