A Novel Valve Concept Including the Valvistor Poppet Valve

Björn ERIKSSON, Jonas LARSSON, Jan-Ove PALMBERG

Abstract: These days, energy efficient mobile fluid power systems are of great interest. A mobile system containing several different cylinder drives supplied with a single load sensing pump (LS-pump) has a number of advantages as well as disadvantages. One of the main advantages is the need only one system pump. This makes the fluid power system compact and cost-effective. A challenge is to keep the hydraulic losses at a low level, especially losses at smaller loads. This paper introduces a fail-safe proportional valve element that is based on the Valvistor poppet valve. Due to the demands of flexibility the poppet valve is bi-directional. The valve has an innovative hydro-mechanical layout that makes it fail-safe, unwanted lowering loads, for example, never occur. The new valve includes simple sensors that are suitable for identification of mode switches, e.g. between normal, differential and regenerative modes. It is also possible to maneuver the system with maintained velocity control in the case of sensor failure. In a less complex system the concept has benefits as well. For example in systems where fail-safe bi-directional on/off valve are needed, then without mode sensing capabilities.

Keywords: fluid power, poppet valve, Valvistor, bi-directional, fail-safe,

1 Introduction

These days, energy efficient mobile fluid power systems are of great interest. A mobile system containing several different cylinder drives supplied with a single LS-pump has a number of advantages as well as disadvantages.

One of the main advantages is the need for only one system pump. This makes the fluid power system compact and cost-effective. A challenge is to keep the hydraulic losses at a low level, especially losses at smaller loads.

Currently there are two main options to avoid these kinds of losses. Those are either supplying each cylinder from different dedicated pumps, or using hydraulic transformers with each cylinder together with one system pump. Both solutions entail undesired increased cost and more usage of space.

Another way to reduce the losses at small loads is to allow the cylinders to operate in differential and regenerative mode when possible. This solution implies a need for more flexible valves. The mechanical link between meter-in and meter-out has to be broken.

This paper proposes a valve, based on the Valvistor seat valve [1], that meets the flexible properties mentioned above. The proposed valve can be sized for a wide flow range that suits most mobile applications.

In a less complex system the concept has benefits as well. For example in systems where fail-safe bi-directional on/off valve are needed, then without mode sensing capabilities, see [2].

PhD-student, Björn Eriksson, Assistant Professor, Jonas Larsson, Prof. Jan-Ove Palmberg; Division of Fluid and Mechanical Engineering Systems, Department of Management and Engineering, Linköping University, Sweden

2 Aims

The aim of this paper is to propose a design of a flexible, robust and fail-safe proportional bi-directional poppet valve. The valve has to be robust and fail-safe. Critical functions such as pressure compensation and load holding can not rely on sensors. Nonetheless, the valve must have good metering properties such as pressure-flow characteristics. It is desirable to have some sensors to be able to determine the operational conditions, especially pressure drop direction.

To make a valve system like this possible to produce, at a reasonable cost, it has to be modular. For instance, opportunities must exist to use the same pilot components for all or most valve sizes. The need for actuation force should be kept at a low level to minimize cost.

3 Related research

There are several ongoing projects around the world in the area of split spool valves, for example Eatons Ultronics™ valve [3], and Huscos INCOVA® [4]. The Ultronics design uses fast high performance pilot operated spool valves. INCOVA consists of poppet valves.

The difference between these concepts lies mainly in the hardware layout. The Eatons Ultronics™ concept uses two three-way spool valves that connect each cylinder chamber to pump line and tank. Huscos INCOVA® concept uses four independent two way poppet valves that connect each cylinder chamber to pump line and tank independently.

4 Fail-safe

Mobile fluid power applications usually handle a large amount of energy. If something in such a system goes wrong, it will presumably cause considerable damage. This is one reason why robust and fail-safe components are needed. When gaining flexibility through making valves bi-directional, there is a risk that new failure modes will be introduced, for instance flow in an unwanted direction.

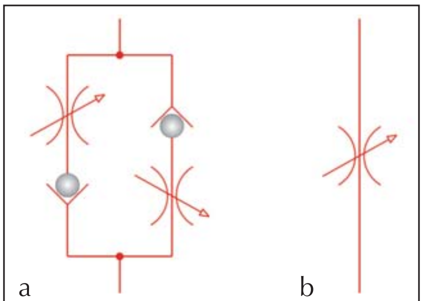

Figure 1. *Schematic sketch of the Fail-safe properties of the Valvistor valve:* (a) *The double pilot concept*, (b) *A single valve concept*

In order to produce a valve that never opens up for flow in the wrong direction, the one proposed here is equipped with two parallel pilot circuits, schematically shown in *figure 1(a)*. If the aim is to achieve a flow from the B-side to the A-side in *figure 2*, then the right-hand pilot in figure 2 valve is used. However, if the pressure p_A is higher than the pressure p_B the result will be that the valve remains closed. This is important as a fail-safe feature.

In a mobile application, a telehandler for example, it is important to prevent falling loads. The schematic valve arrangements in figure 1 are assumed to be meter-in valves, flows upward in the figure. Suppose that the load will be lifted, flow upward in the figure. The left-hand valve in figure 1(a) is used. If the pressure drop is positive in the upward direction a flow will be obtained, otherwise the check valve will prevent a flow. The valve in figure 1(b) would have behaved in the same manner. But if the pressure drop had acted in the other direction, the load would have fallen to the ground. The valve in figure 1(b) is often used together with two pressure sensors to determine the pressure drop and thereby also the possible flow direction. The problem is that if some of the sensors fail, the application can be unpredictable. The demand for pressure sensor accuracy is considerably high. This is because of the wide range of operation. The sensors need to be able to measure pressures in the range of zero up to full system pressure; several mega pascals at the same time as the interesting pressure drop is just a few bars. This means that accuracy has to be extremely high, only a few fractions of a percent.

5 Pilot circuit

In most applications of a valve like this it would be of interest to have pressure compensation. Since the Valvistor valve amplifies the pilot flow it is favorably to pressure compensate the pilot valve. Through this arrangement the compensator can be kept small. One reason for keeping the compensation feature in the hardware is robustness. When using pressure sensors for pressure compensation the sensors need to be extremely accurate. This is because the system has to be able to measure a small pressure difference with two sensors that can manage a large pressure range, see section 4. In this proposed valve, compensators are used instead of pressure sensors.

6 The bi-directional Valvistor

The Valvistor valve would be a suitable choice in a split spool concept. One of the benefits of the Valvistor valve is the high flow gain that can be obtained. To meet the flexibility requirement however the Valvistor valve needs to be bi-directional.

The traditional Valvistor concept is a proportional poppet valve in one direction and acts like a check valve in the other direction [1]. By modifying the original Valvistor concept according to figure 2, it becomes bi-directional and fail-safe, patent pending.

The modified Valvistor consists of a combination of the properties from an A-type and a B-type Valvistor. In an A-type Valvistor the slot in the main poppet connects the A-side with the chamber above the main poppet, in a B-type Valvistor the slot connects the B-side with the chamber above the main poppet; see [1] for more on A- and B-type Valvistor. To make this work it is necessary to add a shuttle valve, or two check valves, inside the main poppet that chooses the highest pressure of p_A and p_B.

If $p_B > p_A$ and the left pilot valve is operated in figure 2, nothing will happen. The valve then acts as a check valve and closes in the direction from A-side to B-side. On the other hand, if the right-hand pilot valve is actuated, flow will start from the B-side to the A-side. It works analogously in the other direction when $p_A > p_B$. This valve is an A-type and a B-type Valvistor at the same time. In one direction it is an A-type Valvistor and in the other direction a B-type.

Used in a system, this valve is not critically dependent on sensors, for instance drifting pressure sensors, to determine flow direction. This is due to the fact that the Valvistor valve is a proportional valve, but acts as a check-valve in the upstream direction. This avoids falling loads. The idea is to use sensors to add intelligence and performance but not reduce robustness and fail-safe properties. The fundamental, critical, function is not allowed to depend on sensors in this valve.

Using this modified Valvistor in a system means a considerable number of elements, in particular pilot valves. One aim of this paper is therefore to find a valve solution where the pilot actuators are independent of the flow capacity of valve. If the same pilot valve elements can be used in almost every size of the valve, it is possible to reduce the manufacturing costs the more that are produced.

To keep the basic functionality independent of sensors, the proposed valve solution must contain a hydraulic pressure compensation in the pilot circuit.

7 Static behavior

The valve has almost the same properties in both flow directions. The difference is the leakage when the valve is closed. In the B-side to A-side flow direction it is leakage-free, a B-type Valvistor. In the A-side to the B-side flow direction there is a small leakage, an A-type Valvistor. There is one possible leakage path in the Valvistor valve, the clearance around the poppet.

When it comes to leakage the difference between the A- and B-type Valvistor is that in the A-type there is a pressure difference between the outlet, B-side, and the chamber above the poppet, and a leakage will occur in the clearance around the poppet. In the B-type case it is different; there is no pressure difference between the inlet, B-side, and the chamber above the poppet, and no leakage flow will be present in the clearance around the poppet.

The static behavior of the valve is described by the equations below. The derivation of the equations used is shown in [5]. It is assumed that the pressure $p_B > p_A$. The right-hand pilot in figure 2 is used. The spring constant in the compensator is ignored.

$$q_m = C_q w_m x_m \sqrt{\frac{2}{\rho}(p_B - p_A)} \quad (1)$$

$$q_p = C_q w_p x_p \sqrt{\frac{2}{\rho}(p_{m2} - p_A)} \quad (2)$$

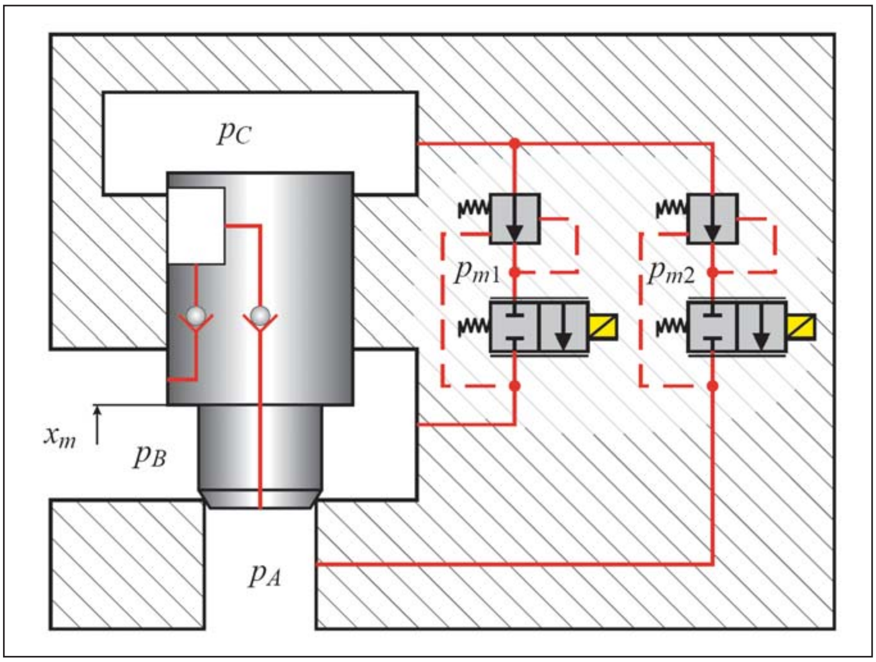

Figure 2. *The modified bi-directional Valvistor valve*

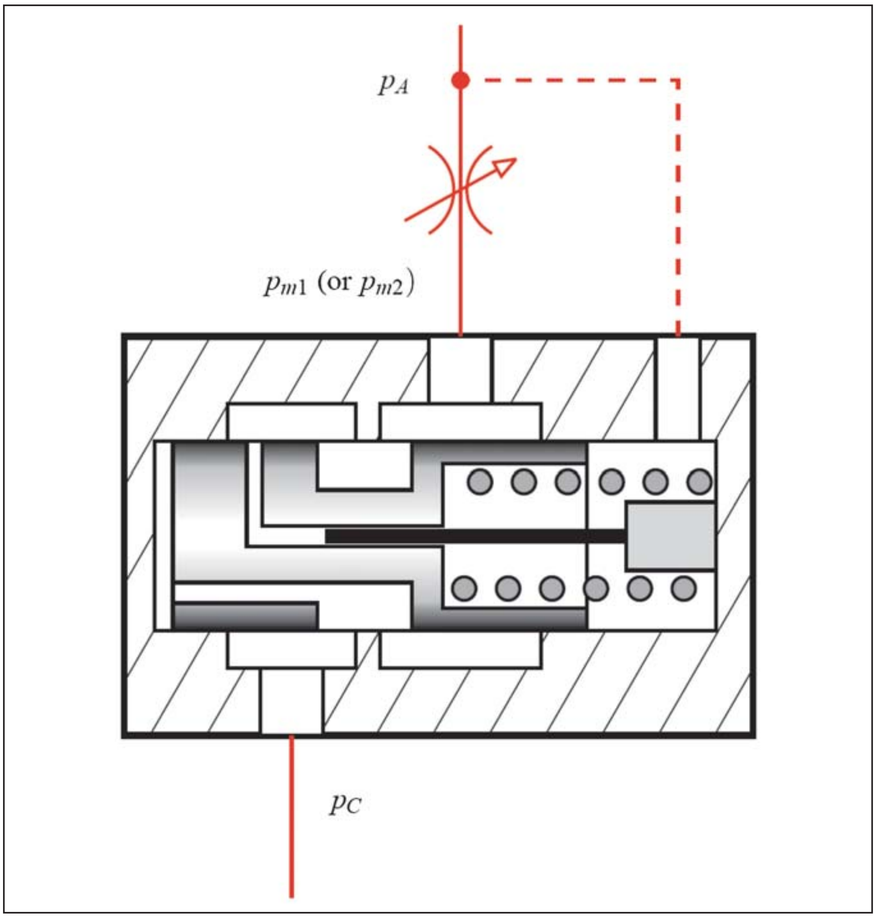

Figure 3. *The compensator used in the pilot circuit. (Observe the pin in the spool)*

$$q_s = C_q w_s (x_m + x_{m0}) \sqrt{\frac{2}{\rho}(p_B - p_C)} \quad (3)$$

$$q_s = q_p \quad (4)$$

$$q_{tot} = q_p + q_m \quad (5)$$

$$A_m p_C + F_s = \kappa A_m p_B + + (1-\kappa) A_m p_A \quad (6)$$

$$F_s = 2 C_q w_m x_m (p_B - p_A) \cos\alpha \quad (7)$$

$$A_C p_{m2} - p_C A_P - p_A (A_C - A_P) - F_0 = 0 \quad (8)$$

If the underlap, x_{m0}, and the flow force, F_s, are ignored the ideal flow gain is

$$g_{ideal} = \frac{q_{tot}}{q_p} = \frac{\frac{w_m}{w_s}}{\sqrt{1-\kappa}} \quad (9)$$

otherwise the gain is

$$g = \frac{q_{tot}}{q_p} = \frac{\frac{w_m}{w_s}}{\sqrt{1-\kappa + \frac{F_s}{A_m (p_B - p_A)}}} - \frac{C_q w_m x_{m0} \sqrt{\frac{2}{\rho}(p_B - p_A)}}{q_p} \approx \frac{\frac{w_m}{w_s}}{\sqrt{1-\kappa}} - \frac{C_q w_m x_{m0} \sqrt{\frac{2}{\rho}(p_B - p_A)}}{q_p} \quad (10)$$

The flow force can often be ignored, and the gain will thereby decrease when the pressure drop increases due to the underlap, the last term in equation (10).

Solving equations (1) to (8) gives the diagrams in *figure 4*. The underlap influences the flow gain in the valve, see equations (9) and (10). The flow pressure coefficient, $K_C = \frac{\partial q_{tot}}{\partial (p_B - p_A)}$ of the valve becomes negative when the pilot is ideally compensated and an underlap is present. Figure 4(c) shows the negative flow pressure coefficient, K_C, in the area where

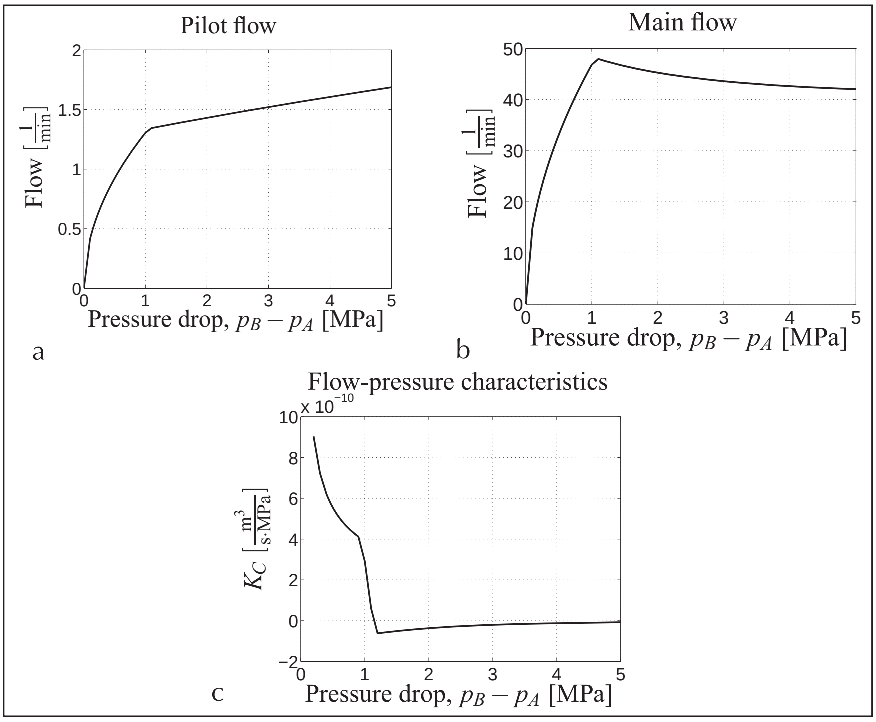

Figure 4. *Flows and flow pressure coefficient as function of pressure drop:* (a) *Compensated pilot flow where pilot opening is held constatnt,* (b) *Total flow, q_{tot}, through the valve with the pilot flow from figure 4(a),* (c) *Flow pressure coefficient for the complete valve*

the compensator is active. It is possible to affect the flow characteristics using the pilot circuit. A compensated valve when an underlap is present can be archived by adding a pin that disturbs the pressure balance in the pilot compensator, see *figures 3* and *4(a)*. The slope of the flow pressure characteristics in the pilot circuit is increased. The pin can be sized so that the K_C - value is always positive.

8 The poppet design

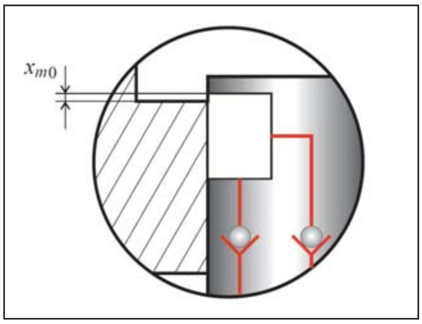

Figure 5. *Underlap in the poppet*

The main design parameters in the poppet concern the slot, primarily the width, which determines the flow gain, see section 7, and the underlap, which over-compensates the valve, see *figures 5* and 4.

There are both dynamic and static aspects to consider when designing the slot in the poppet.

Statically

Flow gain The poppet design determines the flow gain of the valve, see equation (9).

Leakage properties When the pilot is closed the main poppet is also closed. A leakage in the pilot will be amplified and an unwanted valve opening will occur. An underlap prevents the main poppet from opening at small pilot flows, such as leakage.

Dynamically

Dynamic opening At a pressure increase and with no underlap the main poppet will dynamically open and compress the chamber volume above the main poppet. With an underlap the main poppet will stay closed since the chamber pressure will follow the changing pressure.

The mass of the poppet can be ignored dynamically due to the high actuation forces on the poppet. The area ratio (κ) of the poppet should be ½; otherwise the properties become different in the two flow directions.

The slot in the poppet determines the flow gain of the valve, see equation (9), as well as the bandwidth of the valve. The bandwidth is proportional to the slot width, w_s, see equation (11). The influence of flow forces is ignored. [6]

$$\omega_b = \frac{C_q w_s}{A_m} \sqrt{\frac{2}{\rho}(1-\kappa)(p_B - p_A)} \tag{11}$$

Consequently, the design of the slot is a trade-off between flow gain, bandwidth and dynamic properties in closed position.

When designing this kind of valve, there are often demands on both bandwidth and flow capacity at a desired pressure drop. The flow capacity demand determines the size of the main poppet area gradient, w_m. Together, the bandwidth demand and the chosen w_m determine the slot width from the flow gain equation (9). The pilot circuit can now be sized to match the slot orifice since the orifices ideally, ignoring flow forces, have the same pressure drop and the same flow.

Most applications are constituted such that the bandwidth demand decreases when the flow demand increases. This fact fits the valve design. It is a fact that low inertia loads demand high bandwidth. This is often the case for partial loads in a mobile system, not the highest load in the system. The pressure drop is then high and the bandwidth thus also high, see equation (11).

9 Conclusions

This paper proposes an alternative bi-directional design of an existing valve, the Valvistor valve. The valve can be used in a flexible complex system in a fail-safe manner because of the arrangement of the double check valves inside the poppet and the double pilot circuits. It can be designed for a high range of flows due to high flow gain capabilities.

The valve seems to have good metering properties which can be adjusted by modifying the unbalance pin in the compensators.

The fundamental properties of the valve, such as flow gain and bandwidth, allow the same pilot valves to be used in a wide range of flow capacities of the main valve.

Actuation forces of the pilot valve can be kept small because of the pressure compensator in the pilot circuit and the fact that the valve allows a high flow gain and therefore a relatively small pilot flow due to the total flow.

Literature

[1] Bo R. Andersson. *On the Valvistor, a proportionally controlled seat valve*. PhD-thesis, LiTH, 1984. ISBN 91-7372-748-2

[2] K. Heybroek, J. Larsson and J-O. Palmberg. Mode switching and energy recuperation in open-circuit pump control. In *The Tenth Scandinavian International Conference on Fluid Power*, 2007.

[3] Stephen Brian Tuner and David Franz Lakin. Electrohydraulic propotional control valve assemblies. European Patent, 1997. EP 0 809 737.

[4] Keith A. Tabor. A novel method of controlling a hydraulic actuator with four independent metering using load feedback. SAE, 2005

[5] Herbert E. Merritt. *Hydraulic Control Systems*. John Wiley & Sons, New York, 1967. ISBN 0-471-59617-5.

[6] B. Eriksson, B. Andersson and J-O. Palmberg. The dynamic properties of a poppet type hydraulic flow amplifier. In *The Tenth Scandinavian International Conference on Fluid Power*, 2007.

Nov koncept ventila z dodatnim Valvistorjevim sedežnim ventilom

Razširjeni povzetek

V zadnjem času je veliko zanimanje za energijsko učinkovite mobilne hidravlične stroje. Mobilni sistem običajno vsebuje več različnih hidravličnih valjev, napajanih s skupno hidravlično črpalko, ki ima navadno spremenljivo iztisnino in je krmiljena po principu zaznavanja obremenitve. Tak sistem ima veliko prednosti in slabosti. Ena od prednosti je, da je potrebna samo ena črpalka, zato je hidravlični sistem kompakten in cenovno ugoden. Izziv snovanja hidravličnih sestavin je zagotoviti čim manjše hidravlične izgube, še posebej pri majhnih obremenitvah. V splošnem sta za zmanjšanje hidravličnih izgub na razpolago dve možnosti. Prva rešitev je, da ima vsak hidravlični valj svojo črpalko, druga pa je uporaba skupne črpalke in hidravličnega tlačnega pretvornika za vsak posamezen valj. Obe rešitvi sta dragi in zahtevata več prostora za vgradnjo.

Tretja rešitev za zmanjšanje izgub pri manjših obremenitvah pa je, da hidravlični valji, ko je mogoče, delujejo diferencialno in regenerativno. Ta rešitev zahteva fleksibilnejše ventile. Mehanska povezava med vstopom in izstopom mora biti prekinjena. Ta prispevek prikazuje ventil, ki je zasnovan na osnovi Valvistorjevega sedežnega

ventila [1] in v celoti ustreza zgoraj omenjenim zahtevam po fleksibilnosti. Ta ventil je lahko izdelan v različnih velikostih. Uporaben je za večino mobilnih strojev. Sistem takih ventilov ima prednosti v manj kompleksnem sistemu. Na primer v sistemih, kjer so vgrajeni zanesljivi dvopoložajni on/off ventili, ne potrebujemo črpalke z zaznavanjem obremenitve [2].

V prispevku je prikazana konstrukcija fleksibilnega, robustnega in vzdržljivega dvopotnega ventila za vgradnjo v bloke (»kartušni tip« ventila). Kritični funkciji, kot sta kompenzacija tlaka in držanje obremenitve, nista odvisni od zaznaval. Poleg omenjenega je zelo pomembna tudi pretočnotlačna karakteristika ventila.

Hidravlike v mobilnih strojih običajno upravljajo veliko količino energije. Vsakršne napake v takih sistemih lahko povzročijo velike škode. Iz tega razloga mobilni stroji potrebujejo robustne in zanesljive sestavine. Ko zagotavljamo fleksibilnost z vgradnjo dvopotnih ventilov, lahko pride do novih napak, kot npr. pretok v neželeni smeri.

Če želimo izdelati ventil, ki se nikoli ne bi odprl v napačni smeri, je eden od predlogov prikazan na *slikah 1* in *2*. Vsebuje dva paralelna krmilna tokokroga, ki ju shematsko prikazuje slika 1a. Če želimo doseči pretok v smeri B proti A po *sliki 2*, je potrebno vključiti desni elektromagnetni (EM) ventil na sliki 2. Če pa je tlak p_A večji od tlaka p_B, bo ventil ostal zaprt. To je pomembno zaradi zanesljivosti.

V mobilnih strojih, npr. pri teleskopskem viličarju, je pomembno, da preprečimo nevarnost padanja bremena. Zasnova ventilov po shemi na sliki št. 1 je izvedena kot »vtočna« izvedba. Predpostavimo, da je potreben pretok po sliki 1 od spodaj navzgor za dviganje bremena. Odprt je desni protipovratni ventil na sliki 1a. V primeru pozitivne tlačne razlike v smeri navzgor bo omogočen pretok in s tem dviganje bremena. V nasprotnem primeru bo protipovratni ventil preprečil pretok. Ventil na sl. 1b bo deloval podobno, le da bo breme pri negativnem padcu tlaka padlo na tla. Ventil po sl. 1b je običajno uporabljen z dvema tlačnima zaznavaloma, ki zaznata padec tlaka in možno smer pretoka. Problem nastopi, če odpove eno od zaznaval. Takrat se lahko zgodi kaj nepredvidenega. Zahteva po natančnosti tlačnega zaznavala je zelo visoka zaradi možnega širokega področja uporabe. Zaznavalo mora biti sposobno natančnih meritev tlaka v območju od ničle do polnega sistemskega tlaka; več megapaskalov, pri čemer je za nas zanimiv le padec tlaka nekaj barov. Natančnost zaznavala mora biti v nekaj decimalkah odstotka.

V večini primerov, ko se uporabljajo tovrstni ventili, je zaželeno imeti tudi tlačno kompenzacijo. Glede na to, da Valvistorjev ventil ojača krmilni tok, je zaželena tlačna kompenzacija ventila krmilnega toka. S tem je kompenzator lahko majhen. Zaradi robustnosti je zaželeno vključiti kompenzator v ventil. Ko uporabljamo tlačna zaznavala zaradi tlačne kompenzacije, morajo biti ta zelo natančna. Razlog je v tem, da mora biti sistem sposoben meriti majhne tlačne razlike med vstopom in izstopom z dvema zaznavaloma v velikem tlačnem območju (pogl. 4). Zato so v predlaganem ventilu namesto tlačnih zaznaval uporabljeni kompenzatorji.

Valvistorjev ventil je dobra izbira v primeru, da uporabimo konceptualno rešitev z drsniškim batom, ki ima bočno zarezo. Ena od prednosti Valvistorjevega ventila je možnost krmiljenja večjega pretoka. Zaradi večje fleksibilnosti mora biti ventil dvopotni. Pri tradicionalnem Valvistorjevem konceptu je enosmerni proporcionalni ventil, ki deluje kot protipovratni ventil v nasprotni smeri [1]. Z modifikacijo omenjenega koncepta po sliki 2 dobimo zanesljiv dvosmerni ventil. Taka izvedba ventila je patentirana.

Modificirani Valvistorjev ventil vsebuje lastnosti tipa A in B. Pri tipu A reža v glavnem kanalu A povezuje komoro nad batkom. Pri tipu B pa je B-kanal povezan s komoro pod batkom. Da pa se to izvede, je potrebno, dodati izbirni logični (ali) ventil oziroma dva protipovratna ventila znotraj glavnega krmilnega bata. Protipovratna ventila izbirata najvišji tlak med p_A in p_B.

Če je $p_B > p_A$ in vklopljen levi krmilni ventil (slika 2), se ne zgodi nič. Ventil potem deluje kot protipovratni ventil in je zaprt v smeri od A proti B. Po drugi strani, če je desni krmilni ventil odprt, bo omogočen pretok od B proti A. Analogno to deluje tudi v obratni smeri, ko je $p_A > p_B$. Ta ventil je Valvistorjev ventil tipa A in hkrati tudi tipa B – v eno smer je tipa A in v drugo tipa B. Pri uporabi takega ventila v sistemu ta ni kritično odvisen od zaznaval, npr. zaradi odstopanja dejanske vrednosti od izmerjene tlačnega zaznavala, ki določa smer pretoka. Razlog za to je, da je Valvistorjev ventil proporcionalni ventil, ampak v osnovi deluje kot protipovratni ventil, obrnjen nasproti pretoku. To preprečuje nevarnost padanja bremen. Ideja je uporabiti zaznavala, da se doda inteligenca in s tem boljša učinkovitost, vendar se s tem ne zmanjšata robustnost in zanesljivost. V tem ventilu osnovne funkcije niso odvisne od zaznaval in njihovih možnih napak pri merjenju. Uporaba tega prilagojenega ventila v sistemu pomeni uporabo več krmilnih ventilov, ki so vključeni v hidrologični ventil. Eden od namenov tega prispevka je poiskati rešitev ventila, ki je neodvisen od pretoka skozenj. Če se lahko uporabijo enaki krmilni ventili v večini velikostih razredov hidrologičnih ventilov, je s tem mogoče zmanjšati proizvodne stroške, ker se proizvaja večja količina.

Da obdržimo neodvisno delovanje tlačnih zaznaval, moramo v krmilne vode vgraditi tokovne ventile s tlačno kompenzacijo.

Ventil ima skoraj enake lastnosti v obeh smereh. Različna je le količina notranjega puščanja, ko je ventil zaprt. Pri tipu ventila B iz smeri B proti smeri A ni notranjega puščanja, medtem ko nastopi manjše notranje puščanje pri tipu ventila A iz smeri A proti B. Pri Valvistorjevem ventilu je ena od možnosti za notranje puščanje zaradi reže med batom in izvrtino.

Vzrok za različno notranje puščanje med izvedbo tipa A in tipom B je v tem, da je pri tipu A tlačna razlika med izhodom B in komoro nad batom. Do notranjega puščanja prihaja v reži med drsnim delom bata in izvrtino. V izvedbi tipa B je drugače. Med vstopno stranjo B in komoro nad batom C ni tlačne razlike. Posledično ni nobenega notranjega puščanja skozi režo med batom in izvrtino. Statično obnašanje bata je popisano v enačbah spodaj (od 1 do 10). Izpeljava enačb je prikazana v viru [5]. Upoštevano je, da je $p_B > p_A$. Takrat je vklopljen desni krmilni ventil na sliki 2. Vzmetna konstanta v kompenzatorju je zanemarjena.

Pogosto se tudi tokovna sila lahko zanemari in razmerje pretokov (q_{sk} / $q_{krmilni}$), ko padec tlaka narašča zaradi negativnega prekritja (*slika 5*). Rešitev enačb od 1 do 8 da možnost za izris grafov – *slika 4*.

Slika 4c prikazuje negativni tlačni koeficient Kc v področju aktivnega tlačnega kompenzatorja. Dobro je, če je bat oblikovan tako, da je Kc vedno pozitiven.

Statično obnašanje: tokovno razmerje: oblika bata definira tokovno razmerje (en. 9).

Notranje puščanje: ko je krmilni ventil zaprt, je zaprt tudi glavni bat. Notranje puščanje bo takrat povečano in lahko pride do nezaželenega odprtja ventila. Negativno prekritje zaščiti glavni bat pred nezaželenim odprtjem zaradi povečanega toka notranjega puščanja.

Dinamično odpiranje: pri porastu tlaka in brez negativnega prekritja se bo glavni bat dinamično odpiral in stiskal volumen tekočine v zgornji komori. V primeru negativnega prekritja pa bo glavni bat ostal zaprt, vse dokler tlak v komori ne bo začel slediti spremembi tlaka.

Masa bata se lahko pri dinamičnih spremembah zanemari zaradi visoke sile na bat (posledica tlaka). Razmerje površin bata mora biti 1 : 2. Če razmerje odstopa od zaželenega 1 : 2, postane obnašanje ventila v različnih smereh različno (smer pretoka: A→B in B→A). Reža med batom in ohišjem določa razmerje pretokov in delovno območje ventila. To je proporcionalno višini reže W_s (en. 11). Vpliv tokovnih sil je zanemarjen. Oblika reže je odvisna od razmerja med pretoki, delovnim območjem in dinamičnim obnašanjem v zaprtem položaju.

Ko se oblikuje tak tip ventila, so ponavadi znane zahteve po delovnih pretokih v obe smeri pri znanem padcu tlaka. Velikost pretoka je odvisno od višine področja glavnega bata w_m….

Zaključek

Ta prispevek obravnava oblikovanje/modifikacijo alternativnega dvosmernega Valvistorjevega ventila. Ventil se lahko uporablja v fleksibilnih kompleksnih sistemih zaradi zanesljivosti, ki je posledica vgradnje dveh protipovratnih ventilov znotraj krmilnega bata in dvojne krmilne veje. Lahko se uporablja za široko območje pretokov. Ventil ima dobre pretočne lastnosti, ki se lahko korigirajo z nastavitvijo trna v kompenzatorju. Osnovne lastnosti ventila, kot je razmerje pretokov, omogočajo, da krmilne ventile lahko uporabljamo pri različnih velikostih hidrologičnih ventilov.

Aktivacijske sile krmilnih ventilov so majhne zaradi tlačnega kompenzatorja v krmilnem vodu, ki zagotavlja konstantno majhen pretok v krmilnem delu v primerjavi s celotnim pretokom skozi ventil.

Ključne besede: *hidravlika, sedežni ventil, valvistor, dvosmernost, varnost v primeru napak,*

Nomenclature

Quantities and sub indexes used in this paper are listed in *table 1* and *2*.

Table 1. *Quantities*

Quantity	Description	Unity
q	Flow	$\frac{m^3}{s}$
p	Pressure	Pa
C_q	Flow coefficient	-
w	Area gradient	m
ρ	Density	$\frac{kg}{m^3}$
F	Force	N
κ	Area ratio	-
A	Area	m^2
α	Angle	°
ω_b	Break frequency	$\frac{rad}{s}$
g	Flow gain	-
K_C	Flow pressure coefficient	$\frac{m^5}{Ns}$

Table 2. *Sub indexes*

Sub index	Description
m	Main stage
$m1$	Between compensator and pilot valve number 1
$m2$	Between compensator and pilot valve number 2
A	A-side of the Valvistor valve
B	B-side of the Valvistor valve
C	Chamber above the poppet in the Valvistor
p	Pilot
s	Slot in the poppet of the Valvistor
tot	Total
0	Initial
P	Pressure pin in the compensator
c	Compensator
$ideal$	Ideally