Technical Paper

A Review of Check Valves in Unsteady Flow

Behavior, Analysis, and Design

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A REVIEW OF CHECK VALVES IN UNSTEADY FLOW: BEHAVIOR, ANALYSIS, AND DESIGN

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ABSTRACT

Check valves intend to prevent reverse flow with minimal impact to normal forward flow. They respond to transient conditions, and improper selection may result in unmitigated check valve slam – causing massive transient pressure surge that can easily cause damage throughout the connected piping network.

Dynamic characteristics show the dependence of closure velocity on fluid deceleration. This paper aims to clarify, summarize, and qualify the use of these curves in liquid flow. Computer simulation models and examples are described along with comparisons to field data. Basic design principles, guidelines for identifying at-risk systems, and strategies for check valve selection are presented.

Keywords: check valves, non-return valves, surge, water hammer, unsteady flow

NOMENCLATURE

a Wavespeed [m s^{-1}]
B^+, B^- Compatibility impedance [kPa kg^{-1} s]
C^+, C^- Compatibility capacity [kPa]
c Speed of sound; celerity [m s^{-1}]
D Nominal diameter [m]
F_o Net force on fully open valve [N]
F_s Net force on fully closed valve [N]
K_v Flow coefficient [m$^3_{H_2O,STP}$ hr^{-1} bar$^{-0.5}$]
$\dot{m}$ Mass flow rate [kg s^{-1}]
p Static pressure [kPa]
ρ Static density [kg m^{-3}]
u Flow velocity [m s^{-1}]
u_o Minimum flow velocity to fully open valve [m s^{-1}]
u_R Reverse flow [m s^{-1}]
$u_{R,close}$ Reverse flow at closure [m s^{-1}]
x Distance from seat of translating element [m]

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1. INTRODUCTION

Check valves are ubiquitous in industry and provide a very straight-forward function: the prevention of reverse flow. For example, a system may drain in reverse through a shut down pump, causing process issues or risking damage to the pump. This is prevented with the proper check valve.

The check valve is expected to change between open and closed states routinely and automatically – otherwise a manual valve would be used. In many cases, the open position corresponds to normal operation, and the closed position to some special state.

Normal operation generally drives the optimization of system design. For an open check valve, this means minimizing pressure loss. From this steady-state design perspective, any check valve that has low pressure loss when open might be considered a good selection. If there are a variety of options, the secondary optimization parameter may simply be cost.

This steady-state approach overlooks the transient nature of changing the check valve state. Changes of state generate unsteady flow, and sudden changes generate *pressure surge* - large transient pressure waves. Because check valves are generally intended to change position quickly, causing potentially problematic surge, the transient interaction with the system must be considered.

2. FUNDAMENTALS OF UNSTEADY FLOW IN SYSTEMS

2.1 Acoustics

Disturbances in fluids generate waves that travel at some finite speed. Any transition from one steady state to another is such a disturbance, and a check valve state change is no exception.

Fundamental fluid dynamics dictates that infinitesimal disturbances generate acoustic waves which propagate at the speed of sound. An acoustic wave is a coupled pressure and velocity wave, whose magnitudes are related by the simple differential Eq. (1) [1].

$$dp = -\rho\ c\ du \tag{1}$$

2.2 Large Amplitude Waves

Non-infinitesimal disturbances to flow velocity result in large amplitude changes in pressure, often referred to as pressure surge. Surge can be estimated for liquid flow with Eq. (2), a non-infinitesimal version of Eq. (1) known as the Joukowsky Equation [2]. The elasticity of non-fluid system components reduces the speed of an acoustic wave [3], so the speed of sound c is replaced by *wavespeed* a.

$$\Delta p = -\rho \, a \, \Delta u \tag{2}$$

These acoustic surge waves travel throughout the entire interconnected fluid system - meaning that areas very remote from the device that caused the original disturbance will see potentially problematic surge [3, 4].

The Joukowsky Equation neglects many phenomena that impact unsteady flow and is sometimes mistakenly considered to calculate a conservative maximum possible pressure surge when Δu is set to the steady-state velocity. Surge pressures higher than this are possible [5], and proper analysis requires a true unsteady system-level simulation, as discussed in more detail later.

2.3 Finite Duration Waves and Communication Time

An acoustic wave travels at a finite wavespeed, and a device that generates such a wave will not see relief until this wave has reflected from a boundary and returned – a process that requires a finite duration known as *communication time*.

A finite duration event like a valve closure will create a *wave family* – an infinite series of infinitesimal acoustic waves with finite length. However, if the event duration is less than the communication time, the system will still see the full Joukowsky surge. Any sufficiently rapid event is functionally equivalent to an instantaneous event in the context of surge.

2.4 Method of Characteristics

The Method of Characteristics (MOC) is widely regarded as the standard for the simulation of one-dimensional acoustic transients in liquid filled piping networks. This approach is well documented in many excellent references [3, 4], and it is not the intent of this paper to discuss the MOC in detail.

The fundamental equations of unsteady fluid dynamics applied to a piping network result in a system of non-linear hyperbolic partial differential equations. The MOC simplifies this system by manipulating it into ordinary differential equations valid along wave paths. With straightforward and justifiable assumptions for liquid flow, these ODEs result in linear algebraic compatibility equations that relate flow and pressure along paths of fixed wavespeed, as shown in Eq. (3).

$$p = C^{+} - B^{+} \, \dot{m} \qquad p = C^{-} + B^{-} \dot{m} \tag{3}$$

At an arbitrary "black box" device with an upstream and downstream pipe, these equations essentially describe transient response of the system to the device for a particular time step of simulation. The mass flows into and out of simple devices are always equal, which results in the linear relationship Eq. (4) between mass flow through the device and pressure change across it.

$$\Delta p = (C^{-} - C^{+}) + (B^{-} + B^{+})\dot{m} \tag{4}$$

The device itself must provide an additional equation relating the transient mass flow to the transient pressure delta to solve a system of two equations and two unknowns. This is not always straightforward, as the device itself may have additional variables that are transient in nature.

The MOC provides for highly accurate and relatively simple unsteady simulations of an entire network and avoids the deficiencies that arise from rule-of-thumb type approaches like the use of the Joukowsky Equation.

3. CHECK VALVE SLAM

An ideal check valve allows forward flow with zero pressure loss for any positive pressure differential, and prevents all reverse flow, no matter how small in magnitude. It would therefore close instantly at exactly zero flow, meaning the pressure surge indicated by the Joukowsky Equation (2) would be zero.

Real check valves take time to close, and don't react only to flow reversal. They will begin closing before the flow velocity reaches zero and will not come to a complete close until after flow has reversed. Any closure against a non-zero flow ($u_{R,close} \neq 0$) will generate pressure surge ($|\Delta p| > 0$). This sequence of events is referred to herein as *check valve slam.*

If the check valve closes faster than the system communication time, the closure is *effectively undamped* [6], and the Joukowsky Equation often well approximates the surge pressure Δp. In effect, this is equivalent to representing the finite closure event as an instantaneous closure that occurs at the closure velocity. This approach can be found in literature as early as 1959 [7] and is widely used to this day [8, 9]. A simple example of velocity through the check valve over time is shown in Fig. 1.

Even moderate slam, from halting a *reverse flow* of $u_R = 1$ m s^{-1} in water ($\rho = 1000$ kg m^{-3}, $a = 1200$ m s^{-1}), results in a Joukowsky pressure surge of $\Delta p = 1200$ kPa that travels throughout the system. The surge is positive - *upsurge* - on the downstream side of the check valve, raising the pressure, and negative - *downsurge* - on the upstream side. A 0.5 m diameter expansion loop well removed from this check valve could easily see transient longitudinal forces greater than 200 kN [10].

If the closure time is on the order of the communication time or longer, the check valve can be said to be *damped*. In these

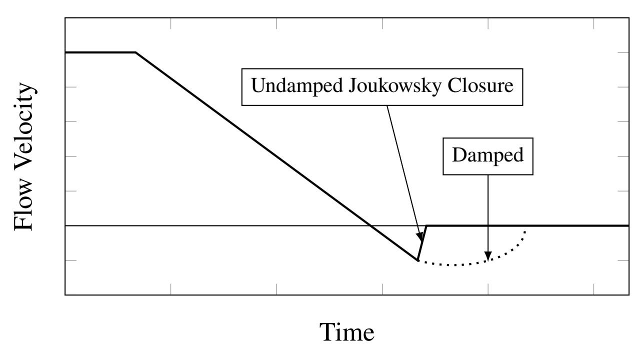


FIGURE 1: SIMPLE VELOCITY BEHAVIOR AT A CHECK VALVE

cases, pressure waves from the generating transient - perhaps a pump trip - begin the valve closure. The closure begins to generate surge waves of its own, which will eventually reflect from system boundaries and re-interact with the check valve. If the valve is still partially open at this time, the reflected wave may serve to reopen the valve, or may encourage it to close faster. Figure 1 shows the velocity behavior after reversal extending for a longer period. The actual velocity variation over this time is not likely to follow a smooth arc, its behavior depends on the system dynamics and cannot be analyzed independently of the piping network. Such systems must be investigated on a case-by-case basis.

Check valves generally close quickly, and can often be treated as undamped. To minimize surge levels this is in fact desirable, even though general wisdom on mitigating surge is to take actions slowly.

4. FORCE BALANCE MODELS

4.1 Multi-dimensional, Multi-physics models

The "gold standard" in simulation might be to construct a multi-dimensional multi-physics model that accounts for as many details of a real check valve's behavior as possible. These models are valuable but difficult and expensive to develop, require enormous amounts of input data, and simulations can take significant time to run [11].

4.2 Practical Simplifications

A force balance model can become much more tractable when simplifying assumptions are made. Namely, the solution should follow the one-dimensional nature of the MOC. This means there is a unique mass flow through the valve at any given time, and there are unique upstream and downstream pressures. Such a model is unable to account for effects like variations in pressure across the valve face, and pressures will always be assumed to act uniformly on the faces of the check valve.

To make the model geometrically simple, pressure and weight forces are assumed to act through specified centers of gravity or area. A uniform material density is used to calculate buoyancy and weight, and additional applied forces functionally dependent on valve position can be supplied.

Still missing from this model is a relationship between flow and pressure loss to allow solution via the MOC. It would again be possible to complete a multidimensional analysis to estimate pressure loss based on valve design and position, but this is overly complicated. Instead, the device can be tested to determine a relationship between flow coefficient K_v and position. The flow coefficient provides the relationship between pressure loss and flow, but position is a new variable – requiring a solution to the force balance described previously.

This simple force balance model is well-described in related literature [3, 6, 12, 13], and will not be discussed in detail here.

4.3 Inertia and Transient Behavior

A unique valve position can be determined for a steady-state flow with the aforementioned force balance. This is insufficient for transient flow because the system has inertia and will thus not respond instantly to changes in the force balance. Fundamentally, this approach amounts to an application of Newton's Second Law in combination with the pressure loss relationship and MOC compatibility equations.

An ideal check valve with zero inertia changes position instantly from fully open to fully closed at the exact moment of flow reversal, preventing slam. The speed at which real valves can react is finite and dependent upon inertia and the applied forces. Mechanisms that aid closure, such as springs or counterweights, can help overcome inertia and reduce closure time. Travel distance must also be considered - a lightweight swing check that must travel a full 90 degrees may take longer to close than one with more inertia that only needs to travel 40 degrees.

In other words, the closure time is a function of, at least, the mass of the physical components and the volume of displaced fluid. This effect can be quantified to some degree by *natural closure time* - the time a valve takes to close when being released from fully open under stagnant conditions [6, 14, 15] .

The movement of the check valve through the process fluid adds additional resistance to valve motion. Accounting for this directly is difficult, and a common approximation is to account for this affect by adding an effective mass to the moving check valve components. For swing check valves, as an example, this mass is assumed equal to the mass of a sphere of fluid of diameter equal to the check valve [16, 17].

4.4 Minimum Velocity to Fully Open Valve

A parameter of interest is the minimum forward steady-state velocity, u_o, required to keep the valve in a fully open position. This value is of interest for check valve closure because only flow velocities below this value will have a dynamic effect. In other words, for the same fluid deceleration, the surge generated by check valve slam would be expected to be the same for any steady-state velocity above u_o.

5. THE DYNAMIC CHARACTERISTIC MODEL

Even the simplified physical model is often impractical and unnecessarily detailed. As discussed in Sect. 3, a key characteristic of the dynamic response is the maximum reverse velocity at closure $u_{R,close}$, so it is logical to build a model using this value as the dependent parameter.

For this model, the check valve remains in the fully open position until $u_{R,close}$ is reached, at which point the valve will instantaneously close, generating a Joukowsky level surge. As discussed in Sect. 2.3, a non-instantaneous closure will show the same surge as long as the closure time is less than the communication time, meaning this assertion is reasonable for any *undamped* check valve system.

Provoost [18] published a paper in 1980 concerning this question. The work details the dynamic testing of a ball and a swing check valve, and Provoost states that "[this result] implies that the fluid velocity gradient dominates the dynamic phenomena during valve closure...", making this the first known publication of a model relating the closure velocity $u_{R,close}$ with fluid deceleration du_R/dt. Provoost published additional papers shortly thereafter [19, 20], strengthening this model. Collier and Hoerner (1983) present an approach for laboratory generation of dynamic characteristics [21], and along with Thorley (1983) [22], present additional dynamic characteristic curves.

5.1 Non-dimensional Dynamic Characteristic Model

Dimensional analysis is a common method for identifying relationships in systems. The Buckingham π theorem requires that the dimensional variables fully characterize the system's behavior if the resulting dimensionless groups are to completely represent the system [23]. The dynamic behavior of a check valve in a piping network is a complicated system with many variables. Some simplification is required for this process to be practical, but it cannot be simplified to only 2 or 3 dimensionless parameters.

Koetzier, Kruisbrink, and Lavooij applied this to the dynamic characteristic model in 1986 [24], relating nominal diameter D and a ratio of material to fluid density ρ_m/ρ_f with du/dt and u_o, resulting in the three dimensionless groups shown in Eq. (5).

$$\frac{u_{R,close}}{u_o} = f\left(\frac{D}{u_o^2}\frac{du}{dt},\ \frac{\rho_m}{\rho_f}\right) \tag{5}$$

Thorley [25] expanded this to 8 dimensionless groups, of which he notes 5 as of "primary importance", providing the relation in Eq. (6). F_o and F_s are net forces when fully open and closed, respectively, and x is the valve opening distance.

$$\frac{u_{R,close}}{u_o} = f\left(\frac{D}{u_o^2}\frac{du}{dt},\ \frac{\rho_m}{\rho_f},\ \frac{F_o}{F_s},\ \frac{x}{D}\right) \tag{6}$$

Throughout the literature, dynamic characteristic curves are presented as traces on the du_R/dt vs. $u_{R,close}$ plane for a particular valve type or configuration - such as "strong nozzle" or "weak split disk." This further simplifies the already simplified model above to Eq. (7).

$$\frac{u_{R,close}}{u_o} = f\left(\frac{D}{u_o^2}\frac{du}{dt},\ \text{valve type}\right) \tag{7}$$

It should be noted that two dimensionless dynamic characteristics may vary substantially between two valves of such simple designation. For this reason, the authors highly recommend the use of dimensional dynamic characteristics from a valve as close in size and design as possible to the valve under consideration - generic dimensionless characteristics are excellent for initial investigation, but should be treated with caution. This is a general issue not strictly related to check valves - the first author has experienced the same issues firsthand in other areas of transient simulation, namely in the non-dimensionalization of four quadrant pump curves [26] and in viscosity corrections for pumps [27].

6. AN OVERVIEW OF HISTORICAL CHARACTERISTICS

6.1 Background

Many dynamic characteristic curves have been generated from the early 1980s to the present day. These curves fall generally into one of three categories:

1. Lab or field measured.
2. Adjusted, perhaps through the non-dimensional model, from measured data to another size, density, etc.
3. Calculated from simulation results.

Because this model is highly simplified and empirical in nature, the authors would argue that the first group covering actual, measured data, should be used whenever possible. If such data for the valve in question is not available, and no existing data set is for a similar valve, the others can be used as an initial starting point.

Dynamic characteristics in the literature are generally presented by valve type and, perhaps, by an additional distinguishing characteristic such as "strong" or "weak" springs. It is uncommon that these curves provide extensive data on the valves - generally any additional data is limited to valve diameter, steady-state pressure loss data, and occasionally minimum velocity to fully open.

6.2 Example Curves

The curves presented in Fig. 3 were collected from as wide a variety of sources as possible. However, this type of testing is not common compared to steady-state testing. The dynamic tests that have been carried out have in many cases been done so by a limited number of individuals, companies, or labs.

The authors cataloged many such curves in an effort to provide clarity to the origin of various test data, to validate the quality of data in common use as much as possible, and to understand the variability in valve behavior. Only a subset of the data cataloged is presented here, selected to demonstrate the high uncertainty generic data imparts to transient hydraulic studies. This uncertainty extends not only to the range of curves available for a particular type of valve, but often to an individual data set. We call attention to two sets of data here for illustration.

The first data set has been widely republished [4, 8, 9, 22, 25, 28], and is arguably the very first dynamic characteristic - a set of four deceleration vs. reverse velocity data points for a 100mm moving ball valve as reported by Provoost in 1980 [18], and noted as curve "A" in Fig. 3a. One concern is that the data tabulated by Provoost does not seem to reflect the values for velocity charted in the same paper. The authors could not determine an explanation for this discrepancy, in particular because the values for initial velocity and deceleration Provoost provides do match the velocity trace. This apparent error has been carried forward through many other publications and is presented as a "standard" dynamic characteristic for ball checks. This concern is echoed throughout the catalog of curves - authors often republish curves, sometimes making errors, or sometimes providing different subsets of information and descriptions of the valves. This can cause significant confusion for the engineer trying to accurately apply the data.

The second data set noted here shows a concern that applies to much if not all of the curves, including the Provoost ball curve. This concern is simply scatter in the original measurements, which should not come as a surprise for any empirically collected data. This data is typically curve fit, and unfortunately the underlying data is often lost over time. This can present a very misleading picture that a particular type of valve follows a strictly defined characteristic. An example data set, shown in Fig. 2, from [14] shows that this is clearly not the case.

Notes on the Presented Curves. The curves in Fig. 3 and Table 1 were selected to show the range of data for different types

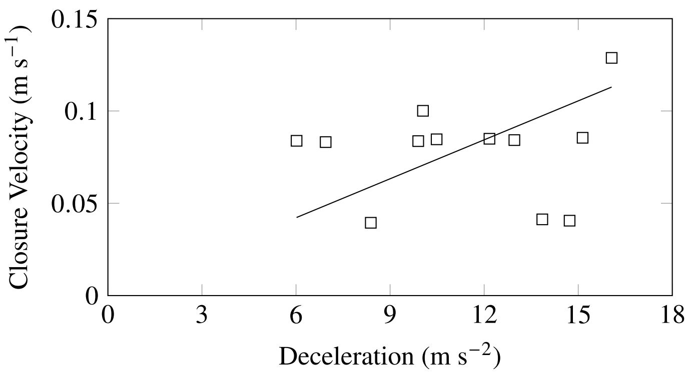


FIGURE 2: SCATTER IN A CHARACTERISTIC [14]

of valves. They cover a variety of sizes, and include features such as strong or weak configurations of identical valves. The fluid in all tests was water, or unspecified and assumed to be water. Only curves that appeared to be from measured data are presented.

The curves shown were fit by the authors either with linear or polynomial regression, and are forced to pass through the origin. In some cases this makes the curve appear to deviate from published curve fits, but the authors would argue the scatter in the data makes this irrelevant in practical application.

Most of the presented charts show a dashed region that approximately covers the domain where measured data is known to exist. These domains are shown together in the final Fig. 3h. The intent of this chart is to conclusively show the large variation in data from valve to valve, even among the same type. It is furthermore intended to show the general difference between different valve types - for example, that swing check valves are generally more likely to cause surge issues than other types.

Finally, readers familiar with the non-dimensionalization process may dispute these charts by arguing that the curves collapse at least somewhat in non-dimensional form. The authors understand and agree with this, but contend that in the context of *pressure surge*, the absolute value of velocity at closure is critical. In surge analysis, the size of a valve is a distant secondary concern compared to the actual magnitude of the halted velocity.

7. SIMULATION

7.1 Force Balance Model

The simulation of the simplified force balance model is carried out more or less as explained in Sect. 4. Every time step of simulation, a coupled system of equations is solved to determine the forces acting on the valve under the current transient conditions, and the valve is moved appropriately, with its motion limited by its inertia. Many descriptions of this model in detail can be found in literature.

7.2 Dynamic Characteristic Model

As this paper has focused on the dynamic characteristic model, some care will be taken here to describe its application in simulation. The actual numerical simulation, in regards to the MOC, is not any different than a regular valve solution. The check valve is assumed to remain at a fully open position until closure velocity is reached, at which point it instantly closes. The primary difficulties in applying the model are in the determination of the deceleration and the behavior of the valve after closure.

Non-linear Deceleration. If the velocity behavior is exceedingly simple - a linear decrease from some steady state value until closure as in Fig. 1 - then the determination of deceleration is clearly straightforward. However, in real systems, the decrease in velocity is never linear. Consider for example the speculative trace shown in Fig. 4. The velocity begins to decrease immediately, then more severely, but rebounds before making its final descent.

There are a few ways the deceleration could be calculated, assuming the final data point or time at which the deceleration is "set" happens at reversal:

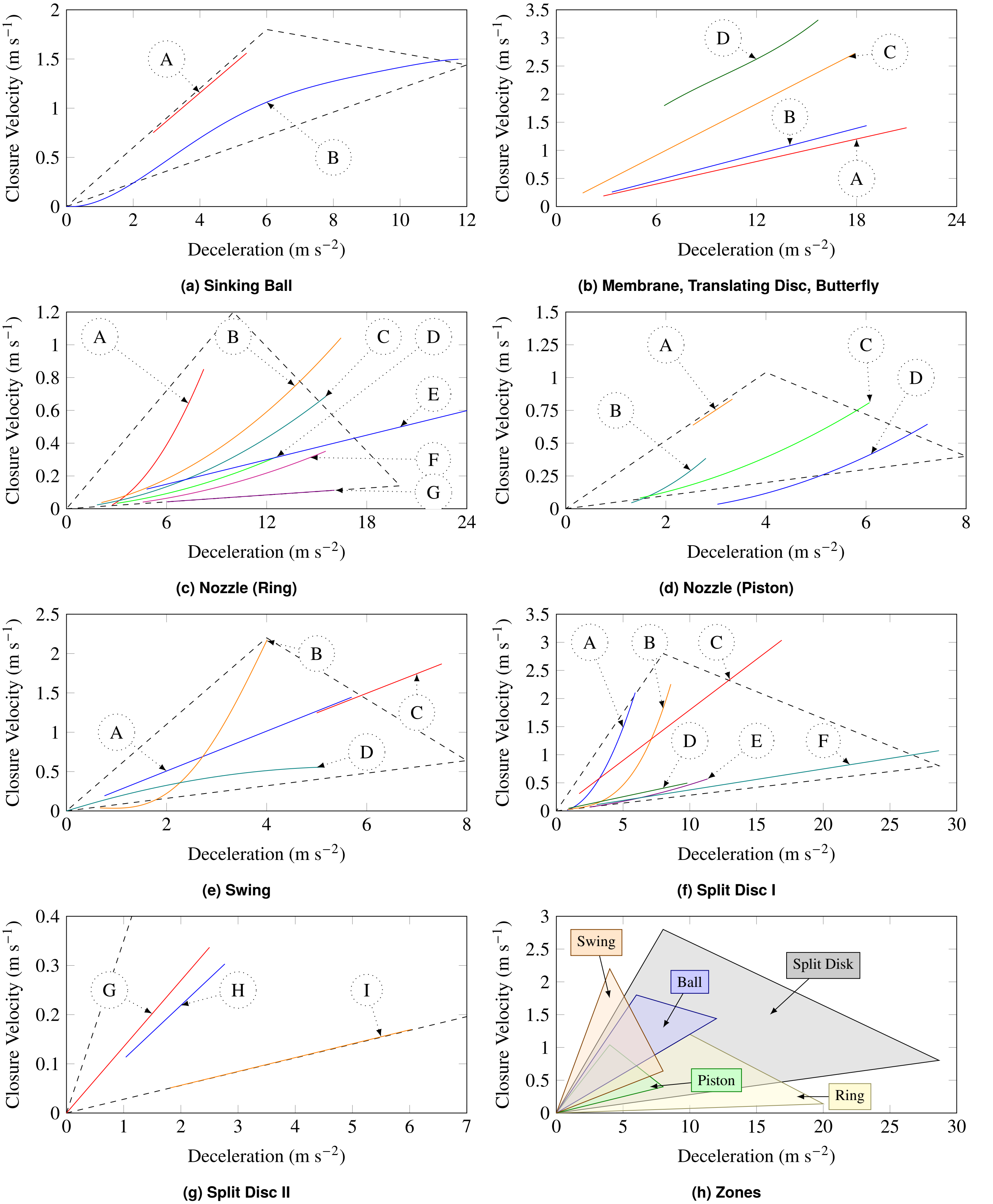
1. Starting at the first decrease in velocity (red).
2. Starting at the first "significant" reversal (blue).
3. Approximating the instantaneous deceleration (teal).
4. Starting at the last "significant" reversal (orange).
5. Starting some specified time *before* reversal by tracking a rolling window (violet).
6. One of the above items, but strictly if the current velocity is below the minimum velocity to fully open the valve (pink).

Clearly, there are a wide variety of options to choose from. The authors would argue that the velocity at the valve around the time of closure is the most physically significant, and should probably influence this decision. However, attempting to approximate instantaneous deceleration in simulation is dangerous - noise in the signal can easily create erroneous values with no meaning. This leads credence to supplying a specified time before closure, but this value is difficult to ascertain. Furthermore, because any variations in velocity above the minimum velocity to fully open have no impact on valve position, that range should be discarded.

An argument may be made that the interval (5, violet) should in general be no shorter than the *natural closure time* if the deceleration from the minimum to fully open option (6, pink) is less severe. Otherwise, the value from the minimum closure velocity should be used if possible, and if the velocity is always below this value, the last significant reversal (4, orange) should be used.

In effect, this encourages the use of the most severe deceleration that can be calculated with stability and is underneath the minimum to fully open velocity. This respects numerical difficulties and the physical nature of the closure.

Post-closure. How to handle the valve after closure is even less clear. A physical model would reopen on a positive pressure differential, and depending on the state of the system, likely flutter open and closed until settling to a final state. The dynamic characteristic model has no obvious way of handling this situation - and the short term oscillations would almost certainly not obey the characteristic behavior. The simplest, and perhaps only feasible option, is to simply leave the valve closed once closure has occurred, understanding that this could underestimate surge pressures but reopening requires a more sophisticated model.



(a) Sinking Ball

(b) Membrane, Translating Disc, Butterfly

(c) Nozzle (Ring)

(d) Nozzle (Piston)

(e) Swing

(f) Split Disc I

(g) Split Disc II

(h) Zones

FIGURE 3: DYNAMIC CHARACTERISTIC CURVES

TABLE 1: DYNAMIC CHARACTERISTIC CURVE SOURCES

Type (Fig. 3 subfigure)	Curve	Nominal Valve Diameter	Source
Sinking Ball (a)	A	100mm	Provoost 1980 [18]
	B	200mm	Provoost 1983 [20]
Membrane (b)	A	200mm	Kruisbrink 1996 [6]
Translating Disc (b)	B	200mm	Provoost 1983 [20]
Butterfly (b)	C	200mm	Kruisbrink 1996 [6]
Butterfly (b)	D	200mm	Kruisbrink 1996 [6]
Nozzle (Ring) (c)	A	32"	Gormley 2002 [29]
	B	32"	Gormley 2002 [29]
	C	32"	Gormley 2002 [29]
	D	300mm	Provoost 1983 [20]
	E	300mm	Perko 1986 [18]
	F	12"	Gormley 2002 [29]
	G	300mm	Perko 1986 [14]
Nozzle (Piston) (d)	A	300mm	Koetzier 1986 [24]
	B	800mm	Thorley 1989 [25]
	C	800mm	Koetzier 1986 [24]
	D	300mm	Koetzier 1986 [24]
Swing (e)	A	200mm	Curtis 1990 [30]
	B	6"	Collier 1983 [21]
	C	100mm	Provoost 1980 [18]
	D	200mm	Provoost 1983 [20]
Split Disc I (f)	A	6"	Collier 1983 [21]
	B	6"	Collier 1983 [21]
	C	6"	Collier 1983 [21]
	D	150mm	Ellis 1986 [31]
	E	150mm	Ellis 1986 [31]
	F	200mm	Provoost 1983 [20]
Split Disc II (g)	G	200mm	Provoost 1983 [20]
	H	200mm	Thorley 1983 [22]
	I	200mm	Thorley 1983 [22]

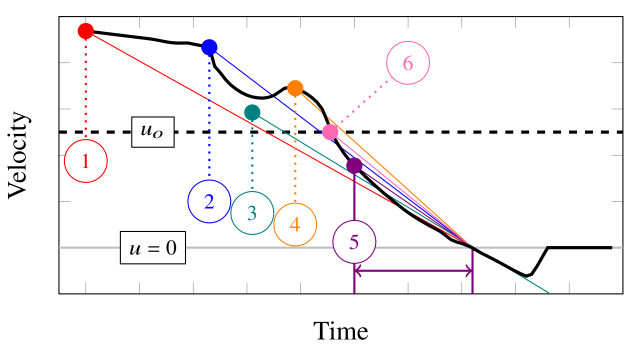


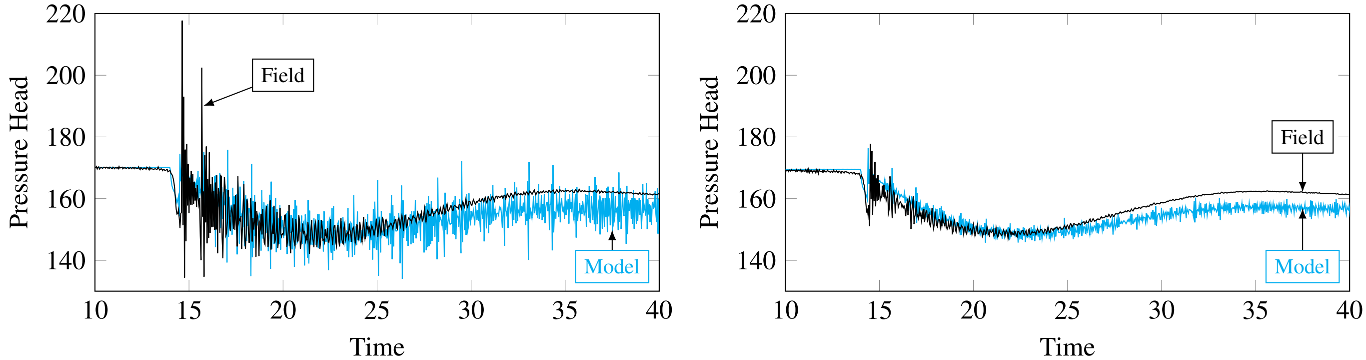
FIGURE 4: SPECULATIVE COMPLEX VELOCITY PROFILE

8. FIELD DATA COMPARISON

Despite the simplicity of the dynamic characteristic model, it can be shown to replicate the field behavior of check valves under surge events quite well. One such study showed the replication of primary surge in a municipal water pumping station outside of Barcelona, Spain [32]. Another example is of a cooling loop in a power station, as described in [30]. Throughout the literature, authors construct characteristic curves with measured data, but this data generally comes from a laboratory setting. While certainly valuable data, it is not as powerful as field behavior validating a simulation built with a model.

The Barcelona study concerned the selection of check valve to prevent the generation of severe surge. The system contained other surge mitigation equipment, which was not sufficient to eliminate the issues caused by a swing check valve. Field tests were taken for the swing check valve and an alternative nozzle check valve. The pressures recorded for the swing check were, predictably, far higher than those recorded for the nozzle check.

As part of this study, a numerical model was constructed to evaluate and understand the system operation. This was completed in the MOC-based liquid network commercial software AFT Impulse [9]. AFT Impulse contains a number of check valve models, including a set of dynamic characteristic curves. The swing and nozzle checks in the case study were modeled with



(a) Swing Check **(b) Nozzle Check**

FIGURE 5: FIELD VS. MODEL RESULTS FROM [32]

the dynamic characteristic approach.

The model peak pressures match within ≈5% for the nozzle check, and within ≈15% for the swing check. In the realm of network surge modeling, this is excellent agreement. Moreover, the primary surge behavior shown by the model clearly reflects the behavior of the measurements, albeit with less noise and stronger initial pressure spikes, as shown in Fig. 5.

9. SYSTEM DYNAMICS

Hydraulic systems are generally unique and require care during design and engineering analysis. Still, several configurations and attributes are known to create increased risk of check valve slam. A primary risk factor that can result in check valve slam is the presence of a high difference in potential energy across the check valve, which drives fluid deceleration at the check valve. Higher differences and/or maintaining it for longer will generate more severe fluid deceleration and thus larger reverse velocity.

In other words, during a transient event, if a large amount of potential energy in the form of pressure can form on the downstream side of a check valve as compared to the upstream side, this energy can serve to slam the valve.

This high differential is common in systems with:

- High static lift or elevation change between the check valve and downstream boundary.
- Parallel pumping arrangements where other pumps continue to provide pressure into a common header.
- Systems protected by hydropneumatic or surge tanks.

In the context of check valve slam, it's important to note high initial velocity as a significant risk factor, as it will likely amplify the severity of fluid deceleration, depending on the transient event. Additionally, large diameter check valves of the same type generally have a worse surge response - the same valve type is not a panacea for all systems.

It is critical to recognize that the entire system must be considered. For example, some systems are equipped with high rotational speed pumps, which likely have lower inertia compared to lower speed pumps of similar performance. During a pump trip, this low inertia lends itself to stronger fluid deceleration, which can worsen check valve slam completely independently of the check valve design.

9.1 Mitigation

Mitigation of check valve slam should be coupled with a comprehensive transient analysis of the system using an MOC based analysis tool. While many systems undergo analysis for typical primary transient events, check valve slam is often overlooked as a secondary transient event resulting from the initial disturbance. For instance, an analysis may focus on an driving event like a pump trip and evaluate the system's risk of low pressure and column separation. The responsive behavior of a check valve may inadvertently be neglected, but even the secondary event of check valve slam can be critically important in surge mitigation.

MOC based analyses are often carried out on a coarse grid. However, when conducting a check valve slam analysis, a refined MOC grid is typically essential to avoid wave dampening near the check valve that can erroneously reduce the fluid deceleration and pressure response when the valve slams. While this mesh refinement significantly increases computational time, it normally can be relieved with a shorter simulation duration. For check valve slam analysis, modeling the initial transient event, flow deceleration, check valve slam, and subsequent wave attenuation is typically confined to a simulation duration of 10 to 15 seconds, as opposed to the several minutes or more required for a more typical transient analysis.

Utilizing the fluid deceleration and other known system factors (maximum allowable pressures, forces, etc.), an engineer can choose from various check valves based on their dynamic performance curves. Some software suites enable the direct input of dynamic performance curves, both dimensional and nondimensional. Alternatively, a valve company may request fluid deceleration data from the analyst to facilitate valve selection.

Engineers can optimize check valve selection by integrating computer simulations for slam analysis with considerations of head loss, energy loss, and capital costs. This approach allows for the identification of the most cost effective design while ensuring protection against check valve slam within the system.

A system may require multiple forms of mitigation, due to the severity of check valve slam or other events, or because any one mitigation technique is not sufficient. Selecting a check valve that performs well in dynamic situations - e.g., low slam - may still show significant surge. In such cases the system as a whole must be considered, and equipment such as a surge vessel or relief system may be required. This investigation is only practical with a computer based simulation tool.

10. DESIGN BASICS

Check valves will always compromise between pressure drop and speed of closure. A proper check valve design should balance cost, including energy loss, and performance - speed of closure and reliability. The best check valve is tailor-made for the application with respect to size, spring, and material selection.

10.1 Steady Flow

In steady flow, the hydraulic force generated by the flowing fluid provides the opening action of the check valve disc. Spring and friction forces oppose the hydraulic force, resulting in a balanced situation in steady-state. In well designed check valves the friction force is very small compared to the spring force and has negligible influence. Depending on type of valve, the weight of the disc is also a concern - helping to open or close the valve depending on position and orientation.

An ideal check valve is fully open at all normal flow conditions. This gives the lowest energy loss due to pressure drop. It is also important that the force required to lift the disc out of the seat - *cracking pressure* - is as low as possible in order to allow a smooth start-up of the pump or compressor.

10.2 Unsteady Flow

Most important for unsteady flow conditions is the check valve dynamic behavior - how it reacts and performs under unsteady flow conditions. A check valve should be designed with this in mind. Ideally, the force acts on the entire area of the check valve disc during the full stroke (movement of the disc), and the stroke is as short as possible. With a rotating disc design, the area in contact with the flow changes during opening of the valve and the stroke is generally longer than valves of linear design.

The best dynamic performance is attained when the area remains the same at all valve positions. This is only possible with an axial flow check valve with a disc connected to a stem which is guided on the central horizontal axis - a nozzle design. A nozzle design with a solid disc - a piston-type nozzle - has an advantage over ring-type nozzles because of the large exposed disc area, allowing for stronger springs to be used. A stronger spring can enable faster closure, lowering the magnitude of slam.

External damping allows additional reverse flow. As a result, the back flow velocity will increase and consequently also the Joukowsky pressure rise will be higher. It is not uncommon to take the approach to "slow down" valve motion to reduce pressure surge, and for many types of transients this is reasonable. For check valve slam, however, adding unnecessary damping will make the response more severe. Some damping may be necessary for other mechanical reasons, but this competes with reducing surge pressures.

10.3 Balancing Design Criteria

It is not possible to design a check valve which has zero pressure drop at steady flow conditions and a perfect dynamic performance for unsteady situations. For liquid applications, the focus should be on the dynamic performance of a check valve as this will help to avoid or limit pressure surge.

11. SELECTION

11.1 Is a Specialized Valve Necessary?

Not all situations call for highly specialized or made-to-order equipment. Critical or otherwise demanding applications are more likely to need an engineered valve solution, tailored to the particular system and transient event. Pump/compressor discharge, parallel pumping, pulsating flows, cryogenic applications, steam lines, and wastewater plants are just a few examples of situations were critical check valves likely exist. There is no one check valve design that is suitable for all applications.

However, not all applications are critical. Furthermore, not all check valves in a given system may be critical. For example, if the flow velocities are low and remain low, the potential for slam is low and the investment in a sophisticated valve is probably not required. In some cases, other existing mitigation equipment may make the likelihood of rapid deceleration very low, also limiting the need for an exceptional check valve. Finally, system tolerances vary. What is an unacceptable pressure surge in one system may not be an issue in another. All of these considerations are system-specific and must be considered. If the system behavior is unknown, it would be wise to err on the side of caution and select a valve with good dynamic characteristics.

11.2 Choosing the Right Valve

A proper check valve sizing should be part of any selection process. The specification should include a graph showing the opening characteristic with cracking pressure and dp vs open position, the flow rate at which valve is fully open, and dp at minimum, normal, and maximum flows. For liquid applications, it should also include the maximum pressure surge at closure when the maximum system deceleration is known.

12. CONCLUSION

Check valve slam in systems can cause dangerously high pressures, high enough to destroy equipment in severe cases, and is sometimes overlooked because it is not considered a "primary transient event." Surge due to slam is strongly related to fluid deceleration, and can be modeled fairly accurately with a simple approach, provided the dynamic characteristic curve in use is appropriate for the check valve under study. Any such analysis must consider the dynamics of the system as a whole, as rule-of-thumb analysis can easily mislead even a seasoned analyst.

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