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NUMERICAL INVESTIGATIONS ON THE WORKING CYCLE OF A HYDRAULIC BREAKER: OFF-DESIGN PERFORMANCE AND INFLUENCE OF DESIGN PARAMETERS

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Abstract

This paper deals with theoretical considerations and numerical simulations concerning the working behaviour of a hydraulic breaker. At first, some formulations coming from sufficiently reasonable considerations on the working principle of the breaker, based on the hypothesis of the motion of the striking mass as uniformly accelerated, are proposed. Later, a previously realized parameterised model is used in order to investigate the influence of the inlet flow rate and of the most important design parameters on the behaviour of the machine. This analysis allows the characterization of these parameters affecting breaker performance, suggesting possible design improvements which may lead to better performance in terms of both impact energy and efficiency.

Keywords: hydraulic breaker, working cycle, simulation, design parameters

1 Introduction

Hydraulic breakers are machines turning the hydraulic energy supplied by a positive displacement pump into mechanical energy in terms of percussions against a chisel. The latter is appointed to crumble a certain material by means of alternating impacts.

A thorough study of the functioning of the machine was developed by Giuffrida and Laforgia (2005) with a recent work oriented to analyse its working behaviour by means of computer modelling and simulation. A very detailed parameterised model, allowing the reproduction of physical phenomena and measurement results, was presented and the functioning of the breaker was discussed with reference to the working pressure and frequency and to the dynamics of the moving masses. That model represented a useful tool to better understand certain physical behaviours characteristic of the functioning of the machine.

The current work follows the previous one and aims at shedding more light on this typology of hydraulic machines. As a matter of fact, scientific literature seems to be very poor in technical papers dealing with hydraulic breakers: only three other papers (Gorodilov, 2000; 2002 and 2005) all by the same author may be found, according to a preliminary bibliographic research. The first two papers were known to the authors and already mentioned in Giuffrida and Laforgia (2005). The third paper (Gorodilov, 2005) is a recent work, carried out by a mathematical point of view, in order to present analysis techniques for hydraulic percussion systems. As in the previous works, the author makes use of similarity criteria to compare systems with different sets of elements.

This work, with respect to the cited ones, i.e. to the state of the art in the current literature first presents the development of simple but original formulations concerning the working characteristics of the breaker, with particular reference to its efficiency. Later, once some design parameters are characterized as relevant, their influence is better investigated by means of a numerical model simulating the functioning of the breaker. Such a model was used to run new simulations in order to investigate both off-design performance of the machine, due to variations in inlet flow rate, and its behaviour as a function of variations of some design parameters.

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2 Theoretical Considerations on the Working Cycle of the Breaker

The model simulating the working behaviour of the hydraulic breaker presented by Giuffrida and Laforgia (2005) was based on the machine dealt with in European Patent no. 0085279 and European Patent no. 0426928. Figure 1 shows a schematic drawing of the breaker.

Here, referring to Giuffrida and Laforgia (2005), European Patent no. 0085279, and European Patent no. 0426928 in regards to the functioning of the hydraulic machine, some considerations will be formulated in order to realize what design parameters really influence the working characteristics and the efficiency of the breaker.

The motion of the striking mass (PIS) is essentially governed by hydraulic forces due to pressure levels in the downward push chamber (V1) and in the upward push chamber (V2). As shown in Fig. 1, whereas high pressure constantly acts on the upward push chamber (V2), the distributor (DIS) alternatively places the downward push chamber in communication with a high pressure circuit, to give it impact power, and with a low pressure circuit, to bring the piston back to its top dead centre. Thus, the distributor is responsible for the alternating motion of the piston.

If pressure oscillations are neglected, also thanks to the presence of the accumulator (ACC), the motion of the piston may be reviewed as it is due to constant pressure at the downward and upward push chambers. Thus, the striking mass has constant acceleration as well and its resulting motion is uniformly accelerated. According to such hypotheses, it is possible to calculate the velocities of the striking mass at every position and, in particular, at the end of the stroke.

Table 1 reports the main breaker working characteristics for application purposes.

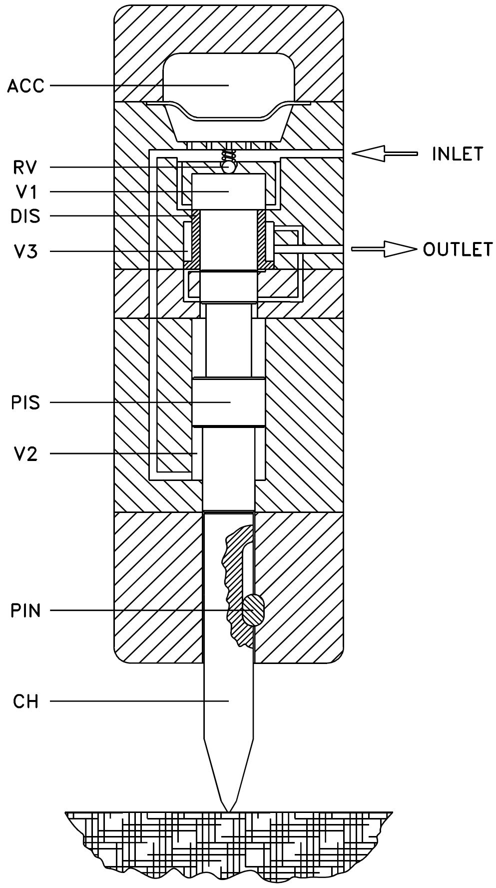

Fig. 1: *Schematic drawing of the hydraulic breaker adapted from European Patent no. 0426928*

Table 1: Breaker working characteristics and their theoretical formulations

Acceleration of the piston during the downward (striking) stroke	$a_\mathrm{p} = \dfrac{S - S'}{m_\mathrm{p}} \cdot p$
Acceleration of the piston during the upward stroke	$a_\mathrm{b} = \dfrac{S'}{m_\mathrm{p}} \cdot p$
Duration of the striking stroke	$t_\mathrm{p} = \sqrt{\dfrac{2 \cdot c}{a_\mathrm{p}}}$
Velocity of the piston at the instant of the impact against the chisel	$v_\mathrm{p,i} = a_\mathrm{p} \cdot t_\mathrm{p} = \sqrt{\dfrac{2 \cdot c \cdot (S - S')}{m_\mathrm{p}}} \cdot \sqrt{p}$
Energy of the piston at the instant of the impact against the chisel	$E_\mathrm{k} = \dfrac{1}{2} \cdot m_\mathrm{p} \cdot v_\mathrm{p,i}^2 = c \cdot (S - S') \cdot p$
Duration of the backward stroke	$t_\mathrm{b} = \sqrt{\dfrac{2 \cdot c}{a_\mathrm{b}}} = \sqrt{\dfrac{2 \cdot c \cdot m_\mathrm{p}}{S'}} \cdot \dfrac{1}{\sqrt{p}}$
Number of impacts in one minute	$n = \dfrac{60}{t_\mathrm{p} + t_\mathrm{b}} = \dfrac{60}{\dfrac{1}{\sqrt{S - S'}} + \dfrac{1}{\sqrt{S'}}} \sqrt{\dfrac{p}{2 \cdot c \cdot m_\mathrm{p}}}$
Power conveyed by the piston	$P_\mathrm{T} = \dfrac{E_\mathrm{k}}{t_\mathrm{p} + t_\mathrm{b}} = \dfrac{c \cdot (S - S')}{\left(\dfrac{1}{\sqrt{S - S'}} + \dfrac{1}{\sqrt{S'}}\right) \cdot \sqrt{2 \cdot c \cdot m_\mathrm{p}}} p^{\frac{3}{2}}$

The upward and downward accelerations were calculated only with reference to the hydraulic force, which is significantly greater than the gravimetric one. It is possible to realize that the pressure of the fluid feeding the breaker is present in all the formulations in Table 1, even if the real input is represented by the oil flow rate supplied by the positive displacement pump of the hydraulic circuit.

However, Table 1 wants another formulation concerning the breaker efficiency, which is defined as

$$\eta_\mathrm{b} = \frac{P_\mathrm{T}}{P_\mathrm{p}} = \frac{E_\mathrm{k} \cdot f}{Q \cdot p} \tag{1}$$

This formulation may be easily simplified, by referring to the two volumes

$$c \cdot S' = \left(Q - Q_\mathrm{b}\right) \cdot t_\mathrm{b} \tag{2}$$

$$c \cdot S = \left(Q + Q_\mathrm{p}\right) \cdot t_\mathrm{p} + c \cdot S' \tag{3}$$

characteristic of the stages of upward and downward stroke of the piston, respectively. Concerning the fluid volume entering the accumulator during the upward stroke of the piston, it is returned during the downward stroke:

$$Q_\mathrm{b} \cdot t_\mathrm{b} = Q_\mathrm{p} \cdot t_\mathrm{p} \tag{4}$$

Thus, considering the system of Eq. 2 to 4, it is possible to formulate the mean flow rate generated by the pump and entering the breaker:

$$Q = \frac{c \cdot S}{t_\mathrm{p} + t_\mathrm{b}} \tag{5}$$

Of course, the numerator in Eq. 5 represents the displacement of the machine

$$V_\mathrm{b} = S \cdot c \tag{6}$$

i.e. the maximum value of the downward push chamber volume as a result of the pumping effect of the piston moving towards the chisel.

According to the previous equations and considering a constant pressure level, the ideal efficiency may be formulated:

$$\eta = 1 - \frac{S'}{S} \tag{7}$$

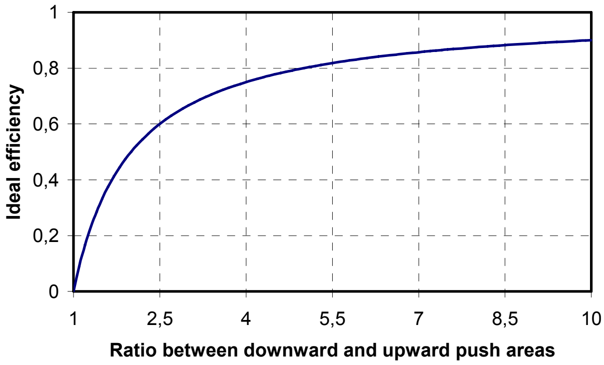

Fig. 2: *Ideal breaker efficiency*

The ideal efficiency is plotted in Fig. 2 which indicates that greater efficiency may be achieved with a considerably high ratio between the areas at the downward and upward push chambers, respectively.

The efficiency formulated with Eq. 7 represents the maximum achievable by a breaker whose configuration is similar to the one proposed in Fig. 1 and with a fixed ratio between the downward and upward push areas. It expresses the limit efficiency for the working cycle, in a way similar to the one suggested by Carnot with reference to thermodynamic cycles.

Nevertheless, this result depends on having considered an ideal distributor managing the dynamics of the striking mass inside the breaker. As a matter of fact, it is to be underlined that part of the hydraulic power supplied by the pump of the hydraulic circuit to the breaker is inevitably thrown away to move the distributor and through the several leakage flow rates. Moreover, the porting geometry with underlapped or overlapped configurations may cause power losses and it should be remembered that the fluid is compressible.

Finally, another noteworthy consideration should be formulated. The breaker dealt with in European Patent no. 0085279 and European Patent no. 0426928 is a machine able to work under a wide range of applications, with reference to both hard and soft materials, since it automatically adjusts the stroke of the piston, which is variable. If the material to crumble is hard, the stroke and the working period will be longer; otherwise, they will be shorter. However, the ratio between the stroke of the piston and the working period may be related to the oil flow rate entering the breaker, which does not depend on the nature of the material to crumble.

$$Q = V_\mathrm{b} \cdot f = S \cdot c \cdot f \tag{8}$$

Thus, longer strokes reflect on high energy releases at the instant of the impact and relatively short working frequencies, and vice versa in regards to shorter strokes. As a matter of fact, it should be realized that due to the constant flow rate, it is impossible to contemporarily achieve high (or low) energy release and long (or short) frequency.

Further theoretical investigations on the performance of the breaker are possible if developed with a numerical model finely simulating the working cycle of the breaker. Simulation is actually useful to study the working behaviour of the breaker, since it does not need all the simplifications and assumptions necessary in a purely theoretical point of view. Moreover, the real input to the breaker consists of the flow rate entering the breaker itself, so it is necessary to replace the pressure with the flow rate, even if the formulations reported in Table 1 already represent a useful tool to identify those design parameters which actually affect breaker performance.

The following sections present the results of several numerical simulations run to investigate the performance of the breaker, according to the model presented in Giuffrida and Laforgia (2005) and developed in the AMESim™ environment (Fig. 3). Each component in the model is a mathematical representation of an

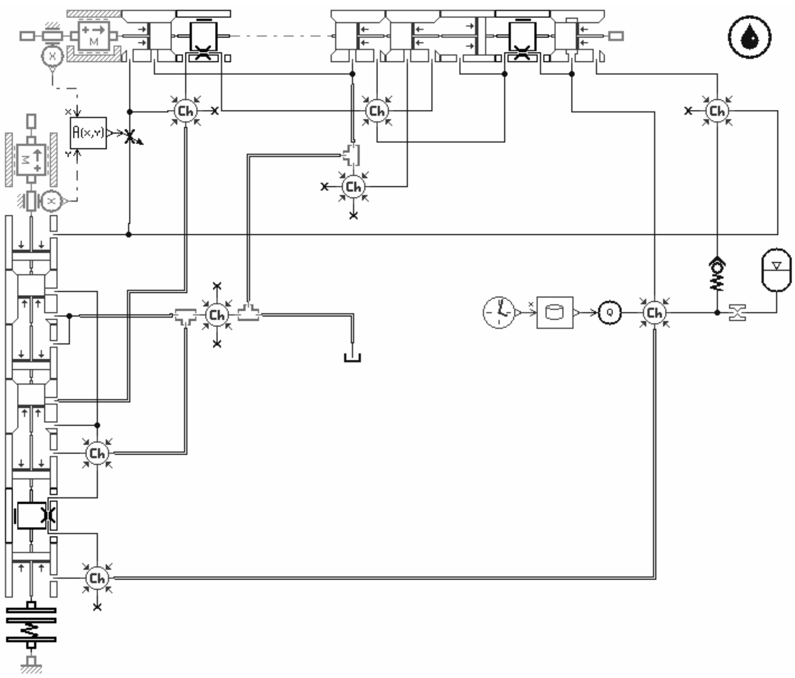

Fig. 3: *Lay-out of the model developed in the AMESim™ environment*

engineering component, whose correct combination allows the simulation of any mechanical-hydraulic system and, in particular, the hydraulic breaker considered in this research activity. Thanks to the parameterisation of the model itself, it was sufficient to change the parameter under investigation each time in order to understand its influence on the performance of the machine. Here, for the sake of brevity, reference to Giuffrida and Laforgia (2005) is made for details about the modelling stages.

3 Influence of the Inlet Flow Rate

The results presented by Giuffrida and Laforgia (2005) were obtained by feeding the breaker with the measured inlet flow rate waveform, as is the real input to the breaker. Nevertheless, it was highlighted that those results did not depend on the particular waveform characteristic of the inlet flow rate. In fact, other simulations were run where the actual inlet flow rate was replaced with its mean value and no considerable variations with respect to the previous results appeared. The only exception consisted of a longer working period with a relative error equal to 2.2 %.

Now, that interesting result allows the running of new simulations to better understand the performance of the breaker. Starting from the inlet flow rate suggested by the manufacturer of the breaker (on-design conditions), its variations in terms of ± 10 %, ± 20 % and ± 30 % are here considered.

Figure 4 shows proportional increases of working pressure and its non-regularity, following increases in inlet flow rate.

Pressure levels refer to the averages of the accumulator simulated pressure, always presenting a saw-toothed shape (Giuffrida and Laforgia, 2005). The mean pressure level is calculated according to the following formula

$$p_{\text{acc,m}} = \frac{1}{T} \cdot \int_T p_{\text{acc}} \cdot dt \qquad (9)$$

whereas its non-regularity is calculated as:

$$\delta_{\text{irr}} = \frac{p_{\text{acc}}\big|_{\max} - p_{\text{acc}}\big|_{\min}}{p_{\text{acc,m}}} \qquad (10)$$

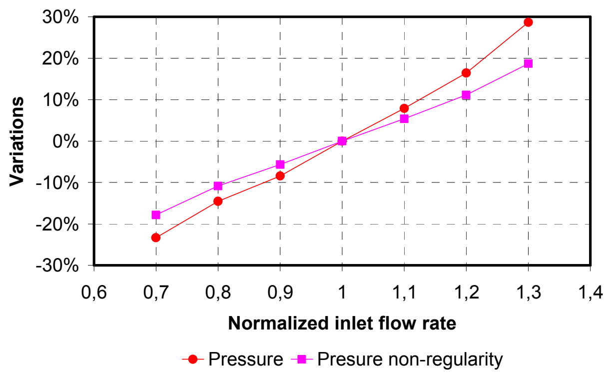

Fig. 4: *Variations in working pressure and its non-regularity depending on variations in inlet flow rate*

These trends are justified if one looks at the breaker as a sort of continuous positioning valve, which obviously leads to higher mean pressure levels at its inlet when a large amount of fluid is directed to it. Greater flow rates result in greater working pressures, which cause increases in velocity of the piston at the instant of the impact against the chisel, as anticipated with the previous considerations and formulations collected in Table 1. In particular, in regards to the working frequency, Fig. 5 shows that it actually depends on the inlet flow rate, as formerly reported in Eq. 8.

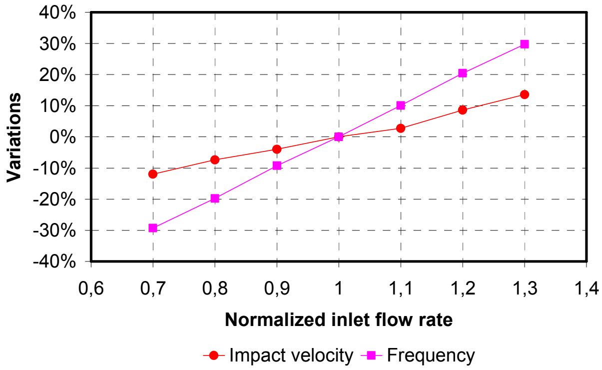

Fig. 5: *Variations in impact velocity and working frequency depending on variations in inlet flow rate*

Finally, trends regarding impact energy and power are shown in Fig. 6: they follow the working pressure trends, even if the power trends seems to be more accentuated, according to the theoretical formulations in Table 1. Moreover, it is to be pointed out that the ratio between impact energy and working pressure is practically constant.

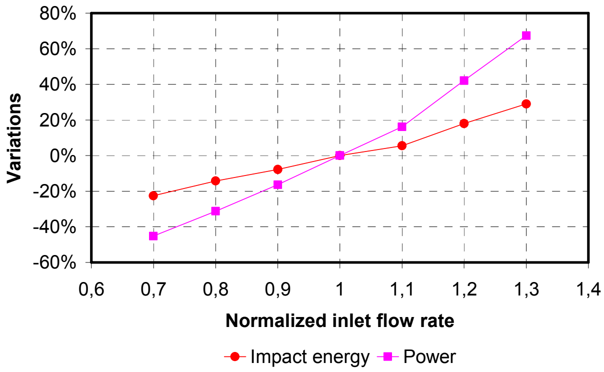

Fig. 6: *Variations in impact energy and power depending on variations in inlet flow rate*

4 Influence of the Design Parameters

According to the previous theoretical considerations leading to the formulations in Table 1, the numerical model simulating the hydraulic behaviour of the breaker is now used to investigate the effects of the variation of some design parameters. In particular, the influences of the mass of the piston, the ratio between the downward and upward push areas, the maximum stroke of the piston and the geometry of the distributor were studied. Finally, various combinations of piston mass and of a diameter characteristic of the geometry of the distributor were investigated.

4.1 Mass of the Piston

When varying the mass of the piston, the numerical model suggests that the most significant variations only concern the impact velocity and breaker efficiency, whose trends are shown in Fig. 7. Efficiency variations are calculated according to the following formula:

$$\Delta\eta_b = \frac{\eta_b - \eta_{b,D}}{\eta_{b,D}} \tag{11}$$

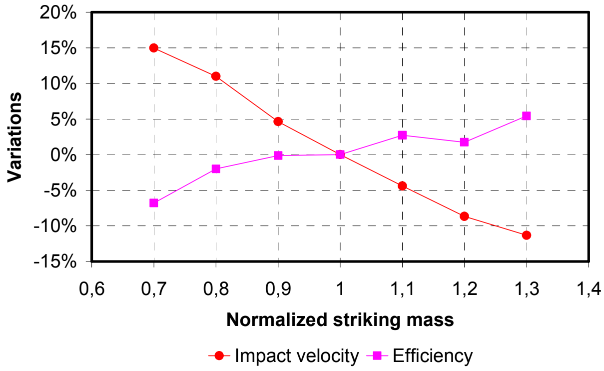

Fig. 7: *Variations in impact velocity and efficiency depending on variations in mass of the piston*

The velocity of the piston at the instant of the impact against the chisel presents an opposite trend with respect to the striking mass, as already reported by a theoretical formulation in Table 1. Nevertheless, variations in velocity are always less than the variations in the striking mass, so that the numerator in Eq. 1 increases with the mass of the piston, even if moderately, suggesting possible improvements in the efficiency of the breaker.

In regards to other features, in particular the impact energy which represents the main specification required by the operator to the breaker, it was not possible to define a significant trend and limited variations (always less than 2%) were predicted.

4.2 Ratio Between the Downward and Upward Push Areas

Another design parameter worthy of investigation consists of the ratio between the downward and upward push areas. These areas are the ones sensing the fluid pressure in the downward and upward push chambers (Fig. 1) and consist of a circle and an annulus, respectively. The areas ratio was changed only by a variation of the upward push area, since the contemporary variation in the downward push area would have caused variations in volumes, clearances and other geometrical dimensions.

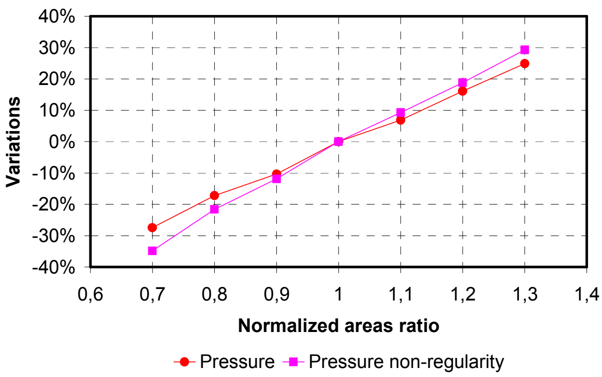

Fig. 8: *Variations in working pressure and its non-regularity depending on variations in the areas ratio*

Figure 8, as Fig. 4, shows that larger mean pressure levels are characterized by larger non-regularities. This trend follows the increase in the areas ratio due to a decrease in the upward push area. As a matter of fact, referring to the unchanged geometry of the breaker, a larger pressure level will be necessary to grant the same upward push to the striking mass.

Impact velocity increases with the areas ratio as well (Fig. 9), even if this trend may be more clearly justified with reference to the increase in working pressure caused by the increase in the areas ratio.

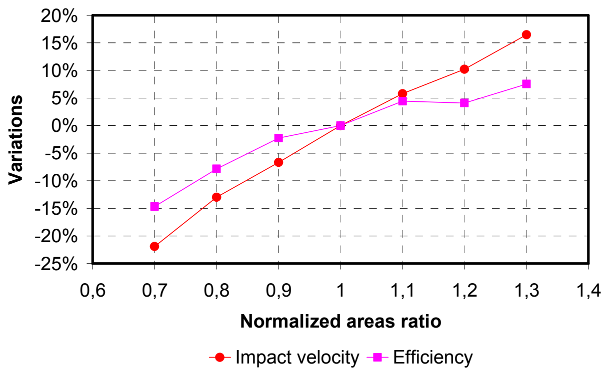

Fig. 9: *Variations in impact velocity and efficiency depending on variations in the areas ratio*

Figure 10 illustrates the efficiency trend increasing with the areas ratio. However, a sort of saturation beyond a certain value of the areas ratio is expected, in full agreement with Eq. 7.

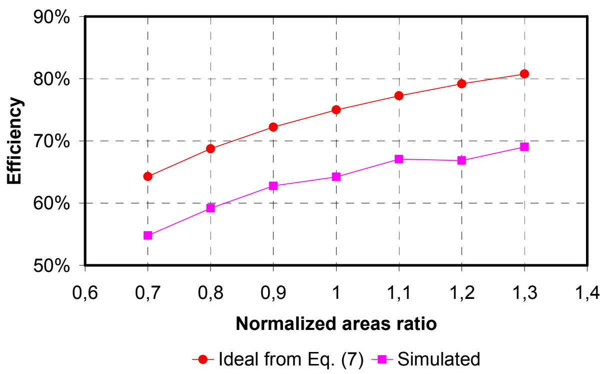

Fig. 10: *Ideal and simulated efficiencies depending on variations in the areas ratio*

It is possible to appreciate that the difference between the two efficiencies is almost constant and equal to 14 % of the value returned by Eq. 7 for all the considered areas ratios. This loss is essentially due to the fluid power required to move the distributor, since it is nearly equal to the overall loss, so that every other source of power loss may be even neglected. Thus, the necessity of spending fluid power to switch the distributor from its bottom dead centre to its top dead centre and vice versa may be reviewed as the main loss in such a hydraulic machine.

Nevertheless, Fig. 8 and 9 show only gross results regarding the breaker performance, because it is not taken into account what occurs inside the machine in terms of possible depressurizations and overshooting pressures. Here, when the areas ratio increases, it is to underline that the previous results are achieved with quick depressurizations inside the downward push chamber. Such events are due to the reduction in flow rate exiting the upward push chamber when the piston moves downwards to strike the chisel, resulting in the reduction in the upward push area. This flow rate, together with the one from the depressurizing accumulator and the one from the pump is not sufficient to prevent the downward push chamber from quick depressurizations up to cavitation.

Thus, an increase in the areas ratio should be fixed together with a correct size of the accumulator. Here, focusing attention on the accumulator, theoretical investigations concerning its volume and pre-charge level were carried out. Figure 11 shows only pressure non-regularity variations, since pressure, frequency, impact velocity and efficiency do not present significant trends when the two considered parameters vary.

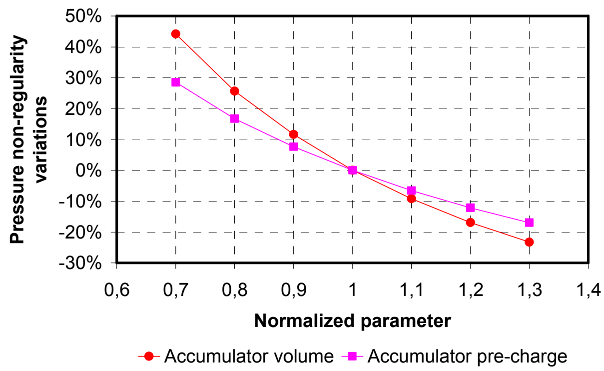

Fig. 11: *Variations in pressure non-regularity at the inlet of the breaker depending on variations in volume and pre-charge of the accumulator*

Pressure non-regularity decreases when both the parameters increase, even if greater variations are achieved with reference to the accumulator volume. Such a result was to be expected, since the nitrogen gas in the accumulator was assumed to obey a polytropic law of the form:

$$p_{\text{gas}} \cdot V_{\text{gas}}^{\gamma} = const \tag{12}$$

where the constant is defined according to the pre-charge pressure and the accumulator volume.

Pressure non-regularity clearly decreases when accumulator volume and pre-charge increase, since these parameters reflect on the mass of pre-charged nitrogen which behaves like a spring with variable stiffness. Of course, when these parameters concerning the accumulator decrease, simulations suggest that pressure evolutions in the downward push chamber are similar to the ones characteristic of a reduction in the upward push area, as previously discussed, i.e. with quick depressurizations up to cavitation.

Thus, in a breaker like the one considered in this study, the accumulator is present not only to limit working pressure fluctuations but, and essentially, to properly fill with pressurized fluid the chamber increasing in volume for the quick acceleration of the striking mass. Nevertheless, the breaker dealt with in this work does not reflect the only possible design solution of a hydraulic percussion machine. As a matter of fact, there are breakers which do not need an accumulator for the oil, as the ones described in European Patent no. 0335994, United States Patent no. 6959967 and United States Patent no. 2001/0022229.

4.3 Stroke of the Piston

As thoroughly described in Giuffrida and Laforgia (2005), the maximum stroke of the striking mass in the modelled breaker depends on the top dead centre reached by the distributor managing the dynamics of the piston inside the breaker. In regards to the breaker under study, this position depends on the material to crumble (European Patent no. 0085279; European Patent no. 0426928).

Now, the mass of the piston may be varied by means of the length of the spool at the middle of the piston (Figure 1), whereas the ratio between the downward and upward push areas may be varied with reference to the annulus at the upward push chamber, as previously adopted. These design modifications are sufficiently simple to be eventually realized. However, variations in the stroke of the distributor, leading to variations in the stroke of the piston, are possible with a new design of the communication with the high and low pressure circuits, at the top of the breaker. Thus, a new and proper choice of porting with correct underlap and overlap regions will be necessary. Considering that the scope of this work lies outside a new design of the machine, here the formulas in Table 1, referring to the influence of the stroke of the piston, are supposed to be sufficient to study its influence on breaker working characteristics.

4.4 Geometry of the Distributor

The fluid power required to move the distributor may be reviewed as the main loss in the breaker working cycle, as previously anticipated in Fig. 10.

The distributor of the studied breaker consists of a cylindrical sleeve valve connected to a flange-type ring plate at its bottom, as shown in Fig. 1. Its geometry is better detailed in Fig. 12, with particular reference to three diameters.

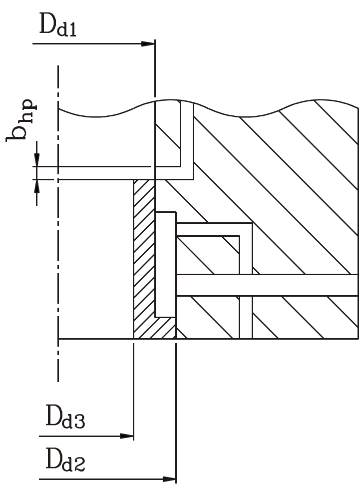

Fig. 12: *Schematic drawing of the distributor ready to leave its bottom dead centre*

If h represents the maximum stroke of the distributor, then the volume which must be filled with high pressure fluid to switch the distributor is equal to:

$$V_h = \frac{\pi}{4}\left(D_{d2}^2 - D_{d1}^2\right) \cdot h \tag{13}$$

Minimizing the volume in Eq. 13 should improve efficiency, so that starting from the original value of the distributor area

$$A_h = \frac{\pi}{4}\left(D_{d2}^2 - D_{d1}^2\right) \tag{14}$$

the diameter D_{d2} was varied in order to achieve variations of the area A_h in terms of ± 10 %, ± 20 % and ± 30 %.

Working pressures and their non-regularities are illustrated in Fig. 13: larger pressure levels are achieved with larger non-regularities, as already shown in Fig. 4 and 8.

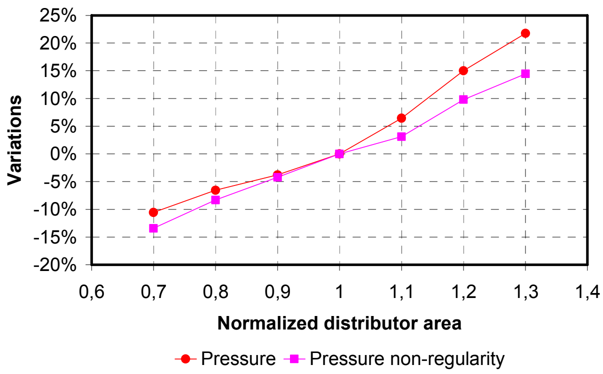

Fig. 13: *Variations in working pressure and its non-regularity depending on variations in the diameter* D_{d2}

Looking at Fig. 14, increases in diameter D_{d2} lead to increases in impact velocity, essentially because of larger working pressures, as shown in Fig. 13.

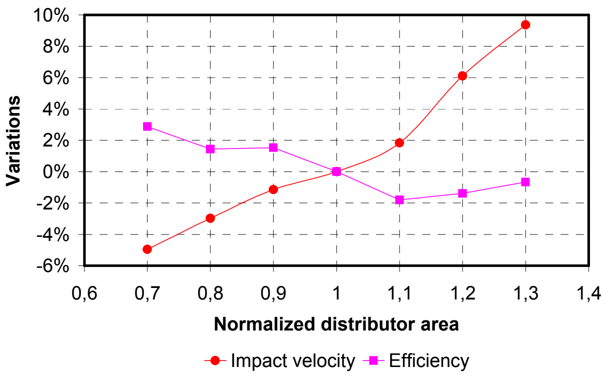

Fig. 14: *Variations in impact velocity and efficiency depending on variations in the diameter* D_{d2}

Moreover, one can get possible efficiency improvements with a reduced diameter D_{d2}. However, efficiency variations, when calculated by Eq. 11, seem to be quite modest, as shown in Fig. 7, 9 and 14.

If a new formulation is adopted to account for the efficiency variations

$$\Delta\eta_{\mathrm{b}}^{\ *} = \frac{\eta_{\mathrm{b}} - \eta_{\mathrm{b,D}}}{\eta - \eta_{\mathrm{b,D}}} \tag{15}$$

it is possible to quantify how much power may be saved with reference to the limit efficiency η, calculated with Eq. 7. Now, according to the last formulation, results proposed in Fig. 7 and 14 must be reviewed.

Results shown in Fig. 15 suggest possible performance improvements with a proper choice of the striking mass and of the diameter D_{d2} of the distributor, as shown in Fig. 12. Of course, it would be interesting to know where an interaction of these two parameters can lead in terms of efficiency and of the other breaker working characteristics.

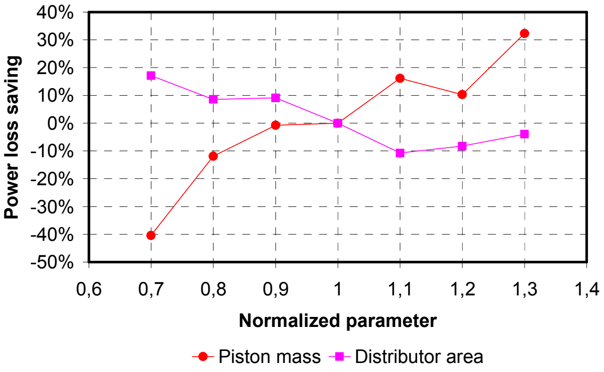

Fig. 15: *Possibility of power loss saving with variations of the piston mass and of the diameter D_{d2} of the distributor*

4.5 Combined Effects

The previous figures showed possible efficiency improvements when:

- the mass of the piston increases,
- the ratio between the downward and upward push areas increases,
- the diameter D_{d2} of the distributor is reduced.

Thus, with reference to the previous simplified analysis, based on uniformly accelerated motion of the piston, other two parameters were found to influence breaker efficiency, together with the areas ratio. Now, considering that an increase in the areas ratio should lead, at least, to a new choice of volume and pre-charge of the accumulator, another analysis is carried out according to the following normalized values:

- 1, 1.1, 1.2 and 1.3 for the mass of the piston,
- 0.7, 0.8, 0.9 and 1 for the distributor area, by means of variation of the diameter D_{d2} (Fig. 12),

leading to $2^4 = 16$ simulation runs. This last analysis aims at finding a combination of these two parameters in order to try to improve breaker efficiency.

Figure 16 clearly shows possible efficiency improvements with a proper combination of striking mass and of diameter D_{d2} of the distributor.

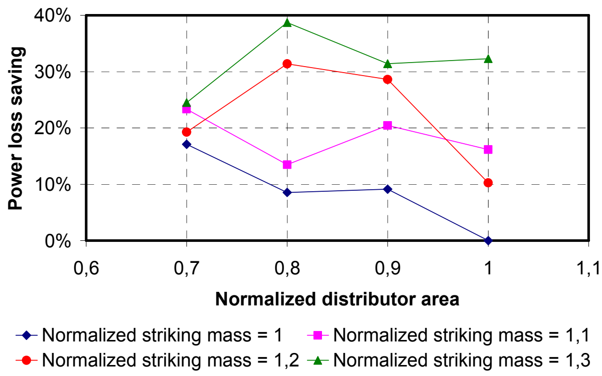

Fig. 16: *Possibility of power saving with combined variations of piston mass and diameter D_{d2} of the distributor*

Nevertheless, if an increase in efficiency seems to be actual, it is necessary to study the variations of the other breaker working characteristics in order to evaluate if improvements in the working cycle of the breaker are significant.

According to the formulations collected in Table 1, working pressure is the variable affecting all the breaker working characteristics. Its variations, with reference to combinations of the two considered parameters, are shown in Fig. 17.

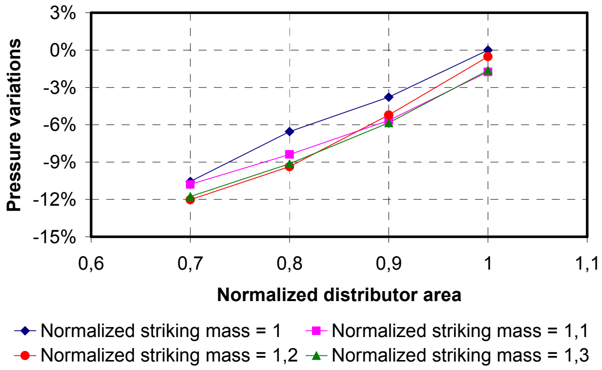

Fig. 17: *Variations in working pressure with combined variations of piston mass and diameter D_{d2} of the distributor*

These variations are essentially caused by reductions in the diameter D_{d2} of the distributor, since piston mass was previously found not to considerably influence the working pressure. Moreover, reductions in working pressure are achieved with reductions in pressure non-regularity, as previously shown in Fig. 4, 8 and 13.

But reductions in working pressure, as suggested up to now by both simulations and formulations collected in Table 1, should cause impact velocity and energy decreases, as shown in Fig. 18 and 19, respectively.

The last trends may be explained considering that impact energy depends on piston mass and impact velocity. Here, the weight of impact velocity is greater than the one of the piston mass in determining impact energy variations.

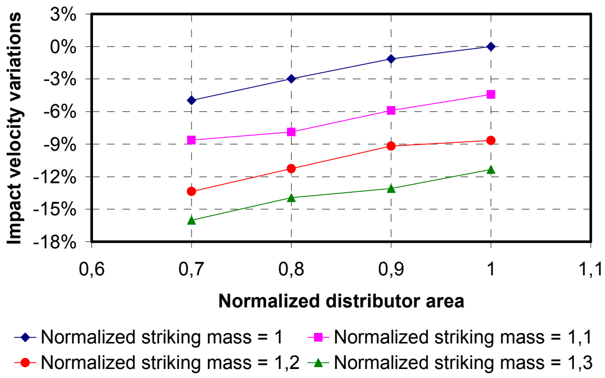

Fig. 18: *Variations in impact velocity with combined variations of piston mass and diameter D_{d2} of the distributor*

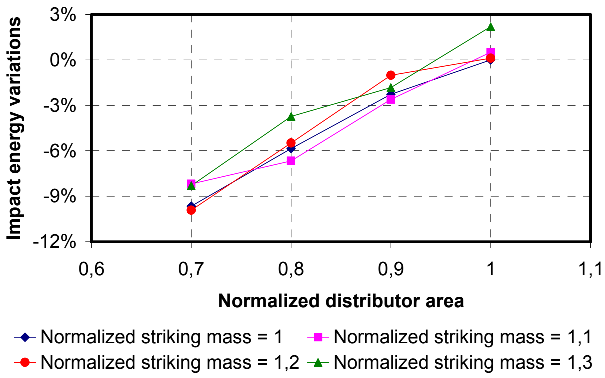

Fig. 19: *Variations in impact energy with combined variations of piston mass and diameter D_{d2} of the distributor*

Figures 16 and 19 show that a power loss saving results in reductions of impact energy, so that the designer of the breaker should adopt a compromise solution. As a matter of fact, if impact energy represents the main performance of the breaker, efficiency actually reflects on the fuel consumption in the engine which moves the pump feeding the breaker.

Finally, in regards to the trends of the impact power, they are similar to the ones proposed in Fig. 19, since simulation suggests little variations of the working frequency (within 3 %) during the 16 simulation runs. Thus, one can conclude that working frequency actually depends on the flow rate feeding the breaker and on the material to crumble, which is supposed to be infinitely stiff as in the previous work (Giuffrida and Laforgia, 2005).

5 Conclusions

This work follows a previous one dealing with the simulation of the working principle of a hydraulic breaker.

Here, a preliminary study of the motion of the striking mass as uniformly accelerated allowed the formulation of several working characteristics of the breaker: duration of the striking stroke, impact energy, etc. Working pressure is the leading variable affecting all the breaker features. The breaker efficiency was formulated as well in an ideal form where the ratio between the downward and upward push areas was found to seriously affect its value in a way similar to the one suggested by Carnot with reference to thermodynamic cycles.

Later, making use of a parameterised numerical model, reproducing physical phenomena inside the breaker, it was possible to study the response of the machine essentially due to variations of inlet flow rate and some design parameters.

The breaker seems to behave like a continuous positioning valve since working pressure and frequency increase when increasing inlet flow rate.

With reference to breaker performance, common result is the increase in impact energy due to increases in working pressure.

A review of power losses was also carried out and possible combinations of two particular design parameters were investigated in order to achieve the greatest energy saving. This result was inevitably related to a small reduction in impact energy. According to the last investigation, showing that achieving energy saving seems to only be possible by penalizing impact energy, a compromise design solution seems to be an necessary choice.

6 Acknowledgements

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Nomenclature

a_b, a_p	Backward and striking piston accelerations	[m·s^{-2}]
c	Piston stroke	[m]
D_{d1}, D_{d2}	Distributor dimensions (Figure 12)	[m]
E_k	Kinetic energy of the piston at the instant of the impact against the chisel	[J]
f	Working frequency	[s^{-1}]
h	Distributor stroke	[m]
m_p	Striking mass	[kg]
n	Number of impacts in one minute	[min^{-1}]
p	Working pressure	[Pa]
p_{acc}	Accumulator pressure	[Pa]
$p_{acc,m}$	Accumulator mean pressure	[Pa]
p_{gas}	Gas pressure inside the accumulator	[Pa]
P_P	Power supplied by the pump	[W]
P_T	Power transmitted to the chisel	[W]
Q	Flow rate generated by the pump	[m^3·s^{-1}]
Q_b	Volumetric flow rate entering the accumulator during the upward stroke of the piston	[m^3·s^{-1}]
Q_p	Volumetric flow rate exiting the accumulator during the downward stroke of the piston	[m^3·s^{-1}]

S	Downward push area	[m^2]
S'	Upward push area	[m^2]
T	Working period	[s]
t_b	Duration of the backward stroke	[s]
t_p	Duration of the striking stroke	[s]
V_b	Breaker displacement	[m^3]
V_{gas}	Gas volume inside the accumulator	[m^3]
$v_{p,i}$	Velocity of the piston at the instant of the impact against the chisel	[$m \cdot s^{-1}$]
δ_{irr}	Pressure non-regularity	[-]
γ	Polytropic index	[-]
η	Breaker limit efficiency	[-]
η_b	Breaker actual efficiency	[-]
$\eta_{b,D}$	Breaker actual efficiency (on-design conditions)	[-]

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