



Characterization of Check Valves Under Steady and Unsteady Conditions [†]

Federico Vincenzo Maraschi * , Niccolò Casati, Matteo Dell'Avo, Giacomo Ferrarese and Stefano Malavasi

Department of Civil and Environmental Engineering, Politecnico di Milano, Piazza Leonardo da Vinci 32, 20133 Milano, Italy; niccolo.casati@polimi.it (N.C.); matteo.dellavo@mail.polimi.it (M.D.); giacomo.ferrarese@polimi.it (G.F.); stefano.malavas@polimi.it (S.M.)

* Correspondence: federicovincenzo.maraschi@polimi.it

[†] Presented at the 6th International Conference on Efficient Water Systems (EWaS6), Thessaloniki, Greece, 11–14 May 2026.

Abstract

Check valves are essential components in hydraulic systems, preventing reverse flow that may lead to critical operating conditions. Simultaneously, their operation can induce system-damaging overpressure phenomena. This study investigates the behaviour of dual-disc check valves through an experimental approach, examining valve performance under quasi-steady and unsteady conditions. In a quasi-steady regime, flow coefficients and fluid–structure interaction instabilities are investigated for partially and fully open configurations. Under unsteady conditions, when the valves act to prevent reverse flow, the analysis focuses on the slam process and the associated overpressure, enhancing the understanding of the mechanisms underlying dual-disc check valve operations.

Keywords: check valves; dual-disc check valves; flow coefficient; slam process; instability

1. Introduction

Check valves are devices widely used in hydraulic systems to allow flow in one direction only and to prevent backflow in case a portion of the system stops or closes [1]. Check valves are opened by the flow passing through them and close if a pressure drop or backpressure occurs, bringing the movable part back into contact with its seat. The movement of the closure parts can be assisted by gravity or by springs. There are several types of these devices, which differ in the geometry of the closure. This study focuses on dual-plate check valves. These valves consist of a solid cylindrical body featuring two semicircular openings in the direction of flow, through which the fluid passes. Beyond these two openings are two semicircular gates, hinged on a pin along the diameter and pushed to close by a torsional spring that works against the flow. The presence of two independently moving discs complicates the analysis, as the motion of the two discs is not necessarily synchronous during closure [2]. Dual-plate check valves have a low slamming effect. Additionally, the total cost of a dual-plate check valve, including initial cost, maintenance cost, and energy cost, is less than that of a swing-check valve. [3]. An ideal check valve, when open, minimizes the pressure loss and closes at the instant the flow velocity reaches zero, not allowing the rise in reverse velocity [4]. This paper investigates dual-disc check valves under both steady and unsteady conditions. The aim is to provide a characterization of their hydraulic performance and dynamic response, with particular focus on the relationship between valve size, flow coefficient, and pressure loss.



Academic Editors: Vasilis Kanakoudis, Evangelos Keramaris, Francesco De Paola, Habib Muhammetoglu and Theodora Papamitsou

Published: 8 July 2026

Copyright: © 2026 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

2. Experimental Investigation Method

The objects of this study are three dual-disc check valves (supplied by OMB valves Cenate Sotto, Bergamo, Italy), such as the one shown in Figure 1a, having a nominal diameter of 2, 3, and 4 inches, which are composed of a valve body, two semicircular discs and a torsional spring, with the discs hinged along the diameter. The dimensions of the three valves are reported in Figure 1b and Table 1.



Figure 1. (a) Dual-plate check valve 3D rendering; (b) schematic drawing of valve front view.

Table 1. Geometric dimensions of the three valves tested.

DN	DN	A	D	h_{center}	b_{center}	A_{eff}
[inch]	[m]	[m ²]	[m]	[m]	[m]	[m ²]
2	0.0508	0.002027	0.0376	0.0349	0.014	0.00060
3	0.0762	0.004560	0.0612	0.0578	0.02	0.00174
4	0.1016	0.008107	0.0750	0.0707	0.025	0.00258

Experimental Setup

The schematic of the test rig, located in Laboratorio Idraulica Fantoli at Politecnico di Milano, is shown in Figure 2. The system is fed from an elevated tank able to maintain a hydrostatic head. The absolute upstream pressure at the test section is approximately 2.2 [barA]. The flow incoming in the system can be controlled by the valve UV1, which was kept fully open for the experiments, and by the downstream valve DV3, which was used during the experiments. Water from the tank enters the system and goes through the test section, where the check valve under study is installed. The system is also fed by a secondary circuit boosted by two pumps. The secondary circuit is shut off during the stationary test, while it is turned on to perform the unsteady, or transient, tests, allowing reverse flow to develop. The valve DP is needed to regulate the discharge of the two pumps and allow water flow when DV3 is closed.

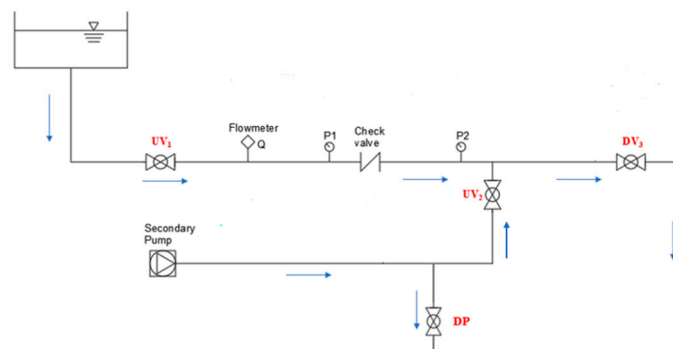


Figure 2. Schematic of the test rig. Blue arrows represent the flow directions. In red the names of the different control valves in the system.

To induce closure of the check valve during normal operation (with pumps running and the main pipe supplied by the tank), valve *DV3* is closed to stop the flow. The resulting pressure transient propagates upstream, superimposes with the pump discharge pressure, and initiates check valve closure upon arrival. During the test, the following measurements are collected: upstream pressure *P1* at *2D* from the tested valve; downstream pressure *P2* at *6D* from the valve (as required by the EN Standard [5]); flow rate *Q*; valve body accelerations via an accelerometer; and fluid temperature.

3. Valve Steady Characterization

To characterize the valves, stationary tests are conducted. Starting from the configuration where the *DV3* valve is fully closed, its opening is increased in steps until it is fully open. For each opening step, after the initial transient, measurements of *P1*, *P2*, *Q*, and *T* were recorded. From the acquired measurements are calculated:

- Reynolds number for the pipe

$$Re = \frac{\rho v D}{\mu}, \quad (1)$$

where *D* is the pipe diameter, *v* = *Q*/*A* the mean flow velocity, with *Q* being the flow rate and *A* = $\pi(D/2)^2$, ρ is the fluid density, and μ is the dynamic viscosity.

- Flow coefficient

$$K_v = Q \sqrt{\frac{\rho/\rho_0}{\Delta p}}, \quad (2)$$

where ρ_0 is the density of water at 25 °C at atmospheric pressure.

- Euler number (pressure loss coefficient)

$$Eu = \frac{\Delta p}{\frac{1}{2}\rho v^2}, \quad (3)$$

where $\Delta p = P2 - P1$ is the pressure differential across the valve.

The results of the stationary tests are presented in the following graphs, Figure 3a,b, reporting the behaviour of the flow coefficient and Euler number against Reynolds number.

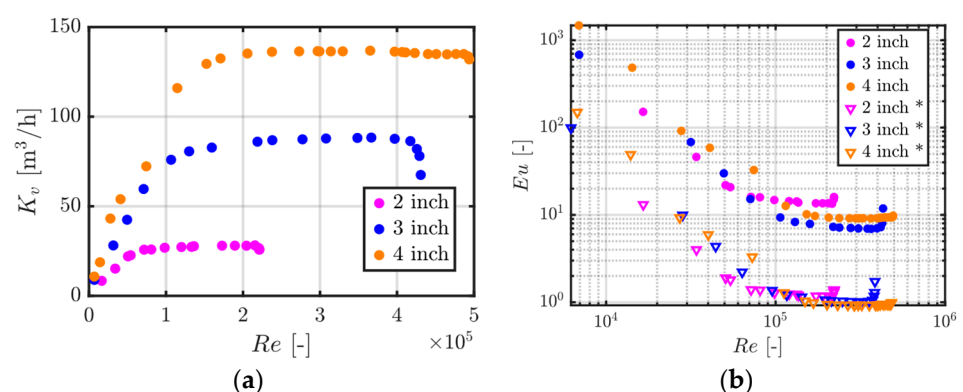


Figure 3. (a) Flow coefficient and (b) Euler number as functions of Reynolds number for 2-inch (magenta), 3-inch (blue), and 4-inch (orange) dual-plate check valves. Panel (a) is shown on a linear scale, displaying the Reynolds number ranges within which the valve operates under fully open conditions: $1\text{--}2 \times 10^5$, $2\text{--}4 \times 10^5$, and $2.4\text{--}4.9 \times 10^5$ respectively. Below these ranges, the valve is not fully open, while beyond the upper bounds, cavitation occurs. The corresponding flow rate ranges are derived from these *Re* values. Panel (b) is shown on a logarithmic scale; Reynolds and Euler numbers are computed using both the nominal diameter and the effective flow area with its hydraulic diameter; the latter are indicated with a star (*) in the legend and triangles in the graph.

The characterization of this type of valve differs from that of control valves, since a specific opening cannot be imposed. The opening depends on the flow, and only the fully open valve condition can be known; this corresponds to the maximum Kv condition. Beyond a certain Δp , the valve will be fully open; for smaller Δp , the opening cannot be determined without direct internal observation of the valve. The valve is in a fully open condition when the Kv becomes independent of the Reynolds number. The maximum flow coefficient was calculated as the average of the Kv values in the self-similar regime. The values are reported in Table 2.

Table 2. Maximum values of flow coefficients, effective area—nominal diameter ratios and cavitation index.

D	Kv_{MAX}	A_{eff}/D^2	σ^*
[inch]	[m ³ /h]	[-]	[-]
2	27.79	0.231	1.45
3	87.31	0.299	1.78
4	135.82	0.250	1.60

Usually, Kv_{MAX} in valves is scaled with D^2 ; however, applying this relationship to the check valves under examination does not yield the correct scaling. A more accurate scaling relationship, yet not perfect, is obtained by considering the effective area A_{eff} , namely the area of the inlet holes in the valve.

By plotting the non-dimensionalized pressure losses, Figure 3b, using the Euler number as a function of Re , it can be observed how these vary with the flow regime. First, it is noted that the three valves do not have the same Eu value for the same Re , as would occur in the case of perfect similarity between the valves; this indicates that they are not perfectly scaled with each other. Additionally, a particular behaviour is observed: the Eu values for the 4-inch valve fall above those of the 3-inch valve. This behaviour occurs because the ratios between effective areas and nominal diameters of the valves do not increase when going from 2 to 4 inches, as reported in Table 2. Figure 3b also reports the Euler number and Reynolds number computed using the effective area to obtain the average flow velocity: $v = Q/A_{eff}$. In the Reynolds number, the hydraulic diameter $D_h = 4A_{eff}/P_w$ is used, computed using the effective area and the wetted perimeter. Using this scaling, the Euler number values for the 4-inch valve fall below the values of the 3-inch valve, as would be expected. Therefore, using the effective area at the inlet as a scaling parameter gives a more reliable scaling. The Eu values are closer to each other, in the self-similarity region with Re_h .

For completeness, the cavitation index σ^* [6] was evaluated, identifying the threshold value at which cavitation begins to affect the valve performance by reducing the effective flow area and, consequently, the Kv . The cavitation index is defined as $\sigma^* = (P_1 - P_{vap})/\Delta p$, where P_{vap} is the vapour pressure of the fluid, and Δp is the pressure drop across the valve. The threshold values are reported in Table 2 and show that the value of σ^* does not increase with valve dimension, suggesting that the scale parameters could be refined to optimize the design of valves of different dimensions.

4. Dynamic Characterization

The transient phenomenon in check valves occurs during flow deceleration, usually following discharge valve closure or pump shutdown, and involves a complex interaction between fluid dynamics and disc motion [1]. A critical characteristic is valve lag: the discs do not close instantaneously when fluid velocity reaches zero, allowing reverse flow to develop. As the discs approach the seats, the reduction in flow area generates a pressure increase that abruptly accelerates the final closure; this phenomenon is known as “check

valve slam" [7]. Water hammer is caused by the sudden arrest of the reverse-flow velocity (V_R) at the moment the discs impact the seats. The magnitude of the pressure peak ΔH can be estimated using the Joukowsky formula:

$$\Delta H = \frac{c V_R}{g}, \quad (4)$$

where ΔH is the magnitude of the pressure peak in [mH₂O], c is the speed of sound in the fluid, and g is the gravitational acceleration.

To predict and mitigate these effects, valves are characterized through different methods:

1. Dynamic Performance Curves (Provoost Method [8]): these relate the maximum reverse velocity (V_R) to the average fluid deceleration (dV/dt).
2. Dimensional Analysis: Dimensionless groups are employed to compare valves of different sizes.

Ideal valves exhibit curves close to the horizontal axis (low V_R even at high decelerations) [7]. For laboratory characterization of check valves, the pressure increase, ΔH , downstream of the valve is measured during the transient. The reverse velocity is then calculated by inverting the Joukowsky equation. This operation appears trivial, but it is not, as it is necessary to distinguish the pressure increase due to the shutdown maneuver from that due to the check valve closure. Even more challenging is identifying the Δt during which flow arrest occurs, which is necessary to compute the deceleration. The procedures found in the literature for identifying points A-B-C-D on the pressure profile downstream of the check valve, such as that shown by Ballun [9], are not always applicable. Therefore, to select these points, which are necessary to calculate reverse velocity and deceleration, different methodologies are used, comparing upstream and downstream pressure data and the data collected with the accelerometer. Figure 4a shows, as an example, one of the graphs obtained from the dynamic tests for the 2-inch valve, showing upstream pressure, downstream pressure, flow rate, and valve body acceleration. From these graphs, the characteristic points are derived for the calculation of reverse velocity and deceleration. In the graphs for the 3- and 4-inch valves, not shown here, a lower pressure peak and increasing coupling between upstream and downstream pressure signals during the transient are observed, particularly for the 4-inch valve. The dynamic test results for deceleration and reverse velocity are plotted in Figure 4b alongside some literature data for dual-disc check valves.

The test rig allowed for a greater variation in the static and dynamic boundary conditions for the 2-inch valve, enabling a larger number of measurements, as can also be seen from the graph in Figure 4b. The present experimental data show considerable dispersion. The key finding is that the tested valves generate maximum reverse velocity values (and consequently pressure surges during closure) that remain relatively constant regardless of deceleration rate and decrease with increasing nominal diameter. The literature data are represented as fitted curves and scatter triangles, and the underlying point data from which these curves were derived are frequently unavailable, as they originate from older studies. Consequently, it is not possible to determine whether the data dispersion observed in the present study was also evident in previous research. The literature data nonetheless highlight that the dynamic behaviour of valves with identical nominal diameters can vary substantially, as demonstrated by comparing the findings of Collier [1] and Ellis [10]. By contrast, those tested by Thorley [11] and Provoost [12] appear to show more consistent behaviour with each other. Regardless, combining different literature datasets in a single plot reveals that these valve responses do not collapse onto a single curve but instead occupy a dispersed area in the parameter space.

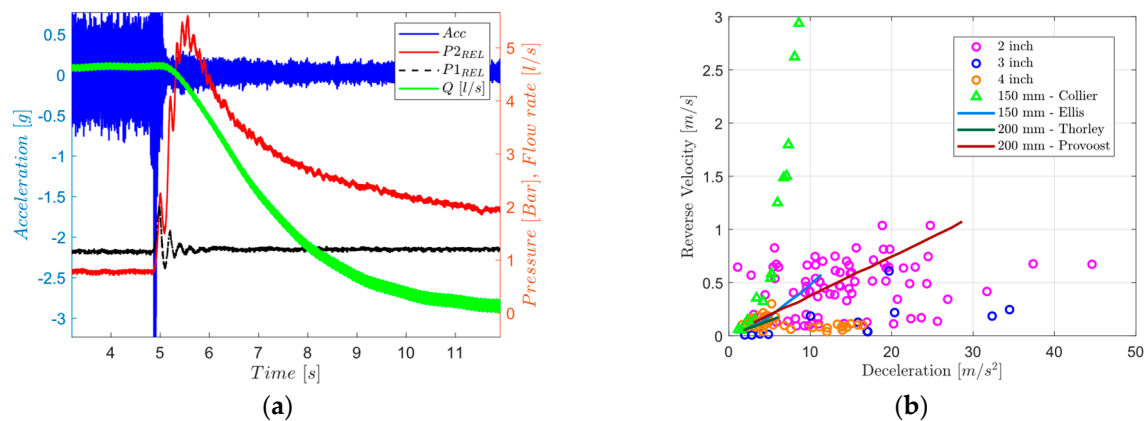


Figure 4. (a) Example of the data acquired during dynamic tests, in this case for the 2-inch check valve. Initial conditions $Q = 4.74$ [l/s], $P1 = 1.17$ [bar], $P2 = 0.77$ [bar]; (b) comparison of reverse velocity and deceleration relationships in dual-plate check valves. Present experimental results (circles: 2-, 3-, 4-inch) are compared with literature data (triangles and line: Collier [1], Ellis [10], Thorley [11], Provoost [12]. Experimental data from past study were extracted from the graphs reported in Reference using PlotDigitizer).

5. Conclusions

This study analyzes the behaviour of dual-disc check valves with three diameters (2, 3, and 4 inches) under steady and unsteady conditions. Within the fully open range, the maximum flow coefficient (K_v) of the valves and their behaviour within the operability range prior to cavitation were experimentally determined. In addition to the K_v , the behaviour of the Euler number as a function of Reynolds number was analyzed. Under maximum K_v conditions, it was found that the nominal pipe diameter is not an appropriate scaling parameter; the effective flow area was identified as a more reliable. Greater accuracy in the scaling parameters is also suggested by the analysis of the cavitation index σ^* . Moreover, the transient tests show that the reverse velocity remains relatively constant, regardless of the deceleration rate. This makes the response of the valves difficult to predict, even if the outcome is positive. Furthermore, the significant effect of fluid-structure interaction on the dynamic response of these valves suggests that refining the design of the valves could improve the stability and predictability of their fluid-dynamic behaviour.

Author Contributions: Conceptualization, S.M. and G.F.; methodology, F.V.M., N.C. and M.D.; software, N.C. and F.V.M.; validation, F.V.M.; formal analysis, F.V.M.; investigation, F.V.M., N.C. and M.D.; resources, S.M.; data curation, F.V.M.; writing—original draft preparation, F.V.M.; writing—review and editing, S.M.; visualization, F.V.M.; supervision, S.M. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: The raw data supporting the conclusions of this article will be made available by the authors on request.

Acknowledgments: We would like to thank the technicians of the hydraulic laboratory of the Politecnico di Milano for their support in the installation of the different experimental set-ups, and OMB Valves for providing the valves that are the subject of this work.

Conflicts of Interest: The authors declare no conflicts of interest.

References

1. Collier, S.L.; Hoerner, C.C. A Facility and Approach to Performance Test of Check Valves. *J. Energy Resour. Technol.* **1983**, *105*, 62–66. [[CrossRef](#)]
2. Chang, Z.; Jiang, J. Study on Transient Flow and Dynamic Characteristics of Dual Disc Check Valve Mounted in Pipeline System during Opening and Closing. *Processes* **2022**, *10*, 1892. [[CrossRef](#)]
3. Sotoodeh, K. Comparing Dual Plate and Swing Check Valves and the Importance of Minimum Flow for Dual Plate Check Valves. *Am. J. Ind. Eng.* **2018**, *5*, 31–45. [[CrossRef](#)]
4. Thorley, A.R.D. *Fluid Transient In Pipeline Systems*, 1st ed.; D. & L. GEORGE LTD: Herts, UK, 1991; pp. 219–243.
5. *EN 60534-2-1*; Industrial-Process Control Valves. International Electrotechnical Commission: Geneva, Switzerland, 2011.
6. Malavasi, S.; Corretto, R. Prediction and handling of cavitation in rotary control ball valves. In Proceedings of the Valve World Conference & Exhibition, Düsseldorf, Germany, 26–29 November 2012.
7. Thorley, A.R.D. Check Valve Behavior Under Transient Flow Conditions: A State-of-the-Art Review. *J. Fluids Eng.* **1989**, *111*, 178–183. [[CrossRef](#)]
8. Provoost, G.A. The Dynamic Behavior of Non-Return Valves. In Proceedings of the International Conference on Pressure Surges, Canterbury, England, 25–27 March 1980.
9. Ballun, J. A methodology for predicting check valve slam. *J. Am. Water Works Assoc.* **2007**, *99*, 60–65. [[CrossRef](#)]
10. Ellis, J.; Mualla, W. Numerical Modeling of Reflux Valve Closure. *J. Press. Vessel. Technol.* **1986**, *108*, 92–96. [[CrossRef](#)]
11. Thorley, A.R.D. Dynamic Response of Check Valves. In Proceedings of the International Conference on Pressure Surges, Bath, England, 21–23 September 1983.
12. Provoost, G.A. A Critical Analysis to Determine Dynamic Characteristics of Non-Return Valves. In Proceedings of the International Conference on Pressure Surges, Bath, England, 21–23 September 1983.

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.