Simulation-Based Design of a Direct-Operated Proportional Pressure Relief Valve

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Abstract

The development of pressure relief valves does not only put requirements on pressure-flow-characteristics, but also requires the maintenance of sufficient dynamics and stability. To accomplish the task the simulation methods support the designing stage and optimization process of the functional subsystems of a direct-operated proportional pressure relief valve. Therefore, an inverse simulation model is built for this valve type and CFD and FEM modeling techniques are applied to parameterize the control edge and solenoid performance. A final comparison between measurement and simulation results demonstrates the applicability and accuracy of used simulation methods for the valve design process.

KEYWORDS: holistic valve simulation, pressure relief valve, CFD, FEM

1. Introduction

Electro-hydraulic valves fulfill important control and safety functions in hydraulic systems and facilities. The above-mentioned pressure relief valves are usually used for pressure limitation or pressure setting. Direct-operated valves are applied to small volume flow applications, while pilot-operated ones are predominantly used for higher volume flow applications. The reason for using pilot-operated valves is the fact that necessary actuator- and counter forces, respectively, increase with rising nominal volume flow. However, application of proportional- or solenoid-controlled valves allows, in contrast to spring-loaded ones, a continuously variable remote adjustment of setting pressure. A direct-operated proportional pressure relief valve, which is closer examined for this article, is outlined in detail in **Figure 1**. Its functionality is essentially characterized by the hydraulic properties of the combination valve seat-valve spool, as well as the displacement-dependent force effects of the solenoid-spring combination. A well-directed geometrical variation of both subsystem designs makes it possible to change and predict the valve's functionality.

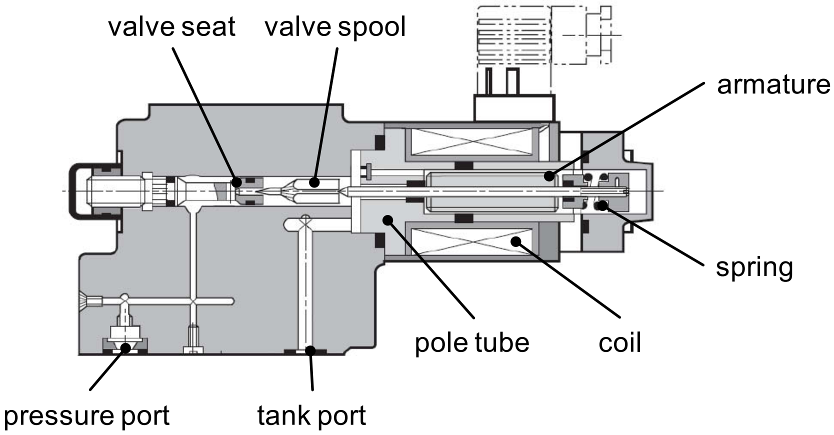

Figure 1: Build-up of a direct-operated proportional pressure relief valve

The physical-based description of pressure relief valves and its subsystems was already part of previous scientific research. The wide range of contents covers detailed mathematical explanations as well as complex 3-dimensional computational fluid dynamics. The mathematical analysis of a direct-operated valve /1/ reveals the fundamental relations and dependences, but the simplified mathematical model does not take the exact geometrical shape of the flow region into consideration. The extension of that initial methodological approach, about a parameter extraction by implementation of computational fluid dynamics (CFD), broadens its range of application. There is now the possibility of carrying out a failure analysis based on geometrical tolerances of the flow region /2/. An integral consideration of all physical domains including the proportional solenoid /3/ completes the methodological approach for investigation of such valve systems. The described methods have a common property: Each focuses on an analysis of existing systems and a validation of the methodological approach. This article takes up these ideas, while extending the established methodologies and adopting them to the synthesis of static valve performance.

2. Valve performance simulation

The static and dynamic properties of a direct-operated proportional pressure relief valve are characterized by the fluid flow through the component and the force effects of the displacement-dependent solenoid-spring combination. Therefore, the valve has a strong mechatronic character, although the electronics is not part of the investigations. **Figure 2** describes the considered subsystems and the interactions among each other.

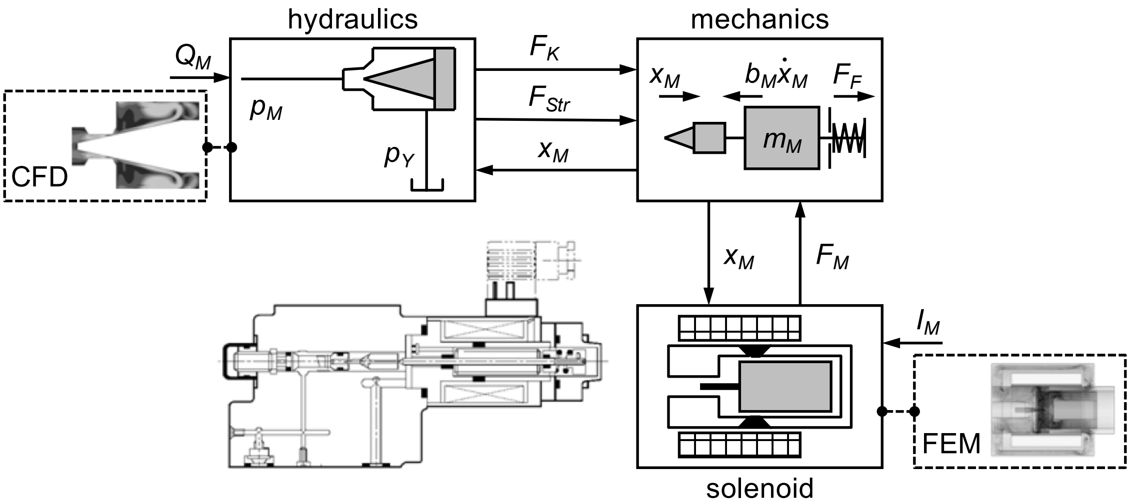

Figure 2: Interactions between the several physical subsystems

Hydraulic subsystem. The description of irreversible energy losses is considered in the hydraulic subsystem. In case of pressure relief valves the energy losses occur predominantly at the control edges. The loss coefficient ζ is calculated with equation (1) according to /4/. Therein $\Delta p = p_M - p_Y$ stands for the total pressure loss over a given system length, averaged over mass flow rate. This pressure difference is applied to the dynamic pressure of the closest flow cross-section, volume flow Q_M and cross-section area A_{Str}.

$$\zeta = \frac{p_M - p_Y}{\frac{\rho}{2}\left(\frac{Q_M}{A_{Str}}\right)^2} \tag{1}$$

With the knowledge of pressure losses the static pressures as well as the force reactions on single boundaries of the fluid region can be calculated. As the fluid is accelerated in proximity to the control edge, it is necessary to adjust equation (2) for the force reaction on the valve spool. Therefore, the force reaction F_{Spool} is split into a theoretical pressure force F_K and a corrective term F_{Str} called flow force.

$$F_{Spool} = F_K - F_{Str} \tag{2}$$

The equation for the flow force F_{Str} can be determined by the conservation of momentum. After some transformations and with the conservation of mass, follows equation (3), in which k_{GF} represents a control edge-dependent geometrical coefficient according to /5/.

$$F_{Str} = \frac{\rho Q_M^{\ 2}}{A_{Str}} k_{GF} \tag{3}$$

Mechanical subsystem. The mechanical subsystem combines all force effects in the equation of motion (4), on that basis the time-dependent displacement of valve spool x_M and solenoid armature is calculated. The difference between solenoid and spring force $F_M - F_F$ expresses the current- and displacement-dependent counterforce.

$$m_M \ddot{x}_M + b_M \dot{x}_M - F_K + F_{Str} + F_M - F_F = 0 \tag{4}$$

Frictional and damping forces are present, but are integrated in a simplified manner by the viscous friction coefficient b_M. The reason for simplifying is that mainly the static valve performance is significant in this article. In case of a stationary state the derivatives with respect to time $\partial/\partial t$ are of no importance.

Solenoid subsystem. The solenoid outlined in **Figure 3** can be regarded as an electromechanical transformer which converts electricity into mechanical energy. Therefore, the coil is set under a current I_M, which causes a magnetic flux in the magnetic circle. An attractive solenoid force F_M arises in the slight gap between armature and pole tube whose current- and displacement-dependence is influenced by the geometrical design, as well as the non-linear material properties of the acting components.

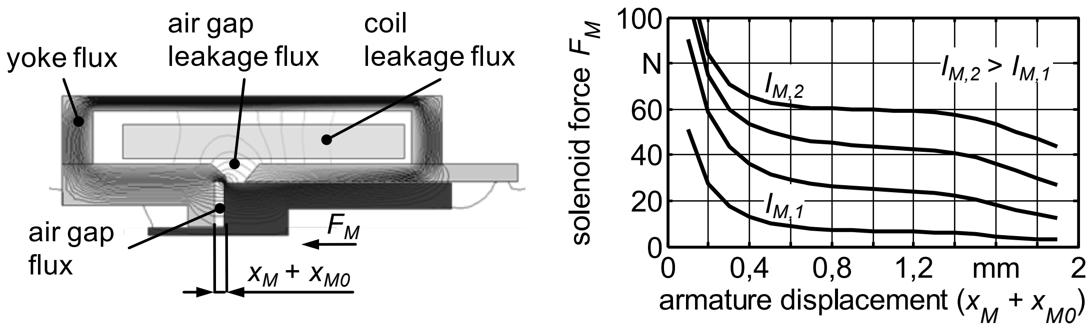

Figure 3: Magnetic flux distribution and force-displacement-curves of a proportional solenoid

For further investigations, the solenoid's force-displacement curve is implemented in the simulation model whose characteristic curves depend on the magnetic flux distribution. Reluctance-networks are not used, because they are not applicable for the functional design on the basis of geometrical configuration adjustments. The time-dependence of the solenoid force is not part of this article. The solenoid dynamics of the initial configuration is sufficient for the treatment of pressure relief valves and is hardly influenced by geometrical design adjustments at the air gap.

3. Design of pressure-flow characteristics

Besides a sufficient valve dynamic the valve's static characteristic is the functional criterion for characterizing these types of valves. This performance is achieved by a complex interaction between the different physical effects explained in short in Figure 2. The resulting forces from these interactions characterize the state triple consisting of volume flow, pressure difference and valve spool displacement. The description of this state triple allows a mathematical relation between the system inputs, the volume flow Q_M, as wells as the solenoid current I_M and the counterforce $F_M - F_F$.

Starting point for the mathematical description of static characteristics is the equation of motion (4) neglecting any time-dependent derivatives $\partial/\partial t$. The pressure force F_K is substituted with the pressure difference Δp and the pressure area A_K, and if $p_Y \approx 0$ bar is assumed, then the flow equation (1) and force equilibrium (2) can be merged. Please note that for the description of the hydraulic subsystem the coefficients ζ and k_{GF} are generally dependent on the current operating point. Therefore, a representation with respect to both state variables Q_M and x_M is necessary. A representation with respect to Reynolds' number Re_M allows an incorporation of fluid properties at this point, too.

$$Re_M = \frac{4\,Q_M}{U_{Str}\left(x_M\right)\dfrac{\eta}{\rho}} \tag{5}$$

In addition to the wetted perimeter U_{Str} this equation additionally contains the dynamic viscosity η as well as the density ρ. The above-mentioned combination of equations, considering the Reynolds' number Re_M, leads to a transcendent equation (6) whose solution can only be computed numerically.

$$Q_M{}^2 = \frac{F_M\left(x_M, I_M\right) - F_F\left(x_M\right)}{\zeta\left[Re_M\left(x_M, Q_M\right)\right]\dfrac{\rho}{2}\dfrac{A_K\left(x_M\right)}{A_{Str}\left(x_M\right)^2} - \dfrac{\rho}{A_{Str}\left(x_M\right)}k_{GF}\left[Re_M\left(x_M, Q_M\right)\right]} \tag{6}$$

Equation (6) forms the basis of all following considerations. With this equation, an implicit relationship between the volume flow Q_M, the control edge coefficients ζ as well as k_{GF} and the solenoid force F_M is defined. The pressure p_M immediately arises from back substitution of both state variables Q_M and x_M into the equation of motion. Variations of the geometrical design change the coefficients and lead to a new system performance. In the present case these dependences are used in a straightforward manner to simulate a chosen system performance through geometrical design adjustments.

3.1. Inverse simulation model

For the realization of a characteristic design the mathematical problem has to be expressed in an inverse manner. Initial and target values are in this case represented by the static pressure characteristic p_M = f(Q_M) for I_M = const. However, these inputs only contain the state variables Q_M and p_M, but for a complete description the state triple is necessary. The mathematical solution takes place in two separate steps. Initially, the flow relation (1) is transformed into a root problem (7). Equation (7) maps the valve spool displacement x_M as a function of both volume flow Q_M and pressure p_M. By solving equation (7) the searched state triples p_M, Q_M and x_M are accessible. The transcendent character of that equation makes it difficult to find a proper solution. This problem is expressed through the dependence of the cross-section area A_{Str} as wells as the Reynolds' number Re_M on the valve spool displacement x_M and volume flow Q_M. Additionally, the Reynolds' number Re_M is indirectly linked with the pressure p_M through the pressure-dependence of the dynamic viscosity η.

$$0=\zeta\left[Re_M\left(x_M,Q_M\right)\right]\frac{\rho}{2}\left(\frac{Q_M}{A_{Str}\left(x_M\right)}\right)^2-p_M \qquad (7)$$

As an iterative problem-solving approach, fix-point iteration may be feasible where a main problem deals with the convergence performance, this has to be estimated in pressure p_M for a fixed volume flow Q_M. The additionally required equation which correlates the independent variable x_M with pressure p_M can be derived from the flow equation (1). As a result a proportional relation of $Q_M \sim {p_M}^{0,5} \sim A_{Str}$ is immediately obtained. Assuming an approximately linear relation between A_{Str} and x_M, $x_M \sim {p_M}^{0,5}$ follows. With its help the missing equation (8) for the determination of a new displacement prediction can be defined which is, strictly speaking, only valid for moderate non-linearity in the opening process.

$${x_M}^{k+1}=\left(\frac{{p_M}^k}{p_M\left(Q_M\right)}\right)^{0,5}\cdot {x_M}^k \qquad (8)$$

With the determination of the state triple, all values for the calculation of the necessary counterforce characteristic $F_M - F_F$ are known. Therefore, equation (6) has to be transformed. Because the state triple is completely given, equation (9) is an explicit expression for the searched counterforce $F_M - F_F$ if all control edge coefficients are available.

$$F_M\left(x_M,I_M\right)-F_F\left(x_M\right)={Q_M}^2\left(\zeta\left[Re_M\left(x_M,Q_M\right)\right]\frac{\rho}{2}\frac{A_K\left(x_M\right)}{A_{Str}\left(x_M\right)^2}-\frac{\rho}{A_{Str}\left(x_M\right)}k_{GF}\left[Re_M\left(x_M,Q_M\right)\right]\right) \qquad (9)$$

It is possible to compute the necessary solenoid force characteristic on basis of equations (7) and (9) without any optimization algorithm. The procedure follows the idea that different control edge concepts require several counter- or solenoid force characteristics. Consequently, a static valve performance is realizable by a suitable choice of the control edge and a downstream solenoid design. The question whether the requirements put on the solenoid by the control edge can generally be achieved remains unanswered so far.

3.2. CFD modeling and simulation of control edge performance

Besides the intended static valve performance $p_M = f(Q_M)$ the inverse valve model also requires the characteristic values of its hydraulic subsystem as initial conditions, which generally consist of four characteristic curves. The closest flow cross-section $A_{Str}(x_M)$, as well as the theoretical pressure area $A_K(x_M)$ are the geometrical values of interest for a special control edge design. The description of energy losses is carried out with the loss coefficient ζ. Local pressure changes and their influences on force reactions at the valve spool are treated by the correcting term F_{Str} or the corresponding coefficient k_{GF}. Correlation of these values on the valve spool displacement x_M and the volume flow Q_M is considered by a relation on the Reynolds' number Re_M.

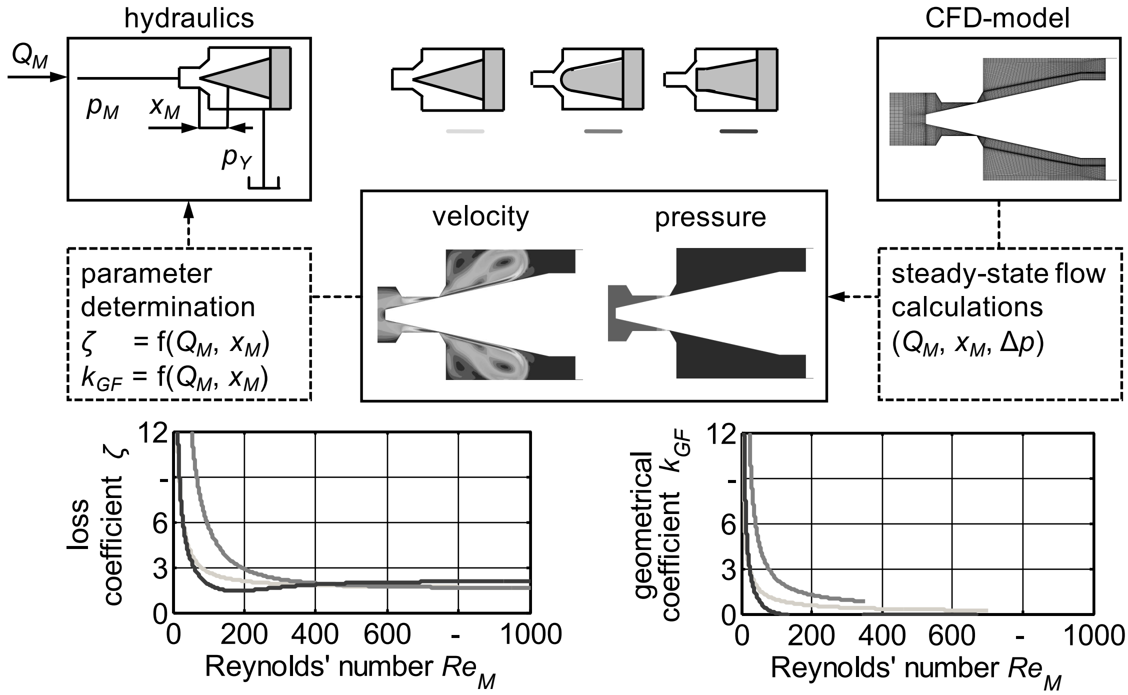

Figure 4: Determination of hydraulic coefficients

A morphological box for basic geometric shapes of valve seat and valve spool is built for demonstrating the influence of different control edge concepts. On this basis, the investigated control edge designs are chosen so that close geometrical similarity is still maintained. This can be achieved with a valve spool shaped in a cone-/sphere-/piston-like manner in combination with a chamfer at the valve seat. The procedure for calculating the hydraulic characteristic values of different basic shapes is explained in **Figure 4**.

Model creation for each control edge concept is accomplished in separated CFD models, whereas the investigations cover various valve spool displacements. With help of these models, pressure losses of a laminar and incompressible flow with a pressure-dependent viscosity $\eta = f(p)$ are computed. The obtained CFD results describe a complex, 3-dimensional fluid flow at different operating points. Assuming that these fluid flows are quite similar at different valve spool displacements, the obtained results are reduced to a 1-dimensional description of the characteristics depicted in Figure 4. It was found that the approximation quality of mathematical regression is influenced by viscosity η used for calculating the Reynolds' number Re_M. This issue was considered and implemented into the inverse valve model.

3.3. Realization of force characteristics

With the hydraulic characteristic values and the inverse valve model in hand, necessary counter forces $F_M - F_F$ can be determined for a given static valve performance $p_M = f(Q_M)$. In **Figure 5** the underlying static pressure performance and the resulting counterforce characteristic map for a lowest and highest pressure setting are explained in dependence to valve spool design. The results of Figure 5 show that the essential valve spool displacement decreases with an increasing pressure setting. Simultaneously, the counterforce slopes of $F_M - F_F$ differ between lowest and highest pressure setting resulting in a spreading. The slope is additionally dependent on the employed valve spool design.

For all following considerations a cone-shaped valve spool is used because of its smallest counterforce slope and its slightly decreasing characteristic at maximum pressure setting. Therefore, the provided solenoid force F_M can completely be applied for closing the valve seat which leads to an advantageous minimum pressure. The necessary spreading in counterforce slopes imposes expedient boundary conditions on solenoid design, because a combination of switching and proportional solenoid performance is looked for. Hence, the necessary counterforce map can be realized by corresponding implementation of the spreading in solenoid force-displacement curves in combination with a spring. Section 4 compares measurement and simulation results of a solenoid fulfilling above-mentioned requirements.

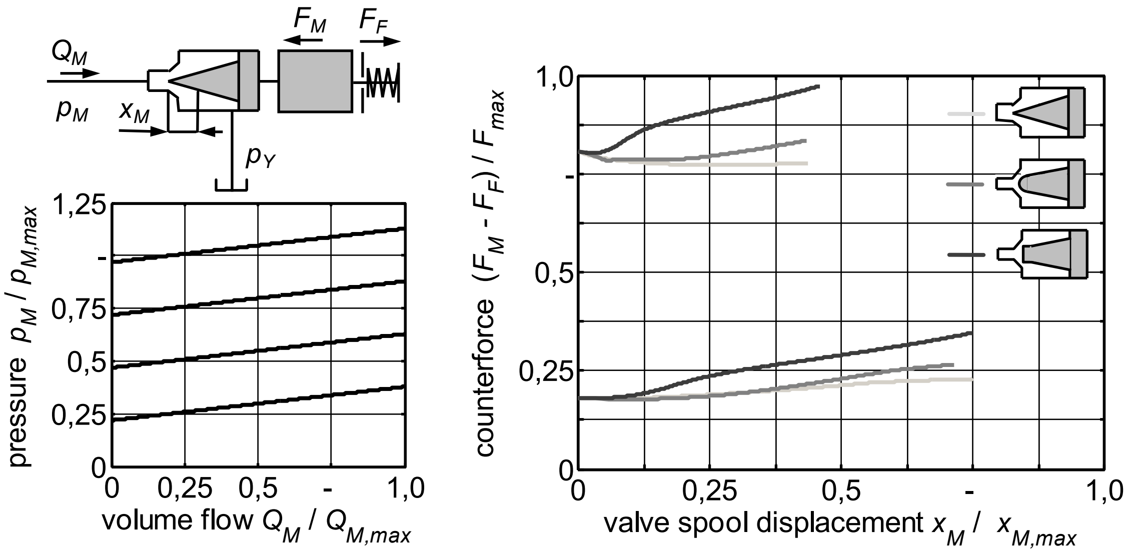

Figure 5: Intended pressure-volume flow performance and calculated counterforce map

4. Measurement and simulation results

In preceding sections the inverse valve model was deduced, the parameterization of hydraulic valve subsystem was explained, and the necessary counterforce map was calculated. This section covers the final validation of computed results based on measurements. The validation process is divided into a static validation of the solenoid performance and a comparison of the entire valve performance. Solenoid and air gap are designed according to conditions specified in section 3.3 for generating the counterforce map.

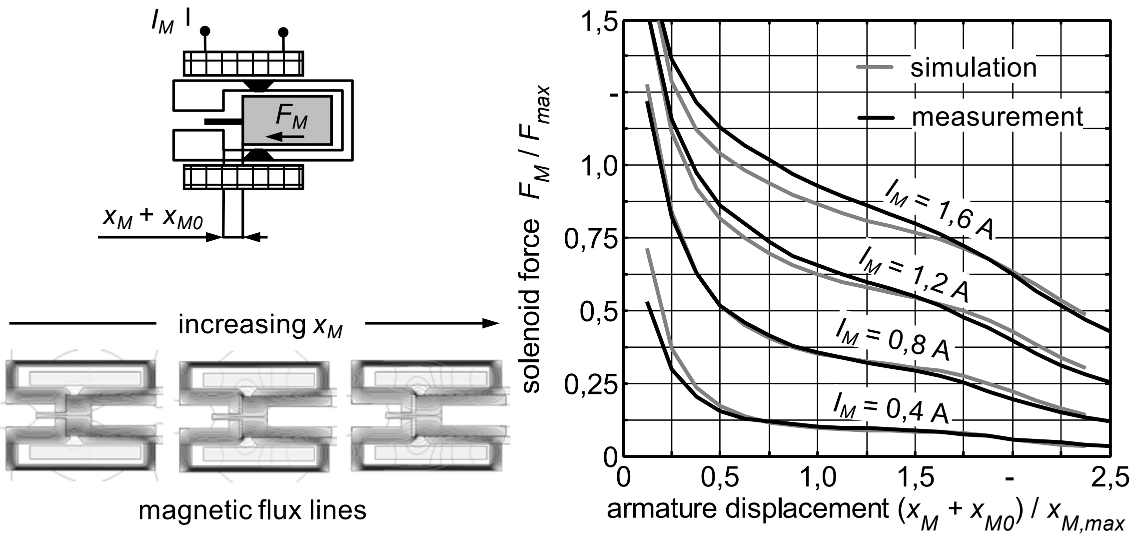

Figure 6: Solenoid force characteristics - simulation and measurement

The measured and simulated force characteristics from **Figure 6** only differ at higher solenoid currents I_M which can be explained by deviations of material behavior in the saturation region. Anyhow, force differences at the solenoid's operating range are below 10 % highlighting the accuracy of the applied prediction.

The entire valve performance is estimated by means of state triple whereby the volume flow Q_M is chosen as independent variable for the drawn characteristic curves in **Figure 7**.

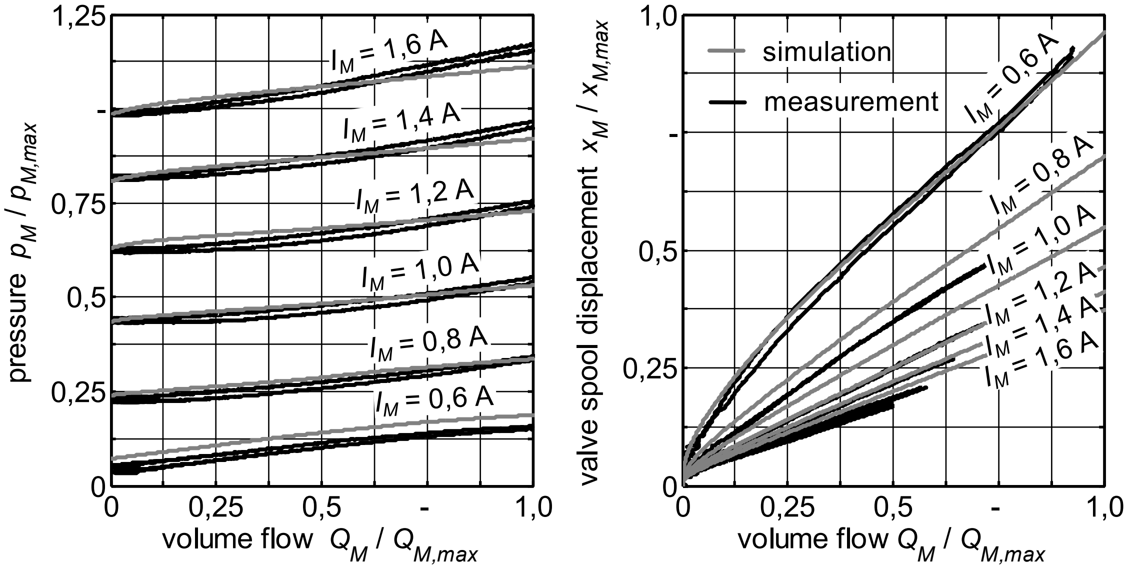

Figure 7: Pressure-volume flow and valve spool displacement-volume flow characteristics - simulation and measurement

Figure 7 compares measurements and simulation results for both characteristic maps. It is evident that a qualitatively and quantitatively accurate accordance is achieved, both in pressure- as well as in displacement-volume flow performances supporting the suitability for this computational prediction. The differences at the beginning of the characteristic curves and at higher volume flows can be attributed to a minor deviation of the applied control edge design and the geometrical shape used in the simulation. Optimizing the parameterization on the basis of the exact geometrical dimensions may improve computational results.

Besides the static performance of pressure relief valves a sufficient damping is also required. **Figure 8** depicts a rectangular jump in nominal current at the solenoid, further proving the desired damping at low volume flows.

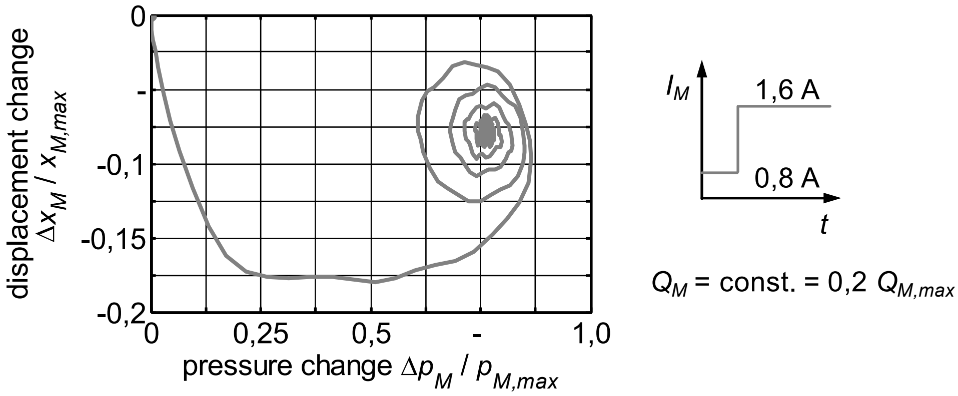

Figure 8: Nominal current jump at the solenoid - measurement

5. Conclusions

This article explains the methodological procedure for designing a static valve performance using the example of a direct-operated proportional pressure relief valve. Existing simulation methods are extended by an inverse valve model. This model benchmarks different control edge concepts and calculates the necessary counterforce characteristics. Computational results establish the basis for the subsequent geometry design of the applied solenoid realizing the necessary force-displacement characteristic. Measurements demonstrate that simulation allows a qualitatively and quantitatively accurate prediction of the static valve performance. It is proven that the performance of electro-hydraulic components can sufficiently be designed and predicted by the methods used in a virtual product development process.

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7. Nomenclature

A_K	pressure area	mm^2	p_M	operating pressure	Pa
A_{Str}	smallest flow cross-section	mm^2	p_Y	tank pressure	Pa
b_M	viscous damping coefficient	Ns/m	Q_M	volume flow	m^3/s
F_F	spring force	N	Re_M	Reynolds‘ number	-
F_K	pressure force	N	U_{Str}	wetted perimeter	mm
F_M	solenoid force	N	x_M	valve spool displacement	mm
F_{Str}	flow force	N	x_{M0}	residual air gap	mm
I_M	solenoid current	A	ζ	loss coefficient	-
k_{GF}	geometrical coefficient	-	η	dynamic viscosity	Pa·s
m_M	mass	g	ρ	density	kg/m^3