First Steps Towards Model-Based Valve Design: Measuring Pressure Drop and Flow Forces of Industrial 2/2 Directional Poppet Valves

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Abstract

This paper introduces a method for the measurement of flow forces on industrial 2/2 directional poppet valves. The valves are both normally open (NO) and normally closed (NC), direct-acting oil hydraulic valves that allow fluid flow in both directions. Previous works have only examined abstracted and reduced geometries that were specially manufactured for measurement purposes. This work provides a detailed insight into the valve actuation and force measuring system for a wide range of geometries as well as elaborates on the conceived quasi-static measurement method with a moving poppet. A unit for the compensation of pressure forces during measurement was developed and both its advantages and disadvantages are discussed. The test bench setup and the assumptions for determining flow forces from two measurements with and without flow are explained. The results for the measurements of five NO valves and four NC valves are shown for one tank pressure and one pressure difference. The results indicate that for most NO valves, the flow direction alters the behavior of the flow forces, and in one valve, the flow force changed its effective direction. In contrast, the flow force behavior in NC valves was qualitatively the same in both flow directions across all four tested valves. These measurements provide a better understanding of the flow forces in industrial NO and NC valves and can be used to validate computational fluid dynamics (CFD) simulations in the future.

Keywords: model-based valve design, poppet valves, flow forces, measurements

1 Introduction

Oil hydraulic 2/2 directional poppet valves are used in a wide range of applications such as safety systems, locking mechanisms and systems with on/off functionalities. The development of valves for specific applications is currently time-consuming and cost intensive. This is due to the fact that valve development is an iterative process between tests and computational fluid dynamic (CFD) simulations, which require expert knowledge, and is usually based on the experience of the companies. In valve development, the geometries of the poppet and sleeve determine the pressure drop and the required actuator force. Both quantities should be as small as possible for 2/2 directional poppet valves. In order to simplify and accelerate the development of such valves in the future, a mathematical model is being developed that can predict the pressure drop and the flow force based on the valve geometry. The calculation of these two quantities could thus be carried out in a fraction of the time required for a CFD simulation, reducing the development times of new valves. In addition, the knowledge is conserved and thus also made accessible to inexperienced engineers.

The model for estimating the pressure drop and flow force based on the valve geometry is data-based. The data required to create this model is determined using CFD simulations. To ensure that the data from the CFD simulations is as accurate as possible, the simulations are validated in advance with measurement data from various valves.

Recent research concerning the measurement of flow forces elaborates measurement methods for spool valves [1], [2]. This paper focuses on the measurement of industrial 2/2 directional poppet valves that are directly actuated, close leak-free in both directions and permit flow in both directions. Previous works have only examined abstracted and reduced geometries that were specially manufactured for measurement. These studies are outlined in the state of the art in chapter 2. Two different units were developed to measure the poppet force. These are presented and their advantages and disadvantages discussed. The test rig developed for the valve measurement is then shown and the measuring cycle explained. The measurement results are presented and differences in the behavior of normally open (NO) and normally closed (NC) valves as well as the influence of the flow direction on NO and NC valves are worked out. Challenges in the measurements of the two valve types are also discussed. Finally, the knowledge gained about the behavior of NO and NC valves is summarized and concluded with an outlook for future works.

2 State of the art

First, this chapter provides a definition of flow forces for a common understanding. Then, previous research about the influence of different poppet valve geometries on the flow force acting on the poppet is summarized. The literature is divided into research, only including valve measurements and research combining valve simulations with measurements. The focus rests on the test setup, the measurement procedure and the test results.

2.1 Definition of flow forces

Flow forces are forces that occur when a fluid flow changes direction and/or velocity. In order to keep the poppet at its position, further external forces are required to balance the flow forces [3]. Figure 1 shows all forces acting on the poppet of a NO valve in axial direction.

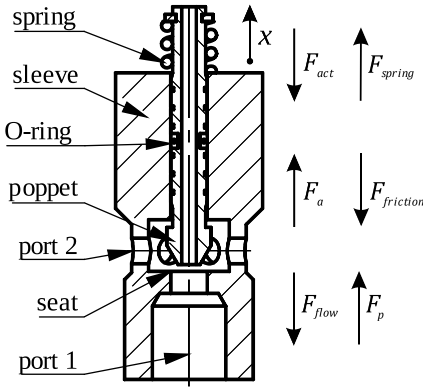


Figure 1: Axial forces on a normally open valve

The poppet is guided in the sleeve and sealed with an O-ring. In the guiding Coulomb friction forces, Newton friction forces and pressure-dependent friction forces caused by the O-ring appear. Hereinafter, all the named friction forces are combined in $F_{friction}$. In a NO valve, the spring opens the valve. The force of the spring F_{spring} is defined by Hooke's law with a pre-load and a position-dependent force. The valve can be closed by the force of an actuator F_{act} which is usually created by a magnet in commercially available valves. Inertia forces F_a are not considered in this work since the valves are not operated dynamically. The flow force F_{flow} results from a combination of static and dynamic forces due to the fluid's momentum, as well as Newton friction forces caused by the flow around the poppet. Static flow forces occur while the poppet is in a static position whereas the dynamic flow forces occur during poppet movement [4]. Since there is no dynamic operation considered in this paper, F_{flow} only contains static flow forces. Furthermore, steady-state and Newton friction forces are merely referred to as flow forces. The pressure forces in axial direction F_p act on every surface that is not parallel to the moving direction. A hole connects the bottom side to the top side of the poppet for pressure compensation. The literature commonly labels the flow direction 1-2 as diverged flow and the flow direction 2-1 as converged flow. This paper uses the numbered label and the convention that a positive flow force acts in closing direction.

2.2 Previous research: Measurements

Especially early research is characterized by measurements and analytical calculations. Stone [5] calculated discharge coefficients and flow forces for an ideal, sharp-edged poppet valve analytically. These calculations were compared to measurements of four different poppet configurations: 45° cone, 45° cone with a shoulder for flow force compensation, 45° sphere and 30° cone. All measurements were conducted at 43°C oil temperature and with an "infinite" volume behind the cone. Additionally, the 45° cone was measured with finite volumes of two different diameters behind the poppet and with the "infinite" volume at 60°C oil temperature. The calculated values were around 25 % below the measured values. It was found that a smaller chamber behind the cone leads to a higher flow force. Furthermore, three different methods for force compensation are discussed: a spring to preload the poppet, a piston with the same diameter as the poppet and a double poppet valve. The test bench used a piston with the same diameter as the poppet to compensate the force induced by the static pressure. The poppet position was controlled by regulating the pressure of the second piston chamber. The flow force was measured indirectly by multiplying the piston area with the pressure in the chamber. A dither was applied by shaking the assembly in order to reduce static frictional forces.

Scheffel [6] investigated the influence of different geometries on the static and dynamic behavior of a direct acting pressure relief valve (PRV) with a cone-shaped poppet. Eight different geometries with cone angles of 30°, 37.5° and 45° as well as different seat widths between 0.7 mm and 2.95 mm were considered. The geometries could be mounted in a variable test valve. The poppet was not pressure-compensated, enabling the attachment of a position and a force sensor directly to the poppet outside the fluid area. The study focusses on the pressure distribution at the poppet with the corresponding valve behavior and not the flow forces. The force sensor was only used to measure the preload of the spring. In the static measurements the Δp/Q-curve was recorded. The dynamic measurements investigated the pressure superelevation caused by the PRV due to a sudden rising system pressure. It was discovered that, with a constant flow rate, both a smaller seat and a smaller poppet angle lead to a larger Δp, while the valve opening was unknown. A mathematical model was developed, describing the correlation between the flow rate and the valve opening depending on the seat width, the poppet angle and the spring pre-load. With this model the valve opening was derived from the stationary measurements. At a constant flow rate, a wider seat and a smaller poppet angle lead to larger valve openings. The results can only give an indirect inference on the flow forces. A larger valve opening with the same external forces would entail lower flow forces.

Wobben [7] extended the investigations on the static and dynamic behavior of PRVs to pilot operated PRVs. The work focuses especially on the flow forces at the poppet of the main stage since these forces influence the dynamic behavior. Three different main stage geometries were tested. The poppets were not pressure-compensated and actuated with a rod which was hydrostatically supported in the housing. The position of the poppet was set by a synchronous cylinder by controlling the chamber pressures. The flow forces on the poppet were measured indirectly by measuring the chamber and the valve pressures and calculating the resulting force with the piston and poppet areas. To address possible friction forces, a dither was applied to the servo valve controlling the chamber pressures. It was found that a sharp-edged poppet with a chamfered seat has lower flow forces than a chamfered poppet with a sharp-edged seat. Furthermore, a larger poppet angle leads to lower flow forces.

Oshima et al. [8], [9], [10] measured the flow forces of five different shaped poppet valves in both flow directions under flow conditions with cavitation using two different methods. The forces on a full-shaped model were measured directly with a load cell and the pressure distribution on the poppet was measured on a half-cut model which could be used to calculate the resulting force on the poppet. Both methods showed good agreement. Possible friction forces in the experimental setup were not taken into account. Valve "A" had a cone-shaped poppet and a chamfered seat with the poppet angle, like in the investigations of Scheffel [6]. Table 1 provides a brief overview of the influences of different parameters on the flow force found by Oshima et al. for poppet "A". Except for a valve with a sharp-edged seat in flow direction 2-1, all flow forces acted in closing direction. The arrows indicate the change of the flow force dependent on the change of the parameter, whereas the tilde represents approximately no change. A general correlation between poppet angle and flow force in flow direction 2-1 was not found, but it could be shown that the magnitude and the behavior of the flow force is strongly dependent on the occurrence of cavitation. The correlation between seat width and flow force is reversed compared to the interpreted findings of Scheffel [6]. This could be due to the fact that the valve opening was derived mathematically and cavitation was not taken into account by Scheffel [6].

Table 1: Influence of different parameters on the flow force

Parameter	Flow direction	Flow force
Valve opening ↑	1-2	$F_{flow} \uparrow$
	2-1	$F_{flow} \downarrow$
Seat width ↑	1-2	$F_{flow} \uparrow$
	2-1	$F_{flow} \uparrow$
Poppet angle ↑	1-2	$F_{flow} \uparrow$
	2-1	$F_{flow} \sim$

The other four investigated valve geometries were based on standard geometries for PRVs at the time. The results of each valve are compared to the results of poppet "A".

Johnston et al. [11] measured flow forces on different shaped poppet and disc valves with a water-based fluid (95:5 water/oil) under non-cavitating, steady-state flow conditions. The poppets were mounted to a rod which was guided in the housing and sealed from its surroundings with a low-friction seal. The force sensor was mounted to the rod on the outside of the housing. Possible friction forces were not considered. Only flow direction 1-2 was investigated. For a sharp-edged seat, the results indicated that a smaller poppet angle leads to a larger flow force while the poppet angle has a bigger influence in large openings. During the tests, different flow patterns occurred while test conditions were maintained. The flow patterns influence the flow force and can also change the force direction. A flow-compensated poppet with an extra shoulder as in [5] was tested and confirmed the results of lower flow forces. The flow forces even assumed negative values. The measurements show discontinuities at the points of changing flow patterns. Investigations on poppets with a chamfered seat showed that the effect of the chamfered seat is neglectable when the opening is large compared to the chamfer.

2.3 Previous research: Simulations and measurements

The more recent research focusses mainly on 2D- or 3D- CFD simulations to investigate poppet valve flow characteristics. In some cases, also measurements were made to validate the CFD simulations.

One of the first people investigating the flow characteristics by measurements and CFD simulations was Dietze [12]. Dietze investigated the flow of simplified poppet valve geometries for static and dynamic operation. It was proven that CFD simulations give the right indication on flow characteristics by comparing a correct and known analytical solution with the numerical solution. Furthermore, a test bench was developed to visualize the flow in the valve. Two valve models scaled with a factor of 2.83 were used. A half cut 2D-model and a full 3D-model. Water was used as the fluid. In the 2D-model plastic particles were added whereas in the 3D-model a color additive was inserted to visualize the flow. The visualized flows match both each other and the simulated flows well. On a second test bench, the flow forces were measured on 4 different valve geometries for flow direction 1-2. The poppets were not pressure-compensated and the flow forces were measured directly with a force sensor. During the setup, the valve was closed first, then the opening position was set with a spindle and then the poppet was clamped at the set position. No seal was used between the poppet and the housing. The friction forces were less than 10 N but not further considered in the evaluation of the flow forces. For a sharp-edged piston with a chamfered seat, it was found, that a larger micro chamfer (range of 100 µm to 200 µm), a smaller opening (with the same flow) and a larger flow (with same opening) lead to a larger flow force. The dynamic investigations showed that the geometry, the opening and the system behind the valve influence flow forces on the poppet as well as the dynamic behavior.

At the same time, Latour [13] investigated four ways of flow force compensation in poppet valves and applied three of them in different valves. On the test bench, the force was measured directly with a rod connecting the poppet and the force sensor. To compensate the force difference caused by the rod guided outside the housing, a second rod was attached to the other side of the poppet and also guided outside the housing. With this setup, the poppet was pressure-compensated like in real valves. Seals were used between the rods and the housing. Possible friction of the seals was not considered in the force measurements. In the investigations, sharp-edged poppets with a chamfered seat were investigated in flow direction 1-2. It was found that a smaller seat angle (range of 15° to 75°) as well as, a larger radius (range of 55 µm to 350 µm) and a larger chamfer (range of 140 µm to 350 µm) at the poppet edge lead

to a larger flow force. A chamfer has higher flow forces than a radius. Both have higher flow forces than a sharp-edged poppet. Methods developed to compensate flow forces are 2D jet deflection, 3D jet deflection, pressure compensation with lowered static pressure and jet angle method with 3D jet deflection. All methods except the second one were applied successfully to different valves. The current state of the art in poppet valves is the 3D jet deflection for NO valves and the 2D jet deflection for NC valves.
Bergada et al. [14] derived analytic formulas for the flow and flow forces for laminar flow conditions in cone shaped poppet valves. The seat had the length of the cone, so that the opening was small compared to the length. It was shown that analytic formulas can be derived for laminar flow conditions.
Simic et al. [4] optimized the geometry of a poppet valve with the aim of lowering flow forces during static operation. The maximum opening was 0.1 mm. The valve was like a spool valve, with the difference that the sealing edge was designed like in a poppet valve. The 2D jet deflection was used to adapt the compensation of flow forces. The influence of five geometrical parameters on the flow forces was investigated via CFD simulations. For one parameter, additional experiments confirmed the accuracy of the CFD simulations with a maximum deviation of 7 %. The spool was not pressure-compensated, and the force sensor was attached directly to the spool with a rod. O-rings were used to seal between the spool and the housing. To take the friction of the seals into account, the force was first measured under static pressure and without flow at all positions. Afterwards, the measurements were repeated at all positions, with flow and a constant pressure difference of 25 bar. The difference in force between these two measurements is the flow force. Every parameter was optimized independently and then combined to an optimal solution.
In summary, previous research only varied single parameters in order to investigate their individual influence on the flow forces. The geometries used for these investigations were mostly simplified and do not occur in currently existing valves. Some experiments took the pressure compensation of the poppet or the friction of seals into consideration, but not at the same time. Mainly NO valves and the flow direction 1-2 were tested.

3 Test setup

This chapter shows the valve and force measuring system as well as the hydraulic test bench developed for measuring industrial poppet valves. Furthermore, the test procedure is explained in detail.

3.1 Valve actuation and force measuring system

One challenge in measuring the forces of the poppet in industrial valves is that the poppets are pressure-compensated. The hole in the poppet transmits the pressure from port 1 to the top side, where the armature pushes on the poppet. The armature is located in the pressurized area. To be able to measure the force on the poppet, the armature, the pole tube and the coil must be removed. In most valves, the pole tube is screwed into the sleeve and can be unscrewed to attach a measuring system. In few valves, the pole tube is welded or pressed to the sleeve and cannot be unmounted. For these valves the pole tube is machined to enable the attachment of a measuring system.
Figure 2 shows a step section view of the actuation and force measuring system developed for industrial poppet valves. The valve shown in the figure is a NC valve in closed position.

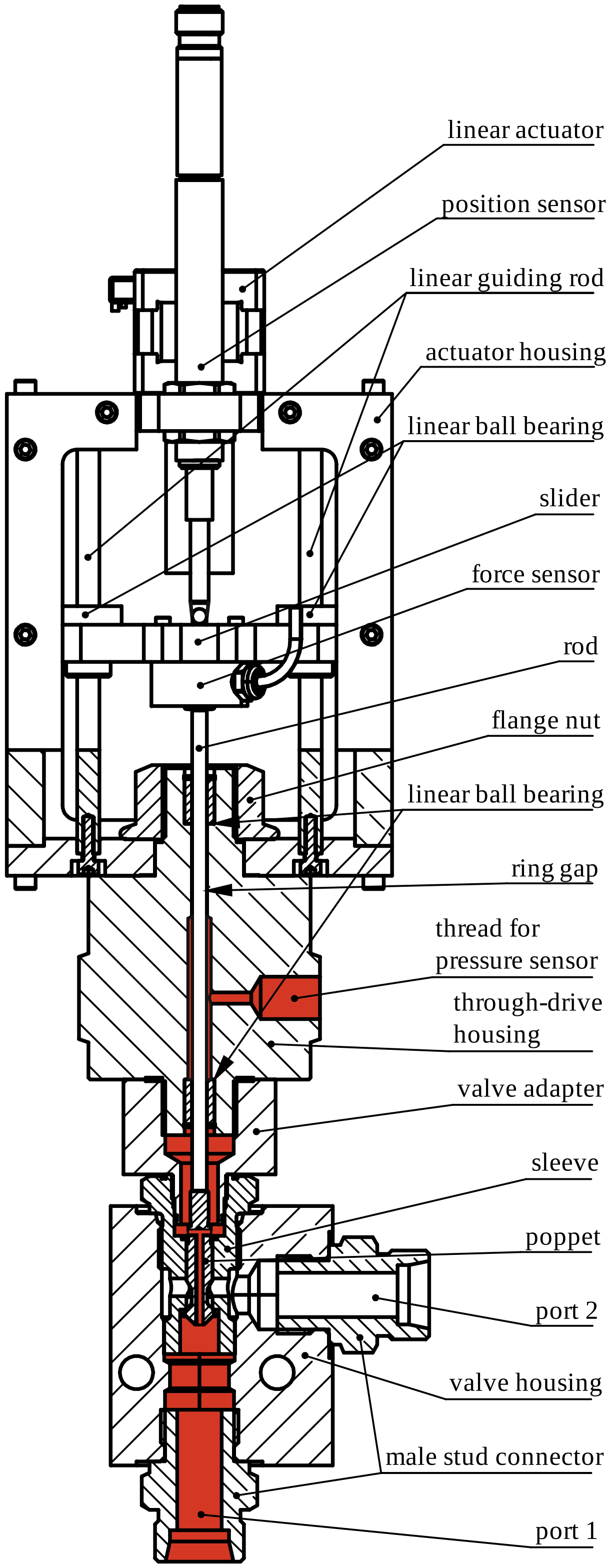


Figure 2: Valve actuation and force measuring system

The pressurized area is marked in red. For simplification, all seals and the spring of the poppet are not shown in the figure. During assembly, the poppet valve is screwed into the valve housing which is connected to the hydraulic circuit via the two ports. The matching valve adapter is screwed into the valve and the through-drive is screwed into the valve adapter. The actuator housing is fit to the through-drive and tightened with a flange nut. The through-drive housing contains linear ball bearings on both sides, guiding the rod that pushes on the poppet. To get a minimal friction force in the through-drive unit, a ring gap with a length of 25 mm is used to seal the pressure to the outside. The fit of the ring gap is 4 mm g6 for the rod and 4.05 mm H7 for the hole. The tolerance chain of the holes for the bearings and the bearings themselves is chosen such that the rod cannot touch the housing. A pressure sensor can be mounted to the unit and measure the compensation pressure of the poppet.
The actuator housing contains two guiding rods. A slider with two linear bearings is mounted on the guiding rods and is freely movable. The force sensor is screwed to the slider. The miniature force sensor HBM C9C, with a maximum force of 500 N and a linearity of 0.2 %, is used. The linear actuator is mounted to the top of the housing in line with the rod of the through-drive unit. It can only apply compressive and no tensile forces to the slider. In fig. 2 the position sensor covers the touching point of the actuator with the slider. The actuator is a Nanotec LGA35 with a maximum force of 242.4 N, a stroke length of 38.1 mm and a resolution of 3 µm per step. The actuator combines a stepper motor with a spindle in one unit. The position sensor is screwed into the housing above the actuator and measures the distance to the slider. The position sensor is a WayCon LVIT with a stroke length of 20 mm and a linearity of 0.5 %.

The maximum pressure all measured valves are rated for is 250 bar. With a rod diameter of 4 mm, the static pressure creates a force of 314 N. The spring forces of the spring opening or closing the poppet as well as the friction forces in the valve were unknown. Due to the size of the original valve coil, the flow forces were expected to be less than 50 N. In general, a force sensor should be selected that has a range equal to the force to be measured in order to obtain the highest possible resolution. In this case the range of the force sensor must cover the force induced by the static pressure, the spring forces and the flow forces, resulting in a force sensor with a capacity of at least 364 N. Consequentially, the measuring range of the force sensor must exceed the flow force that is to be measured by at least a factor of 7. This results in a lower resolution of the flow force measurement due to a lower signal-to-noise ratio. To achieve a more exact force measurement, at first a compensation unit was developed. The compensation unit allows the measurement of the poppet forces without forces induced by static pressure on the rod. It can be installed in the place of the through-drive. Figure 3 shows a section view of the developed compensation unit. The main part of the unit is the spool with three different sized shoulders. The two outer shoulders have a diameter of 8 mm, while the inner shoulder has a diameter of 11.3 mm. Each shoulder has a guided length of 10 mm and contains three pressure compensation grooves. Two of the shoulders are guided in the housing and the third shoulder is guided in the top cover. Not pictured in the section view are three M8 screws fixing the top cover at the housing. To reduce torque on the spool by possible lateral forces, the spool is as short as possible and two rods guided in linear ball bearings are located on both sides of the spool. The rods are rounded on the spool side. A pressure sensor can be screwed to the unit to measure the compensation pressure of the poppet. A leakage pipe can be connected to the unit to catch the leakage at ambient pressure. To explain the functionality, the pressurized area is colored red. The lower rod is pressurized from both sides so that no pressure forces arise. The spool seals the pressure to the outside and has three shoulders to compensate pressure forces. The working principle is that the pressure force acts on the lower shoulder in x direction. A force of the same magnitude occurs in the opposite direction, between the upper 8 mm shoulder and the 11.3 mm shoulder, since the differential area of the two shoulders equates the area of the small shoulder. This allows the measurement of the poppet forces with a more precise force sensor, characterized by a smaller range.

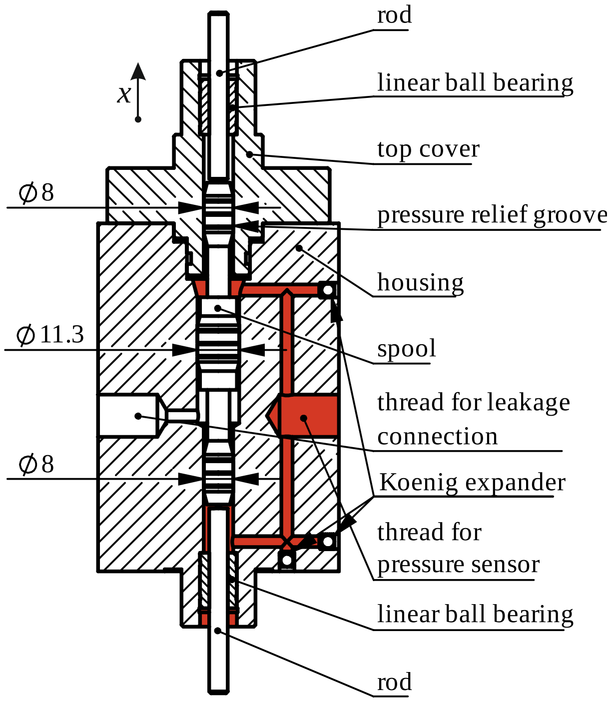
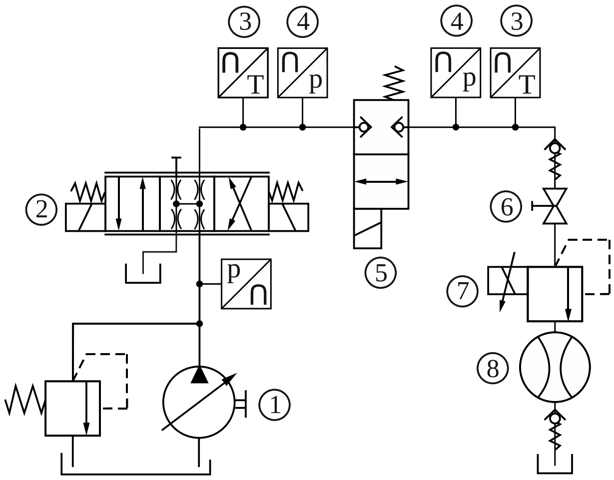


Figure 3: Compensation unit for static pressure forces

Tests showed that the unit works and pressure forces are compensated. A problem that occurred was that the spool jammed at certain positions. The jams could be seen in discontinuous force measurements. The peaks in the force measurements became larger with higher pressure in the compensation unit. Measurements also showed different pathways (force over poppet position) in two runs, when the pressure was changed between the runs. This means that measurements cannot be taken reliably and repeatedly under the same conditions and the compensation unit is unsuitable for measuring flow forces in this application. Possible reasons for the jamming are that the spool is not free from torque or that the housing deforms under pressure. However, the most likely reason is the double overfit of the spool. From a manufacturing perspective, it is generally not optimal to guide more than one diameter on a spool since it leads to an overfit. This overfit exists between the spool and the housing. An additional overfit exists between the spool and the top cover. The top cover itself is fit to the housing. All dimensional, shape and position tolerances added lead to a non-optimal guiding and a jamming of the spool. Summarized, it can be said that the concept of pressure compensation works but for a reliable compensation unit an even more precise manufacturing or a different concept is required. For all measurements presented in this paper, the setup shown in fig. 2 is used.

3.2 Hydraulic test bench

A hydraulic test bench for measuring the poppet valves was developed and built. Figure 4 shows the hydraulic circuit diagram of the test setup.



① central pressure supply
② 4/3 directional proportional valve
③ temperature sensor
④ pressure sensor
⑤ test poppet valve
⑥ ball valve
⑦ proportional pressure relief valve
⑧ flow rate sensor

Figure 4: Hydraulic diagram of the test bench

The experimental setup is connected to the central pressure supply (1) of ifas. The pressure supply is set to a flow rate of 80 l/min with a temperature of 40°C. Both a temperature sensor (3) and a pressure sensor (4) are located in front of and behind the tested industrial valve. The temperature sensors are Danfoss MBT 3270 as PT1000 elements. As pressure sensors, Hydac HDA 4700 series sensors with a measuring range of 0 bar to 400 bar and an accuracy of 0.25 % were used. The distance from the pressure sensor to the inlet of the valve housing is 150 mm. From the outlet of the valve housing to the second pressure sensor, the distance is 230 mm. The distance between the temperature sensor and the pressure sensor is 60 mm. The 4/3 directional proportional valve (2) is a Bosch Rexroth 4WREE10-3X and is used to control the differential pressure over the test valve. The valve has a negative overlap. The proportional pressure relief valve (7) is an Argo-Hytos SR4P2-B2 and is used to control the outlet pressure of the test valve. The flow rate sensor (8) VSE VS2, with a measuring range of 0.1 l/min to 120 l/min and an accuracy of 0.3 %, measures the flow rate through the test valve. The ball valve (6) is used to stop the flow through the test valve to conduct measurements under static pressure. The hydraulic circuit diagram in fig. 4 is simplified. At the real test bench, a second and equal measurement string exists. It is connected to the closed port of the 4/3 directional proportional valve (2) and ends with a ball valve (6) in front of the proportional pressure relief valve (7). The second measurement string enables the user to change faster between valves for measurement. To change the measurement string, the 4/3 directional proportional valve is actuated in the opposite direction. The flow direction through the test valve can be changed by turning the housing of the test valve on the test bench. All signals are recorded with a frequency of 1000 Hz.

3.3 Test procedure

This chapter explains the general assumption that was made to measure the flow forces and describes the test procedure that was used for every valve in both flow directions. The idea is that the actuator force F_{act} can be measured with the previously described actuation and measuring unit. The force is equal to the sum of the four forces F_p, F_{spring}, $F_{friction}$ and F_{flow}. With just one measurement it is not possible to differentiate between the four forces. The assumption is that F_p, F_{spring} and $F_{friction}$ are the same with and without flow. With flow the flow force F_{flow} is added to the system. So, the difference between the measured force with flow and without flow must be the flow force. A restriction is that F_p and $F_{friction}$ depend on the pressure. This is addressed by taking measurements at different pressure levels. Furthermore, the friction coefficient is different when the poppet is at a static position or moves. Static friction would add uncertainty to the measurements. Therefore, the poppet should move at a very slow speed during the measurements to only have dynamic friction. So, all measurements are taken with a moving poppet in quasi-static operation.
Figure 5 shows all steps after mounting the valve until all measurements in one flow direction are finished. The first step is to measure the angle of the valve in the housing. During manufacturing, the start of the thread on the valve is random since it is a turned part. The start of the thread determines the positioning of the radial holes in the sleeve to port 2 in the valve housing. This position is later important for the CFD simulation because the valve angle should be included in the model. Thereafter the ball valve (6) is opened and the pressure supply is started in order to heat all components of the test bench. For this, the test valve is opened and a tank pressure p_t of 75 bar and a pressure difference Δp of 15 bar to 30 bar, depending on the test valve, is set. When the temperature in front of the test valve reaches 45°C, the pressure supply is stopped.
After this, the actuator is calibrated. During the calibration, the closed position of the valve is set in the test bench control. The actuator is moved slowly downwards to push on the poppet with a constant velocity. In the force over time plot, at first, a steep rising of the force can be seen until the spring pre-tension is reached. This is followed by a flatter zone with a linear increase defined by the spring stiffness. After this zone, a second peak can be seen. For NO valves this point is the closed

position. For NC valves, this point is the maximum technically possible opening position which, however, differs from the maximum opening position during operation. The maximum force is limited between 45 N and 95 N, depending on the valve. The closed position of NC valves is at the transition from the first steep rise to the flatter area.

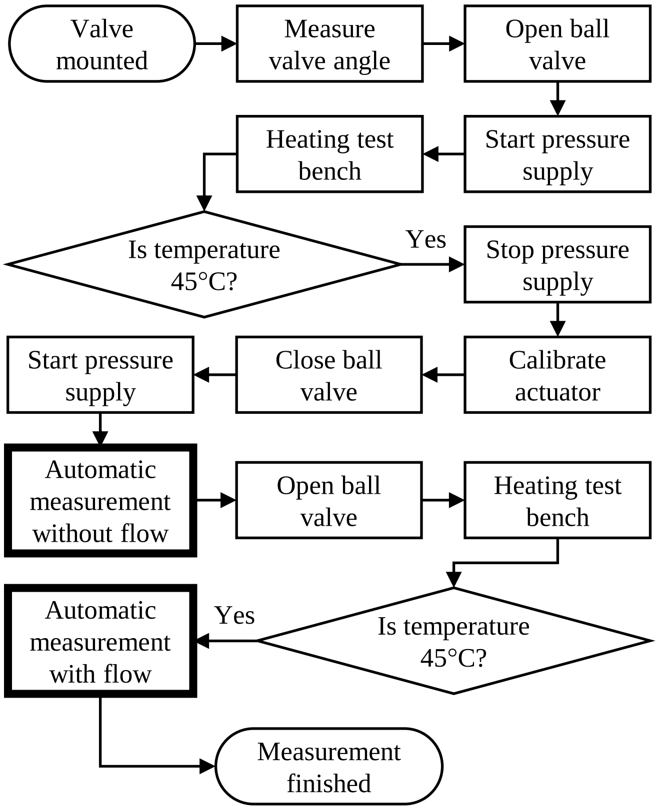


Figure 5: Measuring cycle for one valve in one flow direction

After the calibration, the ball valve (6) is closed and the pressure supply started again to begin with the measurements without flow. Figure 6 shows the measurement cycle without flow in detail. For the measuring cycles a maximum and minimum opening position is defined dependent on the valve. The minimum opening position lies between 2 % and 20 % and the maximum opening position between 60 % and 100 % of the overall stroke. The minimum positions are chosen so that the valve opening is at least 0.1 mm. The maximum opening position is 100 % for all NO valves, while the maximum opening position had to be limited for the NC valves to ensure that the valve can close by itself in all tested operating conditions. This phenomenon is discussed later in this paper. During the cycle, at first the poppet is moved to the maximum opening position and the pressure is set to 10 bar. When the pressure is reached, the poppet is moved to the minimum opening position and then again to the maximum opening position with a velocity of 0.05 mm/s. Before and after each movement, a waiting time of 5 s is included. After the moving cycle, the pressure is raised in 5 bar steps to 60 bar and then in 10 bar steps to 200 bar. Thereafter, the measuring cycle without flow is finished and the ball valve (6) is opened. Then the test bench is heated again to 45°C and the automatic measuring cycle with flow started. Figure 7 shows the cycle in detail.

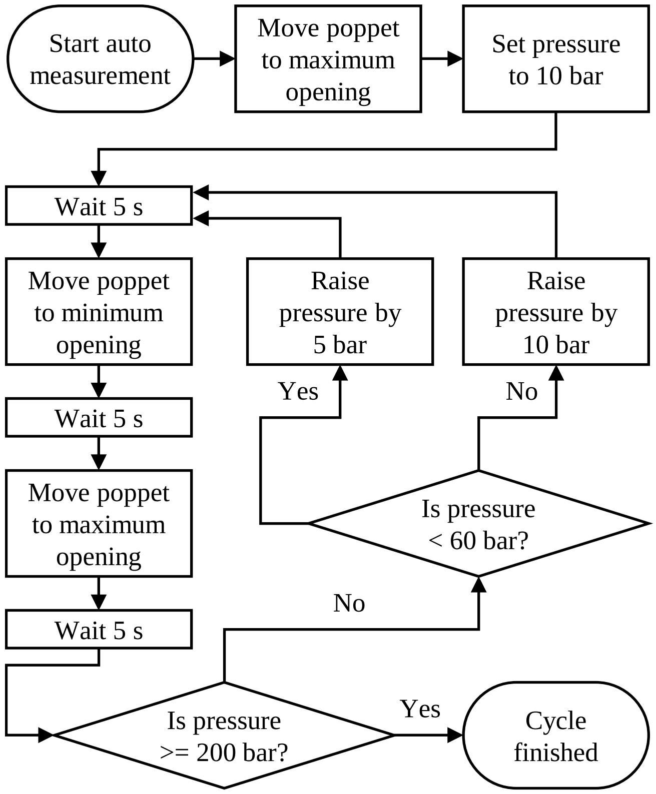


Figure 6: Cycle for automatic measurement without flow

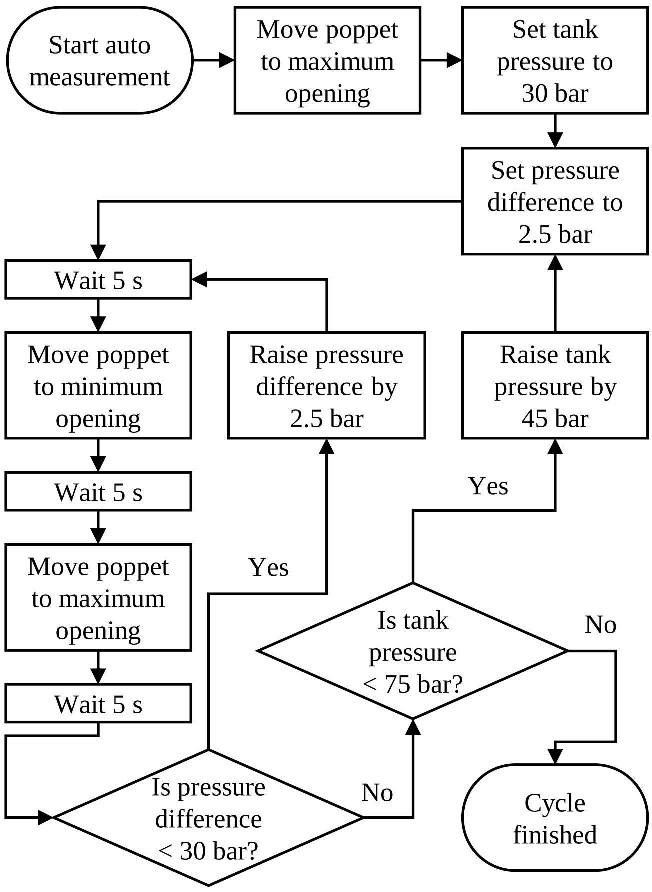


Figure 7: Cycle for automatic measurement with flow

At first, the poppet is moved to the maximum opening position. Next, the tank pressure is set to 30 bar and the pressure difference over the test valve is set to 2.5 bar. When both pressures are reached, the poppet is moved to the minimum opening position and then to the maximum opening position with a velocity of 0.05 mm/s. Before and after each movement, the position was held for 5 s. After each moving cycle, the pressure difference is raised by 2.5 bar until 30 bar is reached. The whole procedure is repeated with the tank pressure of 75 bar. Thereafter, all measurements for one valve in one flow direction are finished.

4 Results and discussion

The first step in the evaluation is to create interpolation (IP) models for the force, based on measurements without flow. For each valve and flow direction, one interpolation model for the opening and one IP model for the closing direction are created. The IP models can predict the actuator force without flow, dependent on the poppet position x and the compensation pressure p_{comp} measured in the through-drive. These IP models are used together with the measured force with flow to calculate the flow forces. For NO valves, the measured force with flow is subtracted from the force predicted by the IP model. For NC valves, the terms of the subtraction switch. This calculation is chosen to obtain the definition of positive flow forces acting in closing direction.
To create the IP model, only the areas of poppet movement from the measurements without flow are used. For the predictors x and p_{comp} continuously rising or falling signals are needed. Since the measured position signal has noise and does not fulfill the criteria, it cannot be used unfiltered. Therefore, a moving average with a range of 0.3 s was applied on x. For all other signals, moving averages with a range of 0.1 s were used. Figure 8 shows the interpolation models in opening and closing direction for a NO valve.

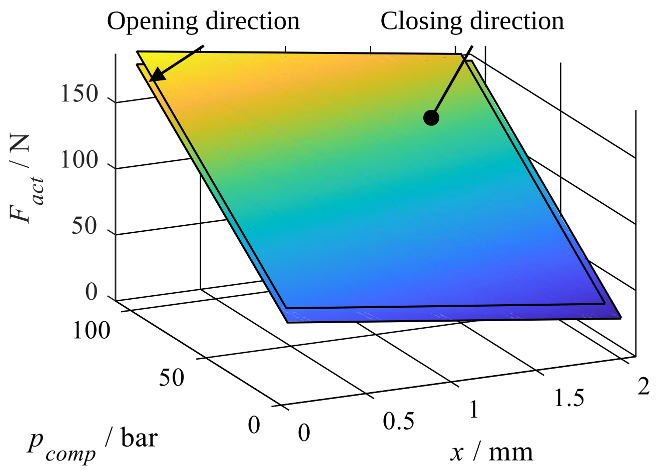


Figure 8: Two interpolation models for a NO valve in closing and opening direction

Both interpolation models are planes that are approximately parallel. The distance between the planes is hysteresis, induced by the friction of the poppet and the through-drive. In the figure, the plane from the closing direction nearly covers the plane from the opening direction completely. The hysteresis for this NO valve lies between 2.8 N and 9.9 N, depending on the compensation pressure and the poppet position. For NO valves at constant pressure, the force rises with a falling position due to the spring. At a constant position the force rises with the pressure due to the pressure acting on the rod of the through-drive. The same behavior under rising pressure can be seen at NC valves. For NC valves at constant pressure, the force rises with a rising position due to the spring. So, in x direction the gradient of the planes would be positive.

With the measurements made, different effects such as the influence of the tank pressure or the flow rate on the flow force can be evaluated for each valve. An influence of the tank pressure on the flow forces could not be seen in any measurements. A higher flow rate, respectively a higher pressure difference, lead to an amplification of the flow forces in general. This influence is not analyzed further in this paper. The aim is to gain a general understanding of flow forces in both flow directions in industrial poppet valves. Therefore, only the measuring cycle at a tank pressure of 30 bar and a pressure difference of 15 bar is shown and discussed in this paper. 15 bar is the maximum pressure difference at which all valves could be measured.
Figure 9 shows the measured flow force and flow rate over the poppet position for the five tested NO valves in both flow directions in opening (op.) and closing (cl.) direction. In flow direction 1-2, the maximum flow forces are between 3.5 N and 24.1 N, depending on the valve. The flow forces are proportional to the maximum flow rate. For valve 5, the maximum opening is not limited by the geometry, which means that this valve has the largest stroke. The behavior of the valves changes in flow direction 2-1. The valves 2, 3 and 5 have an almost constant flow force of 0 N to 4 N over the stroke. Valve 4 shows the same qualitative curve as in direction 1-2 but shifted downwards and flatter. The maximum flow force of this valve is 12 N. Valve 1 only has negative flow forces of up to -26.8 N. This means that the flow forces act in the opening direction. Valve 1 has a 20 % greater flow rate and valve 4 has an 80 % greater flow rate than in the direction 1-2.

Figure 10 shows the measured flow force and flow rate over the poppet position for the four tested NC valves in both flow directions in opening (op.) and closing (cl.) direction. In flow direction 1-2, the maximum flow forces are between 4.6 N and 16.7 N, depending on the valve. The flow force curves are qualitatively the same for all valves. In the first section, the force increases, reaches a maximum and then decreases again. For valves 1, 2 and 4, the maximum flow force is at the same position as the maximum flow rate. The fact that the flow force and the flow rate reach a maximum and then fall again when the valve is opened further is because the smallest cross-section is no longer at the sealing edge, but on the side of the radial holes in the direction of port 2. In this case, the flow force is partially negative and acts in the opening direction. When the opening or the pressure difference gets too large, the valves do not close on their own anymore. The poppet stays at an open position since the actuator cannot pull on the poppet. Then the pressure difference must be lowered. This is a state that does not occur in an original valve since the original magnet cannot open the valve that wide. Thus, preliminary investigations were made to find the maximum pressure

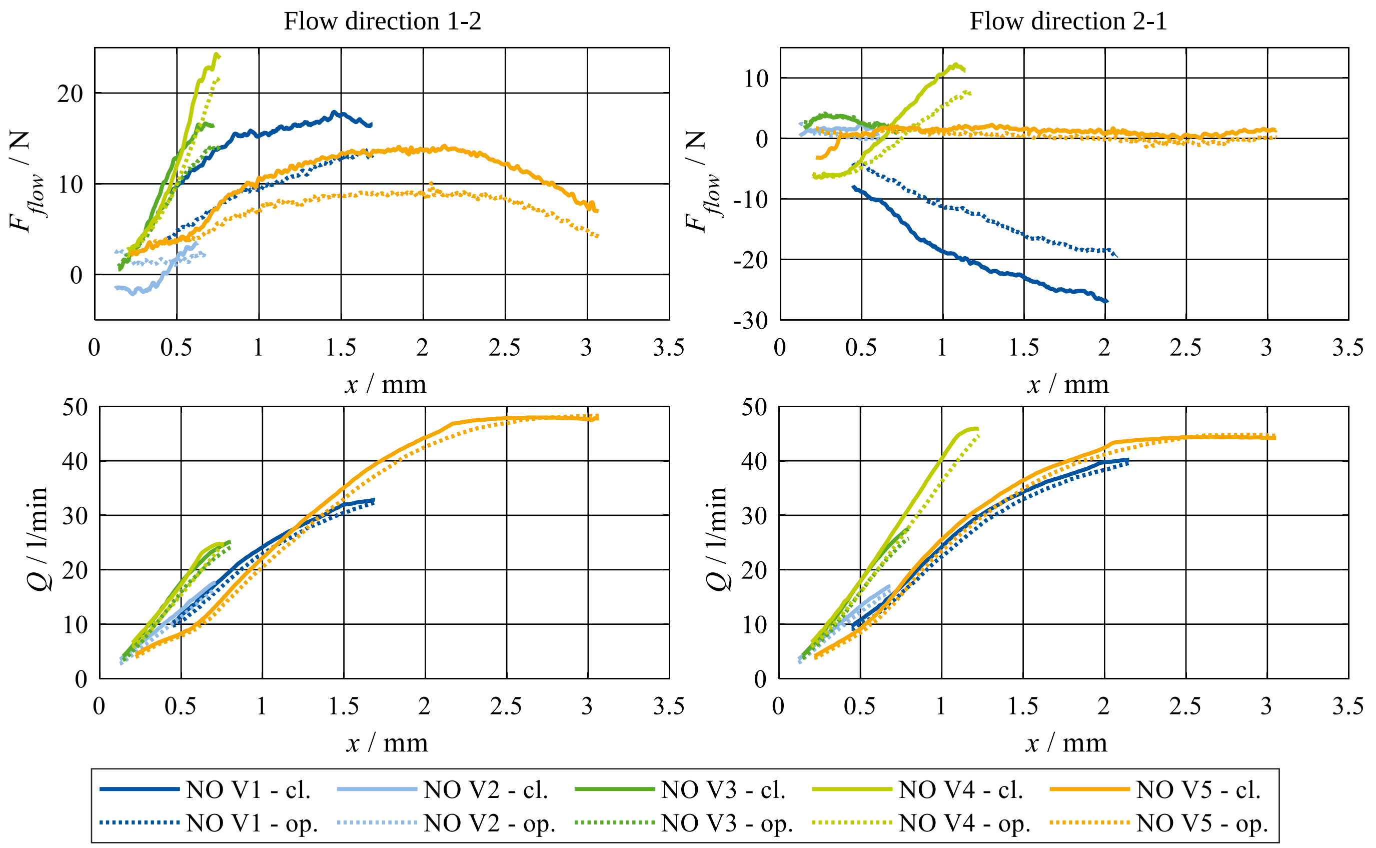


Figure 9: Flow force and flow rate over the poppet position for 5 NO valves at p_t = 30 bar and Δp = 15 bar

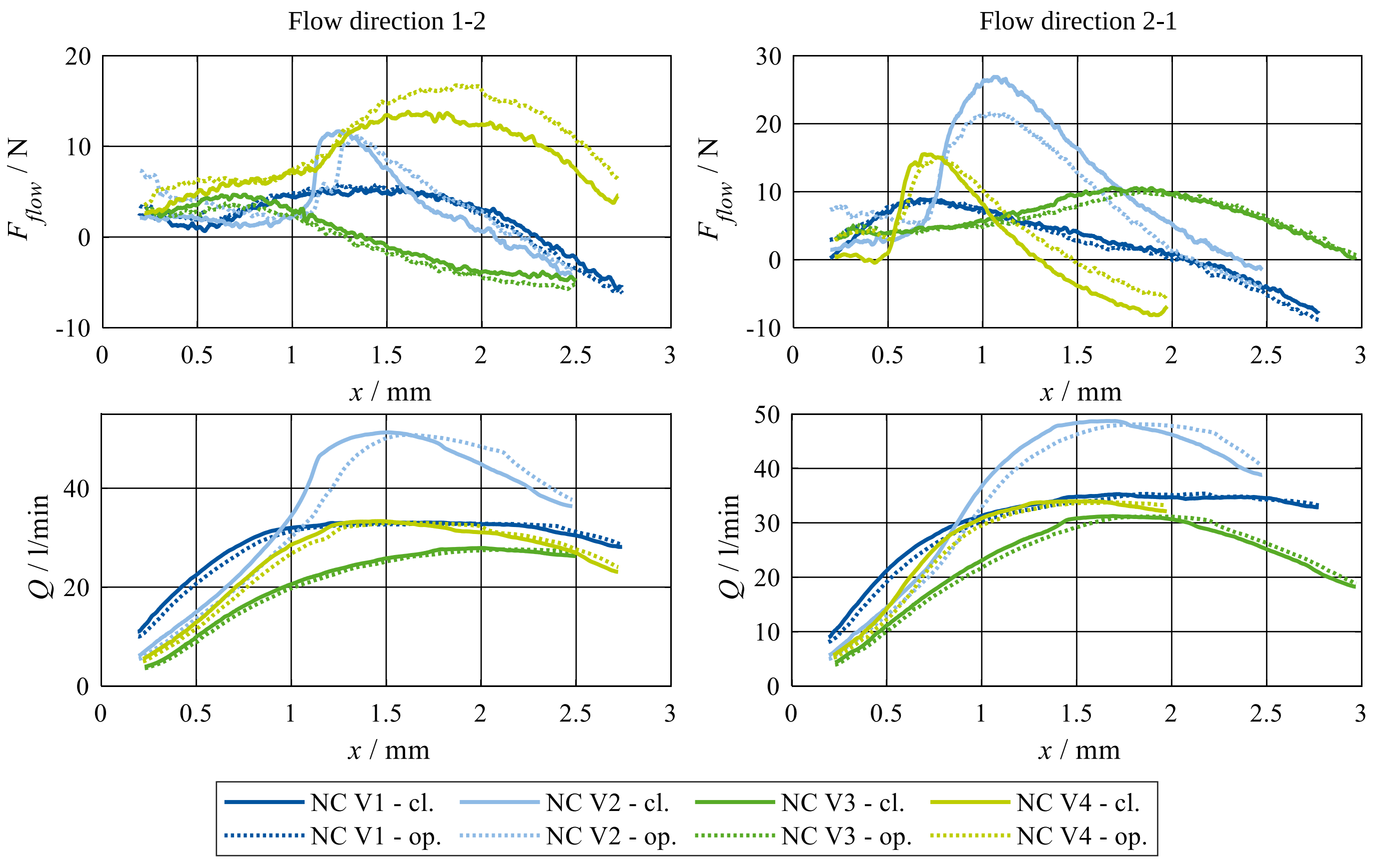


Figure 10: Flow force and flow rate over the poppet position for 4 NC valves at p_t = 30 bar and Δp = 15 bar

difference and opening width for the automatic measuring cycle. In flow direction 2-1, the curves of the flow force and the flow rate match qualitatively to the curves in flow direction 1-2. The peaks in flow force occur in two valves at lower positions and in one valve at a greater position. However, the maximum flow force of three valves is greater by a factor of 2 to 2.5 and reaches a maximum of 26.9 N. The flow rate also remains quantitatively unchanged.
Summarized, it can be said that in NO valves, the behavior of the flow forces changes with the flow direction. In one NO valve, the flow force changes the effective direction. The flow rate tends to be higher in flow direction 2-1. In NC valves, the flow direction does not influence the behavior of flow forces as much. The flow rate is qualitatively and quantitatively the same. The peak of the flow forces changes its position and tends to be higher in flow direction 2-1. Over all measurements made, the highest flow force measured was below 35 N.

Hereinafter, other characteristics that could be observed during the tests and the effect of not using the compensation unit will be discussed. One characteristic that can be seen in fig. 9 and fig. 10 is that the flow force in opening direction is different to the flow force in closing direction over a large range of the poppet position in almost all valves. The expectation is that the flow force is the same during opening and closing since hysteresis due to friction is considered in the IP model. Hence, during the measurements with flow, other forces must occur, or the flow pattern changes during the movement. One indication for the change of flow patterns is that the flow forces are equal in a specific range and then split up (e.g. NO valve 1). The change of flow patterns was investigated amongst others by Masuda et al. [15] and shows the same behavior. The most probable different force is a changing friction force. Due to the flow, an uneven pressure field can occur around the poppet. This leads to a slight tipping of the poppet in the sleeve and higher friction forces that are not considered in the IP model. This theory must be investigated by CFD simulations in the future.
All measurements were conducted as a basis for the validation of CFD simulations. The measurements were made with a moving poppet even though the valves are switching valves and the CFD simulations should be stationary. The aim is to preferably simulate the real operating point of the valve. Therefore, the poppet position during real valve operation must be found. For NO valves, the real operating point can be seen easily in measurements since the rod of the through-drive lifts from the poppet when the poppet is in the force equilibrium and consequently in the real point of operation. In NC valves, the force equilibrium forms together with the force of the magnetic actuator. This force is unknown thus the real point of operation is unknown in NC valves. So, assumptions based on measurements must be made to select one operating point for the validation of CFD simulations for NC valves.

All measurements were made using the through-drive unit and not the compensation unit. Therefore, a force sensor with a maximum force of 500 N had to be used. The resolution of the Beckhoff EL3351, used for data acquisition of the force sensor, is 16 bit. The typical noise measured is 0.05 % of the full-scale value. Using the 500 N sensor, this results in a noise of 0.25 N. The flow forces in fig. 9 and fig. 10 are in a wide range larger than 5 N. In these areas, the signal-to-noise ratio is larger than 20, which is accurate enough. The ability to use a 50 N sensor in combination with the compensation unit would lower the noise to 0.025 N, so that the signal-to-noise ratio is increased by a factor of 10. This would increase the accuracy of the flow force measurements below 5 N. Knowing that the real point of operation of the valves is in the areas of larger flow forces, the measurement method using the through-drive and the 500 N force sensor is valid and accurate enough to validate CFD simulations. Three NO valves in flow direction 2-1 show flow forces around 0 N. These points of operation need special attention when using the measurement data for validation.

5 Conclusion and outlook

The presented valve actuation and force measuring system with a through-drive can be used to measure flow forces. It was attempted to replace the through-drive with a unit that compensates the static pressure. This would have allowed to use a force sensor with a smaller measuring range and higher accuracy. The compensation unit worked, and the static pressure was compensated. Unfortunately, the spool in the compensation unit jammed at certain positions. The most likely reason for the jamming is the double overfit of the spool at the two shoulders in the housing and the one shoulder in the top cover. Since the accuracy of the force measuring system with the through-drive is high enough to validate CFD simulations, the compensation unit is not further improved and considered.
The flow forces are determined by two automatic measuring cycles. In the first cycle the poppet is moved at specific pressure levels up to 200 bar without flow. During the second cycle, the valves were flown through and the tank pressure as well as the pressure difference over the test valve controlled while the poppet was moved. The movement of the poppet was used to avoid static friction forces and the speed of 0.05 mm/s to achieve quasi-static measurements. The tested tank pressures were 30 bar and 75 bar and the maximum pressure difference was dependent on the valve between 15 bar and 30 bar. To determine the flow force it was assumed that, between the two cycles, all forces are identical except for the flow force. So, the difference in the measured force from both cycles must be the flow force. To calculate the flow force, interpolation models were created that can predict the force without flow based on the poppet position and the compensation pressure.
Five NO valves and four NC valves were measured. The results for the flow force and the flow rate over the poppet position were presented in both flow directions for the tank pressure of 30 bar and the pressure difference of 15 bar. The maximum flow force from all tested NO valves was 24.1 N. For NO valves it was found that the flow direction changes the behavior of the flow forces for most valves. In one valve, the flow force changed the effective direction to -26.8 N. For the NC valves the maximum flow force is 26.9 N. The flow force behavior is compared between the four NC valves qualitatively the same in both flow directions.
In future work, the measurements can be used to validate CFD simulations of the valves. With a validated CFD simulation,

the database for the prediction model of the flow forces and the pressure drop can be generated. The prediction model can be validated with measurements on further valve geometries using the valve actuation and force measuring system, the test bench and the test procedure presented in this paper.

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Nomenclature

Designation	Denotation	Unit
Δp	Pressure difference	bar
F_a	Acceleration Force	N
F_{act}	Actuator Force	N
F_{flow}	Flow force	N
$F_{friction}$	Friction force	N
F_p	Pressure force	N
F_{spring}	Spring force	N
p_t	Tank pressure	bar
Q	Flow Rate	l/min
x	Poppet position	mm

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