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# Design analysis of a novel orifice control valve for turbomachinery testing facilities

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**Abstract.** In testing facilities for turbomachinery applications, the main component for the regulation of the flow is the control valve. In order to fulfill the flow rate requirements, the performance of the control valve should be highly accurate in a specified flow range. The robustness against variations or disturbances, is a major issue to be considered. A control valve incorporated in any testing facility has to meet certain requirements. Thus, operating aspects, such as the flow rate range and the valve pressure drop, along with control prerequisites, regarding flow stability, repeatability and robustness of the valve, are considered. The purpose of this study is to establish a new designing approach for control valves. Orifice plates are used as a guideline for this study because of the geometrical similarities with the control valve. A sensitivity analysis determines the most influential parameters and prioritizes the requirements. With emphasis on the cross-sectional area and by neglecting its shape, a simplified 2-D model is optimized. Eventually, a CFD model, verified by experiments, is used for the analysis of the effects of geometrical parameters. The established workflow weighs the requirements according to each application and indicates the method to reach the desired goals. CFD simulations verify the 2-D model and assist in the fine adjustment of the control valve's characteristics. With an average deviation of 3% between the 2-D and the 3-D model, the simplified 2-D model proves to be sufficient for setting the valve's characteristics, setting the CFD simulations necessary, only in applications of extreme flow rates and temperature conditions.

## Nomenclature

| | | |
|---|---|---|
| A | Cross-sectional area | [m²] |
| $C_d$ | Discharge coefficient | [-] |
| $\dot{m}_a$ | Actual mass flow | [kg/s] |
| $\dot{m}_m$ | Measured mass flow | [kg/s] |
| $t^*$ | Dimensionless thickness | [-] |
| $V_a$ | Actual velocity | [m/s] |
| $V_m$ | Measured velocity | [m/s] |
| ζ | Pressure loss coefficient | [-] |
| θ | Angular position % | [-] |

## 1 Introduction

The primary task of control valves is the regulation of a flow. The diverse applications and operational requirements lead to long design and optimization processes, to customize the control valve design to the respective demands.
Solenoid spool valves are a common solution in hydraulic systems, in order to control the flow in a large range. [1] Agh et. al propose a cam-nozzle for a metering unit of the fuel flow of a gas turbine, which achieves reduced sensitivity in the pressure fluctuations of the flow, before and after the unit. And increased control accuracy at the system. [2] Finally, a bell-shaped control valve is used in a steam turbine to regulate the flow. [3]
The purpose of the study is to propose a new designing method, that is able to adapt to different uses and requirements. Rotating-disk control valves are small, have good sealing abilities and enable continuous regulation of the flow. Apart from these characteristics, the designing flexibility, that they provide, facilitates the analysis of the control valve and sets them as the most suitable type. Furthermore, this choice is driven from their ability to implement the principles of orifice plates on their geometry. Although orifice plates are pressure differential flowmeters, their operation and the flow characteristics have many similarities with the rotating-disk control valve. Nevertheless, there are some differences between them. The variable cross-section leads to a non-circular opening shape, in the majority of the positions. Also, the cross-section is not concentric with the disk. Therefore, depending on the position of the rotating disk, the cross-section looks like a segmental or a fully eccentric orifice plate. The operating principles are the same, despite the differences.
This assumption allows the use of the studies on orifice plates, in order to analyze and improve the characteristics

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of the investigated control valve. In this case, the water mass flow rate varies in the range [0.01-1] kg/s.
Specific performance requirements are expressed in terms of linearity, repeatability, robustness against variations and low flow fluctuations. The linearity requirement implies the demand for locally linear behavior and limited flow rate variation. Also, under the same conditions, the performance should be repeatable within an error range of 0.2%. For a robust control valve, its performance should not be affected significantly from the variation of any parameter or a flow-induced disturbance. The flow aspect of this requirement is expressed as a demand to maintain a stable flow with fluctuations lower than 0.2%.
The workflow suggests a sensitivity analysis to, first, estimate the most influencing parameters and prioritize them, according to the aforementioned requirements. Emphasis is given on the geometry of the control valve, because studies have proven the significance of the relationship between the opening area and the position of the control valve on its performance characteristics. [4] [5] Therefore, the effect of geometrical parameters of the proposed design on the characteristics of the control valve are studied. The 2-D analysis of the cross-section is followed by the selection of the parameters' values, which lead to the best characteristics and fulfill the requirements. Finally, a CFD analysis is conducted, to study the flow phenomena and fully determine the performance of the control valve in 3 dimensions. In this last part of the analysis, the accuracy of the simplified, 2-D model is determined. Also, the effect of the thickness of the control valve's disk on the performance is evaluated.
Studies have proven, that as t* value increases, the effect of thickness is negligible and the performance is not affected, both in terms of $\zeta$ and $C_d$. [6] Friction losses become dominant against losses from secondary flows or turbulence. [7] The ability to dampen flow-induced disturbances is improved, and, thus, the performance of thicker orifice plates is less vulnerable to flow disturbances. However, the pressure recovery is delayed and, the inception cavitation number decreases. As a result, cavitation occurs more easily and in larger vapor cavities. [8] The introduction of a chamfer on the edges of the cross-section of the control valve, the pressure field is improved, obstructing cavitation inception. The formation of a 3-D throat at the cross-section, can increase the cavitation inception number up to 70% and, at the same time, reduce the turbulent kinetic energy by approximately 50% for flows with the number Re=35000. [9]

# 2 Methods

## 2.1 Measuring techniques

To evaluate the performance of control valves, a specific testing rig is used. To circulate the fluid in the rig, a pump is used. At the outlet of the pump the mass flow rate is monitored with a flowmeter. Following that, the fluid enters the control valve section. The control valve's position is regulated manually via a shaft. The fluid exits the facility in ambient pressure and through a supplementary tank returns to the storage vessel.
According to the specifications of the installed flowmeter, the measuring and the repeatability errors do not exceed 0.1%, in the mass flow range
[0.01-1] kg/s. [10]
Speed control is applied on the pump. Depending on the control valve's position, the pump's frequency is adjusted to provide the necessary pressure head at the inlet of the control valve, to meet the 2-bar pressure drop requirement.
Manometers and pressure transducers are installed before and after the control valve. The manometers provide a reading for the local pressure. Based on their indications, the speed control is performed accordingly. The pressure transducers monitor the instantaneous flow conditions. Schlienger et al. have proven that the use of a single pressure transducer in a single probe is sufficient to accurately measure the 3-D flow. [11] Therefore, the implemented method to record the pressure fluctuations is capable of providing accurate results for this application. The maximum frequency of the pump is achieved before the control valve position, that corresponds to the maximum flow rate. Thus, the flow rate is measured under different pressure-drop conditions in a number of positions. The pressure loss coefficient is used to recalculate the measured values and estimate the actual flow rate under the required 2-bar pressure drop. The numerical value of the pressure loss coefficient is a function of the dynamic pressure and, therefore, of the cross-section. The calculation method also include the mass conservation equation and it is presented in Eq. 1 & Eq. 2.

$$\zeta = \frac{\Delta P}{\frac{1}{2}\rho V^2} \Rightarrow V_a = V_m \sqrt{\frac{\Delta P_a}{\Delta P_m}} \quad (1)$$

$$\dot{m}_a = \rho A V_a \Rightarrow \dot{m}_a = \dot{m}_m \sqrt{\frac{\Delta P_a}{\Delta P_m}} \quad (2)$$

## 2.2 CFD model

The control valve performance is evaluated with a computational model, too. Pressure boundary conditions are applied in the inlet and the outlet of the domain.
For orifice plates and control valve applications the k-ε models are usually preferred. For the numerical

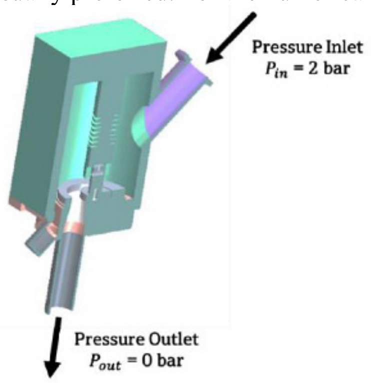


**Fig. 1.** Control valve model

solution, the realizable k-ε model is used. It has been found that this model is more accurate than all of the k-ε models especially for separated flows, boundary layer flows involving high pressure gradients and flows with complex flow structures. [12] These characteristics set it as the most suitable for the current application.

### 2.3 Sensitivity analysis

Based on the requirements of this application, it is important to determine the parameters with the greatest influence on the control valve performance. This analysis is matching performance requirements to parameters of the control valve system. The investigation focuses on the flow conditions, the properties of water and the influence of the geometry of the control valve.

This analysis demands a working control valve to estimate the influences. It is assumed, that the results of the study on the flow conditions are not strongly dependent on the geometry of the control valve. This hypothesis allows the study on flow stability, by monitoring the pressure fluctuations. By detecting noise and vibrations, during the experiments, the occurrence of cavitation can be determined. For the same purpose, the minimum static pressure in the CFD models is recorded, to determine the flow rate, which leads the pressure to drop below the vapor point.

The variation of the water's density, which depends on the temperature, represents the fluid's contribution to the control valve performance. The variation of the ambient temperature or a heating of the water can alter the properties of the fluid. The operating temperature range is 5°C to 40°C. Assuming that the discharge coefficient, which was calculated form the experimental data, remains constant, at each control valve position, it is calculated:

$$\dot{m} = C_d A\sqrt{2\rho\Delta P} \Rightarrow \frac{\partial \dot{m}}{\partial \rho} = \frac{1}{2} C_d A \sqrt{\frac{2\Delta P}{\rho}} \Rightarrow$$
$$\Rightarrow \delta\dot{m} = \left(\frac{C_d A}{2}\sqrt{\frac{2\Delta P}{\rho}}\right)\delta\rho \tag{3}$$

The variation of the cross-section during the rotation of the control valve is strongly influencing the control characteristics. The requirements for linearity and repeatability dictate high resolution on the rotation, so that the mass flow rate of change is very low in the entire range. The effect of the geometry is studied by introducing a 0.1° mispositioning error.

### 2.4 Cross-section analysis

This analysis is performed with a simplified 2-D model. It is assumed that the shape of the cross section does not have a strong effect on the final performance.

The implemented design consists of circular arcs and spirals. For their evaluation, the cross-sectional

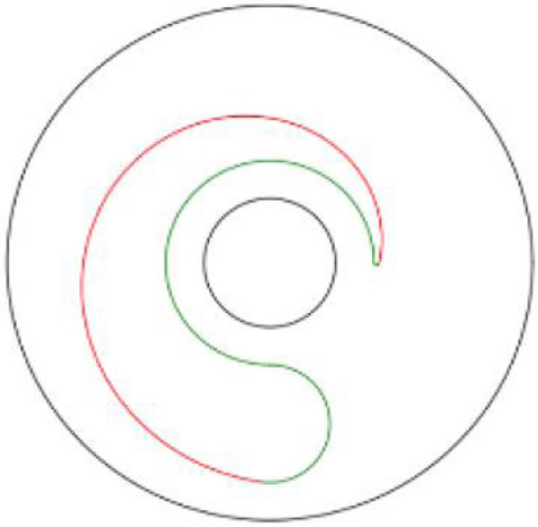

**Fig. 2.** Initial 2-D- control valve design

area and rate of change as a function of the position of the control valve are plotted. In the drawing of Figure 3, the circular arcs are shown in green and the spirals in red color.

To improve the design and meet the requirements for the performance of the control valve, geometrical parameters are investigated and presented in Figure 4. The modification of the geometry alters the area and the cross-sectional sensitivity curves, and changes the characteristics of the control valve. The first parameter is the radius $R_1$ of the circular arc, that corresponds to the smaller cross-sections. Its values vary from 0.2 to 1 mm. The second one is referring to a specific angle $\Delta\theta_c$ related to the circular arc of the larger cross-sections, with the value of the angle varying from 10° to 60°. The last parameter is the order $N$ of the spiral polynomial curve. Linear, square and cubic spirals are analyzed. Although accurate performance of the control valve is required across the full flow rate range, the calculated values are more important. It must be ensured, that in the positions corresponding to these flow rates, the control characteristics will be optimal and, consequently, the sensitivity of the control valve will be minimum.

**Table 1.** Flow rates & areas for design evaluation

| Mass flow rate [kg/s] | Area [$mm^2$] |
|---|---|
| 0.02 | 2 |
| 0.09 | 6 |
| 0.10 | 7 |
| 0.25 | 17 |
| 0.35 | 24 |
| 0.63 | 42 |
| 0.98 | 64 |

For this reason, the flow rates must be matched with the cross-sectional area values and, then, control valve's positions. Based on the hypothesis, that the flow conditions do not differ significantly, the Blum

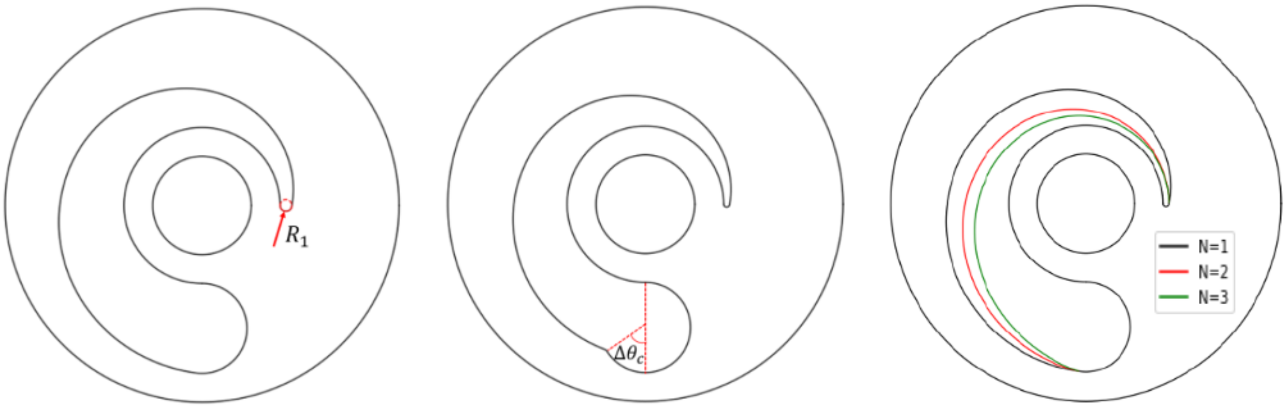


**Fig. 3.** Investigated geometrical parameters

design provides the necessary data. [13] Therefore, according to his design, the data are matched, as it is presented in Table 1.

Knowing the cross-sectional area to which these flow rates correspond, the respective control valve positions are calculated. Then, the cross-sectional sensitivity values of each flow rate are determined. Aiming to minimize the overall sensitivity of the control valve, it was decided that the most appropriate assessment criteria are the average sensitivity of the 7 flow rates and the ratio of the maximum to the minimum sensitivity value.

$$F_1 = \sum_{i=1}^{7} \left(\frac{dA}{d\theta}\right)_i \ \& \ F_2 = \frac{\left(\frac{dA}{d\theta}\right)_{MAX}}{\left(\frac{dA}{d\theta}\right)_{MIN}} \tag{4}$$

## 2.5 CFD analysis

The CFD analysis aims to verify the findings of the cross-section analysis and further investigate the characteristics of the control valve. The performance is greatly influenced by the flow conditions in the system. It has not been yet investigated how the flow interacts with the new control valve design. Therefore, the control valve has to be simulated, in order to fully determine its characteristics. The main issues, which may come up, are related to flow disturbances and cavitation. Both of them can alter the performance of the control valve. To cope with the flow disturbances, the thickness of the disk will be varied. According to the orifice plate studies, this is an effective approach to increase the friction in the control valve and reinforce its antidisturbance abilities. The control valve plate can have a minimum thickness of 5 mm, which ensures the tightness of the system and prevents leakages. Thus, the control valve disk will be simulated for thicknesses varying from 6 mm to 10 mm. In order to evaluate the potential of the higher disk thickness, parameters as the mass flow rate are monitored. Furthermore, the velocity, the pressure field and the flow profiles downstream the control valve are recorded, to estimate the necessary mixing length, which leads to fully developed flow. Finally, it is studied the effect of an edge chamfer on the top edges of the cross-section on the flow conditions. The orifice plate studies suggest this modification, as a method to change the conditions of cavitation inception. This chamfer may be partially or totally occupying the length of the valve's throat. In the CFD simulations, the wall angle is varied from 3° to 65°.

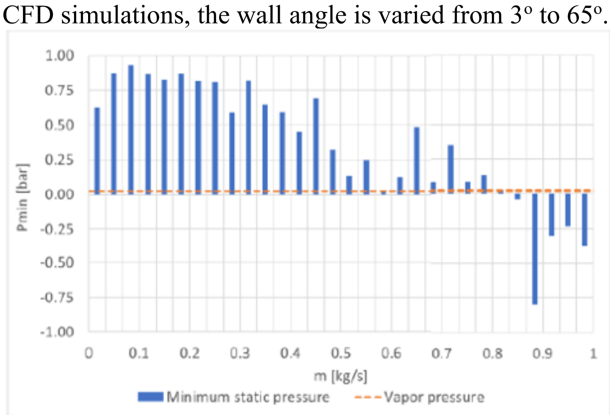


**Fig. 4.** Minimum pressure & Vapor point limit per

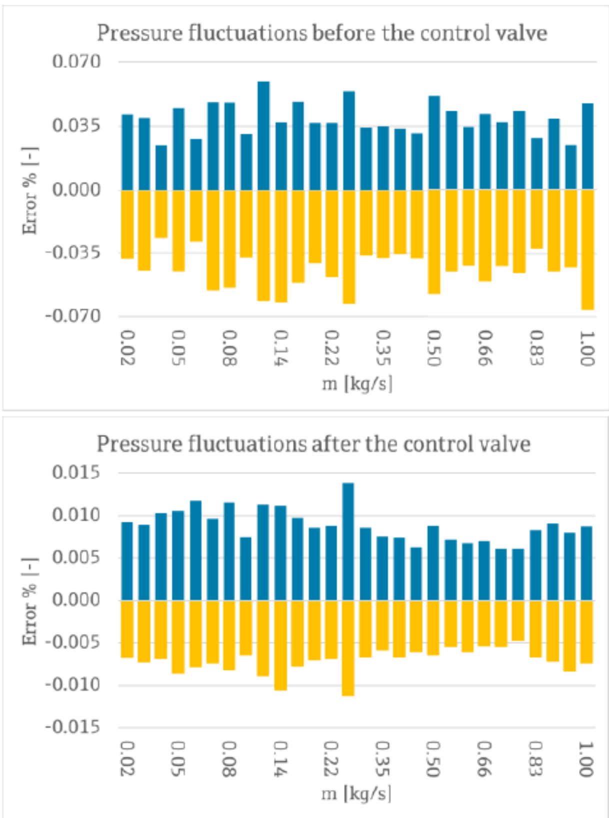


**Fig. 5.** Pressure fluctuations before & after the control valve

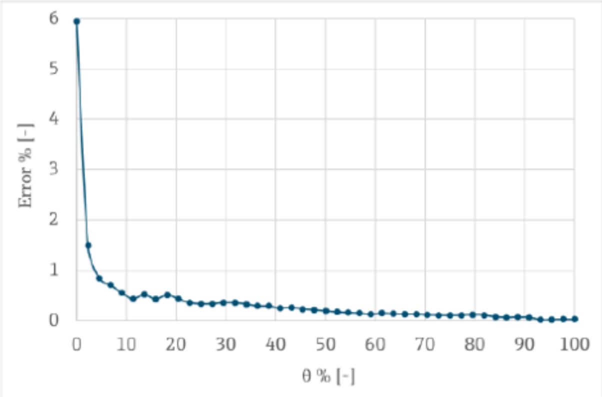


**Fig. 6.** Effect of the geometry on the performance

## 3 Results & discussion

### 3.1 Sensitivity analysis

The minimum pressure in the computational domain varies with the flow rate and, although there is not a clear tendency in the results, it remains above the vapor point for flow rates up to 0.7 kg/s, indicating that cavitation is present in the control valve system. Nevertheless, since neither vibrations nor hissing noises were detected during the experiments it is implied that its effects are mild.
The devitations of the pressure from the local average value are sufficiently low both before and after the control valve. In the studied flow rate range the pressure fluctuates by 0.05% before and approximately 0.01% after the control valve. Therefore, it is clear that the dependency of the performance from the pressure conditions is very low. Despite the increase in the flow rate, the pressure fluctuations remain almost constant, indicating that the flow stability is not related to the varying shape of the cross-section. Also, since the sensor downstream the control valve records lower variations, it is deduced that the control valve has the ability reduce the flow fluctuations. Considering the variation of the density of the water, after the adjustment of the experimental results, it is calculated, that the performance is hardly influenced since the change is approximately 0.015%/K, in the specified range.
The rotation of the control valve in the intial positions, leads to greater and significant crosssectional changes, and, thus, relatively large performance variations. Since the control valve's repeatability has been connected with the crosssectional sensitivity, the performance standards are not met up to 50% of its rotation. Only then, the resulting performance variation drops below 0.2%, setting evident, that the cross-sectional sensitivity can be dominant.

### 3.2 Cross-section analysis

The variation of the radius $R_1$ has an indirect effect on the spiral curve. The increase of $R_1$ reduces the $\Delta R$ of the spiral, with its influence being apparent on the smallest cross-sections. The increase of its value leads to quicker increase of the cross-sectional area in the same range of rotation. The effect on sensitivity is, firstly, identified by the prolongation of the steep descending section of the curve. Secondly, $R_1$ affects the latter part of the sensitivity curves, lowering its value. Nevertheless, the effects fade as the control valve rotates, and, eventually, the curves are coincident.
Similarly with the effect of $R_1$, the increase of $\Delta\theta_c$ indirectly affects the spiral by reducing its $\Delta R$, through the modification of the largest cross-sections' characteristics. Although there are apparent differences in the area curves, since the increase of $\Delta\theta_c$ alters the slope in the middle section, the variation of $\Delta\theta_c$ has limited effects on the control characteristics of the valve. The sensitivity curves

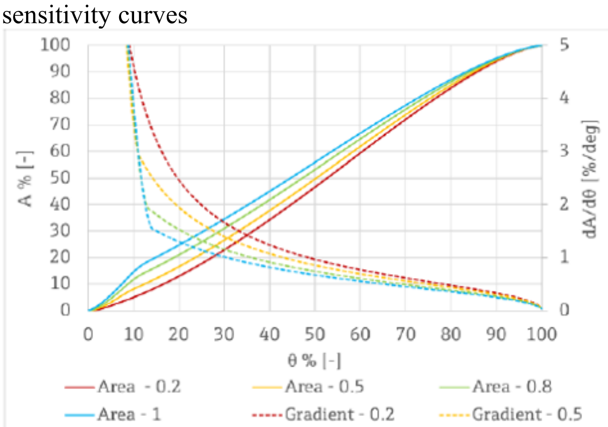


**Fig. 7.** Effect of Radius $R_1$ on the control valve's characteristics

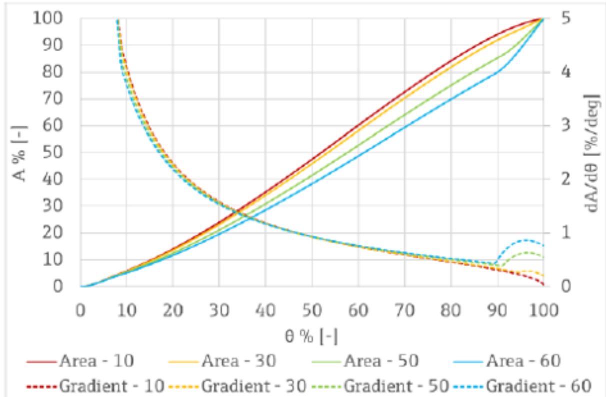


**Fig. 8.** Effect of radius $\Delta\theta_c$ on the control valve's characteristics

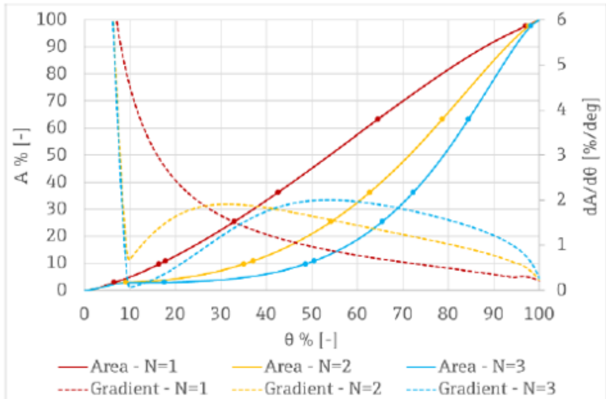


**Fig. 9.** Effect of polynomial's order N on the control valve's characteristics

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are almost coincident for 90% of the control valve's rotation. The effect of $\Delta\theta_c$ is only apparent in the last positions, where the sensitivity might change up to 0.6%. The cross-sectional area curves are linear, with the two aforementioned parameters introducing, locally, non-linearities in the characteristics of the control valve. Their magnitude is determined by the values of the parameters. In the same way, the hyperbolic sensitivity curves are locally distorted and the sensitivity reduces in an exponential manner. Greater effects are applied on the cross-sectional area and its sensitivity from the variation of the order of the spiral's polynomial, due to the overall change in the relationship between the position of the control valve and the spiral's radius. By increasing the order of the spiral, the parabolic area curves lead to slow increase of the cross-section at the initial positions, which becomes sharper as the control valve's rotation progresses. Furthermore, the initial, high-sensitivity section is avoided, due to parabolic shape of the area curves. The sensitivity has dropped below 2%/° before 9% of the rotation of the control valve. The sensitivity values of the dotted points correspond to the cross-sections for the evaluation of the designs. Higher order spirals also affect the curve by slightly lifting it in its latter section. Towards the end of the curves, the higher flow rates have increased their sensitivity, by approximately 1-2%.

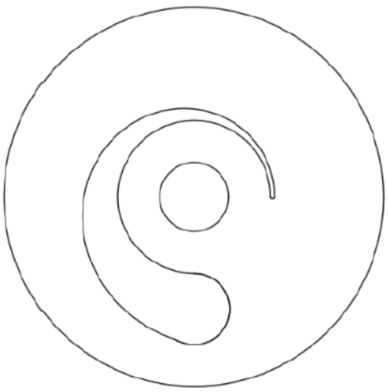

**Fig. 10.** Final 2-D control valve's design

Despite the local increase of sensitivity, in the higher-order spirals, the characteristics of the control valve are significantly improved, due to their dependence from the cross-sections corresponding to the lower flow rates. The desired characteristics, within the limits of the proposed restrictions, are achieved with a set of parameters' values, where $R_1$=0.3 mm, $\Delta\theta_c$=30° and N=2. The design leads to an average sensitivity of 1.4%/°, with $F_2$=4.3.

### 3.3 Effect of thickness of the disk

Table 2 indicates, that the increase of the thickness from 5 mm to 6 mm is leading to a change of approximately 3.11%. For thicknesses higher than 6 mm the effect in the mass flow and, the discharge coefficient is negligible, as it is expected. Table 3 is

**Table 2.** Effect of thickness on the mean mass flow change

| Flow rate range [%] | Δm % | | | | |
|---|---|---|---|---|---|
| | t=6 | t=7 | t=8 | t=9 | t=10 |
| [0-10] | 2.44 | 2.17 | 2.08 | 1.9 | 2.04 |
| [10-50] | 2.69 | 2.73 | 2.72 | 2.52 | 2.39 |
| [50-100] | 4.4 | 3.99 | 3.9 | 3.91 | 4.13 |
| [0-100] | 3.11 | 2.9 | 2.84 | 2.7 | 2.78 |

**Table 3.** Effect of thickness on the highest flow rates

| Thickness [mm] | Mass flow rate [kg/s] | | |
|---|---|---|---|
| | θ=96% | θ=98% | θ=100% |
| t=6 | 1.1144 | 1.1408 | 1.1315 |
| t=7 | 1.1045 | 1.1292 | 1.1247 |
| t=8 | 1.1003 | 1.1260 | 1.1222 |
| t=9 | 1.0998 | 1.1226 | 1.1226 |
| t=10 | 1.0960 | 1.1207 | 1.1210 |

displaying the effect of the thickness of the disk on the control valve's performance at the higher flow rates. As the disk's thickness increases, the shape of the mass flow curve changes in the last positions. Thicker walls dampen flow disturbances. As a result, for t=10 mm, the maximum flow rate is achieved in the maximum cross-sectional area. The position, the cross-sectional shape and the area influence the characteristics of the flow downstream the control valve. The asymmetry in the velocity profiles of the thinner discs is shown in Figure 11. For disks wih a

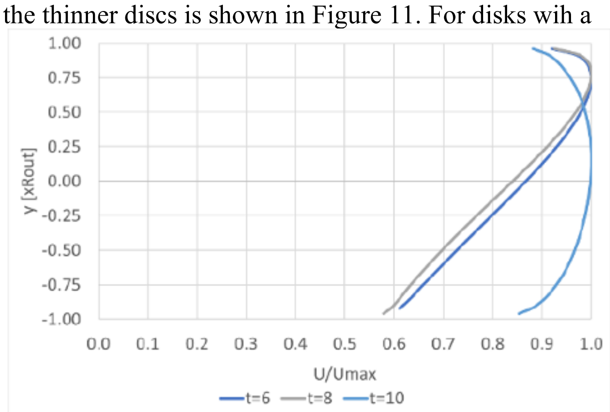


**Fig. 11.** Effect of thickness on the flow profile 10D downstream the control valve

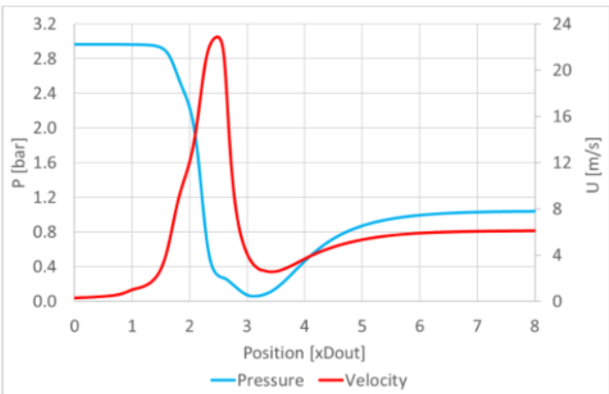


**Fig. 12.** Pressure and velocity variation across the control valve

thickness up to 8 mm, the flow asymmetries are strong in the outflow. However, as the thickness of the disk of the control valve further increases, the flow recovers develops fully in a shorter pipe length. Nevertheless, the increase in the thickness of the disc is intensifying the occurrence of cavitation, as the high peak velocity and the variation of the absolute pressure on the region of the vapor point demonstrate.

### 3.4 Effect of chamfer of the disk

Introducing a chamfer on the edges of the control valve is merely a successful technique. The results suggest, that cavitation is only avoided for flow rates lower than 0.3 kg/s. Figure 13 demonstrates that the minimum pressure in the computational domain drops below the vapor point for higher mass flow rates. It is also constantly reducing as the flow rate increases. However, at the largest flow rate an increased pressure value is recorded, due to the fully-circular cross-section, whose uniform shape prevents further flow acceleration.

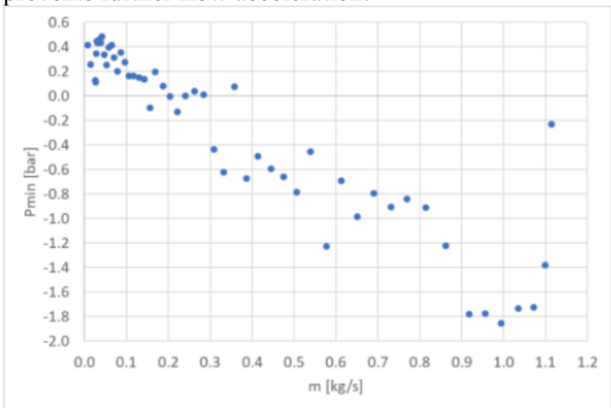


**Fig. 13.** Minimum pressure for various flow rates

### 3.5 2-D model validation

The mass flow rates of the control valve design are displayed in Figure 10. The targeted maximum value of 1 kg/s is considered as 100%. The mass flow curve resembles the area curve of the 2-D analysis, with the average deviation between the two approaches being 3%. The 2-D and 3-D characteristics of the control valve are not identical, too. The control valve is oversized,

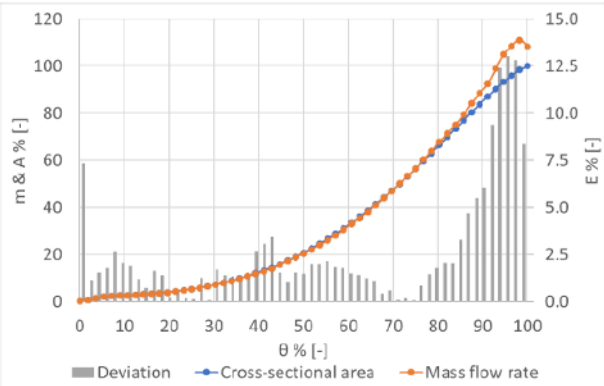


**Fig. 14.** Percentagewise comparison of cross-sectional area and mass flow of the control valve

since it has the capability to deliver slightly higher flow rates than its specifications. Sencondly, although the performance of the control valve agrees with the two-dimensional model up to 1 kg/s, there is a mismatching in the positions of the maximum cross-section and maximum flow rate, which is achieved at 98% of the control valve's rotation. After its peak value, the flow rate is reducing. Therefore, it is considered, that the performance is altered, due to a flow disturbance, deriving from the interaction of the flow with the geometry of the control valve.

## 4 Conclusions

The results of the analysis on the control valve's performance are indicating the effectiveness of the proposed method to improve its characteristics. By establishing a flow field, that displays pressure fluctuations of approximately 0.01% downstream the control valve, the requirement for a stable flow is met. Furthermore, its performance has proven to be nearly insensitive to variations of the density of the water in the operating temperature range. With the necessary design modifications, the final geometry has linear characteristics and reduced sensitivity to cross-sectional variations, meeting the repeatability requirement of 0.2%. Finally, an even more robust control valve performance is achieved, by applying design modifications for flow disturbances.

With the sensitivity analysis, it is proven that the flow fluctuations are independent of the varying shape of the control valve's cross-section. The fluid properties, for limited temperature variations, are hardly influencing, altering the valve's performance by approximately 0.015 $\%/K$.

The analysis of the cross-section proved, that only one core-design is enough to produce control valves with different inherent characteristics, which can be used in entirely diverse applications. The role of the function of the cross-sectional area and the position of the control valve is critical.

Higher-thickness disks display greater benefits in flow-related characteristics. The flow recovers fast downstream the control valve. Its performance is not affected from flow-induced, upstream disturbances, and, hence, leads to a more unsensitive and robust performance.

Geometrical modifications, such as edge chamfer contribute to advantageous conditions, that increase cavitation resilience. However, their effectiveness is small. To operate under non-cavitating conditions, the reduction of the pressure drop is necessary. Finally, designing a control valve, is a process, that can be decoupled from CFD simulations. In this study, the design was determined in two dimensions, with an error of 3%. Such deviations are considered satisfactory for the simplicity of 2-D model. This leads to a significant reduction of the duration of the designing process.

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