STATIC AND DYNAMIC BEHAVIOUR OF SAFETY VALVES IN THE HYDRAULIC CYCLES OF AGRICULTURAL MACHINES

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Introduction

The safety valve is an indispensable element of hydraulic cycles. Should it function also as an overflow valve, it must meet rigorous requirements.

In their layout, safety and overflow valves are similar or only very slightly different. The difference lies mostly in the role they play in the cycle. The overflow valve is practically always open, its task is to discharge the excess oil supplied by the pump and not used up by the system. Overflow valves must keep practically constant pressure over a wide range of through-flow rates.

The job of the safety valve is to protect the system against non-desirable overloads. Under normal working conditions, therefore, they stop the flow of the liquid. The pressure at which the valve opens must at all times be higher than the maximum working pressure.

With the steadily increasing pressures applied in hydraulic systems, an incorrectly designed or erroneously selected safety valve may cause very considerable losses. As its name implies, the safety valve must provide safe conditions and protect the system. An incorrectly chosen safety valve, instead of protecting it, would cause excessive overloads.

No safety valve can be judged unequivocally to be good or bad. A valve which may be suitable under certain working conditions may fail under a different set of conditions.

General characteristics

To ensure the proper operation of a safety valve it must be statically and dynamically matched to the system in which it is to operate. These requirements are obviously partly contradictory; whether static or dynamic adjustment is of greater importance depends on the characteristics of the cycle concerned.

To judge the merit of safety valves unequivocally, the following tests are required:

a) to plot the blowdown characteristics of the valve at different opening pressures; this curve is also termed the static characteristic of the valve.

b) to plot the pressure curve of the valve as a function of time with sudden opening at different pressures. This curve is known as the dynamic characteristic of the valve;

c) testing with sinusoidal excitation; this consists in determining, as a function of frequency, of the amplitude of the pressure fluctuations, with sinusoidal variations in the liquid stream across the valve; the determination of the frequency of resonance.

After completion of the three tests the valve may be qualified, and conclusions may be drawn as to its likely behaviour in a given cycle.

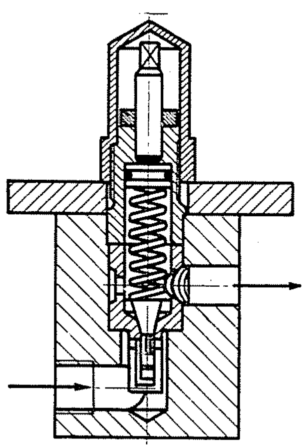

Fig. 1. Direct controlled safety valve

There are a great many safety valves of different design and layout in practical use. They may differ by design, by the layout of their lock or the way in which they provide for the passage of flow, and whether or not the two elements are combined, and also in the method of the damping they apply. Neglecting the non-essential differences, safety and overflow valves fall into two main categories:

1. Direct controlled
2. Pre-controlled

safety valves.

Fig. 1 shows a direct controlled safety valve; Fig. 2 the layout of a pre-controlled type.

The essential difference between the two lies in the structural layout which clearly indicate the dissimilarity of the theories underlying their design.

In the valve shown in Fig. 1, pressure acts directly on the lock. If the acting pressure is at or over the value set by the spring force, the element providing for passage rises to let a defined quantity of liquid pass. The force equilibrium is described by the following relationship

$$pA = F_r \tag{1}$$

where

$p = p_b$ and p_b	designate the adjusting or opening pressure,
$A = d^2\pi/4$	the effective surface
$F_r = c_r X_0$	the spring force
c_r	the spring constant
X_0	the pretensioning of the spring.

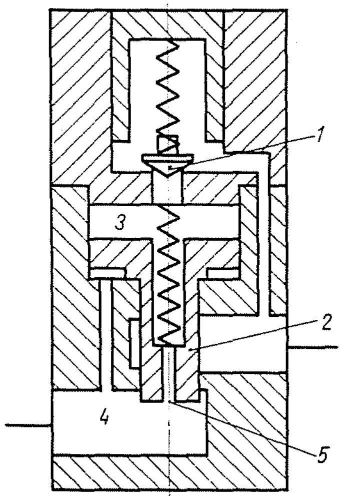

Fig. 2. Pre-controlled safety valve

With the valve lifted, the spring force tends to increase. To keep balance with a greater spring force a higher pressure is required. The larger the quantity of liquid streaming across the valve, the greater its lift and the higher the pressure in the system. After opening, a direct controlled valve increases the load on the system. The precontrolled safety valve seen in Fig. 2 consists essentially of two parallel connected valves in which the opening of the precontrol valve (1) permits the opening of the main valve (2). Until the precontrolling valve is closed, equal pressure prevails in the spaces marked 3 and 4. Since the effective surface of the main valve in both spaces is equal, it remains closed. Another factor acting in closing direction is the relatively soft spring located in the space marked 3. Should the pressure in the system exceed the value set on the pre-control valve, it will open letting the liquid pass. Since the liquid must flow also through the narrow bore of the main valve marked 5, a pressure difference will come about between the spaces 3 and 4, equal in magnitude to the pressure drop caused by the narrow bore in the main valve. This will upset the static equilibrium in the main valve and cause it to shift in the direction of opening. This new state of equilibrium will be determined by the increased spring force and the force arising through the pressure difference. Since the main valve is fitted with a soft spring, pressure with increasing throughflow will grow only to a negligible degree. Precontrolled valves, accordingly, have a very small blowdown range and they do not cause pressure rise in the system, over a wide range of throughflow rates. In this property lies the difference between the two types of valves. Precontrolled valves are, therefore, particularly well suited to work as overflow valves.

In what follows we shall investigate into the static and dynamic behaviour of these valves and into their predictable performance in hydraulic cycles.

Determination of the blowdown range of direct controlled safety valves

For the purpose of our examinations the valve with conic seat, a type in wide use, has been selected. The examination results can be readily generalised.

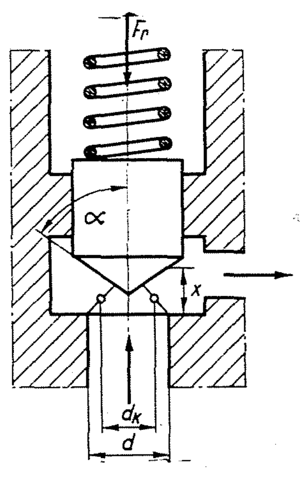

Fig. 3. Scheme of a valve with conical seat

Fig. 3 is a sketch of the valve. For flows across gaps with a small 1/D ratio the following relationship can be written:

$$Q = \mu \cdot A \sqrt{\frac{2g}{\gamma} \cdot p} \tag{2}$$

where

μ is the coefficient of throughflow
A the cross section
γ the specific gravity of the medium
p the difference between pressures ahead and behind the valve.

The flow cross section of the conical valve shown in Fig. 3 is as follows:

$$A = x\pi (d \cdot \sin\alpha - x \cdot \sin^2\alpha \cdot \cos\alpha) \tag{3}$$

Since the second member (in brackets) is negligibly small we may write down at a fair approximation that

$$A = \pi x \cdot d \cdot \sin\alpha \tag{4}$$

where

d the diameter of the pipe ahead of the valve
x the degree of valve lift
α the magnitude of the half cone angle.

Taking Eq. (4) into consideration:

$$Q = \mu\, d \cdot \pi \cdot \sin\alpha \cdot \sqrt{\frac{2g}{\gamma}} \cdot x \cdot \sqrt{p}\,. \tag{5}$$

As shown by the tests, the coefficient of throughflow is constant over a wide range of flow. For the given valve, therefore, we may write down that

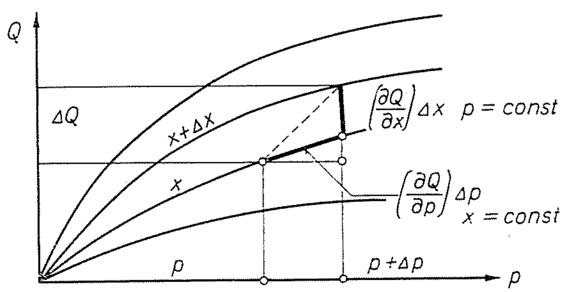

Fig. 4. Set of characteristics $Q = f(p)$ of a safety valve

$$\mu d \cdot \pi \cdot \sin\alpha \cdot \sqrt{\frac{2g}{\gamma}} = B = \text{const}. \tag{6}$$

Substituting:

$$Q = B \cdot x \cdot \sqrt{p}\,. \tag{7}$$

As shown by relationship (7), increased rate of throughflow will influence the valve lift x and the pressure p. The set of curves describing this relationship is illustrated in Fig. 4.

With linear approximation, the variations of throughflow are defined by the relationship

$$\Delta Q = \left(\frac{\partial Q}{\partial x}\right)_{p=\text{const}} \cdot \Delta x + \left(\frac{\partial Q}{\partial p}\right)_{p=\text{const}} \cdot \Delta p \tag{8}$$

where

$$\frac{\partial Q}{\partial x} = B \cdot \sqrt{p} \quad \text{and} \quad \frac{\partial Q}{\partial p} = \frac{B \cdot x}{2\sqrt{p}} \,. \tag{9}$$

Since, compared to $\partial Q/\partial x$, $\partial Q/\partial p$ is negligible,

$$\Delta Q = B \cdot \sqrt{p} \cdot \Delta x \,. \tag{10}$$

From the static equilibrium of the spring we have

$$\Delta p \cdot A = c_r \cdot \Delta x \tag{11}$$

where

Δp the pressure increase
Δx the valve lift.

Accordingly, the pressure increase will be

$$\Delta p = \frac{\Delta Q}{B \cdot \sqrt{p}} \cdot \frac{c_r}{A} \,. \tag{12}$$

In deriving Eq. (12) we have disregarded the resistance of the valve casing which, too, increases with increasing throughflow. The resistance of the casing is

$$\Delta p_h = K_h(Q_2^2 - Q_1^2) \tag{13}$$

where K_h designates a coefficient proportional to the resistance of the valve casing.

With (13) and the resubstitution of the B value:

$$\Delta p = \frac{\Delta Q}{\mu d \cdot \pi \cdot \sin\alpha \sqrt{\frac{2g}{\gamma}} \cdot \sqrt{p}} \cdot \frac{c_r}{A} + \Delta p_h \,. \tag{14}$$

Relationship (14) shows clearly the effect of the different valve characteristics upon the spray range p.

The greater the cone angle, the smaller the range of blowdown. With increasing spring constant it increases proportionately, increasing pressure of opening or adjustment will diminish the blowdown range.

According to our previous deliberations Eq. (14) will enable a good approximation for slight changes in ΔQ only. To determine the range of blowdown also for higher rates of throughflow, the following method should be chosen:

Dividing the nominal discharge rate into n equal parts we obtain the value of ΔQ (see Fig. 5).

We then determine the opening pressure p_{be} of the valve.

Having calculated the constant B in advance and neglecting the resistance the casing, with the aid of Eq. (12) Δp_1 can be calculated as:

$$\Delta p_1 = \frac{\Delta Q}{B \cdot \sqrt{p_{be}}} \cdot \frac{c_r}{A} .$$

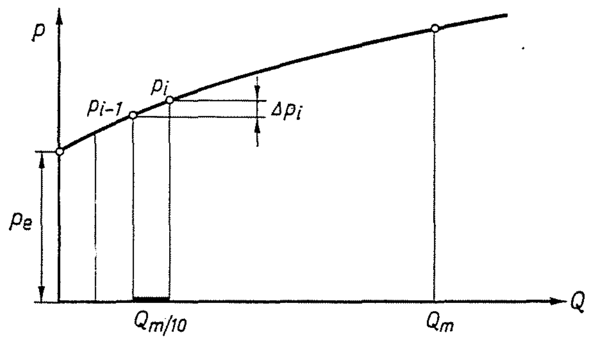

Fig. 5. The blowdown range

This means also the determination of the point 1 of the curve, since

$$p_1 = p_{be} + \Delta p_1 .$$

To calculate point 2, pressure p_1 must be taken into consideration.

$$\Delta p_2 = \frac{\Delta Q}{B\sqrt{p_{be} + p_1}} \cdot \frac{c_r}{A} = \frac{\Delta Q}{B \cdot \sqrt{p_1}} \cdot \frac{c_r}{A}$$

and

$$p_2 = p_{be} + \Delta p_1 + \Delta p_2 = p_1 + \Delta p_2$$

yielding in turn all points of the curve.

Better to understand the sequence of calculation, let us determine the static characteristics of the conical safety valve which had previously been measured.

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Example

Spring constant of the valve: $C_r = 39.2$ kp/cm.

Diameter of the pipe ahead of the valve $d = 6$ mm.

The half cone angle of the lock $\alpha = 20°$.

One of the static curves of the valve was plotted at an opening pressure $p_{b_e} = 38$ kp/dm². For a better comparison of the two curves, the calculations are based on this value.

Let us now determine the constants:

$$B = \mu \cdot d\pi \cdot \sin\alpha \sqrt{\frac{2g}{\gamma}} = 0{,}65 \cdot 0{,}6 \cdot 0{,}342 \cdot \sqrt{\frac{2{,}981}{0{,}9 \cdot 10^{-3}}} = 616 .$$

If

$$Q = 5 \frac{\text{dm}^3}{\text{sec}} = 83{,}3 \frac{\text{cm}^3}{\text{sec}} \quad \text{és} \quad A = \frac{d^2 \pi}{4} = \frac{0{,}6^2 \pi}{4} = 0{,}283 \, [\text{cm}^2]$$

then

$$\frac{\Delta Q \cdot c_r}{B \cdot A} = \frac{83{,}3 \cdot 39{,}2}{0{,}283} = 18{,}67$$

$$\Delta p_1 = \frac{18{,}67}{\sqrt{38}} = 3{,}02 \, [\text{kp/cm}^2]$$

$$p_1 = p_{be} + \Delta p_1 = 38 + 3{,}02 = 41{,}2 \, [\text{kp/cm}^2]$$

$$\Delta p_2 = \frac{18{,}67}{\sqrt{p_1}} = \frac{18{,}67}{\sqrt{41{,}2}} = \frac{18{,}67}{6{,}44} = 2{,}9 \, [\text{kp/cm}^2]$$

$$p_2 = p_1 + \Delta p_2 = 41{,}02 + 2{,}9 = 43{,}92 \, [\text{kp/cm}^2] .$$

We abstain from presenting here the rest of the calculations but give the pertinent value pairs compiled in the table hereunder, together with what has been dealt with above.

Table I

Q dm³/min	0	5	10	15	20	25	30	35	40	50	60	70	80
p kp/cm²	38	412	4392	4672	4945	521	5468	572	5966	6468	6927	7371	7585

For calculation and measurement results see Fig. 6.

In the course of our previous examinations the effect of the momentum of the liquid flowing at high velocity had been disregarded. Opinions vary: some find it negligible, others exaggerate its importance. The force of momentum depends on the design of the valve. If necessary, its effect can be utilised partly or in whole to influence the static characteristics of the valve. Taking the direction of closing to be positive, the momentum with designations used in Fig. 7 will be as follows:

$$F_i = F_2 \cos \alpha - F_1 \tag{15}$$

$$F_i = \frac{\gamma}{g} \cdot Q \, (w \cdot \cos \alpha - v) \, . \tag{16}$$

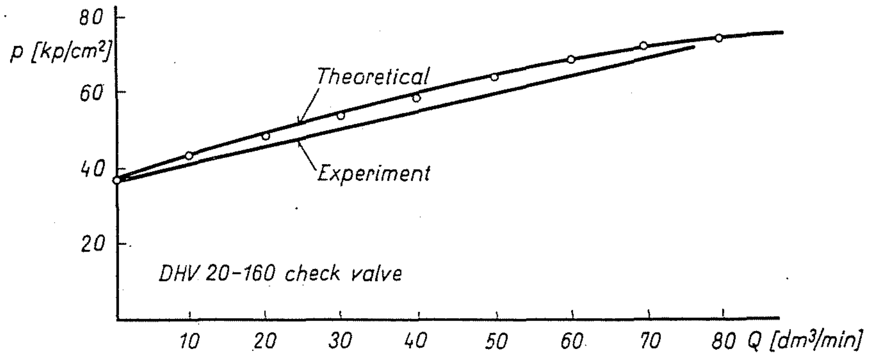

Fig. 6. Calculated and measured curves for conical safety valve

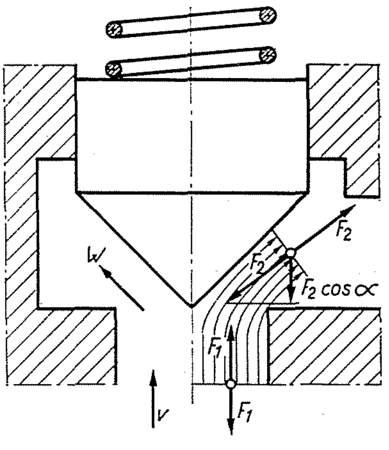

Fig. 7. The effect of momentum upon the element providing for the passage of the liquid

Since the flow velocity of the medium in the narrowest cross section is considerably higher than in the pipe ahead of the valve, its effect can be disregarded. Thereby:

$$F_i = \frac{\gamma}{g} \cdot Q \cdot w \cdot \cos \alpha \tag{17}$$

where

$$w = \sqrt{\frac{2g}{\gamma} \cdot \Delta p} \tag{18}$$

and

$$Q = \mu \cdot A \cdot w = \mu \cdot A \cdot \sqrt{\frac{2g}{\gamma} \cdot \Delta p} \, . \tag{19}$$

Thus:

$$F_i = \frac{\gamma}{g} \mu \cdot A \sqrt{\frac{2g}{\gamma} \cdot \Delta p} \cdot \sqrt{\frac{2g}{\gamma} \Delta p} \cdot \cos \alpha \, . \tag{20}$$

After simplifications:

$$F_i = 2\mu A \cdot \cos \alpha \cdot \Delta p \, . \tag{21}$$

The throughflow cross section can be written down as the function of the valve lift, viz. in the form of:

$$A = K \cdot x = d_k \cdot \pi \cdot x \tag{22}$$

where

K circumference
x valve lift
d_k mean diameter.

Accordingly:

$$F_i = 2\mu d_k \cdot \pi \cos \alpha \cdot \Delta p \cdot x \, . \tag{22}$$

Disregarding the pressure rise upon the lifting of the valve, we have

$$2\mu \cdot d_k \pi \cdot \cos \alpha \cdot \Delta p = c_f = \text{const} \tag{23}$$

and thereby

$$F_i = c_f \cdot X \tag{23}$$

where c_f is the spring constant of the hydraulic spring. The effect of momentum is the same as the effect of mechanical springs. In steady state conditions, after opening the valve, the pressure ahead of it must overcome not only the mechanical but also the hydraulic spring constant. The resultant spring constant will accordingly become:

$$c_e = c_r \pm c_f \, . \tag{24}$$

To determine the magnitude of the spring constant c_f and to relate it to the mechanical spring constant c_r, let us calculate the hydraulic spring constant of the valve described in the example (in design, the momentum had been compensated).

Example

From the previous example $\alpha = 20°$ and $d_k = 0.6$ cm.

The opening pressure of the valve, $P_{be} = 38$ kp/cm². Neglecting the pressure loss in the pipe behind the valve we have

$$p = p_{be} = 38 \text{ kp/cm}^2 .$$

Be μ 0.65. Thereby

$$c_f = 2\mu d_k \cdot \pi \cdot \Delta p \cdot \cos \alpha$$

$$c_f = 2 \cdot 0.65 \cdot 0.6 \cdot \pi \cdot 38 \cdot \cos 20° = 87{,}5 \text{ [kp/cm]}$$

$$c_f = 87{,}5 \left[\frac{\text{kp}}{\text{cm}}\right].$$

In this case the hydraulic spring constant is greater than the mechanical one.

The resultant spring constant is

$$c_e = 39{,}2 + 87{,}5 = 126{,}7 \left[\frac{\text{kp}}{\text{cm}}\right].$$

With this taken into consideration the static characteristic of the valve will be modified to a considerable degree.

Without going into details, see the corresponding value pairs compiled in Table II:

Table II

Q dm²/min	0	10	20	30	40	50	60
p kp/cm²	38	52,5	64,9	76,1	84,6	96,07	105,24

Comparison of static and dynamic characteristics of safety valves

We do not deal with the precontrolled safety valves in greater detail. The comparison of the characteristic curves of the two types permits, however, conclusions to be drawn for their applicability. Figs 8 and 9 show the static and dynamic characteristics resp., of the direct controlled valves discussed above.

The dynamic characteristic curve shows the pattern of pressure with the valve suddenly opened. Since the pressure acts directly upon the element which provides for the liquid passage, opening takes place practically without any delay. If sufficient damping is provided, the pressure fluctuations will be negligible. The static characteristic of the direct controlled valve is unfavourable due to its excessive blowdown range Δp. Its dynamic characteristic, on the other hand, is favourable since the valve reacts to pressure changes without any time lag, preventing pressure peaks to build up.

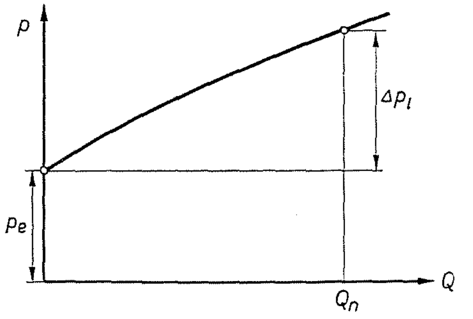

Fig. 8. Direct controlled valve: static characteristics

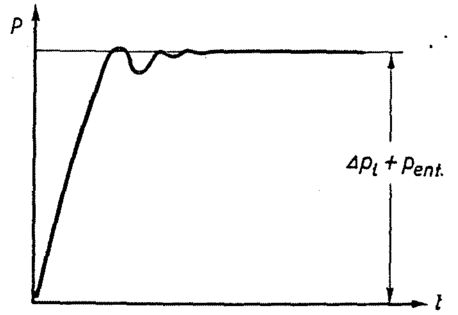

Fig. 9. Direct controlled valve: dynamic characteristics

Fig. 10 illustrates the static characteristic of the precontrolled safety valve. At low rates of throughflow only the precontrolled valve is open, designed with a steeply ascending characteristic curve. Should the pressure drop on the main valve, due to the flow across the precontrolled valve, be sufficient to open it, then it will open but, owing to the soft spring, have a very small blowdown range.

The precontrolled valves exhibit very favourable static characteristics but rather poor dynamic ones, due to their high time constant. With a sudden leapwise input signal the main valve cannot open until the precontrolled valve has opened. A choke is always inserted ahead of the precontrolled valve, to provide for a more favourable static characteristic.

As said, the opening of the main valve cannot take place before that of the precontrolled valve. Meanwhile the pump continues to deliver the medium, which cannot pass through the valve, but accumulates in the space ahead, increasing its pressure to a degree depending on its capacity.

With the sudden opening of precontrolled valves, considerable pressure peaks tend to build up, resulting in rather unfavourable dynamic characteristics.

Summing up, the static characteristics of direct controlled valves are unfavourable, their dynamic characteristics are good. The case with precontrolled valves is the reverse: The static characteristics are good, the dynamic ones poor. However, a good safety valve must have equally favourable static

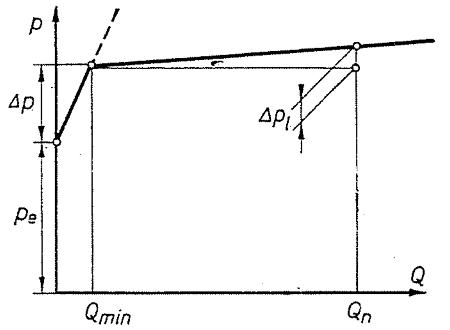

Fig. 10. Static characteristics of precontrolled valve

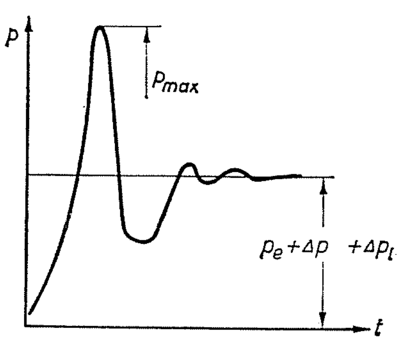

Fig. 11. Dynamic characteristics of precontrolled valve

and dynamic characteristics. Such valves can be designed, even such with negative blowdown characteristics. Agricultural machines in fact need this latter type.

As stated earlier, to operate safety valves to satisfaction they should be both statically and dynamically matched to the given system.

Static matching is particularly important when the flow rate is at or over the nominal value. In pumps driven by internal combustion engines where the highest revolution speed may be rised to the multiple of the basic rpm through the gas feed, the safety valve may cause a considerable overload on the pump, and thus, on the entire system. Such a case is illustrated in Fig. 12.

If the opening pressure is adjusted so that even at lowest rpm it is above the working pressure, at maximum speed a significant overload on the pump must be reckoned with, since the working pressure is generally close to the permissible maximum load of the pump. This case is shown by the curve drawn in a dashed line.

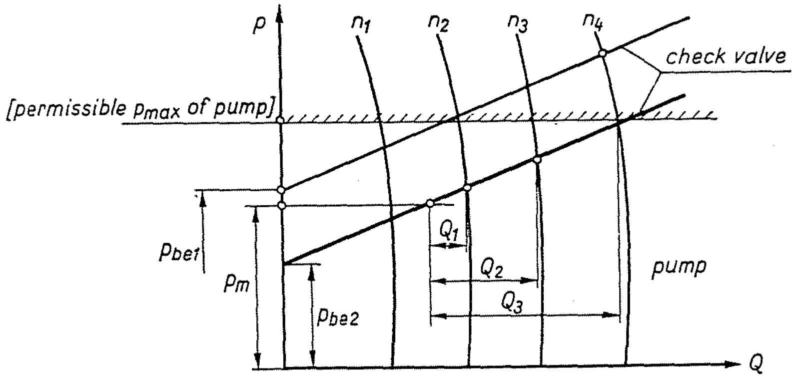

Fig. 12. Co-running of pump and safety valve

If overloading at maximum rpm is to be prevented, the opening pressure must be set at a very low value (the curve in staggered line), and the working pressure of the pump remains unutilised. A comparison of the static characteristics of safety valves has shown that in such cases the use of precontrolled valves is advisable.

If the valve must be cut in at a high frequency, the system — or the pump — might be affected by the pressure peaks arising with the sudden openings (see Fig. 11). The calculation processes and designs now available enable the manufacture of valves with optional static and dynamic characteristics.

The matching of the dynamics of safety valves to the cycle means the prevention of sudden pressure peaks at sudden valve openings. The peaks value is generally 2 to 2.5 times but sometimes 4 to 6 times the opening pressure. With suitable precautionary measures pressure jumps may be kept within permissible limits.

For the appropriate dynamic adjustment the amplitude-frequency curve of the valve must also be known. Such curve for a direct controlled valve can be seen in Fig. 13. Its knowledge is important because the pump delivery is pulsating and, should the frequency of pulsation fall near or coincide with the natural frequency of the valve, pressure pulsation might assume an excessive degree.

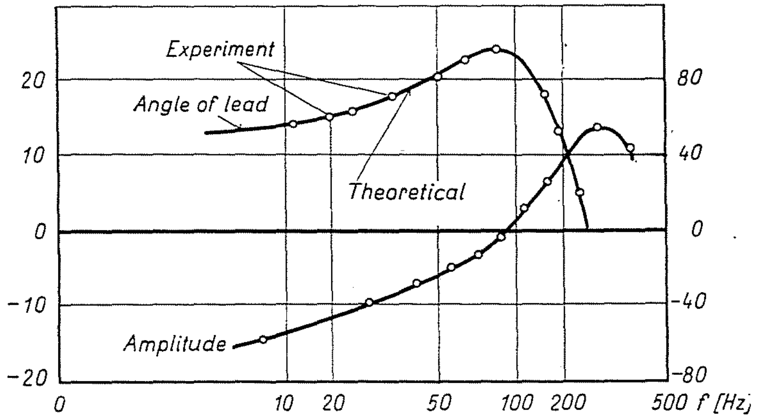

Fig. 13. Amplitude-frequency curve of direct controlled safety valve

Summary

Hydraulic cycles are strongly affected by safety valves operating also as overflow valves. The blowdown range of the overflow valve may be excessive if the gap element is affected by momentum in the direction of closing. Validity of the equation describing the static characteristic of the valve has been proven by measurements. Expected behaviour of safety valves in a hydraulic cycle depends on their static characteristic, amplitude-frequency characteristic and jump function. Commercial safety valves are not equally good in all hydraulic cycles. In a given case, the valve has to be matched to the cycle.

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