International Journal of Fluid Power, 2014
Vol. 15, No. 2, 55–67, http://dx.doi.org/10.1080/14399776.2014.897500

Coupled Physics Modelling for Bi-Directional Check Valve System

Shaoping Xiong* and John H. Lumkes Jr*

Purdue University, Indiana, USA

This paper describes the development of a high speed on/off bi-directional check valve. The design was characterized using coupled-physics modelling tools, and a prototype was constructed and tested in the laboratory. The simulated and experimental results were compared. The high speed on/off bi-directional check valve (BDCV) utilizes positive feedback of flow forces and differential port pressure to quickly open and close the primary poppet. The coupled-physics model incorporates mechanical and fluid domains, which was solved through conducting finite element analysis (FEA) on a 2D planar model. After characterizing the BDCV, the system model was expanded for a single piston pumping system using two BDCVs and the simulation on system full displacement pumping was conducted. The modelling results showed a moderate agreement with measurements, which demonstrated the capability of the coupled physics model to effectively investigate the dynamic performances of a BDCV operating in a digital pump system.

Keywords: high speed on/off valve; coupled physics model; FEA; digital hydraulics; digital pump/motor

1. Introduction

Conventional hydraulic systems utilize metering valves for smooth control; however, metering valves cause throttling losses during operation and cause unnecessary energy losses in the system. Various studies have proposed the concept of a digital hydraulic system that allows discrete and accurate control with improved efficiency. proposed the digital flow control unit (DFCU) consisting of multiple on/off valves in a parallel configuration. Researchers also have developed virtual variable displacement pump (VVDP) technologies using a fixed displacement pump with pulse width modulating the on/off valves to achieve variable flow (Rannow et al., 2006; Lumkes et al., 2009). Another technology is piston-by-piston control using on/off valves to achieve variable control of pump/motor displacement (Rampen, 2006; Merrill et al., 2010). These studies indicate that the successful development of competitive digital hydraulic systems requires low cost valves with fast response and large nominal flow ratings.

Many valve designs have been proposed to enable digital hydraulics. Sturman Industries presented a 3-way, spool type, high speed latching valve with 0.45 ms switching and 17 L/min nominal flow @ 5bar pressure drop (Johnson et al., 2001). This valve utilizes the latching force from the residual magnetism of electromagnets to maintain the valve position without requiring a hold current (power loss). Tu et al. (2011) utilized a rotary high speed on/off valve and achieved about 3ms in on/off switching times and 35L/min nominal flow @ 5bar pressure. This valve was integrated into a PWM based virtual variable displacement pump system (VVDP), demonstrating system efficiency improvements from 7% to 15%. Winkler et al. (2010) presented a piloted fast switching valve utilizing a multi-poppet concept and achieved 2ms switching times and 85L/min @ 5bar pressure drop. Mahrenholz and Lumkes (2010) studied a 3-way 2-position valve for a VVDP application and achieved 0.2-1.5ms switching times with 25 L/min nominal flow and 0.2J energy consumption per switch; Wilfong (2011) designed and fabricated a prototype two stage bi-directional check valve (BDCV) with a 30L/min nominal flow @ 5bar pressure drop for digital pump/motor applications. The response time of this valve ranged from 2 to 8ms. Branson et al. (2011) developed a piezo-electrically actuated valve utilizing the Horbiger plate principle and achieved less than a 1.5 ms on/off switching time but only 20L/min nominal flow @ 5bar pressure drop.

An accurate, generic, and computationally efficient modelling tool is needed to more quickly assess and design high speed on/off valves. To deal with multi-physics coupling existing in high speed on/off valve systems, some researchers have focused on lumped parameter models. Lumkes et al. (2009) presented a lumped parameter model for VVDP systems using on/off valves, which included the dynamics of the pump, on/off valve, pressure load valve, accumulator, etc. Tu et al. (2009) developed a lumped parameter dynamic model for a rotary valve VVDP, which captured the dynamic effects of fluid compressibility, accumulator dynamics, valve spool operation, as well as the transition and orifice throttling effects. Reuter et al. (2010) proposed a lumped parameter reluctance model to reflect the electrical and magnetic properties of a dual coil high speed solenoid digital valve. Lumped parameter models are often used to reduce computational requirements, but this often comes with trade-offs in

*Corresponding authors. Email: xshaopin@purdue.edu; lumkes@purdue.edu

modelling accuracy. Some researchers utilize a look-up table for certain parameters within a distributed element model and then incorporate the table into a coupled lumped parameter model. Uusitalo et al. (2010) developed a static 2-D axisymmetric finite element model for an electromagnetic domain of a novel bi-stable hammer type digital valve to evaluate electromagnetic force. It was found that the simulated model accurately predicted power consumption values but failed to capture the accurate response time; Wilfong (2011) and Branson et al. (2011) both developed the static computational fluid dynamics (CFD) models for the flow domains of their high speed valve systems to acquire flow force profiles at different valve positions. However, this approach does not include the dynamic of the valve poppet/spool, which causes modelling discrepancies with the actual valve performances.

This research, that continues the work about the BDCV design from Wilfong (2011), models its valve characteristics and its operation in a digital pumping system using 2-D finite element, coupled physics (fluid dynamics and mechanical motion) models. The model provides simulation results of important parameters that help to evaluate and optimize the design of BDCV and digital pumps/motors.

2. Bi-directional Check Valve Design

2.1 Design Description

The bi-directional check valve (BDCV) developed by Wilfong et al. (2010) is a pilot operated, two-stage seat type valve. From the BDCV diagram shown in Fig. 1, flow from Port A and Port B will both be directed into actuation chambers around the main stage valve through the pilot stage valve. Other than a small bias spring, the main stage poppet is pressure balanced and the differential pressure between the actuation chambers will cause the valve to open or close. There is an internal leakage path across the actuation chambers, which will be sealed by an O-ring. The pilot stage valve is responsible for directing flow from Port A and Port B into the actuation chambers. Depending on whether Port A or Port B is at a higher pressure, the main stage valve position can be controlled by switching the position of the pilot stage valve. Therefore, unlike a conventional check valve which only allows flow in one direction while checking another, the BDCV allows flow in both directions: forward flow (Port A → Port B) and reverse flow (Port B → Port A), i.e. bi-directional checking.

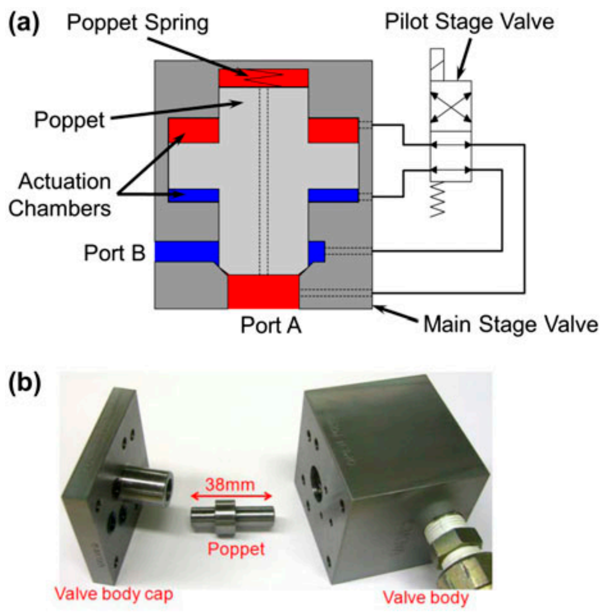

Fig. 1: BDCV design and main stage prototype (Wilfong, 2011). (a) BDCV design concept, (b) main stage prototype.

The BDCV design contains several characteristics which allow it to be used for multiple applications. First, as flow forces increase, the pressure difference across the actuation chambers also increases. With this feature, actuation force from differential pressure is always sufficient to overcome the flow force; hence, fast on/off switching is possible. Second, there is no electrical control signal required at full displacement digital pumping when the BDCV operates as a passive check valve and its dynamic capabilities are influenced by the fluid and mechanical forces. Even if an electrical signal is required, it only applies to the pilot stage (active checking) where the electrical energy consumption is much smaller than the power required to move the main stage. Third, the actuation forces increase proportionally to the actuation chamber area, so BDCV can be scaled for very large flow applications, such as wind turbines. Hansen et al. (2013) reported a similar valve design for the Power Take Off (PTO) system of ocean wave energy converters. This valve uses a 3/2 switching valve as the pilot stage valve to achieve bi-directional checking for the energy saving from the passive checking. It is rated for 1000L/min @ 5 bar pressure drop and simulations indicated over 15ms valve opening during passive checking.

2.2 Primary Application

The primary application of BDCV presented here is to enable digital pump/motors. The digital pump/motor is a 25-year-old technology that uses digital valves to control the commutation between the individual displacement chamber and working ports (Rampen and Salter, 1990; Ehsan et al., 1996). A digital pump/motor is built upon a multi-piston unit with each piston displacement chamber equipped with high speed on/off valves instead of having a port plate like traditional piston pumps/motors. This design enables the leakage and friction mechanisms to scale with displacement and leads to maintaining a relatively high efficiency over a wide range of operational displacements (Holland et al., 2011). Rampen (2006) from Artemis Intelligent Power developed and tested a radial digital pump/motor prototype for wind turbine transmission as shown in Fig. 2; Holland (2012) and Merrill (2012) developed a three piston digital pump/motor actively controlled by three couples of on/off valves as shown in Fig. 3. Both prototypes have demonstrated improved energy efficiency over conventional axial piston machines.

The digital pump/motor operation is based on a set of operation strategies-flow diverting and flow limited displacement control (Merrill and Lumkes, 2010), which are controlled by timing when each on/off valve opens and closes. Fig. 4 shows how two BDCVs are positioned into a single piston pumping system. The configuration enables four-quadrant pump/motor operation. By sending a signal to the pilot stage valve to actively control the BDCVs, this system can enable different flow strategies at full or partial displacement.

3. Model Development of BDCV

3.1 Coupled Physics Model Description

The BDCV system presented here is a single piston pumping system for a digital pump/motor feasibility study, which can be modelled as a fluid dynamic, mechanical motion, and electromagnetic multi-physics coupled system. The hydraulic valve system will iteratively include two fundamental domains: fluid dynamics and mechanical motion. Since the BDCV may require an electrical signal to control the pilot stage valve position using a solenoid, electromagnetic force calculations can also be included in BDCV system modelling. However, the primary focus of this work is modelling and testing the main poppet dynamics while switching under external influences due to the geometrically constrained pump piston motion. The main stage of the BDCV was fabricated at Purdue University and two Parker Hannifin Pulsar valves were used to replace one 4-way 2-position valve for the pilot stage.

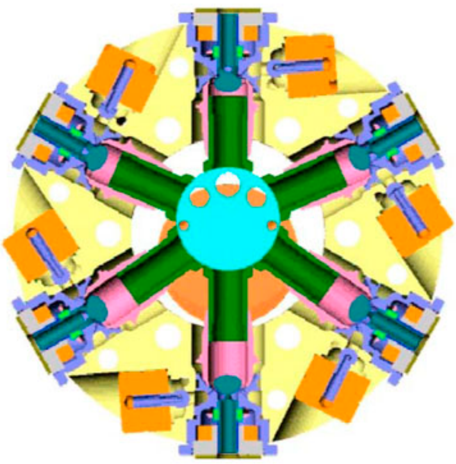

Fig. 2: Artemis digital pump/motor (Rampen, 2006).

Fig. 5 illustrates the coupling mechanism of these two domains. First by solving the Navier-Stokes equation, the velocity $\boldsymbol{U}$ for flow rate evaluation and pressure p of fluid domain is obtained. Then the pressure profile will be used to calculate the actuation forces of valve motion to acquire position and velocity of the valve poppet according to Newton's Law of Motion, which will have an impact on the boundary condition and geometry of the fluid domain.

3.2 Fluid Domain

To describe the fluid dynamics of valve system, Navier-Stokes equations are utilized in a finite element model:

$$\rho\frac{\partial \vec{U}}{\partial t} + \rho(\vec{U}\cdot\nabla)\vec{U} = \nabla\cdot\left[-p+\mu\left(\nabla\vec{U}+(\nabla\vec{U})^T\right)-\frac{2}{3}\mu(\nabla\cdot\vec{U})\right]+\vec{F}_{body} \tag{1}$$

$$\frac{\partial\rho}{\partial t}+\nabla\cdot(\rho\vec{U})=0 \tag{2}$$

Here $\vec{U}$ represents the velocity vector. Eq. 1 is the momentum transport equation; Eq. 2 is the equation of continuity for compressible fluids. Isothermal fluid compressibility will be defined using bulk modulus K in Eq. 3, which is related to fluid density ρ and pressure p:

$$\frac{1}{K}=\frac{1}{\rho}\left(\frac{d\rho}{dp}\right)_T \tag{3}$$

If ρ_0 is defined as fluid density at reference pressure 0 bar, then Eq. 3 can be rewritten as:

$$\rho=\rho_0 e^{\frac{p}{K}} \tag{4}$$

Then, the Navier-Stokes equations combined with Eq. 4 represent the fluid domain model for BDCV system.

After solving Eqs. 1 through 4, the resulting pressure forces and viscous friction forces can be calculated respectively using Eq. 5 and Eq. 6:

$$F_p=\iint pdS \tag{5}$$

$$F_{vf}=\iint\left[\mu\left(\nabla\vec{U}+(\nabla\vec{U})^T\right)-\frac{2}{3}\mu(\nabla\cdot\vec{U})\right]dS \tag{6}$$

3.3 Mechanical Domain

The mechanical domain of the BDCV system refers to the main stage poppet motion which depends on the resultant force acting upon it. First, the flow forces including pressure forces and viscous friction forces need to be calculated:

$$F_{flow}=F_p+F_{vf} \tag{7}$$

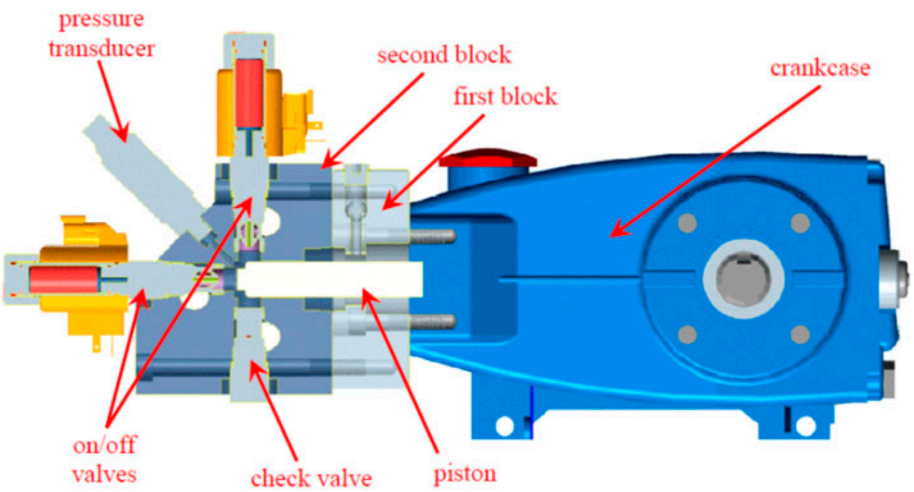

Fig. 3: Purdue digital pump/motor (Holland, 2012).

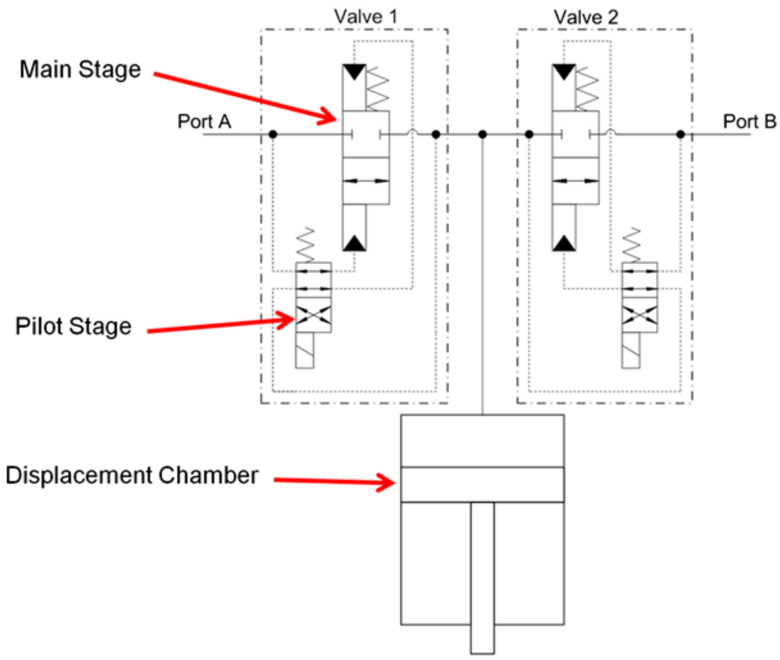

Fig. 4: Configuration of single piston pump/motor with BDCVs (Wilfong, 2011).

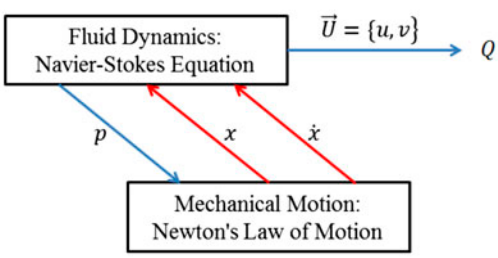

Fig. 5: Coupling mechanisms of multi-domain model.

The spring force which normally closes the valve is expressed by Hooke's Law:

$$F_{spring} = F_{s,o} + k_{spring} x_s \tag{8}$$

The walls in the valve body for restricting the poppet motion are modelled as a stiff spring with damper when the valve poppet is in contact with the wall:

$$F_{wall} = \begin{cases} k_{wall} x_{wall} + b_{wall} \frac{dx_{poppet}}{dt}, & x_{wall} < 0 \\ 0 \quad , & x_{wall} \geqslant 0 \end{cases} \tag{9}$$

There is an O-ring seal to prevent internal leakage in the prototype main stage valve, which will introduce undesired friction. The O-ring friction force F_{oring} can be modelled by the methods introduced by Thoman (1992) and the Parker O-ring Handbook (Parker, 2007):

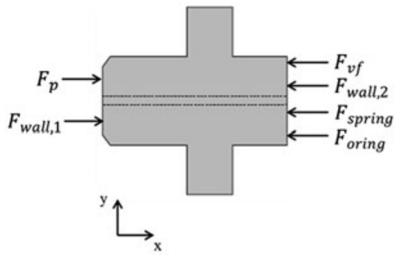

Fig. 6: BDCV main stage free body diagram.

$$F_{oring} = F_C + F_H = f_c L_p + f_h A_p \tag{10}$$

A sign function should be added into Eq. 10 to deal with direction changing of friction forces, as shown in Eq. 11:

$$\begin{cases} F_{oring} = \mathrm{sgn}(v) \cdot (f_c L_p + f_h A_p) \\ v = \frac{dx_{poppet}}{dt} \end{cases} \tag{11}$$

The empirical value of 630.4 N·m^{-1} (Wilfong, 2011) is used for the friction coefficient from O-ring compression f_c, and the friction coefficient from fluid pressure f_h is evaluated using a function of pressure in Eq. 12:

$$f_h = \frac{1}{375} p + 68947.5 \tag{12}$$

The seal contact length L_p is 39.9mm and the projected area of seal A_p is 51.6mm^2.

According to the previous force analysis, the motion of the main stage poppet can be expressed by Eq. 13; Fig. 6 represents the free body diagram of the BDCV:

$$m_{poppet} a_{poppet} = F_{flow} + F_{wall} + F_{spring} + F_{oring} \tag{13}$$

Once main stage poppet acceleration is obtained, its velocity can be evaluated by integrating acceleration over time and its position can be determined by integrating the resulting velocity:

$$\begin{cases} v_{poppet} = \int a_{poppet} dt \\ x_{poppet} = \int v_{poppet} dt \end{cases} \tag{14}$$

4. Modelling Implementation

4.1 Valve Characteristic Test

Wilfong et al. (2011) conducted the valve characteristics tests on the prototype BDCV and Fig. 7 shows the test stand. During the tests, pilot stage valves were kept stationary to provide fixed flow paths between the working ports and actuation chambers. In this case, the BDCV operates as a standard check valve. This mode was called passive checking. The finite element modelling was applied to this BDCV passive checking test.

As the BDCV is not an axisymmetric geometry, the cross section of BDCV was used to develop a 2D planar model for the valve characteristics test, as shown in Fig. 8. This model was developed and simulated using the commercial FEA software COMSOL, whose meshing technique was detailed in Section 4.2. The simulated BDCV valve characteristics will be compared with the measurement results. This step was used to justify the 2D planar model implementation on the pumping tests when the BDCV operates in a digital pump/motor. The model includes the functional components of BDCV: the main stage poppet, pilot stage valve and main stage returning spring. The system pressure boundary conditions were applied on Port A and Port B, and were simulated at two levels, 21 bar and 69 bar, matching the experimental test conditions. System flow rate was calculated using Eq. 15:

$$Q = \frac{\pi d^2}{4} \cdot v_{average} \tag{15}$$

Where d is the inlet/outlet boundary length and v_{average} stands for the average fluid velocity of the boundary, which was evaluated by Eq. 16:

$$v_{average} = \frac{\int_L v dl}{d} \tag{16}$$

4.2 Full Displacement Pumping Test

This BDCV multi-physics model will be coupled with a single piston pumping model for the digital pump/motor application as shown in Fig. 4. The electromagnetic domain is not included in this model, only the full displacement pumping mode where the BDCV stays in passive checking mode, will be studied in this paper. Fig. 9 illustrates the operation mode of a single piston pumping system at 100% displacement. In the suction portion fluid is drawn into the piston chamber from the low pressure line through the inlet BDCV while the outlet BDCV is checking the flow; then when the delivery portion begins, the inlet BDCV starts to check the flow while fluid is discharged at high pressure to the working port through the outlet BDCV. Note that there can be undesired backflow through the inlet port BDCV during the transition between suction and delivery.

A test rig was constructed for a single piston pumping system incorporating a BDCV as shown in Fig. 10. Since the primary goal was to evaluate the dynamic characteristics of a BDCV in a digital pump/motor, the outlet BDCV was replaced with a normal check valve, so only one prototype BDCV was needed. Experiments to validate the BDCV in different digital pump/motor operating modes were conducted and a sinusoidal actuation mechanism was applied to the single piston. The inlet was maintained at a constant pressure and the outlet pressure was determined by the load from the relief at the outlet. The significant parameters of the test stand are listed in Table 1. The key parameters of this prototype BDCV also apply to the valve characteristics test.

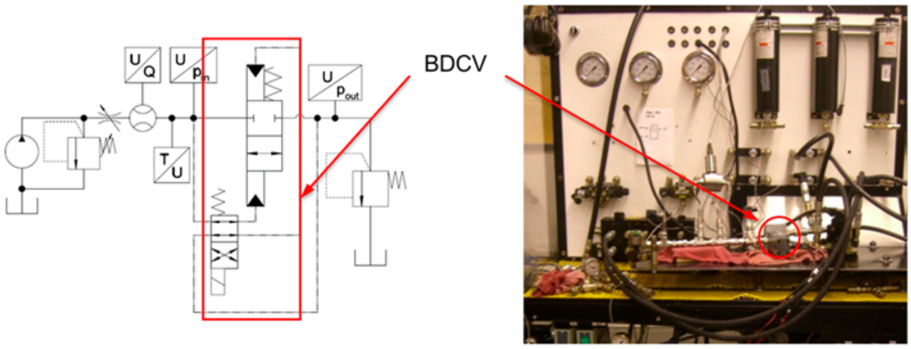

Fig. 7: BDCV valve characteristics test stand (Wilfong, 2012) (a) Geometric model, (b) finite element model.

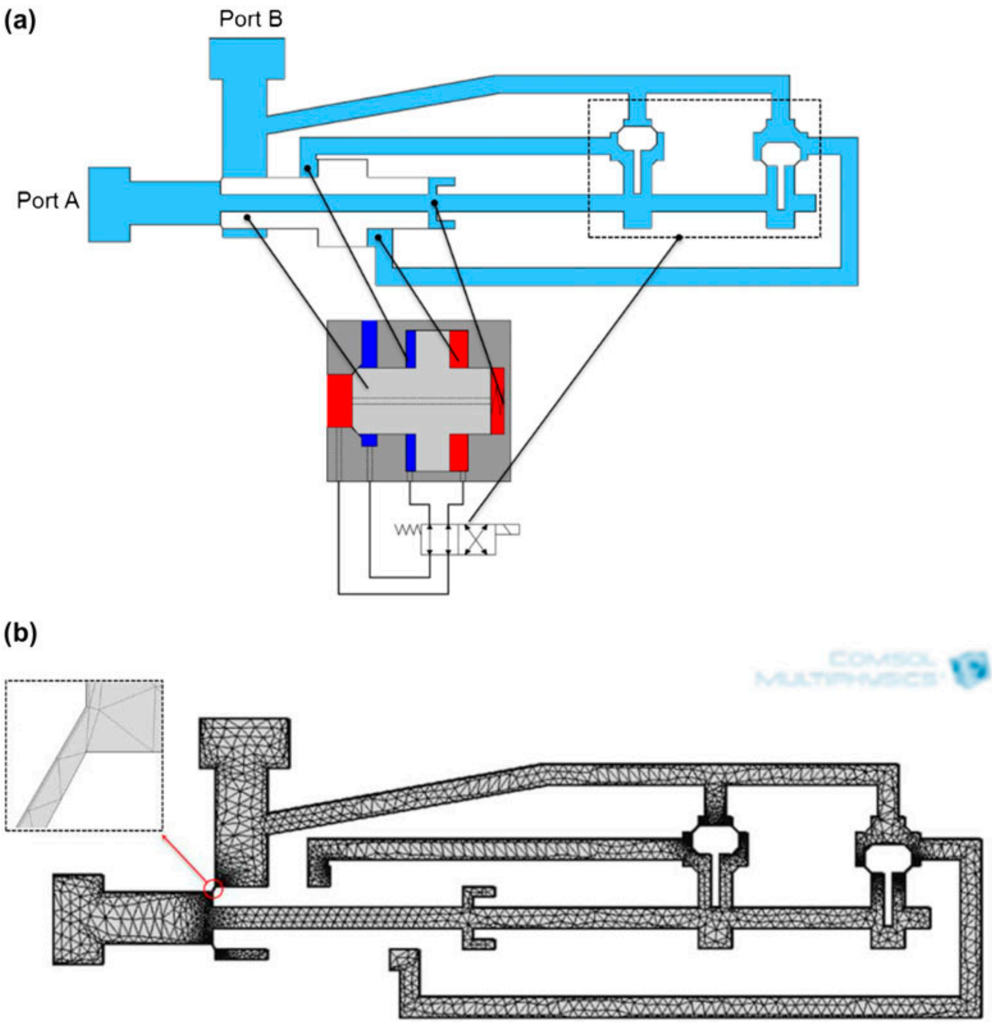

Fig. 8: BDCV 2D-plane model.

More detailed information about the test stand and relevant experiments can be found in a paper from Holland et al. (2011).

The finite element model for a single piston pumping system was developed using the commercial software COMSOL. As shown in Fig. 11, a 2-D model was again used since the system is not axis-symmetric. The setup of this model was based on the single piston pumping test stand shown in Fig. 10. Parameter values and boundary conditions for the model were determined by the actual test stand parameters as shown in Table 1. Only the load pressure at the outlet was varied according to characteristics of the outlet pressure relief valve, as shown in Eq. 15.

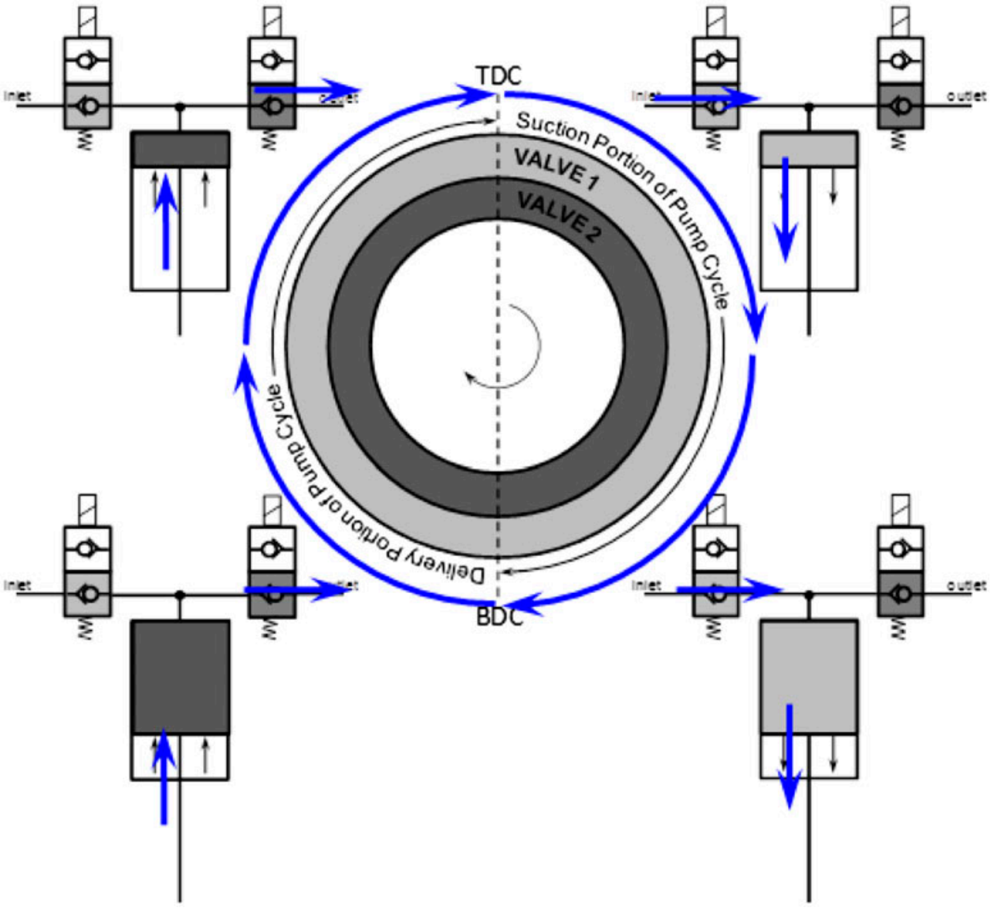

Fig. 9: Full displacement pumping cycle (Holland et al., 2011).

Geometric dimensions were sized to the actual test stand except the pilot stage valves, which were approximated by two 2-way poppet valves. Note that the check valve at the outlet on the actual test stand was replaced by a BDCV in the model.

Fig. 12 shows how the 2-D model was discretized for finite element analysis. Triangle elements were utilized for the entire domain meshing while different mesh sizes were manually distributed for better convergence speed and stability. Layer meshes were implemented on the wall boundaries where large flow gradients may occur, such as the main stage valve opening edge; otherwise significant error can be introduced during simulation.

Navier-Stokes equations for fluid domain are embedded into the COMSOL solver. Dynamic equations for the main stage poppet motion with all boundary conditions were manually added. To directly couple the fluid domain and mechanical motion in the finite element model, the Moving Mesh mode in COMSOL was utilized. At each time step, velocity and position of the main stage poppet were computed to setup the conditions for the moving boundaries. The moving boundary displacement was extended throughout the model to obtain a smooth mesh deformation everywhere. This was accomplished by solving the PDEs for the mesh displacements as incorporated in the Moving Mesh mode. Mesh movement described above is based on the technique-arbitrary Lagrangian-Eulerian (ALE) method. By conducting a coordinate transformation between the deformed frame and reference (fixed) frame. The ALE method allows for moving boundaries without the need for the mesh movement to follow the material, which is suitable for a fluid flow model that includes moving mechanical parts.

In addition, large boundary displacements may cause the mesh to become too deformed/inverted to obtain a converged solution. Therefore, an automatic remeshing technique was utilized to help the simulation proceed. Specifically, a threshold condition to determine whether the mesh gets inverted was added into the model. Once the threshold condition is satisfied during simulation, the solver stops and the 2-D model will be remeshed in reference to the initial mesh settings and then simulation continues. A similar technique called "dynamic mesh" was adopted to investigate the unsteady forces in a spool valve (Vescovo and Lippolis, 2006). More detailed descriptions of mesh movement and automatic meshing can be found in the COMSOL Multiphysics Users Guide (2012).

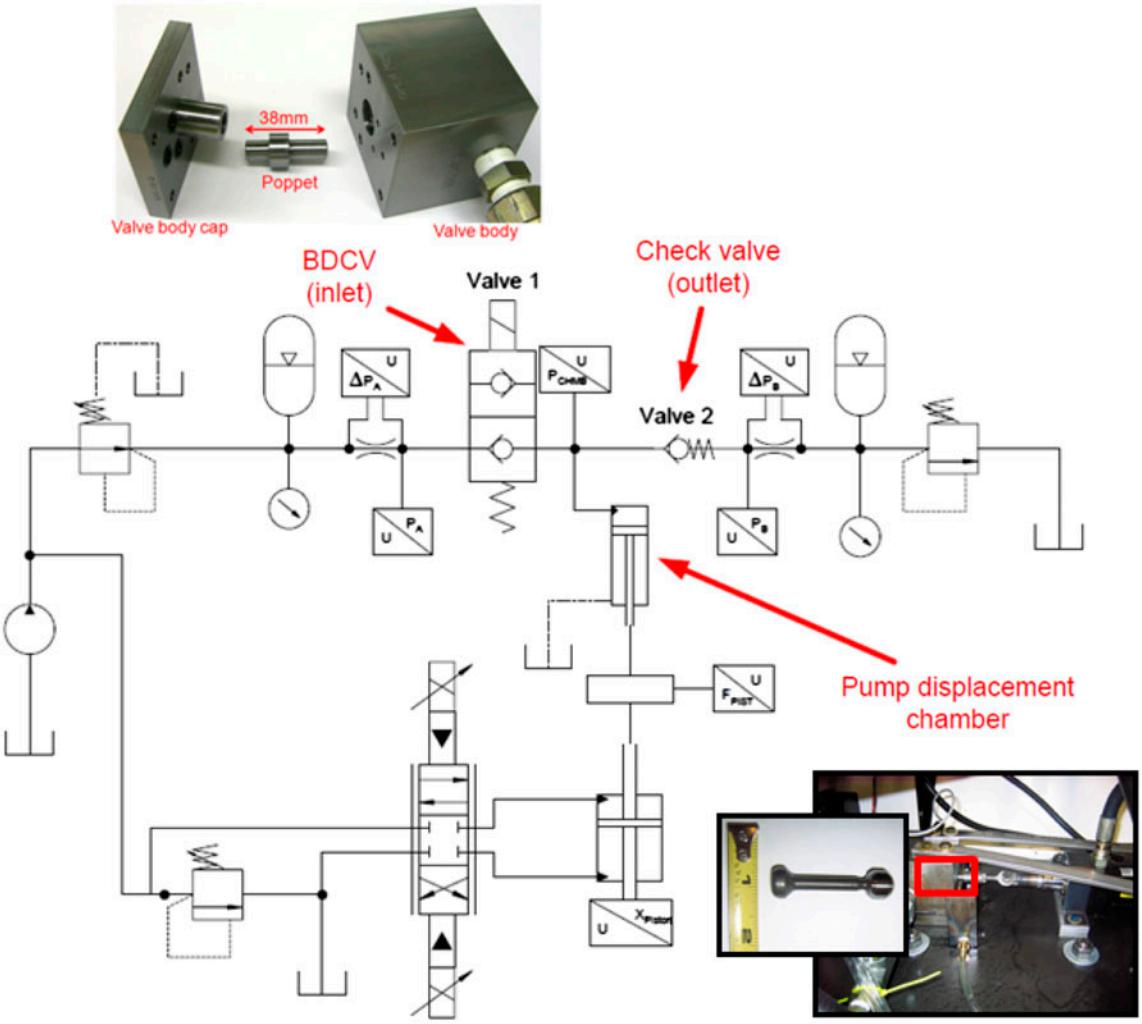

Fig. 10: Single piston pumping test stand circuit (Holland et al., 2011).

Table 1. Test Stand Parameters.

Description	Symbol	Value
Wall damping ratio	b_{wall}	4.6×10^4 N/m s^{-1}
Piston diameter	d_{piston}	13.53mm
Main stage poppet diameter	d_{poppet}	9.42mm
Piston motion frequency	f_{piston}	500strokes/min
Spring preload force	$F_{s,o}$	20N
Spring coefficient	k_{spring}	11730N/m
Valve body wall stiffness	k_{wall}	2×10^{10} N/m
Fluid bulk modulus	K	1.2×10^9 Pa
Stroke length	L_{stroke}	23mm
Main stage poppet mass	m_{poppet}	0.027kg
Inlet pressure	p_{in}	14bar
Fluid viscosity	μ	0.01575Pa•s
Fluid density @40C	ρ_0	875kg/m^3

5. Simulation and Validation

5.1 Valve Characteristics

Fig. 13 shows the BDCV steady-state characteristics curves in both forward flow (FF) and reverse flow (RF) scenarios. The simulation results agree well with the experimental results. Note that an approximately linear relationship, instead of the square root relationships of normal valves, between the pressure drop and flow rate presents because of the spring force linear to the valve stroke length.

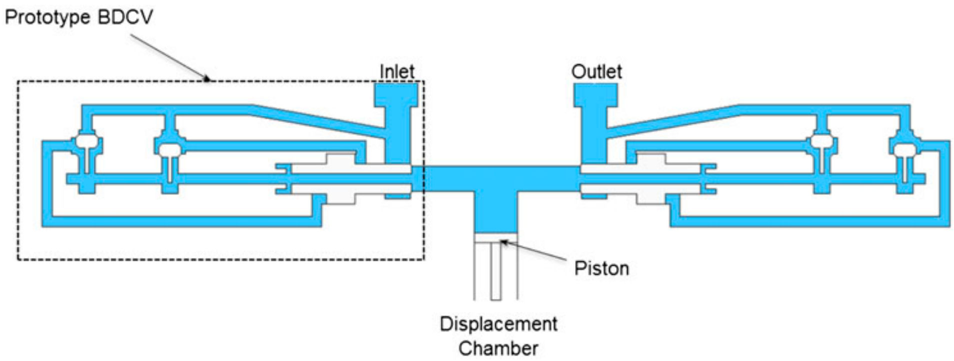

Fig. 11: 2-D plane model for single piston pumping system.

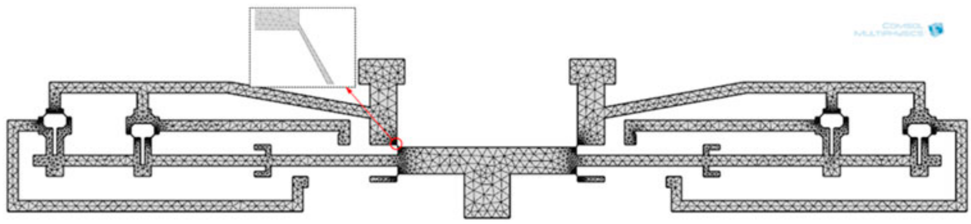

Fig. 12: Finite element model for single piston pumping system.

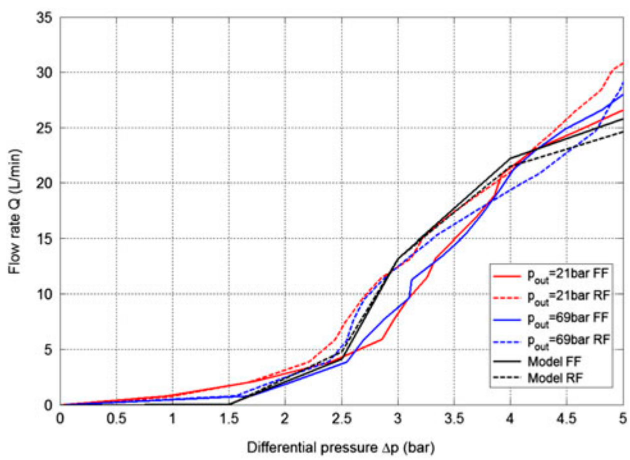

Fig. 13: BDCV valve characteristics curve (a) Suction stroke, (b) delivery stroke.

5.2 *Full Displacement Pumping Results*

Fig. 14 shows the FEA solution domain of flow and pressure in one 100% pumping cycle. During the suction stroke, the pressure at the displacement chamber keeps decreasing while the piston is moving downward. Differential pressure increases between the actuation chambers of inlet BDCV, actuating the inlet BDCV and allowing fluid flow into the displacement chamber. High fluid velocities are momentarily observed as the valve begins to open. The BDCV orifice is small at the beginning of valve transition, which leads to a pressure loss and high velocities of orifice flow. This is not a steady state loss but occurs in a short valve transition period (several milliseconds). Orifice flow velocities decrease as the orifice opening becomes larger. At the delivery stroke, pressure in the displacement chamber starts to build up with the piston pressurizing the chamber fluid, when the inlet BDCV is still open. Once the displacement chamber pressure surpasses the inlet pressure, differential pressure between the inlet actuation chambers will close the inlet BDCV. It can be noticed that certain amount of flow is delivered back to the inlet low pressure side during this process, which causes a loss (backflow leakage). As the displacement chamber pressure continues to increase, it will eventually exceed the outlet pressure. Then the differential pressure between the outlet actuation chambers will open the outlet BDCV so that the fluid stored in displacement chamber can be delivered to outlet working port. Here it can be observed that there was no pilot stage motion or electrical signal input during the entire cycle, demonstrating passive checking using the BDCVs at full displacement pumping.

The piston position profile follows the standard sinusoidal variation in one full displacement pumping cycle (without pilot valve action). Simulated inlet/outlet flow rates were evaluated using Eq. 16 and pressures at the inlet, outlet and the piston chamber were investigated. Simulation and experimental results in Fig. 15 shows that the sinusoidal piston motion causes the cylinder pressure to change, where the flow is redirected from the inlet into the cylinder chamber through the BDCV at Port A in the suction stroke and completely discharged to the outlet working port through the BDCV at Port B during the delivery stroke.

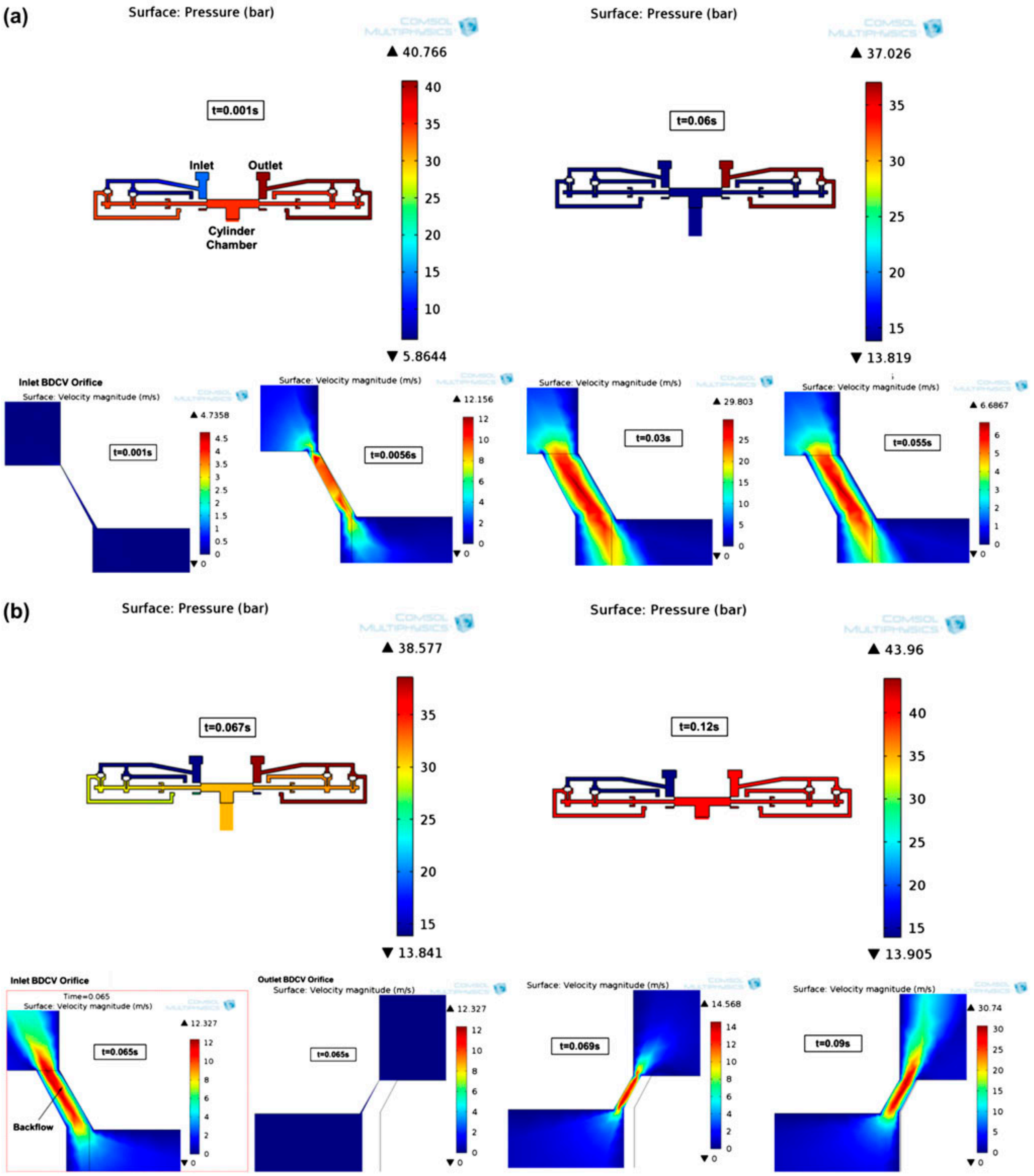

Fig. 14: Solution domain of pressure and velocity (a) Flow trend plot, (b) pressure plot.

Flow and pressure simulation results generally correlate well with the measurement results. The flow modelling plots successfully capture the backflow rate, which is about 2L/min, and which causes about a 5ms transition delay. But there are some discrepancies between the measurements and simulations. For the flow rates, simulations didn't reflect a high frequency oscillation seen in the measured flow trace. This can be explained by an indirect instrumentation of flow rates. During the experimental tests, the flow rate measurements were accomplished by orifice flow meters constructed from the Honeywell HL-Z high line differential pressure transducers and orifices. The orifices were calibrated to evaluate the flow rates from the dynamic pressure measurements, which

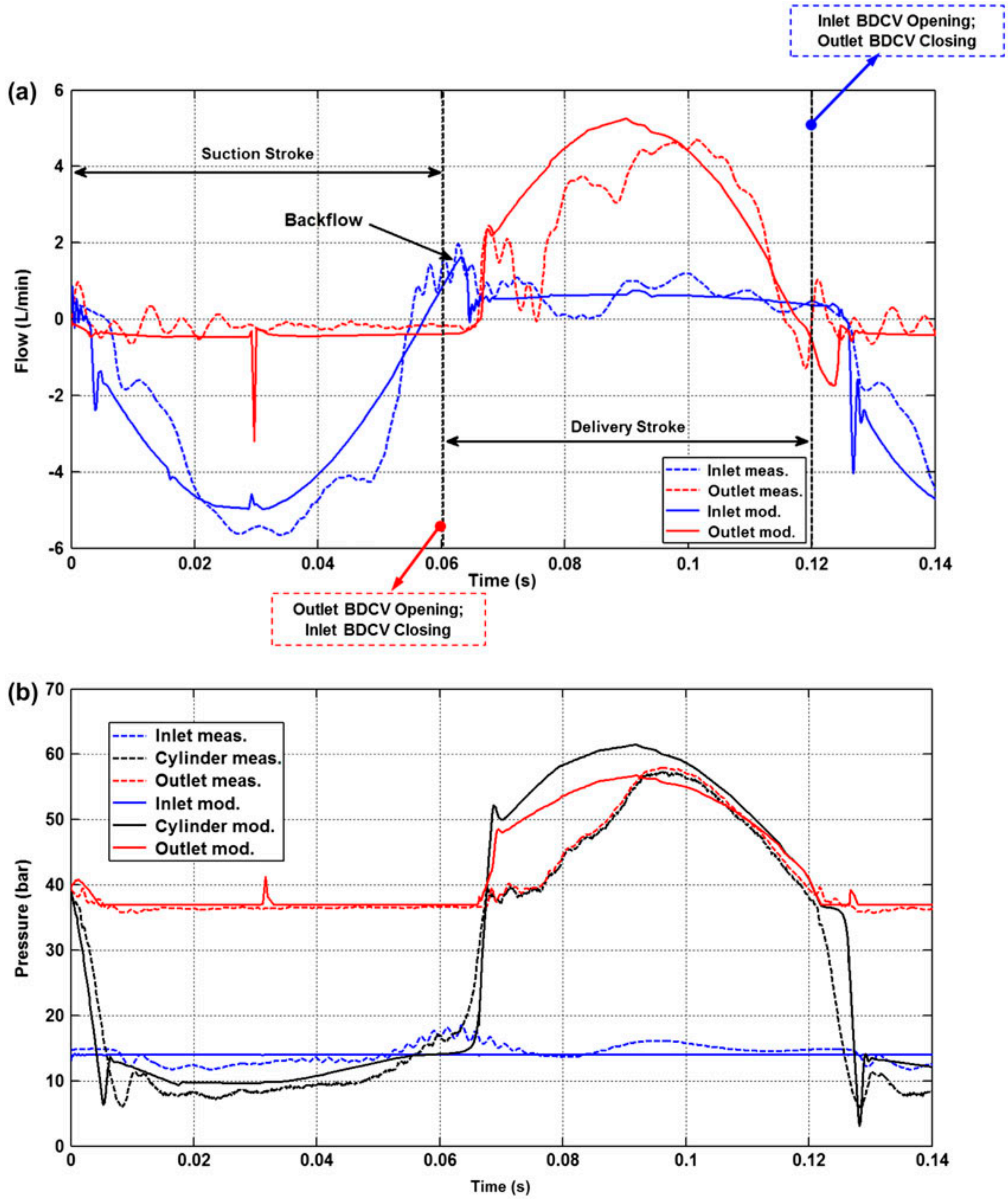

Fig. 15: System pressure and flow profiles of full displacement pumping.

produced some oscillation. As for the pressure discrepancy during the delivery stroke, this can be caused by replacing the outlet normal check valve with a BDCV in the experimental tests. The return spring of BDCV is larger than that of the check valve. This leads to a simulated 5bar, larger than the measured 0.34bar, pressure difference between the cylinder and outlet port at maximum system flow.

Simulation time for the entire pumping cycle was less than 50 minutes on a common desktop computer with Intel(R) Core™ i5-2400 CPU @ 3.10GHz. This computing time range allows this model to conduct multiple simulations for valve design optimization.

6. Conclusion

A coupled physics 2-D planar FEA model was developed for a bi-directional check valve (BDCV) system. This model was built by coupling the fluid dynamics and mechanical motion, and their interactions. The model was implemented to study BDCV's valve characteristics and its application in a single piston pumping system for a digital pump/motor. Simulation work showed a moderate performance in terms of both accuracy and computational costs, which proved this model to be a reasonable approach to analyze the valve characteristics and dynamic system performances of a similar valve design.

Meanwhile, simulation results also revealed some limitations of this model. This 2-D model didn't model a true system geometry, which may cause an inherent modelling error. The simulation costs are still expensive compared with a lumped parameter model. The applicability of this model can be affected due to these limitations. In the future, modifications on model will be studied to improve the predication quality and efficiency. Moreover, the electromagnetic domain will be included into the current model to complete a more comprehensive multiphysics coupled model for high speed valve systems.

Nomenclature

A_p	Projected area of seal	in^2
a_{poppet}	Acceleration value of main stage poppet	m/s^2
b_{wall}	Damping coefficient of wall spring system	$N/(m \cdot s^{-1})$
d	Inlet/Outlet boundary length	mm
d_{poppet}	Main stage poppet diameter	mm
d_{piston}	Piston diameter	mm
f_c	Friction coefficient due to O-ring compression	N/m
f_h	Friction coefficient due to fluid pressure	Pa
f_{piston}	Piston motion frequency	Hz
F_{body}	Body force term	Pa
F_c	Friction force due to O-ring compression	N
F_h	Friction force due to fluid pressure	N
F_{oring}	Friction force from O-ring	N
F_p	Fluid pressure force	N
F_{spring}	Return Spring force	N
$F_{s,o}$	Spring force at original position	N
F_{vf}	Fluid viscous friction force	N
F_{wall}	Wall force	N
K	Fluid bulk modulus	Pa
k_{relief}	Relief valve coefficient	bar·min/L
k_{spring}	Return spring coefficient	N/m
k_{wall}	Valve body wall stiffness	N/m
L_p	O-ring seal contact length	in
L_{stroke}	Piston stroke length	mm
m_{poppet}	Main stage poppet mass	kg
p	Fluid pressure	bar, Pa
p_{in}	Inlet fluid Pressure	bar
p_{out}	Outlet fluid Pressure	bar
p_{relief}	Pressure due to relief valve spring	bar
Q	System inlet/outlet flow rate	L/m
U	Flow velocity	m/s
$v_{average}$	Average flow velocity at inlet/outlet boundary	m/s
v_{poppet}	Main stage poppet moving velocity	m/s
x	Displacement	m
x_{poppet}	Main stage poppet displacement	m
x_{wall}	Depth valve poppet intruding into the wall	m
μ	Fluid viscosity	Pa·s
ρ	Fluid density	kg/m^3
ρ_0	Fluid density at 40°C, 0bar(barometer)	kg/m^3

Acknowledgements

This research was supported by the Center for Compact and Efficient Fluid Power, a National Science Foundation Engineering Research Center funded under cooperative agreement number EEC-0540834.

Notes on Contributors

Shaoping Xiong Born on December 1, 1986 in Nanchang, Jiangxi, China. He received the BSE degree in 2008 and the MSME in 2010 from China Agricultural University. He is studying towards his PhD in the area of high speed on/off valves modeling and design for digital hydraulics at Purdue University.

John H. Lumkes, Jr. John received his BSE degree from Calvin College in 1990, MSE from the University of Michigan-Ann Arbor in 1992, and PhD from the University of Wisconsin-Madison in 1997. From 1997 to 2004 he was an Assistant and Associate Professor at Milwaukee School of Engineering. In 2004 he joined Purdue University and is active in digital hydraulics, modeling and controls, mechatronics, and advising senior projects.

References

Branson, D. T., Wang, F. C., Johnston, D. N., Tilley, D. G., Bowen, C. R. and Keogh, P. S. 2011. Piezoelectrically Actuated Hydraulic Valve Design for High Bandwidth and Flow Performance. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 2011, 225–345.

COMSOL Multiphysics. 2012. *COMSOL Multiphysics Users' Guide*.

Ehsan, M., Rampen, W. H. S. and Salter, S. H. 1996. Computer Simulation of the Performance of Digital-Displacement Pump-motors. *Fluid Power Systems and Technology: Collected Papers,* Vol. 3, pp. 19–24. (Proceedings of the 1996 International Mechanical Engineering Congress & Exposition.)

Hansen A., Pederson H. and Anderson T. 2013. Design of Bidirectional Check Valve for Discrete Fluid Power Force System for Wave Energy Converters. *Proceedings of the ASME/BATH 2013 Symposium on Fluid Power & Motion Control,* October 6–9, 2013, Sarasota, Florida, USA.

Holland, M. 2012. Design of Digital Pump/Motors and Experimental Validation of Operation Strategies. PhD Thesis, Purdue University.

Holland, M., Wilfong, G., Merrill, K. and Lumkes, J. 2011. Experimental Evaluation of Digital Pump/motor Operating Strategies with a Single-piston Pump/motor. *Proceedings of the 52nd National Conference on Fluid Power*. March 23–25, 2011. Las Vegas, NV, USA.

Johnson, B., Massey, S. and Sturman, O. 2001. Sturman Digital Latching Valve. *Proceedings of the 7th Scandinavian International Conference on Fluid Power*, Linkoping, Sweden, pp. 299–314.

Linjama, M. and Vilenius, M. 2007. Digital Hydraulics - Towards Perfect Valve Technology. *The Tenth Scandinavian International Conference on Fluid Power*, Tampere, Finland.

Lumkes, J., Bartdoff, M. A. and Mahrenholz, J. 2009. Model Development and Experimental Analysis of a Virtually Variable Displacement Pump System. *International Journal of Fluid Power,* Vol. 10, No.3, pp. 17–27.

Mahrenholz, J. and Lumkes, J. 2010. Coupled Dynamic Model for a High Speed Pressure Balanced 3-way On/Off Hydraulic Valve. *Journal of Dynamic Sysems, Measurement, and Control,* Vol. 132.

Merrill, K. 2012. Modeling and Analysis of Active Valve Control of A Digital Pump-Motor. PhD Thesis, Purdue University.

Merrill, K. and Lumkes, J. 2010. Operating Strategies and Valve Requirements for Digital Pump/Motors. *Proceedings of the 6th Fluid Power Net International PhD Symposium*, West Lafayette, 2010, pp. 249–258.

Merrill, K. J., Holland, M. A. and Lumkes, J. H. 2010. Efficiency Analysis of a Digital Pump/Motor as Compared to a Valve Plate Design. *Proceedings of the 7th International Fluid Power Conference. (7.IFK),* Aachen, Germany, March 2010, pp. 22.–24.

Parker Hannifin Corporation 2007.*Parker O-Ring Handbook: ORD 5700*.

Rampen, W. 2006. Gearless Transmissions for Large Wind Turbines – The History and Future of Hydraulic Drives. Retrieved from http://www.artemisip.com/Pictures/GearlesstransmissionsBremenNov06.pdf

Rampen, W. and Salter, S. 1990. The Digital Displacement Hydraulic Piston Pump. *Proceedings of the 9th International Symposium on Fluid Power*, Cambridge.

Rannow, M., Tu, H., Li, P. and Chase, T. R. 2006. Software Enabled Variable Displacement Pumps - Experimental Studies. *Proceedings of the 2006 ASME-IMECE*, No. IMECE2006-14973.

Reuter, J., Maerkl, S. and Jaekle, M. 2010. Optimized Control Strategies for Fast Switching Solenoid Valves. *International Journal of Fluid Power,* Vol. 11, No. 3, pp. 23–33.

Thoman, R. 1992. An empirical approach to seal friction. *Aerotech '92 Conference*, Anaheim, CA, October 5–8, 1992.

Tu, H., Rannow, M., Wang, M., Li, P. Y. and Chase, T. R. 2011. The Advantages and Feasibility of Externally Actuating a High-speed Rotary On/Off Valve. *Proceedings of the 52nd National Conference on Fluid Power*, Las Vegas, NV, p. 20.3.

Tu, H. C., Rannow, M. B., Wang, M., Li, P. Y. and Chase, T. R. 2009. Modeling and Validation of A High Speed Rotary PWM On/Off Valve. *Proceedings of the ASME 2009 Dynamic Systems and Control Conference*, DSCC2009-2763.

Uusitalo, J., Ahola, V., Soederlund, L., Linjama, M., Juhola, M. and Kettunen, L. 2010.Novel Bistable Hammer Valve for Digital Hydraulics. *International Journal of Fluid Power,* Vol. 11, No. 3, pp. 35–44.

Vescovo, G. D. and Lippolis, A. 2006. A Review Analysis of Unsteady Forces in Hydraulic Valves. *International Journal of Fluid Power,* Vol. 7, No. 3, pp. 29–39.

Wilfong G. 2011. Design and Dynamic Analysis of High Speed On/Off Poppet Valves for Digital Pump/Motors. Master Thesis, Purdue University.

Wilfong, G. J., Holland, M. A. and Lumkes, J. H. 2011. Design and Analysis of Pilot Operated High Speed On/Off Valves for Digital Pump/Motors. *Proceedings of the 52nd National Conference on Fluid Power,* 2011, pp. 539–543.

Winkler, B., Ploeckinger, A. and Scheidl, R. 2010. A Novel Piloted Fast Switching Multi Poppet Valve. *International Journal of Fluid Power*, Vol. 11, No. 3, pp. 7–14.