Development of a fast seat type switching valve for big flow rates*

Bernd WINKLER, Rudolf SCHEIDL

Abstract: Highly accurate and fast response drives for fast and precise positioning for instance, currently rely on big servo or proportional valves. Such valves are costly and are applied to resistance control with its inevitable energetic losses. A promising method to get rid of such losses and to reduce the valve costs is to use appropriate switching valves in combination with switching control. Former investigations showed that switching valves with flow rates of about 100 l/min at 5 bar and switching times of 1 to 2 ms can cover a reasonable range of applications. Commercial switching valves don't meet such requirements. In this paper, a novel, hydraulically piloted, seat type switching valve which approximately fulfils the mentioned requirements on switching time and flow rate is presented. The high flow rate is accomplished by multiple metering edges in a plate type valve, just like the well known Hörbiger compressor valve. The fastest switching time which is strongly pressure dependent is about 1.5 ms. Its seat valve properties make it highly suitable for emergency applications and mobile hydraulic applications were absence of leakage is required.

Keywords: switching valves, big flow rate, fast switching, seat type valves,

1 Introduction

Cost reduction and improvement of energy efficiency are of capital importance to keep hydraulic systems competitive with other drive technologies. The application of the hydraulic switching technology is a promising attempt to meet these demands.

Today, fast and precise hydraulic motion control can only be realized by the use of big and costly servo or proportional valves. They are typically applied to resistance principle with its considerable losses. Hydraulic switching control which requires fast and big switching valves can increase energy efficiency and lower costs.

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A poppet valve based realisation of this principle has been presented in [1, 2]. This spool type switching valve is directly operated by a solenoid and switches on and off within 1 ms at a nominal flow rate of about 45 l/min at 5 bar pressure drop.

Such a directly operated switching valves have two main shortcomings:

- The nominal flow rate is practically limited by the flow forces. Own investigations showed that a flow rate of about 50 l/min is a reasonable limit for a robust performance.
- A leaking valve is unacceptable in some cases, like for instance in many mobile machinery applications.

To avoid these shortcomings of spool valves a seat valve was developed. It is hydraulically piloted and should fulfil the following demands:

- Switching time of about 1 to 2 ms
- Nominal flow rate of about 100 l/min at 5 bar pressure drop
- No leakage
- Low production costs
- Low electric power consumption of the pilot stage

2 Basic concept

High flow rates need a large flow passage area. This can be realised either by a big stroke or by a big diameter of a spool or poppet valve respectively. But, both measures tend to increase the switching time which hinders to achieve the performance data mentioned above.

The new valve presented in this paper utilises the Hörbiger plate valve principle [3,4] which has several annular rings at two opposite valve plates to form multiple metering edges. This principle facilitates a very big flow passage area at a given poppet diameter and stroke. The Hörbiger plate valve which is used since about a hundred years as a compressor valve and a simplified scheme of the presented hydraulic valve are shown in *Figure 1*. The passage area is controlled by the distance of both plates.

The flow saturates if the plates' distance is approximately one half of the groove width. The smaller this

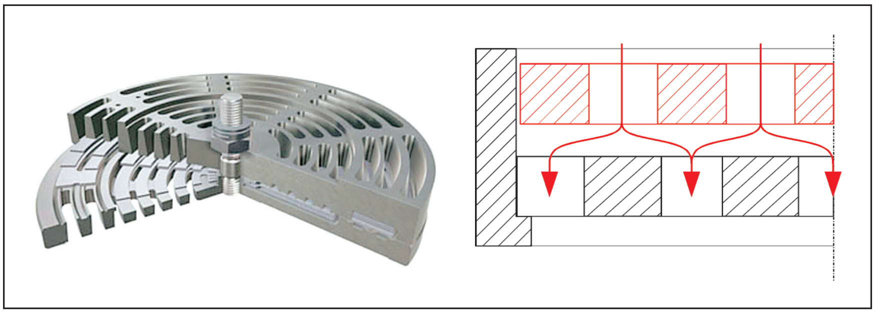

Figure 1. *Hörbiger plate valve (left) and simple scheme of the flow path (right)*

width the smaller the necessary plate stroke and the faster the switching time of the valve. Exploitation of this measure is limited by the manufacturing process, by oil contamination, or by fluid friction effects which may destroy the positive effect narrow grooves if they become too small.

To make it a fully controllable valve it must be equipped with some actuator to open and close the poppet. For this purpose, the Hörbiger principle is combined with the 2/2 way cartridge valve principle which applies a plunger to control the poppet position. Thus, the new valve can be considered a modified cartridge valve which applies the multiple metering edges plate valve principle instead of the conventional conical seat valve.

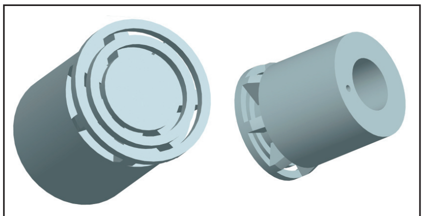

Figure 2. *Poppet with coaxial metering edges*

Figure 2 shows the realized poppet with its multiple metering edges. The opposite fixed plate is designed accordingly.

3 Valve design

As already mentioned, the coaxial rings forming the metering edges are tiny structures to achieve small poppet strokes and, hence, a small switching time. The limitations in the feasible and affordable manufacturing techniques of the prototype valve resulted in a groove width of 1mm. Thus, the saturation stroke limit is about 0.5 mm.

Besides the number of the needed metering edges for a nominal flow rate of 100 l/min at 5 bar it has to be ensured that rigidity and strength of the rings are sufficient and the surface pressure at the contact areas between poppet and opposite plate is not too high. Stresses and distortions have been analyzed, both, by an analytical model based on curved beam theory and by a Finite Element model.

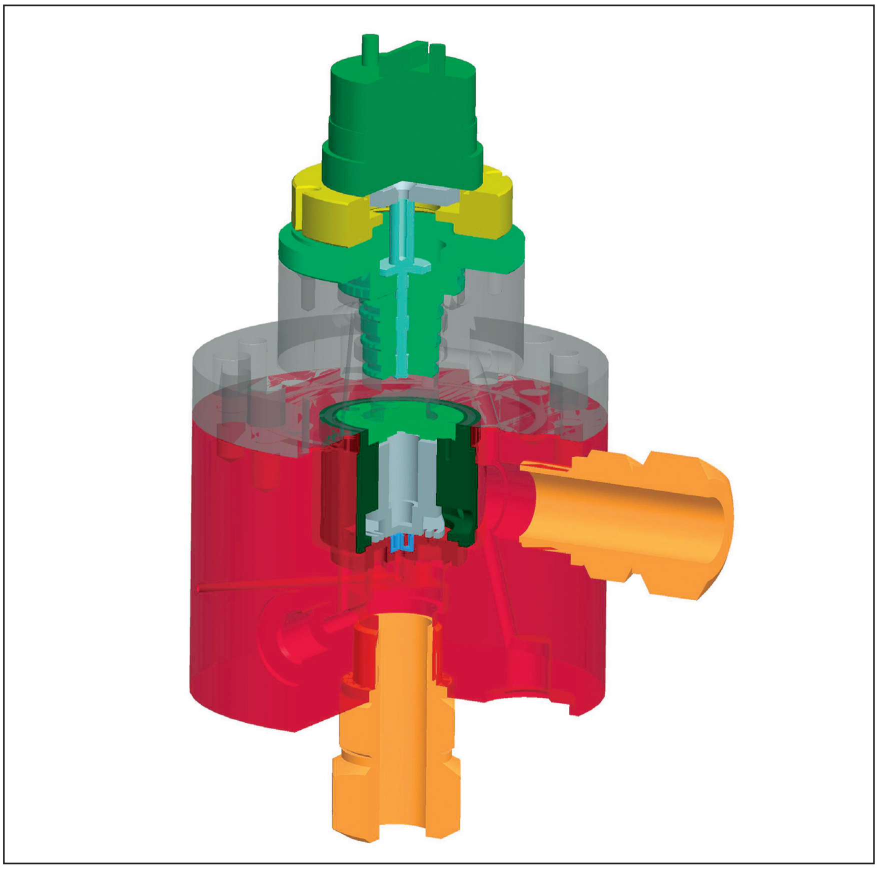

Figure 3. *Design of the seat type switching valve*

Figure 3 shows the final design of the valve. As pilot valve a 3/2 way valve developed at LCM years ago for another application was used. The nominal flow rate of this pilot valve is about 3.5 l/min at 5 bar pressure drop and an oil temperature of 45°C. The switching time of this valve is about 1.6ms (oil temperature: 23°C).

The poppet and its opposite plate are depicted in detail on the right side of Figure 3. In the center bore of the poppet the counterbalancing spring for closing the valve is arranged (not shown in Figure 3).

The poppet is guided within the cartridge which fixes also the opposite plate. Above the poppet a plate for limiting the poppet stroke is arranged.

4 Experimental Results

Both, the pilot valve and the main stage have been measured in detail. The following experimental results are focusing on the main stage, since the pilot valve has not been a proper subject of this development.

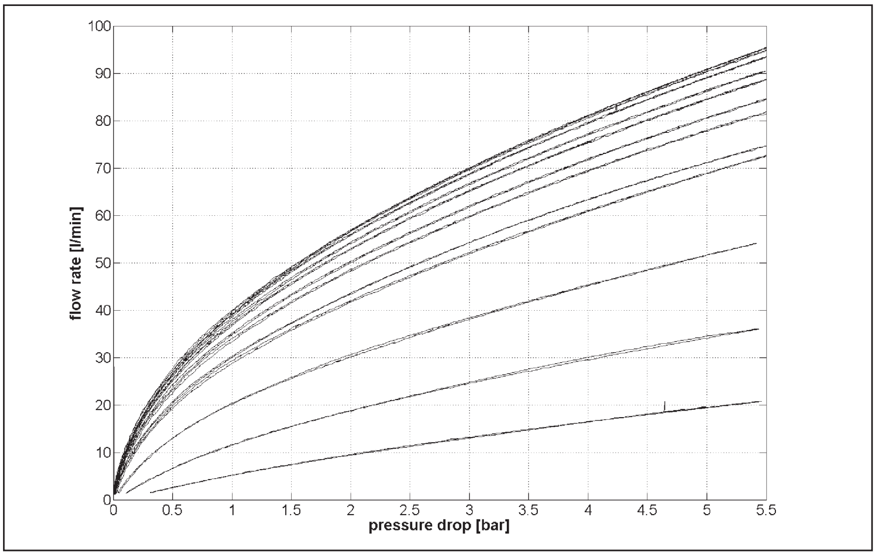

Figure 4. *Steady state flow characteristic of the valve at 32°C*

4.1 Steady State Flow Characteristics

Figure 4 shows the steady state flow characteristic of the valve (main stage) for different poppet positions which have been adjusted by a special adjustment device in the range of 0.05 mm to 0.6 mm in steps of 0.05 mm.

The expected nominal flow of 100 l/min at 5 bar is not fully achieved. The maximum measured flow rate is about 90 l/min at 5bar. This results from additional losses at the metering device which have not been taken into account in the basic dimensioning. Of course, with these experimental findings the model for calculating the nominal flow rate can be updated.

4.2 Dynamic Experiments

It is well known that the switching time of hydraulically piloted 2/2 way valves which exploit the two main stage pressures and don't apply a separate pilot pressure source depend strongly on the pressure drop over the valve. Thus the switching time has to be assessed in conjunction with the pressures at the two ports of the valve.

For this valve the flow rate at 200 bar difference pressure is about 600 l/min. Such high instantaneous flow rates can normally only be realized by very dynamical accumulators right at the ports of the valve (see *Figure 5*) and with very low parasitic inductivities between these two accumulators.

In the experiments the accumulators have been attached outside the valve, resulting in some sub-optimal dynamical performance of the system. With integrated accumulators, however, the switching times can be considerably decreased.

Figure 6 depicts the results for the valve opening at different supply pressures and with different counterbalancing springs. At time = 0 the PWM-signal for the solenoid of the pilot valve is set (PWM=1 means 24V and PWM=0.5 means 0V). It has to be pointed out that in the diagrams only the voltage signal of the sensor is depicted and not the exact curve of the poppet position. A sound calibration of the position sensor with a reasonable effort was not possible due the special placement of the sensor in the valve. But this shortcoming is acceptable since the switching time is the main performance measure.

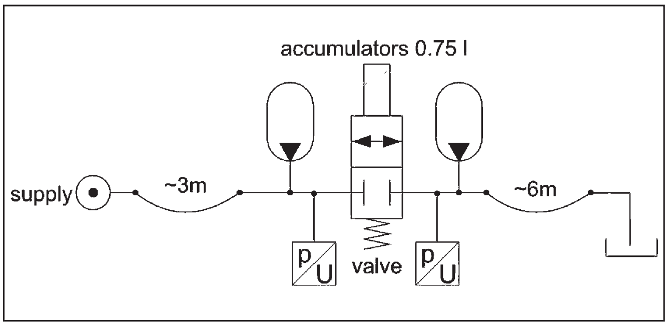

Figure 5. *Hydraulic circuit for measurements*

It takes almost 2 ms before the main stage starts to move. The switching time depends only marginally on the counterbalance spring stiffness. As expected, the pressure drop over the valve has

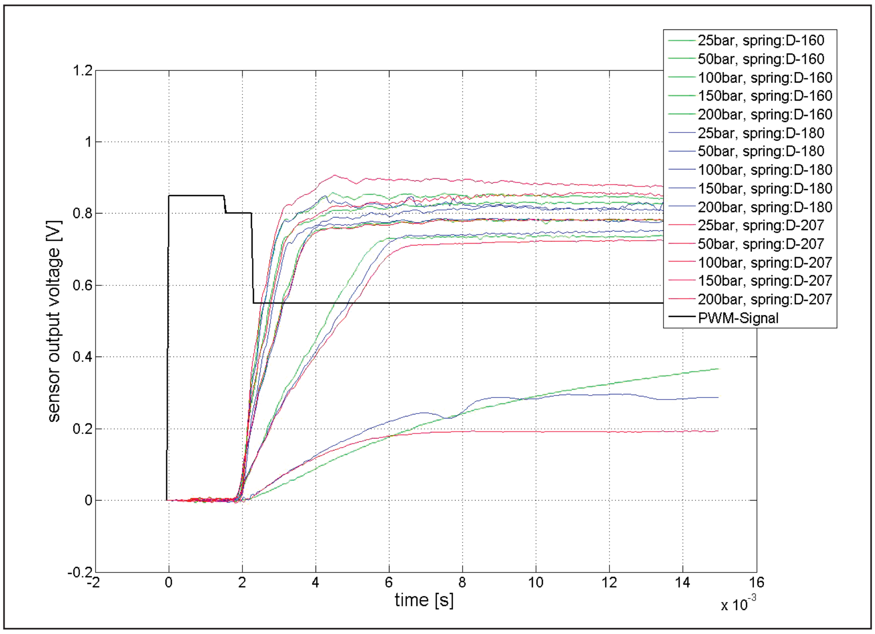

Figure 6. *Opening curves of the valve's main stage (consider: switching of the pilot stage after the on signal takes about 1.6 ms)*

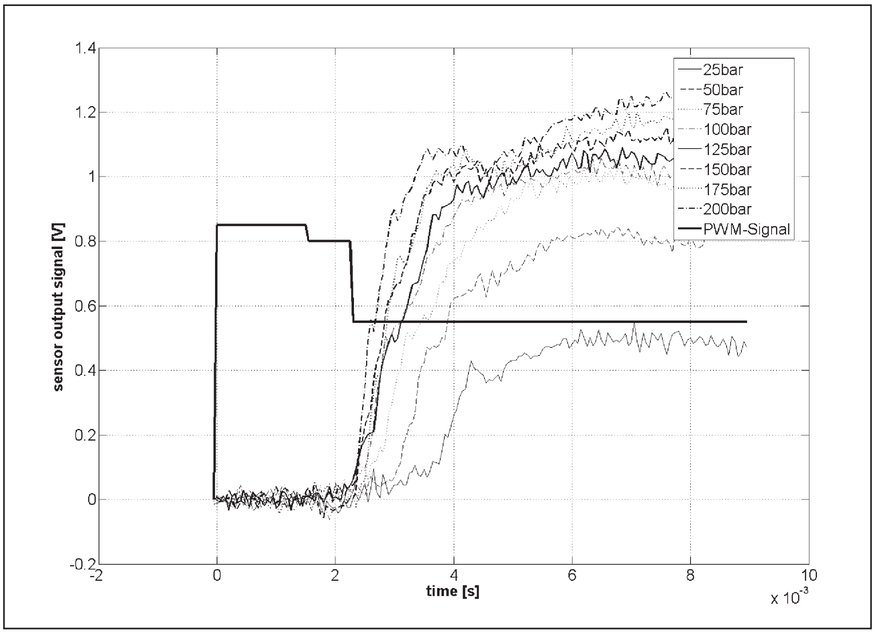

Figure 7. *Valve opening curves without accumulators*

the most important influence. For low pressures the switching times (which is defined as the time between 5% and 95% of total poppet stroke) are about 4 to 6 ms. For higher pressures (200 bar) the switching time of the main stage is reduced to 1.1 ms in the fastest configuration. But it must be pointed out that the rated pressure difference is only pending as long as the valve is closed. Once the valve starts opening a rapid drop of the pressure difference over the valves takes place which has a strong influence on the further valve motion.

To single out the influence of the supply system the switching times without accumulators have been recorded and are depicted in *Figure 7*.

The pressure evolution and the valve opening and closing motion depend on the dynamics of the whole system unless a perfect decoupling of the switching process from the system is achieved.

Figure 7 shows oscillations in the opening process resulting from pressure oscillations in the hydraulic system. In contrast to switching valves with a separate actuation system this valve has no proper characteristic switching time.

Besides the opening time of the main stage the closing time is a significant performance value. The results of the corresponding measurements are shown in *Figure 8*. Like for the opening of the valve the influence of the pressure drop over the valve is essential for the closing time. Again, the used counterbalance spring is not significant.

After the valve is closed one short reopening can be observed. The reason for that are negative pressure differences as can be seen in *Figure 9*. This valve acts as a check valve in the negative flow direction. When the pressure at the low pressure side rises above the high pressure side, what can easily occur via pressure pulsations in the system, the main stage will open.

For the measurement with 200 bar

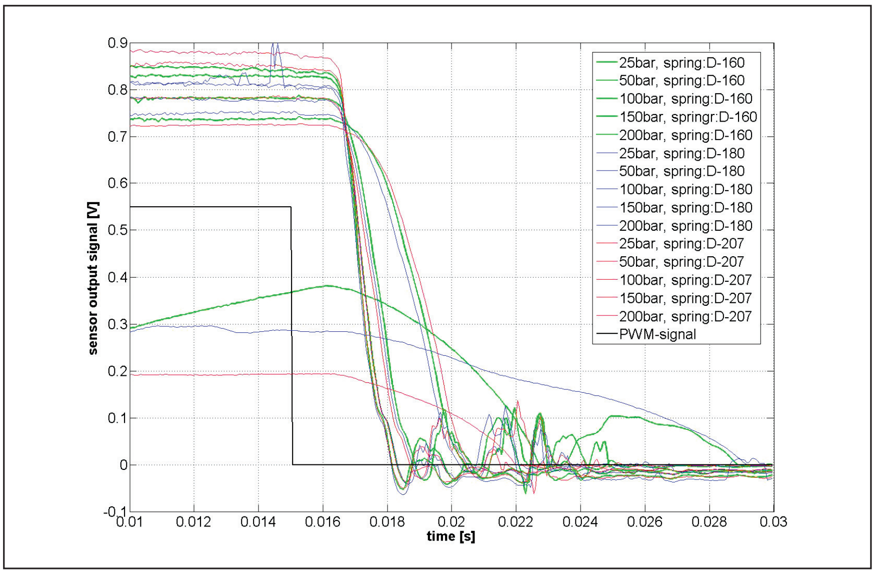

Figure 8. *Closing time of the main stage*

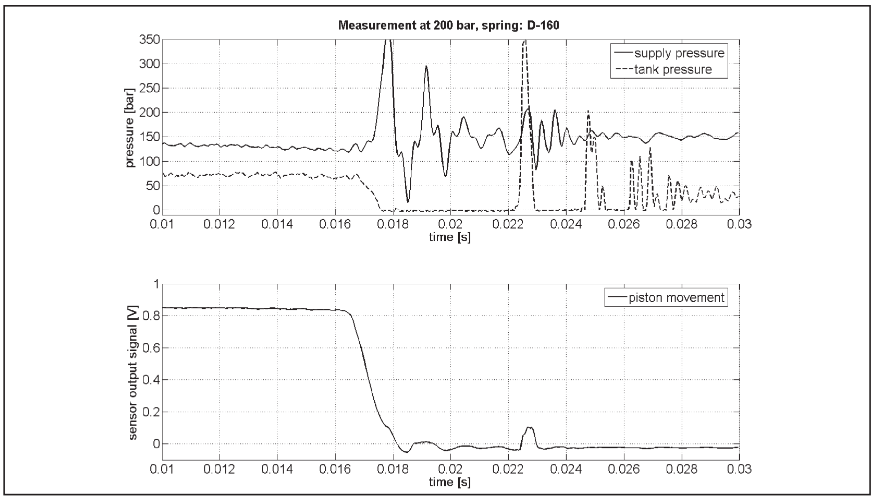

Figure 9. *Pressure and poppet position for valve closing at 200 bar supply pressure*

supply pressure the valve closing and the corresponding pressures before and after the metering edge are depicted in Figure 9. After the closing of the valve (about 2ms) a pressure peak occurs at the low pressure side at about 0.022s. This pressure peak is responsible for the short reopening of the valve as can be seen in the lower diagram.

4.3 Leakage

Seat type valves normally are leakage free in the closed position. But this valve showed some small leakage as indicated in Table 1. For the measurement the inlet port is set under supply pressure and the leakage flow at the outlet port is measured in a measuring beaker. Sealing is provided by several bands along which the poppet and the opposite plate are in contact. The optimal width of these bands must assure a high enough contact pressure to avoid leakage but must avoid a damage of the surface by a too high contact pressure. The quality of this seat has to be improved in further versions of the valve. The main reason for the unexpectedly high leakage flow is that the sealing seat was slightly harmed by some hand grinding carried out to remove some adhesive on top of the position gauge.

Table 1. *Leakage at different supply pressures*

Pressure [bar]	Leakage [ml/min]	Oil Temperature [°C]
50	0.45	27
100	0.8	28
200	2.8	28

5 Conclusion

A new seat type valve has been presented which basically fulfils the demands on switching time (1ms) and nominal flow rate (100 l/min). Due to the dependency of the switching time on to the pressure drop over the valve the demanded switching time of about 1ms can only be achieved with a sufficient pressure drop prevailing at the valve even when it is open. The measured nominal flow rate of this valve (90 l/min) can easily be increased to higher values with some small modifications in the valve geometry.

For the presented prototype of the valve the annular grooves are 1 mm. This design can be realised with standard manufacturing processes like turning and milling without trouble. A thorough manufacturing analysis would be necessary prior to reducing this groove width considerably.

With modern production processes like etching or laser cutting tinier structures can be realised. This could reduce the needed poppet stroke and thus reduce the switching time.

The used pilot valve in the presented prototype is not properly aligned with the main stage. A bigger pilot valve would reduce the valve's switching time.

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Razvoj hitrega sedežnega preklopnega ventila za velike tokove fluida

Razširjeni povzetek

Zmanjšanje cene in izboljšava energetske učinkovitosti sta izredno pomembna dejavnika pri ohranjanju konkurenčnosti hidravličnih sistemov v primerjavi z drugimi pogonskimi tehnologijami. Uporaba hidravličnih sedežnih 2/2-ventilov je obetajoč poskus pri izpolnjevanju teh zahtev.

Visoko natančni in hitro odzivni pogoni, npr. za hitro in natančno pozicioniranje, so trenutno odvisni od velikih servo- ali proporcionalnih ventilov. Takšni ventili so dragi in temeljijo na principu uporovnega krmiljenja z visokimi energetskimi izgubami. Hidravlična preklopna tehnika z uporabo 2/2-ventilov, ki zahtevajo hitre in velike preklopne ventile, lahko pomaga pri izboljšanju energetske učinkovitosti in pri zmanjševanju cene. Realizacija takšnega ventila z uporabo predkrmilnega ventila je obravnavana v [1, 2]. Uporabljeni vzdolžni batni preklopni ventil je direktno vkrmiljen z elektromagnetom, ki vklaplja ter izklaplja ventil v času 1 ms pri imenskem volumskem toku 45 l/min in tlačnem padcu 5 bar na krmilni rob. Takšni direktno vkrmiljeni preklopni ventili imajo dve glavni pomanjkljivosti:

- Imenski volumski tok je praktično omejen s tokovnimi silami.
- Zaradi lekaže v ventilu je takšen ventil nesprejemljiv v nekaterih aplikacijah, kot je npr. mobilna hidravlika.

Novorazviti sedežni ventil, ki je predstavljen v tem prispevku, odpravlja zgoraj omenjene pomanjkljivosti z vzdolžnim batom. Ta ventil je

vkrmiljen hidravlično in dimenzioniran za izpolnitev naslednjih zahtev:
- preklopni čas okrog 1 do 2 ms,
- imenski volumski tok okrog 100 l/min pri tlačnem padcu na krmilnem robu 5 bar,
- ni lekaže v ventilu,
- nizki proizvodni stroški,
- nizka poraba električne energije predkrmilnega ventila.

Novi ventil temelji na znanem kompresorskem ventilu s krmilno ploščo proizvajalca Hörbigerja (*slika 1*), katerega princip delovanja je predstavljen v [3, 4]. Ta ventil vsebuje dve nasprotno postavljeni krmilni plošči z več utori, ki tvorijo krmilne robove. Ta princip omogoča velik presek odprtja ventila ob določenem premeru in pomiku krmilnega bata in s tem tudi velik volumski tok skozi ventil. Kontrolni bat ventila s krmilno ploščo in več krmilnimi robovi na koncu je prikazan na *sliki 2*.

Slika 3 prikazuje novorazviti sedežni ventil, predstavljen in analiziran v tem prispevku, ki temelji na principu Hörbigerejevega ventila. Za dosego majhnih preklopnih časov je bistven pogoj, da pomik preklopnega bata ni večji od 0,5 mm glede na to, da zaradi tehnologije, ki je bila uporabljena pri izdelavi ventila, obodni utori nimajo večjih dimenzij od 1 mm. Poleg tega ima ventil zadostno število krmilnih robov za izpolnjevanje zahteve po volumskem toku 100 l/min pri 5 bar tlačnega padca. Trdnost in togost krmilnih robov oz. utorov kakor tudi površinski pritisk na kontaktnih površinah med krmilnim batom in nasprotno krmilno ploščo so dimenzionirani in analizirani s pomočjo analitičnih modelov in metode končnih elementov. Na ta način površinski pritisk med krmilno ploščo in krmilnim batom ni previsok, zato ne pride do mehanskih poškodb, in tudi ni prenizek, zato ne pride do lekaže.

Eksperimentalni rezultati izvedenih meritev statičnih in dinamičnih lastnosti ventila so prikazani na slikah 4 do 9, kot je razvidno s *slike 4*, volumski tok pri 5 bar tlačnega padca doseže 90 l/min, kar je nekoliko manj od zahtevanih 100 l/min. Vzrok za to so dodatne lekažne izgube znotraj predkrmilnega ventila, kar je bilo prezrto pri dimenzioniranju glavnega ventila in bo v bodoče izboljšano s pomočjo teh raziskav.

Analiza dinamičnih karakteristik je bila izvedena s pomočjo zasnove hidravličnega vezja, prikazane na *sliki 5*, ki zagotavlja dovolj velik kratek čas potreben volumski tok do 600 l/min pri tlačni razliki 200 bar. Na *sliki 6* in *7* so prikazane meritve odprtja ventila na skočno funkcijo. Iz rezultatov je razvidno, da pride do zakasnitve pri reakciji glavnega ventila, ki znaša 2 ms, celotni preklopni čas pa znaša 4 do 6 ms, kar je močno odvisno od tlačne razlike in zanemarljivo od togosti vzmeti, ki deluje nasproti krmilnemu batu ventila. Preklopni čas ventila se zmanjša na 1,1 ms pri tlačni razliki 200 bar. Oscilacije, ki so vidne na merilnih rezultatih na *sliki 7*, so posledica oscilacij tlaka v hidravličnem sistemu, ker niso uporabljeni hidravlični akumulatorji, medtem ko rezultati, prikazani na sliki 6, ne vsebujejo oscilacij, ker so bili v tem primeru uporabljeni hidravlični akumulatorji. Na *sliki 8* so prikazane meritve zapiranja ventila na skočno funkcijo. Najpomembnejši vplivni dejavnik na kratek zapiralni čas je padec tlaka v ventilu. Po zaprtju ventila pride do ponovnega krajšega odprtja ventila, kar je posledica negativnih tlačnih razlik v sistemu, kot je razvidno s *slike 9*, kjer se pojavi tlačni vrh pri času 0,022 s. Raziskave so tudi pokazale, da je nujna dodatna optimizacija ventila glede notranje lekaže, ki se pojavila pri meritvah. Rezultati meritev lekaže so predstavljeni v *tabeli 1*.

Zaključimo lahko, da predstavljeni ventil v osnovi izpolnjuje zahteve in pričakovanja, vendar si bo treba še nadalje prizadevati za njegov razvoj in optimizacijo, pri izdelavi pa uporabiti nekatere napredne izdelovalne tehnologije.

Izvleček: Visoko natančni in hitro odzivni pogoni, npr. za hitro in natančno pozicioniranje, so zdaj odvisni od velikih servo- ali proporcionalnih ventilov, ki so dragi in temeljijo na principu uporovnega krmiljenja z visokimi energetskimi izgubami. Obetajoča možnost za odpravo teh slabosti in za zmanjšanje cene ventila je uporaba ustreznih preklopnih ventilov z volumskimi tokovi okoli 100 l/min pri tlačnem padcu 5 bar na krmilni rob in preklopnih časih 1 do 2 ms, ki lahko pokrijejo znaten obseg aplikacij. Komercialni preklopni ventili ne ustrezajo takšnim zahtevam. V prispevku je predstavljen nov, hidravlično vkrmiljen sedežni ventil, ki približno izpolnjuje omenjene zahteve glede preklopnega časa in volumskega toka. Visok volumski tok je dosežen z uporabo več krmilnih robov v ventilu s krmilno ploščo, podobno kot pri dobro znanem kompresorskem ventilu proizvajalca Hörbiger. Najhitrejši preklopni čas novega ventila je močno odvisen od tlaka fluida in dosega 1,5 ms. Zaradi svojih lastnosti je novi sedežni ventil primeren za uporabo v kritičnih in mobilnih aplikacijah, kjer ni dovoljena prisotnost lekaže.

Ključne besede: preklopni ventili, velik volumski tok, hitro preklapljanje, sedežni ventili,