DESIGN OF A HIGH-SPEED ON-OFF VALVE

Allan A. Katz
Mechanical Energy and Power Systems Lab
Worcester Polytechnic Institute
Worcester, MA, USA

James D. Van de Ven†
Mechanical Energy and Power Systems Lab
Worcester Polytechnic Institute
Worcester, MA, USA

ABSTRACT

On-off control of hydraulic circuits enables significant improvements in efficiency compared with throttling valve control. A key enabling technology to on-off control is an efficient high-speed on-off valve. This paper documents the design of an on-off hydraulic valve that minimizes input power requirements and increases operating frequency over existing technology by utilizing a continuously rotating valve design. This is accomplished through use of spinning port discs, which divides the flow into pulses, with the relative phase between these discs determining the pulse duration. A mathematical model for determining system efficiency is developed with a focus on the throttling, leakage, compressibility, and viscous friction power losses of the valve. Parameters affecting these losses were optimized to produce the most efficient design under the chosen disc-style architecture. The experimental valve matched predicted output pressure and flows well, but suffered from larger than expected torque requirements and leakage. In addition, due to motor limitations, the valve was only able to achieve a 64Hz switching frequency versus the designed 100Hz frequency. Future research will focus on improving the prototype valve and improving the analytical model based on the experimental results.

1 INTRODUCTION

The power of hydraulic pumps, motors, and actuators are traditionally controlled by one of two methods. The first method throttles the fluid through a valve until the desired output pressure is reached; however, this throttling converts excess power into heat and is very inefficient. The second method uses a variable displacement pump or motor to achieve the desired output flow rate, but these units are bulky, expensive, and relatively complicated.

An alternative to conventional methods is switch-mode control, Figure 1, where a high-speed valve is used to rapidly switch the system between efficient on and off states, creating virtually variable displacement functionality. By varying the ratio of the on-time to the total cycle time, defined as the duty ratio, a variable output pressure is produced. On the source side of the circuit, an accumulator minimizes the pressure pulses.

A review of the current technologies shows a wide array of methodologies for creating a high speed on/off valve. Some methods use a solenoid [1-5] or piezoelectric actuator [6; 7] to oscillate a poppet or spool valve. One method uses solenoids to actuate a series of 3-way check valves [8]. Another attractive architecture uses a rotary spool [9-11] to avoid the power loss of accelerating a control mass in the oscillatory designs. Other rotary designs use the phase angle [12; 13] between components to generate a variable duty ratio. This architecture
is the most closely related to the proposed high-speed switching valve.

**2 PHASE CONTROLLED VALVE CONCEPT**

The novel phase-shift valve, the topic of this paper, is shown schematic in Figure 2. The valve consists of three sub-valves labeled Section 1A, 1B, and 2, which all rotate at the same constant angular velocity. Each sub-valve is composed of a half cylinder spool inside a sleeve with two radial inlet ports and one axial outlet port. The outlet port of each sub-valve is connected to each inlet port for half of a revolution. As seen in Figure 2a) the only open flow path is from Port P (supply pressure) to inlet port 1a, to the outlet port of section 1A, then to inlet port 2a and finally exiting section 2 to Port A (to a hydraulic actuator). Figure 2b) shows the same system after a π/2 rotation. The previous flow path is now blocked at inlet port 2a, but the flow path from Port T (tank) to Port A is now open. By varying the relative phase between the Tier 1 sub-valves with respect to the Tier 2 sub-valve, the duty ratio, can be continuously varied from 0 to 1.

The flow diversion of the valve during a single cycle is shown in Figure 3. Note that Section 1A and 1B remain synchronized at π phase with respect to each other, while section 2 can vary from 0 to π phase with respect to section 1. This creates a continuously variable duty ratio from 0 to 1, where zero is defined as full flow from/to tank, Port T, and a duty ratio of 1 is defined as full flow from/to supply pressure, Port P. A negative phase shift or a phase shift beyond π will also result in a duty ratio between 0 and 1.

A beneficial characteristic of the phase-shift architecture is that 2 pulsed segments are generated for every rotation of the valve. This creates a switching frequency that is 2X the operating frequency of the valve, an advantage compared to other continuously rotating valve methods.

Taking the kinematic inversion of the conceptual valve shown in Figure 2 and redirecting the flow to move axially, a disc style architecture is created. In Figure 4, the three discs are stacked on top of each other. Note that section 1A and 1B have been combined into Tier 1. In this setup, varying the duty ratio is achieved by changing the phase of Tier 1 relative to Tier 2, which is fixed. A continuously rotating valve plate is used to produce the switching cycles.

![Figure 2: Schematic of the phase shift valve for two different positions.](image)

![Figure 3: Plot of the flow diversion in the valve for a given phase shift. Shaded regions denote when Section 2 is receiving flow from Section 1. Labeling is consistent with Figure 2.](image)

![Figure 4: Disc architecture with Section 1 sub-valves combined into Tier 1. Tier 1 can change phase relative to Tier 2, which is fixed, to change the duty ratio. A new component, the continuously rotating valve plate generates the switching pulses.](image)

The disc style architecture offers several advantages over the radial flow spool style architecture.

1) The pressure forces on rotating components in the valve can be easily balanced.
2) Manufacturing tolerances can be looser, by using axial thrust bearings to maintain valve clearance instead of spool to sleeve manufacturing tolerances.
3) In this configuration, the phase only needs to be maintained between two stationary components.
4) The integration of components minimizes the switched volume, decreasing compressibility losses.
5) Flow travels axially through the valve, so there will not be a centrifugal pumping effect.

3 POWER LOSS MODELING AND ANALYSIS

To better understand the pressure drops and flow rates of the high-speed phase-shift valve, a mathematical model is developed. The energy losses of the valve are of primary interest and include the throttling losses of the fully open and transitioning phases, the internal leakage losses of the valve, the compressibility losses due to compliance in the fluid, and the viscous friction losses between rotating components. Losses that will not be analyzed include hysteresis losses in the accumulator, inefficiencies in the pump/motor, compressibility losses due to compliance in the valve and pump/motor structure, and viscous pipe flow losses. Once the equations for energy loss are developed, the operating parameters are optimized for the highest efficiency.

Referring to Figure 4, it can be seen that for the instant shown, the flow path from Tier 1 to Tier 2 is momentarily blocked. This occurs whenever the valve transitions from one state to the next, which is twice per cycle. A typical application for the valve would be the control of a fixed displacement pump/motor on a hydraulic hybrid vehicle. As the valve transitions, fluid flow will be blocked. The motor will continue to rotate and draw a constant flow, causing the motor inlet to vacuum and cavitate. If the hydraulic unit were acting as a pump, then this momentary blockage would create a large pressure spike at the outlet of the pump. To alleviate these issues, two check valves are placed in the hydraulic circuit shown in Figure 5. The right check valve prevents cavitation during motoring, and the top check valve prevents pressure spikes during pumping.

Figure 5: Simplified high-speed valve circuit used for analysis purposes. The two check valves have been added to avoid extreme pressure fluctuations occurring when flow is completely blocked during valve transitions.

The key geometry features of the valve are shown in Figure 6. First, we can define the number of replications of ports on the valve components by the variable N (Figure 6 shows N = 1). The ports of the valve are defined by the inner radius R_i and the outer radius R_o. The ports on the valve plate span an angle of δ while the larger ports of Tier 1 and 2 span an angle of γ. Note that δ + γ = π/N, thus the valve will be completely blocked twice each switching cycle. Keeping Tier 1 fixed, the angular position of the valve plate θ is defined as zero degrees when the valve ports are completely blocked. Looking axially at the valve, the phase angle α is referenced as zero degrees when Port A is aligned with Port T. The phase can vary from 0 to π/N for a 0 to 1 duty ratio, respectively. A_1 is the variable orifice created by the overlap of Tier 1 with the valve plate, while A_2 is the variable orifice created by the overlap of Tier 2 with the valve plate.

Figure 6: Schematic of disc style valve with key features defined

3.1 VALVE THROTTLING LOSSES

Despite the fact that a switch-mode valve was chosen to avoid the inefficient throttling loss of common valves, a significant source of power loss is from throttling within the high-speed valve. The dominant throttling loss is incurred when the valve transitions from one state to another. Since there are two full on-off periods for each switching cycle of the valve, there are 4*N switches per revolution. For each of these switches, the area of one of the internal valve ports changes from fully open to fully closed or vice versa, creating throttling across a variable area orifice. At low or high duty ratios, throttling across two simultaneously varying area orifices occurs.

Before calculating the energy loss due to throttling, expressions for the internal port areas of the valve must be developed. Referring back to Figure 6, the first variable orifice area A_1 is defined as the port area created by the overlap of Tier 1 and the valve plate. Defining the valve plate angle θ as zero degrees when the valve plate ports are fully blocked and about to transition to Port T, A_1 is given by:

\[ A_1(\theta) = \frac{\theta \cdot N}{2} \left( R_i^2 - R_o^2 \right) \text{ for } 0 \leq \theta \mod \frac{\pi}{N} < \delta \] (1)

\[ A_2(\theta) = \frac{\delta \cdot N}{2} \left( R_i^2 - R_o^2 \right) \text{ for } \delta \leq \theta \mod \frac{\pi}{N} < \gamma \] (2)

\[ A_3(\theta) = \frac{\left( \gamma \cdot N - \theta \right) \cdot N}{2} \left( R_i^2 - R_o^2 \right) \text{ for } \gamma \leq \theta \mod \frac{\pi}{N} < \frac{\pi}{N} \] (3)
where $\theta$ modulo $\pi/N$ maintains the evaluated angle between 0 and $\pi/N$ for multiple rotations. The symmetry of the valve allows the use of $\pi/N$ of 2$\pi/N$. Similarly, the second variable orifice area $A_2$ is defined as the port area created by the overlap of the valve plate and Tier 2. Creating a new variable $\theta^* = 0 - \alpha$, $A_2$ is given by Eqs. (1-3) with $\theta$ replaced by $\theta^*$.

Note that the valve plate has 2*N ports, but only N ports have flow. Also, besides the instantaneous moment when the valve is completely blocked, N ports on the valve plate will always have flow. The internal port areas as a function of rotation angle for a phase shift of 30º are shown in Figure 7.

The addition of check valves to the system means that for a portion of time during each transition, flow will be split between the valve and check valve pathways. For instance, referencing Figure 5, when the high-speed valve is in motoring mode and switching from Port P to Port T, the internal variable orifices begin to close, causing a large pressure drop. Eventually the pressure at the output of the valve reaches the tank pressure plus the cracking pressure of the tank side check valve, which causes it to open. Flow through the check valve increases until all flow passes through the check valve when the high-speed valve passageways become completely blocked. As the valve plate continues to rotate, the internal variable orifices begin to open to Port T. The check valve will hold the output pressure steady as it begins to close. Eventually, the output pressure of the valve reaches the cracking pressure of the check valve and it closes. Full flow from Port T to the hydraulic motor is now going through the valve. To begin the throttling loss analysis it is necessary to first develop an expression for the full flow pressure drop across the high-speed valve. By assuming the valve as two orifices in series the pressure drop is given by [14]:

$$\Delta P_{valve} = \Delta P_{Tier1} + \Delta P_{Tier2} = \frac{\rho}{2} \left( \frac{Q}{C_d A_1} \right)^2 + \frac{\rho}{2} \left( \frac{Q}{C_d A_2} \right)^2$$

(4)

where $\Delta P_{valve}$, $\Delta P_{Tier1}$, and $\Delta P_{Tier2}$ are the pressure drop due to full flow through the high-speed valve, Tier 1 of the valve, and Tier 2 of the valve respectively, $\rho$ is the mass density of the fluid, $Q$ is the flow rate, $C_d$ is the discharge coefficient of the orifice, and $A_1$ and $A_2$ are the current area of the first and second Tier orifices respectively.

Next, it is necessary to determine when flow will be split between the high-speed valve and the check valves. It is assumed that the pressure drop across the check valve is always $P_{check}$. When the hydraulic unit is in motoring mode, the low pressure check valve will have flow when:

$$\Delta P_{valve} > P_i - P_{tank} + \Delta P_{check}$$

(5)

Where $P_i$ is the input pressure of the valve, either $P_{High}$ or $P_{Tank}$ and $\Delta P_{check}$ is the cracking pressure of the check valve. If this condition is met, then the pressure drop across the valve is held constant by the check valve and is given by:

$$\Delta P_{valve} = P_i - P_{tank} + \Delta P_{check}$$

(6)

When the hydraulic unit is in pumping mode, the high pressure check valve will have flow once:

$$\Delta P_{valve} > P_{High} - P_i + \Delta P_{check}$$

(7)

If this condition is met, then the pressure drop across the valve is held constant by the check valve and is given by:

$$\Delta P_{valve} = P_{High} - P_i + \Delta P_{check}$$

(8)

If these conditions are met, note that the pressure drop across the high-speed valve remains constant while the flow rate through the valve becomes variable. If these conditions are not met, then the pressure drop across the high-speed valve is given by Eq. (4). The calculation of the flow through the high-speed valve is attained by rearranging Eq. (4), which gives:

$$Q_{valve} = C_d A_1 A_2 \sqrt{\frac{2\Delta P_{valve}}{\rho (A_1^2 + A_2^2)}}$$

(9)

By assuming a constant flow through the external hydraulic unit, the flow through the check valve is always described by:

$$Q_{check} = Q - Q_{valve}$$

(10)

The pressure drop across and flow through the high-speed valve when the hydraulic unit is acting as a motor are shown in Figure 8. Parameter values used to generate these plots are given in Table 1. In the figure, note the four large pressure drop spikes corresponding to the four transition events, along with the
decrease in flow through the valve as a result of the check valves opening. Looking at the middle plot, the effect of the check valve can be seen when the valve transitions to Tank pressure around 0.4 radians. The output pressure is held constant until the threat of cavitation is averted.

![Figure 8: Pressure drop across and flow rate through the valve vs. angular position](image)

Table 1: Values used to generate throttling pressure drop and flow plots

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_i$</td>
<td>0.0025m</td>
</tr>
<tr>
<td>$R_o$</td>
<td>0.015m</td>
</tr>
<tr>
<td>$\delta$</td>
<td>22.5°</td>
</tr>
<tr>
<td>$N$</td>
<td>2</td>
</tr>
<tr>
<td>$Q$</td>
<td>$5 \times 10^{-4}$ m$^3$/s</td>
</tr>
<tr>
<td>$P_c$</td>
<td>7.77MPa</td>
</tr>
<tr>
<td>$P_{check}$</td>
<td>0.2MPa</td>
</tr>
<tr>
<td>$P_{tank}$</td>
<td>0.1MPa</td>
</tr>
</tbody>
</table>

Once the pressure drop and flow rate through the valve is determined, the instantaneous power loss due to throttling can be calculated from:

$$P_{throttling} = \Delta P_{valve} Q_{valve} + \Delta P_{check} Q_{check}$$  \hspace{1cm} (11)

3.2 VALVE LEAKAGE LOSSES

Another form of energy loss is from the internal leakage of the valve. Starting from the high-pressure port of Tier 1, the two primary leakage paths are radially outward to the bore, which is held at tank pressure, and circumferentially to the tank pressure ports. These leakage paths exist between two parallel surfaces, so parallel plate leakage is assumed [14].

$$Q_{leak} = \frac{\text{Perimeter} \cdot c^3 \cdot \Delta P}{12 \cdot \mu \cdot L}$$  \hspace{1cm} (12)

where Perimeter is the perimeter of the leakage path given by the average arc length $\gamma \cdot (R_{bore} + R_o) / 2$, $c$ is the clearance between the plates, $\Delta P$ is the pressure differential, $\mu$ is the fluid viscosity, and $L$ is the length of the leakage path given by $(R_{bore} - R_o)$. The radial leakage on the front side of the valve plate, the region between the valve plate and Tier 1, is given by:

$$Q_{\text{leak},f} = \frac{N \cdot \gamma \cdot c^3 \cdot \Delta P \cdot (R_{bore} + R_o)}{24 \cdot \mu \cdot (R_{bore} - R_o)}$$  \hspace{1cm} (13)

where $R_{bore}$ is the radius to the bore of the valve, and $c_i$ is the clearance between the front face of the valve plate and the Tier 1 sub-valve. The rear side of the valve plate, the region between the valve plate and Tier 2, will also experience leakage losses, but this loss is affected by the duty ratio. When the high-pressure ports are blocked, the perimeter of the rear side of the valve plate will be determined by the valve plate port angle $\delta$, but when the ports are unblocked, the perimeter will be determined by the Tier 2 port angle $\gamma$. This gives:

$$Duty = \frac{\alpha \cdot N}{\pi}$$  \hspace{1cm} (14)

$$Q_{\text{leak},b} = \frac{N \cdot c_b^3 \cdot \Delta P \cdot (R_{bore} + R_o)}{24 \cdot \mu \cdot (R_{bore} - R_o)} \cdot \left(\gamma \cdot Duty + \delta \cdot (1 - Duty)\right)$$  \hspace{1cm} (15)

where Duty is the duty ratio and $c_b$ is the clearance between the back face of the valve plate and the Tier 2 sub-valve. Once the leakage flow rate is calculated, the power loss due to radial leakage is simply:

$$P_{Leak,rad} = (Q_{\text{leak},f} + Q_{\text{leak},b}) \cdot \Delta P$$  \hspace{1cm} (16)

The circumferential leakage analysis is complicated by the fact that the rotating valve plate creates variable leakage lengths. Furthermore, at the start of the cycle when the ports are completely blocked, the leakage length $L$ from Equation 15 is zero, predicting infinite flow. To simplify the analysis, orifice flow will be assumed around transition events and parallel plate flow will be assumed once the leakage path length increases. The cycle of leakage modes is shown in Figure 9. Initially, the leakage length is near zero and orifice flow is assumed. As the valve plate advances the leakage path length $L$ becomes sufficiently long for parallel plate flow to be used, but $L$ is dependent upon the valve plate position $\theta$. Once the valve plate port advances beyond the Tier 1 land at an angle of $\delta$, the leakage length remains constant until the start of the next transition event. During the next transition event, the process is reversed.
Substituting Eq. (20) and Eq. (21) into Eq. (19) and integrating yields Eq. (25), the circumferential leakage volume per revolution:

\[
V_{\text{circum,f}} = \frac{4 \cdot N^2}{\omega} \left[ \int_{\alpha \cdot \theta_{\text{var}}^{\text{trans}}}^{\alpha \cdot \theta_{\text{var}}^{\text{trans}} + \alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{orifice}} \, d\theta + \int_{\theta_{\text{var}}^{\text{trans}}}^{\theta + \alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{plate}, \var 1} \, d\theta + \int_{\theta_{\text{var}}^{\text{trans}}}^{\theta + \alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{plate}, \var 2} \, d\theta \right] \tag{23}
\]

where \( Q_{\text{orifice}} \, d\theta \) and \( Q_{\text{plate}, \var 1} \, d\theta \) are the same as Equation 20, but with \( \theta \) replaced by \( \alpha - \theta \) and \( \theta - \alpha \), respectively. For the range \( 0 \leq \alpha \leq \delta \)

\[
V_{\text{circum,f}} = \frac{4 \cdot N^2}{\omega} \left[ \int_{\alpha \cdot \theta_{\text{var}}^{\text{trans}}}^{\alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{orifice}} \, d\theta + \int_{\theta_{\text{var}}^{\text{trans}}}^{\theta + \alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{plate}, \var 1} \, d\theta + \int_{\theta_{\text{var}}^{\text{trans}}}^{\theta + \alpha \cdot \theta_{\text{var}}^{\text{trans}}} Q_{\text{plate}, \var 2} \, d\theta \right] \tag{24}
\]

Where \( Q_{\text{plate}, \var 1} \, d\theta \) and \( Q_{\text{plate}, \var 2} \, d\theta \) are the same as Equation 20, but with \( \theta \) replaced by \( \alpha - \theta \) and \( \theta - \alpha \), respectively and \( Q_{\text{plate}, \var \delta} \) is the same as Equation 21 but with \( \delta \) replaced by \( \gamma \). Due to symmetry, Equation 24 can also be used for the range \( \frac{\pi}{N} - \delta \leq \alpha \leq \frac{\pi}{N} \) with the simple...
conversion $\alpha = \frac{\pi}{N} - \alpha$. Note that Equations 23 and 24 will use the back side clearance $c_b$ instead of $c_f$. Finally, the energy loss from circumferential leakage is:

$$E_{\text{Leak, circum}} = \left(V_{\text{circum, } f} + V_{\text{circum, } b}\right) \cdot \Delta P \quad (25)$$

### 3.3 VALVE COMPRESSIBILITY LOSSES

The next major form of energy loss of the high-speed valve is due to the compressibility of the fluid subjected to a fluctuating pressure. Every time the valve switches from low to high pressure, the fluid is compressed, increasing its density. As the valve switches to the tank port, the energy put into compressing the fluid is lost as it decompresses to tank pressure. The volume of fluid subjected to this fluctuating pressure includes the internal volume of the pump/motor, the internal volume of one path of the directional control valve, the output porting of the high speed valve, the port volume of the valve plate and Tier 2 of the high speed valve, and the volume of any passages leading to the check valves.

The bulk modulus $\beta$ is defined as the pressure increase needed to cause a given relative decrease in volume, $\beta = -\frac{\Delta P}{\Delta V}$. The effective bulk modulus can be described by:

$$1 = \frac{1}{\beta_e} \frac{1}{\beta_{oil}} \frac{R}{P_{high} \cdot \kappa} \quad (26)$$

where $\beta_e$ is the effective bulk modulus, $\beta_{oil}$ is the bulk modulus of air free oil, $R$ is the entrained air content by volume at atmospheric pressure, and $\kappa$ is the ratio of specific heats for air. From the definition of bulk modulus, the change in volume due to every switch from the tank branch to the pressure branch can be described by:

$$\Delta V = \left(P_{\text{high}} - P_{\text{tan} \cdot \kappa}\right) \cdot V_{\text{switch}} \frac{1}{\beta_e} \quad (27)$$

where $\Delta V$ is the change in volume and $V_{\text{switch}}$ is the switched volume. Finally, the energy loss during each switch due to fluid compression is:

$$E_{\text{comp}} = \left(P_{\text{high}} - P_{\text{tan} \cdot \kappa}\right) \cdot \Delta V \quad (28)$$

### 3.4 VALVE VISCOUS LOSSES

The last form of energy loss to be considered is caused by viscous friction, which is the friction caused by the shearing of fluid. Viscous friction occurs between the face of the valve plate and the Tier 1 & Tier 2 port faces. The area on the face of the valve plate can be divided into two sections: the annular region outside the port switching area and the annular region within the port switching area.

Viscous friction forces are developed on the front and rear face of the valve plate. From Newton’s postulate, the frictional torque on the valve plate outside the switching area between the valve plate and Tier 1 is:

$$T_{f, \text{outer}} = F_r = \frac{\mu \cdot A \cdot u}{c_f} \cdot \Delta P$$

where

$$\Delta P = \int_{\theta=0}^{\theta=2\pi} \int_{r=R_b}^{r=R_o} \mu(r \cdot d\theta \cdot d\theta) \cdot (\omega r)$$

$$\omega = \frac{\pi \tau_o u}{2c_f} \left(R_{\text{bore}}^4 - R_o^4\right) \quad (29)$$

The outer friction face torque on the rear side of the valve plate, between the valve plate and Tier 2, $T_{\text{plate, b, outer}}$, is of the same form of Eq. (29), but with $c_f$ replaced by $c_b$. The viscous friction within the switching area is complicated by transition events. A simplification of Eq. (29) will be used with the limits of integration from $\theta=0$ to $2\pi$ for the front face switching torque, $T_{f, \text{switch}}$, and $\theta=0$ to $\pi$ for the rear face switching torque, $T_{b, \text{switch}}$. These new limits of integration are because the ports prevent a full annular region between the valve plate and the Tier 1 & 2 sections in which viscous losses can have a significant effect. The power loss from these frictional torques is then:

$$P_{\text{friction}} = \frac{\Omega}{\omega} \left(T_{f, \text{switch}} + T_{b, \text{switch}}\right) \quad (30)$$

### 3.5 POWER LOSS SUMMARY

After expressions for the energy losses of the high-speed valve were determined, Matlab was used to optimize critical parameters to achieve a maximum efficiency. The model parameters and resulting optimized parameters are shown in Table 2 and Table 3 with the estimated efficiency vs. duty ratio shown in Figure 10.

<table>
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<tr>
<td>$V_{\text{switch}}$</td>
<td>$1.0 \times 10^{-5} \text{ m}^3$</td>
</tr>
<tr>
<td>$f_{\text{switch}}$</td>
<td>100 Hz</td>
</tr>
<tr>
<td>$\omega$</td>
<td>$1500 \text{ rpm}$</td>
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<td>Duty</td>
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</tr>
<tr>
<td>$P_{\text{tank}}$</td>
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</tr>
<tr>
<td>$\Delta P_{\text{check}}$</td>
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<tr>
<td>$C_f$</td>
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<tr>
<td>$P$</td>
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<td>$\mu$ (DTE28)</td>
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<table>
<thead>
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<tr>
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<tr>
<td>$\rho_{\text{bore}}$</td>
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<td>$B_{\text{tan}}$</td>
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<tr>
<td>$P_{\text{fuel}}$</td>
<td>16 MPa</td>
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Table 2: Model parameters  Table 3: Optimized parameters
PROTOTYPE DESIGN AND IMPLEMENTATION

After the valve parameters were optimized, a prototype was designed and built. Sensors were used to measure the pressure at the inlet and outlet of the valve, the flow into and out of the valve, and the torque required to spin the valve plate.

4.1 VALVE DESIGN

The prototype design is shown in Figure 11. In the interest of brevity, a complete discussion of the mechanical design is not presented here, but can be found in the this reference [15]. Multiple sensors are used to measure the valve operation. A schematic of the location of each pressure sensor is shown in Figure 12. Note that unlike the model, the load for the experimental setup is controlled with a variable orifice. The input flow is quantified using two pressure transducers to measure the pressure drop across a fixed orifice. The downstream transducer was also used to quantify the input pressure. A fast response time pressure transducer is used to measure the pressure output of the valve. The output flow is quantified by passing the flow through a fixed displacement motor and monitoring the shaft rotation with an optical encoder. Finally, a force transducer mounted at a known radius from the center of the freely-rotating motor measures the torque created by the valve plate motor.

Figure 11: Solid model of the final prototype design.

Figure 12: Schematic of experimental setup showing location of sensors, orifices, and other hydraulic components

4.2 EXPERIMENTAL SETUP

The laboratory setup is shown in Figure 13 with a close up of the prototype valve shown in Figure 14. The optimization procedure determined the sizes of all the key geometric features; however, limitations of the available hydraulic flow bench required the optimized values to be recalculated. In particular, the available hydraulic supply is only capable of 8 liters per minute at 6.9 MPa, which is a 1/5th of the desired flow rate and ½ of the desired pressure. Also, Mobil DTE28 hydraulic oil was not readily available so Mobil 15M was used. This oil has a viscosity roughly a third of what was desired. Optimizing with these new properties changed the inner slot radius $R_i$ and $R_o$ to 4mm and 8mm, respectively. These changes had the following detrimental effect on the valve performance. As can be seen in Table 4, the efficiency of the prototype valve is expected to be lower overall, especially at lower duty ratios.

Table 4: Efficiency at selected duty ratios for new parameters

<table>
<thead>
<tr>
<th>Duty Ratio</th>
<th>0.25</th>
<th>0.5</th>
<th>0.75</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency</td>
<td>11.8%</td>
<td>55.3%</td>
<td>70.3%</td>
<td>79%</td>
</tr>
</tbody>
</table>
4.4 EXPERIMENTAL RESULTS

While certainly not complete, preliminary experimental results are presented to provide perspective on the prototype valve operation. The prototype valve was operated at varying duty ratios from 0 to 1 at frequencies up to 64 Hz. As seen from the output pressure plots, Figure 15, as the duty ratio is increased, the width of the pressure pulses also increases. This functionality demonstrates the basic desired functionality of creating virtually variable displacement behavior.
The preliminary maximum efficiency achieved by the valve is 38% at 1.0 duty ratio and 0.001 in valve plate clearance. The discrepancy between model and experimental results is attributed to a few sources. As observed in the zero duty ratio plot, the first subplot of Figure 15, pressure spikes can still be seen suggesting that the valve is experiencing unforeseen leakage from pressure to the output port. Also, in this configuration, the flow to the valve is much higher than the minimal output flow, suggesting that there are significant unforeseen leakage losses. Physical inspection of the valve prototype reveals small gaps between the tier 1 and tier 2 ports and the valve plate when the valve is supposed to be blocked. Furthermore, rounds were machined on the port surfaces, creating additional leakage paths.

Friction losses, which were calculated to be the least significant of the power losses, contributed to more than half of the total power loss in the experimental system. This discrepancy is believed to be caused by misalignment of components within the valve and greater than expected friction coefficients in the thrust bearings.

5 CONCLUSIONS

This paper presented the design of a high-speed phase-shift valve to enable switch-mode control of hydraulic. Switch-mode control allows any fixed displacement hydraulic pump, motor, or actuator to have virtually variable displacement by rapidly switching between efficient on and off states. The novel valve design uses a phase shift between two switching tiers to create a constant frequency pulsed flow where the duty ratio is controlled by change the phase between the switching tiers. The design uses a disc architecture with axial flow and a single constantly rotating valve plate.

Through a numerical model, the various forms of energy loss in the valve were computed. Using the model and an optimization procedure valve parameters were selected to minimize the total energy loss. These results were used to develop a prototype valve. Limited experiments demonstrated the correct switch-mode function of the valve at switching frequencies up to 64 Hz. However, due to larger leakage and torque requirements than predicted, the demonstrated efficiency was significantly less than expected.

Multiple areas of future work exist related to the phase-shift high-speed hydraulic valve. First, the experimental device will be refined to reduce leakage losses and torque requirements. This work will hopefully result in better alignment of the predicted and experimental efficiency results. Based on the experimental results, the analytical model will be improved, with specific focus on the viscous force equations and fluid dynamics. A next generation prototype will then be integrated into a hydraulic pump/motor to minimize the switched volume and provide real-world results.

In conclusion, many aspects of the high-speed hydraulic valve were a success, yet many areas of future research exist. Particularly in making the valve more compact, more efficient, improve the bandwidth, and developing control systems.

REFERENCES