Novel Poppet Valve for Digital Hydraulic Transformer

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Abstract.

Need for an efficient and robust hydraulic transformer for secondary controlled drives is well established. The 4-bit, linear-acting Digital Hydraulic Transformer (DHT) approach has given promising results, but in development of more advanced functionality, valves failed due to high demands. No valve was found that met requirements, so effort was undertaken to design a valve, specifically for DHT, from first principles, that shifts reliably in extreme conditions. This novel poppet valve is conceived, manufactured, tested and evaluated as potential DHT Bit-1 valve. Apparatus is set up to establish p-Q curve, and to measure C_v and leakage. As well, apparatus is set up to explore critical flow and response at critical flow but found to be inadequate for the flows and pressure spikes involved. Valve p-Q curve and leakage are applied to existing DHT model and impact on performance is examined, including efficiencies and End Of Stroke strategies.

Keywords. Digital hydraulic, hydraulic transformer, 2/2 valve, balanced, double piloted.

1. INTRODUCTION

1.1. Background

It is well known that hydraulic transformers operating on a constant pressure net can eliminate many losses associated with conventional hydraulics and increase the overall efficiency of machines. First, they eliminate valve losses. Second, they allow energy regeneration from aiding and braking loads. Third, the secondary controlled architecture allows the prime mover to operate at it's point of highest efficiency. The main reason hydraulic transformers are not generally used is the market is still waiting on a winning design[1].

The Digital Hydraulic Transformer (DHT) concept was first disclosed in a U.S. patent application in 2005[2], and first presented academically at the Eleventh Scandinavian International Conference on Fluid Power, Linköping, Sweden, in 2009[3]. The first technology of its kind, it represented the nexus of secondary controlled drives and parallel channel digital hydraulics, both of which were still new technologies, with relatively low industry acceptance, at that point[4][5].

The central concept of the DHT involves dividing the working area of a piston into axially-offset, binary weighted annular areas, Fig. 1. Selectively pressurizing the individual annular chambers results in a cumulative output force that can be incrementally raised or lowered, which is essentially the ability to change piston size on the fly. With just four bits, this digital piston is capable of a wide range of cumulative force output illustrated by Fig. 2.

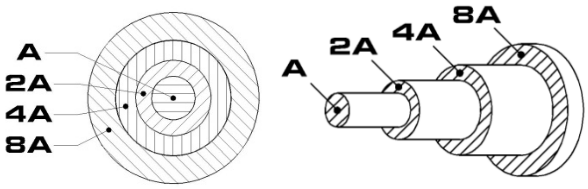

Figure 1. Axially-offset binary-weighted, areas

Figure 2. Range of four-bit system output

Inside the DHT is a special displacer piston, the transtatic bridge (TSB), which is two of these digital pistons oriented back to back. A matrix of 2/2 valves controls which of three pressures acts on each individual area. Thus, it is possible to vary displacement on both sides such that hydraulic power transformation according to discrete ratios may occur. In the case of the 4-bit DHT, a total of 143 unique transform ratios are possible when all combinations are used. Regardless the output state, the product of input pressure and flow equals the product of output pressure and flow, as shown in Equation 1.

$$Q_{in} \times p_{in} = Q_{out} \times p_{out} \qquad (1)$$

Fundamentally, the DHT combines high pressure fluid, low pressure fluid and transform fluid in specific volume ratios to vary output pressure and ultimately displace the fluid volume required by the actuator. Figure 3 shows a simplified DHT schematic.

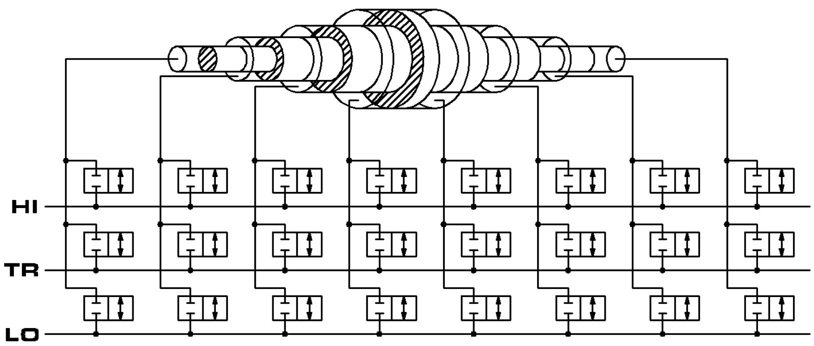

Figure 3. 4-bit DHT schematic

The TSB travels axially in its housing during transformation. When it nears its mechanical travel limit the pressures applied to each side of the TSB are exchanged end for end so that transformation can continue, but with the TSB now moving in the opposite direction. By these reciprocation means unlimited flows are made possible.

During this End of Stroke (EOS) event, output flow and pressure momentarily drop to zero and then the entire system is accelerated back towards steady state. This happens quickly enough that, in systems with non-negligible mass, inertia damps velocity fluctuation, but it cannot completely eliminate it. While this pressure and flow discontinuity may be acceptable for some applications not requiring fine control, such as bulk transfer, most applications are not and so work has gone into developing strategies to combat and minimize pressure and flow fluctuation at EOS.

One approach to mitigate the EOS output fluctuation is to inject high pressure fluid directly into the transform rail to supplement transform flow while bridge changes direction. This could be accomplished using a 2/2 servo valve plumbed between high rail and transform rail, but with 24 valves already, it is the view of the author that the DHT cannot endure any more complexity and expect commercial success, so a servo valve is ruled out. As design constraint, the function must be achieved using only the bit valves already making up the transformer.

1.2. Problem

In the DHT matrix schematic architecture, flow directly from high rail to transform rail is possible through any of the eight bit rails with both HI and TR valves open. Which bit rails are chosen to provide cancellation flow depends on output state. Opportunity exists for motoring EOS cancellation in chambers that are contracting to their minimum volume at TR pressure transitioning to expanding at HI pressure. In this case the HI valve can be turned on in advance so that it is open to flow through bit rail and through TR valve into TR rail. In addition to providing cancellation flow, the additional axial force on the bridge in the time from HI valve opening to TSB velocity zero serves to accelerate the TSB in the new direction faster.

A pulse of high pressure hydraulic fluid through HI and TR valves in series can satisfy the volume requirement that is the difference between steady state DHT output and what the bridge is actually delivering. But because the volume injection time is shorter duration than the period it is compensating for the flow rate must be higher than steady state. For instance, if Bit-1 steady state is 5 gpm, instantaneous injection flow rate might be 20 gpm or higher and the valves must still close reliably.

Turning one valve on before another valve on the same bit rail turns off creates an opening overlap. The opening overlap has to be carefully controlled to inject precise amounts of high pressure fluid into the output rail to cancel EOS flow discontinuity. During this overlap valves experience pressure differences upwards of several thousand psi and flow increases sharply.

In development of this phased overlap control it was discovered that the direct acting solenoid operated spool valves of the DHTM475 transformer were being subjected to conditions beyond what they were designed to accommodate and were failing to close.

In this case, high enough Δp acting for even a few milliseconds accelerates enough flow through valves for inertial flow forces on spools to dominate return spring force and the

NC valves stay open when de-energized. In conventional systems this overflowing condition rarely or never occurs, but in the DHT application this occurs at every EOS.

Avoiding overflowing valves is especially critical for the DHT application because of the mode of failure. Unpredictability of transform ratio is not something that can be tolerated in a system with a transform ratio as high as 15x. If the wrong valve or valves overflow and pressurize, with high pressure, chambers that are to be pressurized with low pressure, the output pressure generated from a typical 3,000 psi accumulator pressure could be transformed to as high as 45,000 psi, which would cause damage to equipment and potentially harm people. This necessitates valves with reliable, consistent response, both on and off.

Demanding even more of valves, when a 2/2 valve switches at high pressure feeding a finite chamber volume through a tube, water-hammer pressure peaks can reach 2x to 3x rail pressure. Across many valves switching in coordination, the cumulative transient environment is hostile. Valves must not only survive 5,000 psi static; they must survive the dynamic peaks, and resultant instantaneous flows, created by neighbouring valves switching.

Also, conventional valves restrict flow too much. The DHT is especially sensitive to valve flow resistance because through each bit oil flows through two valves on its path between entering and exiting the transformer. Because we want the work done by the hydraulic system doing work on the load rather than internally through valves, efficient valves with low pressure drop are needed.

In addition to the overflowing and efficiency issues, the conventional spool valves used in DHTM475 exhibit an overall amount of leakage that is large enough to be of concern. Although not measured directly, overall energy expenditure due to leakage was observed to be significant. A valve with reliable, consistent response, both on and off, with low restriction and that exhibits low leakage, is clearly necessary for continued development.

Over several years, numerous distributors, applications engineers, company technical representatives, engineers and leaders from the many names in hydraulic valves from the Americas, Europe and Asia were tasked by the author with providing bidirectional 5, 10, 20 and 40 gpm 2/2 valves that can close reliably at or near critical flow.

Many valves were recommended and reviewed and several were considered, including internally piloted logic valves with high flows, but with response that is highly flow and pressure dependent. Also considered were 4/2 spool valves with combined flow and opposing flow forces, but leakage would be even worse than already with 2/2 spool valves, response would be slower and there would be extra plumbing involved. Several piloted poppet valves were considered, but pilot sections were found to be intentionally throttled for slow opening and closing, which is opposite of what is required by DHT. Servo valves were ruled out due to cost and complexity. There were no satisfactory candidates found.

1.3. Solution

After searching but finding no satisfactory results, a decision was made in August 2024 to attempt the design and manufacture of a new valve, from first principles, specifically for the DHT. The approach chosen was that of brute force: hydraulically force the poppet closed against flow forces with as large of pilot area as necessary. Ideally, the valve will close reliably in flows all the way up to and including critical flow at burst pressure, but that is a lot to ask.

There are also downsides to increasing pilot area such as slower response and increased size that must be weighed. This is in addition to the already slower response time and overall increase in size owing to the jump from direct acting to pilot actuated valves. On the positive, because closing function is achieved hydraulically there is no need for a return spring so it may be eliminated. A poppet concept was chosen over spool due to the possibility of double blocking, minimized leakage and simplified manufacturability.

The overall concept includes a double balanced poppet valve that is continuously piloted either on or off, with return pilot piston area increasable to meet arbitrary pilot force requirement independent of poppet size or shape. No return spring is necessary because the closing function is achieved hydraulically.

Of the four valve sizes required, the smallest valve was selected to be attempted first due to it being the most difficult to design and manufacture, as well as requiring the minimum of test rig flow.

The requirements on the new valve concept are summarized as follows:

- Ability to close regardless of flow or Δp
- Poppet with 45° conical seat seal – No leakage
- Double blocking – Hydrostatically balanced at each port
- Double piloted – No spring
- Nominal flow rate: 5 gpm @ ≤20 psid
- 0.375”Ø flow path (for ≤20’/s at rated flow)
- 800 psi pilot pressure
- Working pressure: 5,000 psi with 4:1 safety factor

In addition to the above requirements, demands on the new valve concept as follows:

- Switching time of ≤50 ms
- 5 Hz at 20,000 psi fatigue-proof
- Compact
- Scalable
- Higher efficiency
- Increased reliability
- Decreased electrical demand
- Low cost
- Easy to manufacture

2. JANUS VALVE

Because a valve capable of meeting the above requirements and demands must have a fitting name, this valve was given the name Janus, for the Roman god of gates, transitions, beginnings and endings. The 5 gpm Janus valve, designated JV05, shown in section in Figure 4, is conceived, manufactured, tested and evaluated as potential DHT Bit-1 valve.

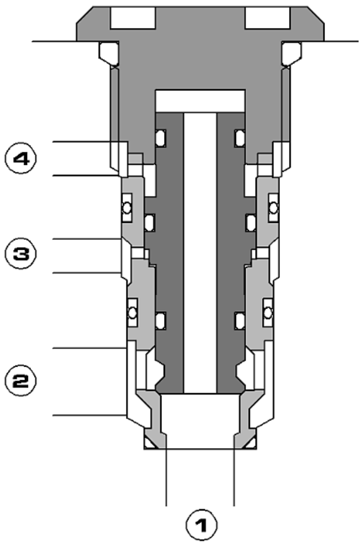

Figure 4. JV05 valve mid-line section

2.1. Design

Design of the Janus valve is an exercise in pragmatism that began with a simple cylindrical poppet having a small 45° bevel around one end. This bevel is lapped with a conical seat in the valve body and is self centering for minimized port to port leakage. The end opposite of bevel is the balance piston, which cooperates with the balance cylinder integrated into the cap. A hole for pressure equalization runs through the poppet axially from the face of one end to the face of the other end so that poppet is hydrostatically balanced. This means pressure levels and fluctuations at nose port 1 have little influence on poppet behaviour. The diameter of the balance hole may be increased for minimized fluid restriction and poppet mass.

Pressure at side port 2 acts radially on poppet and is therefore self-cancelling and balanced such that pressure and flow fluctuations at side port 2 have little effect on poppet behaviour.

In order to shift the poppet axially, on and off, a pilot piston is integrated around the poppet and chambers cooperating with pilot piston for actuation in each direction are integrated into the body. Pilot port 3 is pressurized to open the valve. Pilot port 4 is pressurized to close the valve. The area of the pilot piston, for a given pilot pressure, must be sufficient as to produce enough closing force to overcome the highest flow force the poppet can theoretically ever see. Practically, this is at critical flow at maximum input pressure.

Fluid paths in pilot section are unrestricted for fast response. Free acceleration of poppet from pilot pressure can, even over a short stroke, produce poppet velocity high enough to cause damage to poppet and/or seat at closing. To avoid damage to poppet and seat at closing, a cushion is integrated into the pilot-on chamber to decrease poppet velocity just before metal to metal impact.

This cushion traps a large portion of swept area of the pilot-on piston in a cushion chamber from which the path out is through a progressively thinner annular orifice created between poppet cushion piston and the body cushion cylinder. There is no check valve.

Instead of a check valve, the cushion piston is wide with a shallow, short stroke such that it acts a short distance with high pressure. At the beginning of valve opening, because initial flow into pilot-on chamber, rising from zero, is low, the annular orifice restriction is of little influence. Additionally, the orifice becomes less restrictive as the poppet moves away from the seat, which allows progressively greater flow over the few thousandths of an inch the cushion acts on.

Because of the short stroke, the poppet can move just a small distance to be out of range of cushion. This arrangement allows the cushion to sufficiently brake the poppet at closing yet not significantly impede immediate response upon opening. In addition, the check-valve-less cushion design allow a cleaner, more compact, less complex overall unit compared to a design with check valve.

Another cushion is designed into the balance chamber for decelerating poppet at the end of opening that is even simpler and shorter stroke than the closing cushion. Rudimentary but effective, two broad, flat faces coming into contact in oil squeeze oil out from between them at progressively higher pressure, which acts on and quickly brings to rest the poppet.

The Janus valve and its cavity were designed as one. The cavity is designed to be machined with standard tooling and require no form tool. One particular departure from standard valve cavities is its flat bottom at a specific depth.

The seal around the nose of the valve, separating port 1 from port 2, is of corner crush design. An o-ring is placed in the flat bottom of the cavity before valve is inserted. Assembly compresses the o-ring in the corner between valve nose and the flat bottom of the valve cavity. Axial dimensions of the sleeve and cap are held closely so that the nose just touches bottom at full torque. This corner seal is chosen over the traditional o-ring with back up rings piston seal due to the elimination of the requirement to fasten the valve's body parts. With no threads, pins, wires or other method of fastening the two body parts, the overall size and mass of the valve, as well as manufacturing effort, is minimized. The parts are simply held in compression by the o-ring. In addition to minimized complexity, the corner compression seal allows a shorter overall installed length.

All three valve components are designed to be chucked in the lathe just once to machine all their critical features, which allows closer concentricity and greater precision due to everything turning about the exact same axis. Only non-critical features are machined in secondary operations.

2.2. *Pilot valve*

The SV08-40 4-way, 2-position solenoid operated spool valve from Hydraforce was selected to pilot the JV05 prototype. It handles a maximum flow of 3 gpm at pressures up to 3,000 psi. With this SV08-40/JV05 arrangement it is possible configure the valve to be normally closed or normally open by switching the pilot line connections.

2.3. *Prototypes*

A prototype JV05 valve, Figure 5, was manufactured on a manual lathe with standard tooling. Each of the three pieces was chucked just once to machine all of their critical features. Of note is that after the body is roughed out, a drill on the tool post comes in radially to drill cross holes, which allows the inside to then be finished, eliminating any burrs left from drilling, while never un-chucking the part.

A body to accommodate both the JV05 and SV08-40 as one unit, shown in Figure 6, was manufactured on a manual Bridgeport mill with standard tooling. This is the combination used for testing and characterization.

Figure 5. JV05

Figure 6. Valve body

2.4. *Flow*

Effort was put into ensuring valve flows freely. As a starting point, both ports are sized to match nominal size fluid path in block. In this 5 gpm instance the diameter is .375”, which keeps fluid velocity just under 20 feet per second at rated flow. This flow area is maintained or exceeded throughout the entire flow path.

Flow through piloted open valve was simulated in SolidWorks, with assistance from Dr Donald Mueller and Dr Zhuming Bi in the Civil and Mechanical Engineering Dept. of Purdue University Fort Wayne, to verify flow forces and estimate critical flow. A flow trajectory simulation of critical flow from port 2 to port 1, at 5,000 psi pressure difference, Figure 7, shows no critical point at which pressure suddenly changes. Pressure drop is fairly consistent over the entire fluid path through valve.

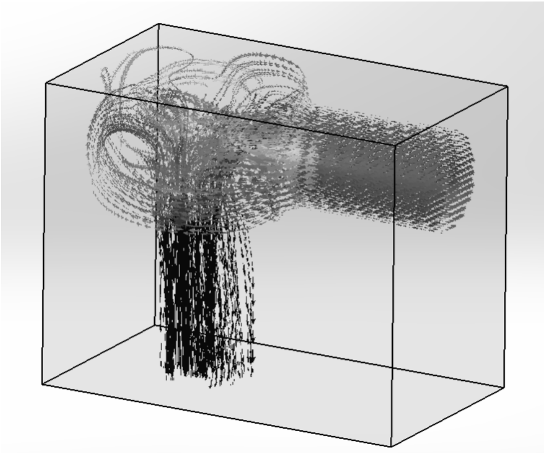

Figure 7. Flow Simulation

2.5. p-Q Curve

To relate flow through valve and port to port pressure differential, JV05 was plumbed to a hydraulic test stand equipped with power supply, reducing valve, variable area flow meter, gauge port test cells, pressure gages and differential pressure gages. Flow meter and differential pressure gages shown in Figure 8. A needle valve plumbed between output port and tank served as variable load. By varying needle valve, flow through valve was adjusted and readings of flow and pressure differential were recorded. JV05 p-Q curve, flow 1 to 2, with ISO 32 oil around 65°F, is given in Figure 9.

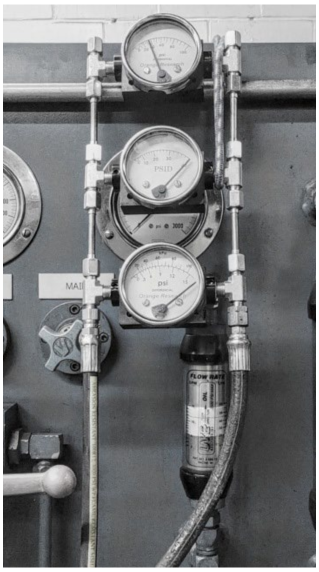

Figure 8. Test rig

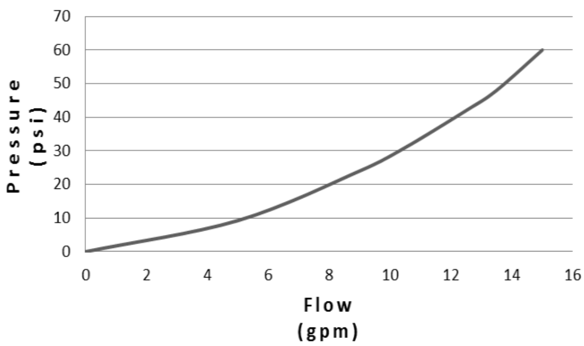

Figure 9. Pressure differential vs. flow, flow 1 to 2

With measured pressure differential of 9 psi at 5 gpm, the design goal of ≤20 psi at 5 gpm was clearly surpassed. 15 gpm is 3x nominal flow.

2.6. C_v

The valve coefficient, C_v, was measured by applying exactly 1 psi water pressure at the inlet port while the valve was piloted open with outlet port open to atmospheric pressure.

Water supply was adjusted to achieve exactly 1 psi, read from a new, quality 0 - 3 psi gauge plumbed at valve inlet port. This setup is shown in Figure 10. Once flow and pressure were observed to be correct and steady state, output flow was directed into a level, graduated container and a stopwatch was simultaneously started. At precisely 60 seconds, flow was directed away from the bucket. Amount of water in the container was read and recorded.

Figure 10. Cumulative flow and pressure measurement apparatus

This experiment was carried out twice in each direction of flow. With volume resolution of the container only down to the quart, volumes of water collected were interpolated using digital calipers measuring between surface of water and the closest graduations.

The average amounts of water that flowed through the valve in one minute are recorded as follows:

$$C_v^{1\text{-}2}=2.35 \text{ gpm}$$

$$C_v^{2\text{-}1}=2.20 \text{ gpm}$$

2.7. Critical Flow

Critical flow through a hydraulic valve is the maximum flow rate achievable for a given upstream pressure, which occurs when the downstream pressure drops to roughly 50 - 55% or less of the upstream pressure. In this state, reducing the downstream pressure further will not increase the flow rate. This critical flow in hydraulic valve is due to cavitation or flashing.

In order to fully understand critical flow one must understand the pressure distribution throughout the valve. Typically, flow through hydraulic valves must get past a main restriction, directly after which is a point of maximum fluid velocity and minimum pressure called the vena contracta, and in these cases standard equations exist to relate pressure drop, pressure recovery, specific gravity, C_v, vapour pressure, etc. However, through the Janus valve there is no such point of restriction, which makes prediction of critical flow difficult.

An attempt at achieving and measuring critical flow through the Janus valve was made using the stopwatch, graduated barrel and pressure gages at hand. Operating on combined flow from full accumulator and 18 hp prime mover input, with input pressure held constant by reducing valve, it was possible to make 20 second runs without running out of adequately pressurized fluid. With input pressure of 1,000 psi, JV05 was found to be flowing around 52 gpm at around 600 psid. Past this point further reduction in outlet pressure did not visibly or audibly increase flow, so there is some belief this was in or at critical flow. However, it was not possible to verify this numerically in real-time with the equipment being used. Due to the difficulty of returning the fluid back to tank and the mess it made the experiment was performed only once. Better flow meters, as well as a higher flow test bench, are needed to verify and characterize the critical flow behaviour. Because of this uncertainty, it cannot be claimed that critical flow was achieved or observed with publication-level confidence.

It should be noted that even at this extreme flow point the valve opened and closed reliably the eight times it was cycled. This would have been explored further but the jumping, jarring and flexing of the test rig from pressure spikes at switching presented danger too great to ignore. At minimum, accumulators must be fitted to each port and the low pressure return line replaced with larger one of higher pressure rating before any additional exploration of critical switching is attempted.

2.8. Response

In order to understand dynamic aspects the valve was connected to test rig with input pressure held constant by a reducing valve and flow controlled by a needle valve. A pressure sensor was fitted between valve output and needle valve. Plumbing volume between valve, sensor and needle valve was kept to a minimum to minimize compression effects and wave propagation issues. A pressure sensor was also applied to each of the pilot lines between pilot valve and main stage. All three pressure sensors are Omega PX613 thin film transducers with a published 5 ms output delay.

A Dataq DI-1100 4-channel data recorder and Windaq software was used for data acquisition. The following images, Figures 11 - 14, show traces of command signal, pilot-on chamber pressure, pilot-off chamber pressure and main stage output pressure over time. Input pressure was held at 1,000 psi, flow was adjusted to 5 gpm and pilot pressure was set at 800 psi. Each channel recorded 1,000 samples per second. Time between divisions is 2 ms. When interpreting the traces it is important to remember that while the command signal trace in channel 1 gets its input directly from pilot valve solenoid wiring, and is thus an immediate representation of state, the three pressure traces are delayed by 5 ms due to sensor limitations.

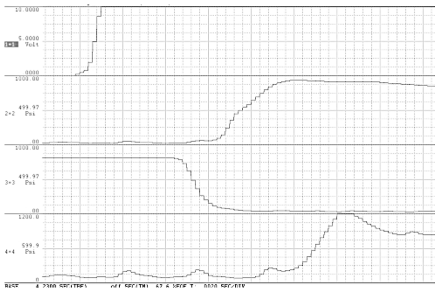

Figure 11. Off-to-On, Flow 1-2

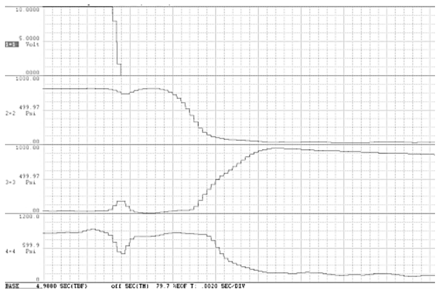

Figure 12. On-to-Off, Flow 1-2

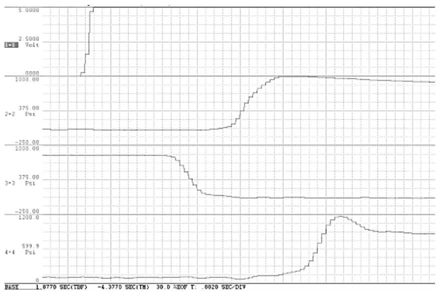

Figure 13. Off-to-On, Flow 2-1

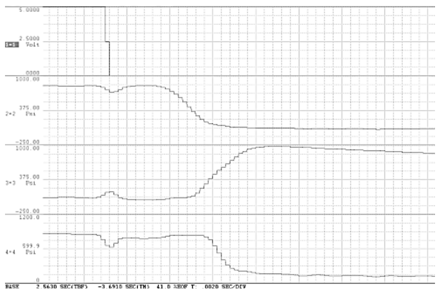

Figure 14. On-to-Off, Flow 2-1

With flow 1 to 2, transitioning Off-to-On, Fig. 11, the first indication of change after the command signal goes high is the release of pressure holding the poppet closed in the pilot-off chamber, which starts at around 14 ms. This pressure falls to its minimum at around 24 ms. Within this timeframe the pilot-on chamber pressure begins to increase starting at around 22 ms and reaches its maximum at around 36 ms. The changes in pilot chamber pressures act on the pilot piston areas causing the poppet to open. While somewhat difficult to disambiguate from the noise in channel 4 data, the first indication of main stage opening is at around 40 ms. Center crossing occurs at around 47 ms.

When the pilot valve is de-energized, Fig. 12, the flyback voltage momentarily affected pilot and output pressure readings so those fluctuations should be ignored. Pilot-on pressure begins to fall at 5 ms and reaches its minimum at around 20 ms. Within that timeframe pilot-off chamber pressure starts to increase at around 13 ms and reaches its maximum at around 28 ms. Changing pilot pressures force the poppet closed. Output pressure begins to decrease at around 15 ms and center crossing occurs at roughly 20 ms.

With flow 2 to 1, transitioning Off-to-On, Fig. 13, performance is similar to opening with flow 1 to 2, but with a center crossing at around 48 ms.

Upon closing, Fig. 14, performance is again similar to that of flow 1 to 2, but with a center crossing at around 22 ms.

With opening response of 47 or 48 ms, and closing response of just 20 or 22 ms, the response of this JV05/SV08-40 valve combination meets the goal of 50 ms.

But while a fast response is good, more important is consistency, reliability and a fast transition. The actual response time is not all that critical because, as EOS is initiated in control well before metal to metal impact in the bridge, the timing of each valve can be determined in advance of execution, and signals to valves with the longest delay are sent first, valves with shortest delay, last, such that the crossings can be timed to millisecond precision regardless of a magnitude of valve response delay.

2.9. Leakage

When the prototype JV05 valve was first assembled and an attempt was made to assess its port-to-port leakage, not even one drop of leakage was observed, with 3,000 psi at either port, over any time period. Of course, this was with a new precision lapped poppet seat with no damage.

After hours of operation, testing and pushing physical limitations, valve port-to-port leakage was measured again by pressurizing one port of the piloted off valve and leaving the other port open to atmospheric pressure and directed into a beaker. With input pressure at port 1 held constant at 3,000 psi, 38 drops were observed to leak from port 2 over a 10 minute period. With pressure applied at port 2, 34 drops in ten minutes leaked from port 1. Expressed with a single digit of precision over a minute, this equates to:

$$Q_{leak}^{1\text{-}2} = 4 \text{ drops/min}$$

$$Q_{leak}^{2\text{-}1} = 3 \text{ drops/min}$$

While this is more leakage than desired it is still less leakage than through the comparable direct acting spool valves the DHT has used up to now. Cushion design will continue to improve to better protect poppet and seat from damage at closing, and it should be possible to decrease leakage with improved geometry, material specification, machining, lapping and fluid cleanliness.

2.10. DHT Model

To understand how the Janus valve can impact DHT performance, an Excel-based, semi-empirical DHT model, originally reported in the Fifty-third National Conference on Fluid Power (NCFP'14) paper[6)], was updated with actual JV05 p-Q and leakage data. The original model was tuned to third party testing data of the DHTM475 and the only things changed in the model are valve p-Q relationships and leakage rates. Everything about the bridge and tubes remained unchanged.

Modeled schematic of digital hydraulic transformer with double piloted, double blocking poppet valves and 4/2 solenoid pilot valves is shown in Figure 15.

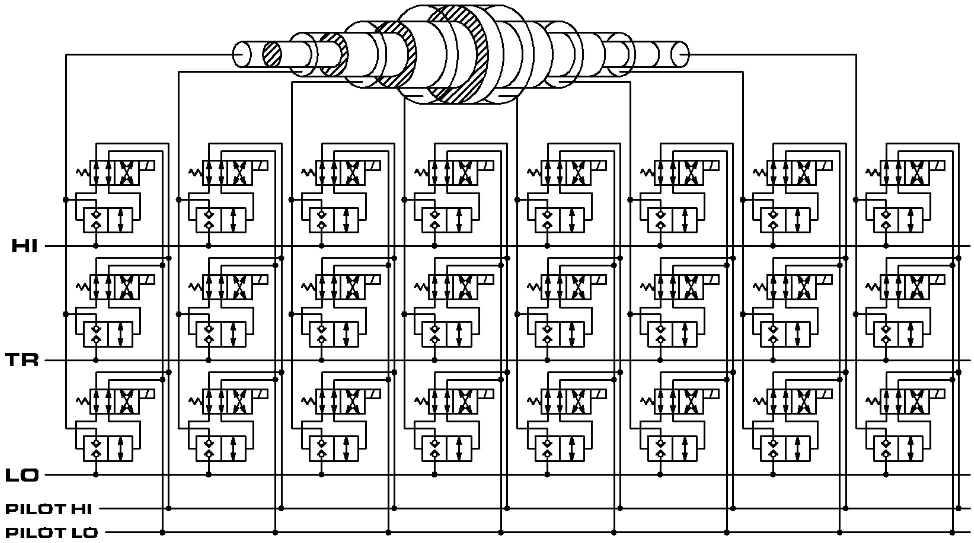

Figure 15. DHT schematic with Janus valves

2.10.1. Mechanical Efficiency

Mechanical efficiency at a given flow and pressure is mainly dependent upon the pressure drops across valves and sliding friction of the transtatic bridge. Beginning assumptions were made:

- p-Q curves, 1 to 2 and 2 to 1, are identical.
- p-Q curves of 10 gpm, 20 gpm and 40 gpm valves are identical to JV05 when scaled to nominal flow.
- Accumulator pressure is fixed at 3000 psi.
- Static friction is not considered in this model.
- Seal friction is estimated using a two-dimensional, second order, lumped parameter model, with input parameters Δp and TSB velocity.
- Coefficients are tuned manually to match experimental data.

The mechanical efficiency map of the DHT with Janus valves is given in Figure 16. The main horizontal axis scale is flow rate from zero to rated flow (75 gpm), the depth axis scale is output pressure from zero to maximum (accumulator pressure) and the vertical axis scale is percentage efficiency from zero to 100%.

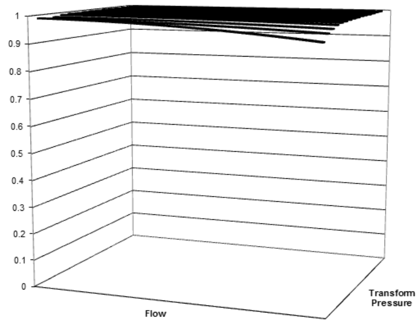

Figure 16. Mechanical Efficiency of DHT

As can be seen in Fig. 16, mechanical efficiency is well above 90% throughout the entire operating range. This is marked improvement over performance with previous valves. Only at lower transform ratios and higher flows does the mechanical efficiency decrease toward 90%. Worst case corresponds to maximum flow at zero pressure. It should be noted, however, that at and around this extreme operating point there is little power transmitted, so lower efficiency is of little consequence.

2.10.2. Volumetric Efficiency

Volumetric efficiency at a given flow and pressure is mainly dependent upon valve leakage and leakage past seals in the transtatic bridge. Beginning assumptions were made:

- Seal leakage is assessed using a two-dimensional, second order lumped parameter model.
- The input parameters are Δp and velocity.
- Coefficients are tuned manually to match manufacturer data.
- Pilot valve leakage is not considered.

The volumetric efficiency map of the DHT with Janus valves is given in Fig. 17. The main horizontal axis scale is flow rate from zero to rated flow (75 gpm), the depth axis scale is output pressure from zero to maximum (accumulator pressure) and the vertical axis scale is percentage efficiency from zero to 100%.

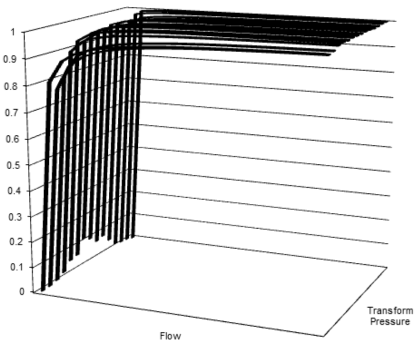

Figure 17. Volumetric Efficiency of DHT

As can be seen in Fig. 17 volumetric efficiency of the DHT is above 90% for the majority of the operating range and over 95% for at least half of it. This is improvement over previous design. Only at low flow conditions does the volumetric efficiency decrease, with worst case at zero flow. It should be noted that at and around zero flow there is very little power transmitted so lower efficiency is of little overall consequence.

2.10.3. Overall Efficiency

Multiplying mechanical efficiency by volumetric efficiency yields overall efficiency. This overall efficiency is also understood as a ratio of power out divided by power in. The modelled overall efficiency map of the DHT with Janus valves is given in Fig.18. The main horizontal axis scale is flow rate from zero to rated flow (75 gpm), the depth axis scale is output pressure from zero to maximum (accumulator pressure) and the vertical axis scale is percentage efficiency from zero to 100%.

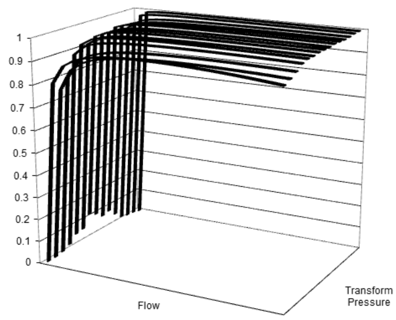

Figure 18. Overall Efficiency of DHT

As can be seen in Fig. 18 overall efficiency for the majority of the output range is above 90%, and over half the output range is above 95%. This is significant improvement over previous design with spool valves. At lower transform ratios and both higher and lower flow conditions the overall efficiency decreases slightly towards 90%, and, as is the case with volumetric efficiency, overall efficiency falls to zero at zero flow. It should again be noted that little power is transmitted through the system at and around these extreme points, so lower efficiency is of less consequence to the system overall. This is especially true at low flows.

2.11. Impact on EOS and DHT

The Janus valve's ability to turn off at high flow means there is potential for it to reliably support direct rail to rail flow at EOS in the DHT application. So long as valves continue to respond to commands in a consistent way, development of this EOS approach can continue.

In addition to its functionality, the Janus valve's low restriction means lower mechanical losses and higher overall efficiency in the DHT.

Beyond this, because every detail of the valve is aimed at solving a problem for just one application, and because all of the design decision-making and manufacturing are collocated and in parallel with DHT development, potential exists that the Janus valve truly meets a need that no other valve can.

3. CONCLUSION

The JV05 double piloted, balanced, 2-way, 2-position poppet valve, designed specifically for the 4-bit, linear-acting Digital Hydraulic Transformer, is conceived, manufactured, tested and evaluated as potential DHT Bit-1 valve. The valve is designed around a simple cylindrical poppet that is actuated in each direction by separate pilot pressures. Because the closing function is achieved hydraulically there is no need for a return spring so it is omitted. An integrated cushion slows the poppet just before impact with seat to eliminate damage to poppet and seat.

One departure from conventional cartridge valve form is a corner nose seal that holds parts in compression while assembled in their cavity. Another departure is that the flow area through the valve is greater than through lines, which is opposite most valves.

All of the requirements are met with a 3-piece design, and each component is designed for ease of manufacture.

Apparatus was set up to establish p-Q curve, which included achieving a differential pressure of just 9 psi at 5 gpm flow, measuring a favorable C_v and measuring a small amount of leakage. As well, apparatus was set up to explore critical flow and response at critical flow but found to be inadequate for the flows and pressure spikes involved.

Opening response delay was found to be around 48 ms, while closing response was found to be around 22 ms. This overall response of the JV05/SV08-40 valve combination satisfies the goal of 50 ms.

Valve p-Q curve and leakage data were applied to existing DHT model and impact on performance was examined, including improved efficiencies and End Of Stroke strategies over previous DHT iterations with conventional spool valves. The model indicates possibility of overall efficiency over 90% for the majority of the operating range. However, the most critical aspect is the valve's ability to shift reliably at high flows because it is required for successful rail to rail flow at EOS.

There is also potential for the Janus valve to find application outside the DHT, as a general purpose, configurable, bi-directional, two-way, two-position directional control valve, operating in either a normally open or normally closed state, to start, stop or isolate fluid flow in industrial and mobile hydraulic systems. While there are many 2/2 valves on the market already with excellent performance and low cost, how many can close reliably at critical flow? Especially where mode of failure is catastrophic and/or a mission is critical, the Janus valve aims to perform as commanded regardless any condition thrown at it.

While the results of testing so far are promising, there is a large amount of work still needed to bring the JV05 forward in development in areas such as:

- geometry optimization
- material specification
- fatigue testing

- proof/burst testing
- critical performance characterization
- development of 10 gpm JV10

The Janus valve, a bidirectional, double piloted, balanced poppet valve, with its conical seat and integrated cushion, appears to be a decent option to pursue as part of DHT development going forward. The valve's ability to close at high flow sets it apart from others. Because the Janus valve is so far proving it can close even at high flows, it appears development of DHT EOS functionality may continue.

In addition to the benefit of being able to close, with pressure drop just a fraction that of equivalent nominal flow valves, it appears the Janus valve will allow the DHT higher mechanical efficiency throughout the operating range than was possible previously. And with leakage just a fraction that of previous spool valves, the Janus valve should also allow the DHT much higher volumetric efficiency throughout the operating range. With both higher mechanical efficiency and higher volumetric efficiency the overall efficiency can only be higher as well. These are results of a semi-empirical DHT model using measured JV05 data.

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