Numerical investigation of reciprocating valve characteristics on pressure pulsations

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ABSTRACT

This paper aims to identify the difference between three methods for modelling reciprocating pumps. It compares an ideal flow model, an ideal valve model, and a detailed valve model that explicitly includes the valve dynamics. The paper describes how these three models, together with a case study, have been used to determine how the predicted pressure pulsations within a pipeline system depend on the relevant parameters. The results show that for typical situations, the method employing the ideal flow model is sufficiently accurate. If valve design or life is an issue, a detailed valve model may be required.

NOMENCLATURE

A_{suc}	valve port area suction side
A_{dis}	valve port area discharge side
b_n	MOC integration constants
C_d	valve discharge coefficient
c	wave speed
D	valve area
d	valve damping coefficient
dt	computational time step
e_n	MOC integration constants
F	force on the valve
f	running frequency of the pump
g	gravitational constant
h	pressure head at a node
K	bulk modulus
k	valve spring stiffness
L	pipe length
l	con. rod length
MOC	Method of Characteristics
m	valve mass
p	pressure in the cylinder
p_{dis}	pressure downstream of discharge valve
p_{suc}	pressure upstream of suction valve
Q_{dis}	flow discharge side reciprocating pump
Q_{suc}	flow suction side reciprocating pump
q	flow through an element
R	crank arm radius
S	piston area
V	volume in the cylinder
x	valve travel
θ	phase angle
ρ	fluid density

INTRODUCTION

From petrochemical industries there is a growing demand for accurate modelling of pressure pulsations generated by reciprocating pumps. The modelling of a reciprocating pump can be done on several levels of accuracy, ranging from a simplified superposition of periodic waves for each cylinder up to a detailed evaluation of the local, temporal pressure changes at the valves of the pump. This paper describes and evaluates a detailed

numerical model for reciprocating pumps that explicitly includes the motion of the valves, and their interaction with the flow through the pump cylinders and the connected pipeline system. In particular, this paper shows how the detailed valve model affects the pressure pulsation amplitudes, and it identifies any possible critical valve parameters.

Numerous reciprocating pump systems are designed according to the code named API674 *Positive Displacement Pumps – reciprocating.* This code describes how to assess the pressure pulsations generated by a reciprocating pump and provides general guidance on the modelling and prediction of pressure pulsations. However, details of what model to use in simulating the flow fluctuations generated by a reciprocating pump are not specified. As the code is being used increasingly often as a basic requirement by operators, engineers and OEM's are starting to recognize the need to better understand the different methods and tools available to predict the pressure pulsations as required by API674. This need is due to two factors: failures and design cost. If the pressure pulsations are underpredicted due to (potential) simplifications during the analysis phase, failures can occur during operations due to, for example, excessive shaking forces. On the other hand, the same simplification could result in an overly conservative prediction of the pulsations resulting in excessive mitigation costs.

Furthermore, as pump valve parameters are not always well defined or even known, it is important to assess the sensitivity of the predicted pressure pulsations to variations and inaccuracies in the model parameters.

This research uses two pipeline systems to study how the predicted pressure pulsations depend on the pump valve parameters and the applied pump model. In both systems the pressure pulsations are computed by means of the Method of Characteristics (MOC) as implemented in the surge analysis software BOSfluids®. The first pipeline system involves a pump cylinder that is connected to an anechoic boundary condition. The second system involves a pressure boundary condition allowing for acoustic resonance in the system. Using these two systems a good comparison between the valve response with and without acoustical resonance is identified and described in the paper. Based on the numerical results, additional guidance in the setup of a pulsation analysis model is given. The paper concludes with an example based on a real-life industrial project using actual reciprocating valve data to illustrate how the detailed valve model can be applied and increase the fidelity of the pressure pulsation calculations.

THEORETICAL MODEL

The Method of Characteristics (MOC) is used for calculating the time-dependent flow and pressure field within a piping system. This method is combined with three different models for the determination of the flow generated by a reciprocating pump: the ideal flow model, the ideal valve model and the detailed valve model.

The three pump models act as a boundary condition within the piping model. The ideal flow model essentially imposes a time-dependent flow rate at an end node in the piping model. The ideal and detailed valve models, on the other hand, use the motion of the piston, and the pressure and characteristics lines of the connecting nodes/elements as input to calculate the flow through the pump cylinder. Note that the current implementation of the latter two models can only simulate a single cylinder at a time.

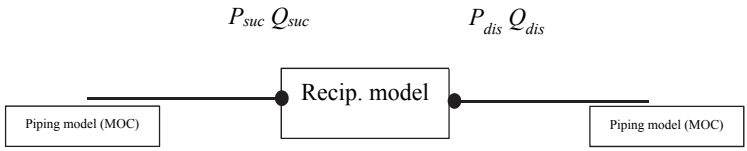


Fig 1 | Schematic overview of reciprocating pump model inside the piping model using MOC. Q_{suc} is positive for flow towards recip. model, Q_{dis} is positive for flow away from recip. model.

Method of Characteristics

The numerical investigation is conducted using an implementation of the Method of Characteristics as described by Wylie [3] based on the surge analysis software BOSfluids®. The method solves the Navier-Stokes equation using the following assumptions:

- The pressure perturbations are one dimensional
- The density is constant (no thermal effects)
- The Mach number is small (no convective term of velocity)
- The friction losses are based on (quasi-)steady conditions.

Using these assumptions, after some rearrangements, a typical equation according to the solution of the MOC for the pressure head at a specific node can be derived:

$$h = \frac{e_n - q}{b_n} \qquad 1$$

For more details on the MOC and its workings the reader is referred to Wylie [3].

Reciprocating pump with valve dynamics

The reciprocating pump is introduced as an element which imposes a flow at connected node. Each time step, a separate set of equations is solved which result in an updated boundary condition on the nodes of the reciprocating element.

The equation below is used to determine the pressure inside the cylinder:

$$\frac{p - p_0}{dt} = \frac{K}{V_0}\left(-\dot{V} + Q_{suc} - Q_{dis}\right) \qquad 2$$

where subscript 'o' denotes the value from the previous time step. The rate of change in volume of the cylinder can be obtained by combining the velocity of the piston together with the area of the cylinder:

$$\dot{V} = -2\pi f S R sin(\theta)(1 + \frac{R cos(\theta)}{\sqrt{l^2 - (R sin(\theta))^2}} \qquad 3$$

An additional set of equations is used to describe the flow at suction and discharge side of the pump. These equations are derived through combining (1) with the equation of pressure drop vs flow through a valve.

$$C_d A_{suc}\sqrt{\frac{2(p_{suc} - p)}{\rho}} = e_n - p_{suc}\frac{b_n}{g\rho} \qquad 4$$

$$C_d A_{dis} \sqrt{\frac{2(p-p_{dis})}{\rho}} = p_{dis} \frac{b_n}{g\rho} - e_n \qquad 5$$

The valve effective port opening A is implemented using a linear relation with the maximum valve travel:

$$A = A_o \frac{x}{x_o} \qquad 6$$

Where the subscripts 'o' denote the maximum port area at the maximum valve travel.

Valve dynamics
The dynamics of the valves are described using a second order ODE of a dampened spring-mass system:

$$m\ddot{x} + d\dot{x} + kx = F \qquad 7$$

Various studies propose to take into account a range of different aspects with regards to the force F on the valve [1]. The aspects included in this study are selected based on a combination of the expected importance of the aspects based on literature as well as the practical feasibility of selecting sensible input values.

Aspects that are taken into account in the current paper with regards to the valve dynamics are the following:

- Discharge coefficient
- Port/valve area
- Valve mass
- Spring stiffness
- First order dampening coefficient
- Maximum travel
- Preload on valve

More complex aspects for the valve dynamics could be included such as non-linear varying valve/port area, stiction force and valve bounce. Equations 1 through 7 are solved numerically using a combined iterative procedure involving the bisectional method.

Ideal Valve
The ideal valve model is identical to the detailed valve model but it assumes that the valve has no mass, no stiffness, and no damping. This means that the valve instantly adjusts its position according to the pressure differential across the valve.

Simplified reciprocating pump model
The ideal flow model does not take the valve dynamics into account; it simply calculates the flow rate through the pump cylinder on the assumption that the pressure difference between the suction and discharge port is constant and known in advance. This model essentially uses (5) in order to determine the flow in/out of the pump cylinder. Typically, the moment of valve opening is delayed based on the compressibility of the liquid inside the cylinder. Fig 2 shows an example of a flow profile as generated us the ideal flow model.

The ideal flow model is computationally cheaper than the ideal and detailed valve models described in the previous paragraphs. Its effectiveness and limitations in predicting the pressure pulsations will be analyzed by comparing it to the ideal and detailed valve models.

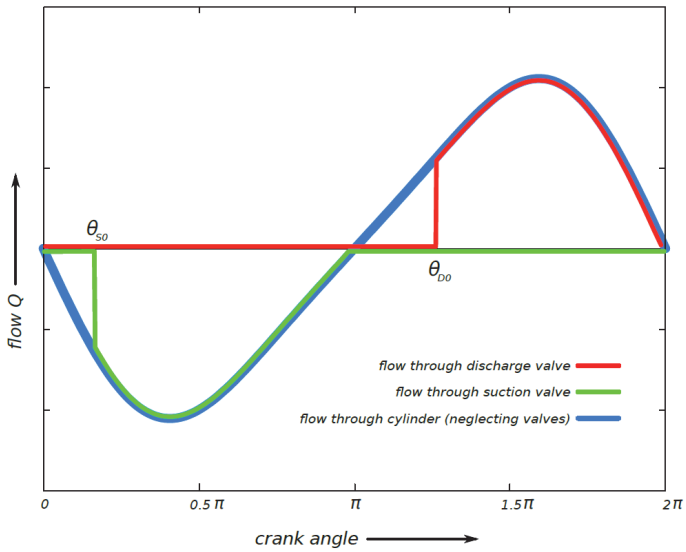


Fig 2 | Example flow profile as generated using a fixed flow model.

Calculation of pulsations

As the MOC solves the pressure and flow field in the time domain, several post-processing steps are required to calculate the amplitudes of the pressure pulsations. The research presented in this paper makes use of a Discrete Fourier Transform (DFT) as implemented by the software package BOSpulse. Note that the initial part of a simulation is not used in the DFT in order to exclude non-periodic flow and pressure fields that occur only during the transition from steady-state flow conditions to periodic flow conditions.

PIPING SYSTEM SETUP

The pressure pulsations generated by a reciprocating pump are a result of a combination of a (large) number of parameters. These parameters describe the geometry of the pump cylinder and piston, the fluid properties, the valve properties, the piping geometry and the boundary conditions. In order to simplify the exploration of this design space, a simplified piping system is created as a base. Perturbations of the parameters with regards to the valve only will be applied to this system for a single parameter at the time.

Two base systems are used: one with an anechoic boundary condition and another with a fixed pressure boundary condition. The anechoic boundary condition is used to eliminate any effects of acoustical resonance on the valve dynamics. For the second model, the length of the pipe sections between the pump and pressure boundary condition are selected such that acoustic resonance occurs at the frequency of the pump. The length is calculated based on a quarter wave resonance condition:

$$\boldsymbol{L = \frac{c}{4} f} \qquad 8$$

An overview of the base models used is shown in Fig 3.

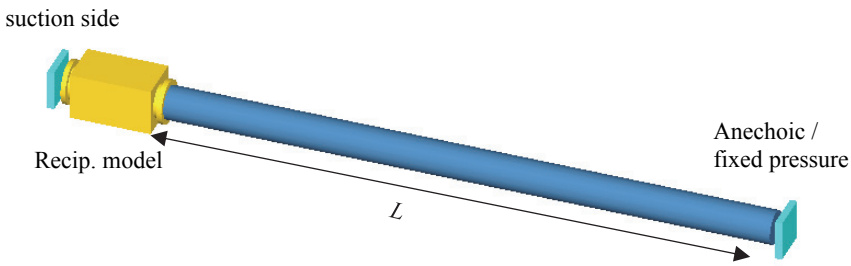


Fig 3 | Overview of base system.

Both systems are based on the following assumptions:

- Constant pressure directly at the suction side of the pump.
- Reciprocating Equipment values as shown in Appendix A.
- Base valve parameters estimated as 'typical' values[1] (shown in Table 1).
- Single pump cylinder model discharging into a constant diameter pipeline.

The constant pressure at the suction side has been chosen to highlight relations between model parameters and the computed pressure pulsations. Table 1 shows the parameters range studied. 50 perturbations are used for each parameter resulting in a total of 300 computational runs per model. The range of the parameter is taken, perhaps unrealistically, large with the purpose of magnifying any possible effect on the pulsations at low/high parameter values.

Table 1 | Range of parameters studied.

Parameter	unit	Lower bound	Upper bound	Base value
Cd	[-]	0.1	100*	0.7
d	[N.s/m]	0	20	1
m	[kg]	0.01	10	0.15
D	[mm]	0.5	500	70
Preload	[N]	0	1000	110
K	[N/mm]	0.1	100	5

* even though unrealistically high, this can show effect of large port area and low valve resistance

RESULTS

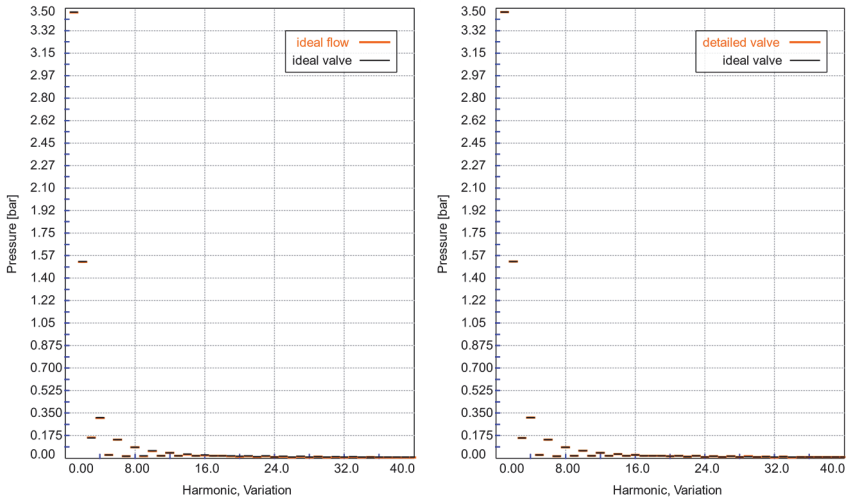
Anechoic boundary model

First the results of the anechoic boundary model are presented. The results of this model allow for a good identification of root/cause analysis with regards to the valve parameters and predicted pulsations.

[1] Values estimated based on experience and a combination of different consulting projects, but cannot be assumed to be typical for each valve manufacturer/pumping system.

Ideal flow model vs ideal/detailed valve model

Fig 4 shows a comparison of the predicted pulsations for the ideal valve model with the ideal flow model and the detailed valve model. From this figure it can be noted that all different modelling methods predict an equivalent pressure pulsation. Taking a closer look at the flow profile generated by the models, as shown in Fig 6, small difference in flow rate can be noted at the moment the valve opens. Especially for the detailed valve model, a short spike in flow rate is observed. The effect of the spike on the predicted pulsations includes a slight increase in pulsations at higher harmonics but is not noticeable in the results as shown Fig 4. This effect on higher harmonics is illustrated in Fig 5.



(a) Ideal valve vs ideal flow (b) Ideal valve vs detailed valve

Fig 4 | Predicted (peak-to-peak) pressure pulsations in the anechoic system comparing ideal valve model with the fixed flow model (a) and the typical valve model (b). x-axis contains harmonic number where the variation of the parameter is plotted around the harmonic in the range between ±0.4.

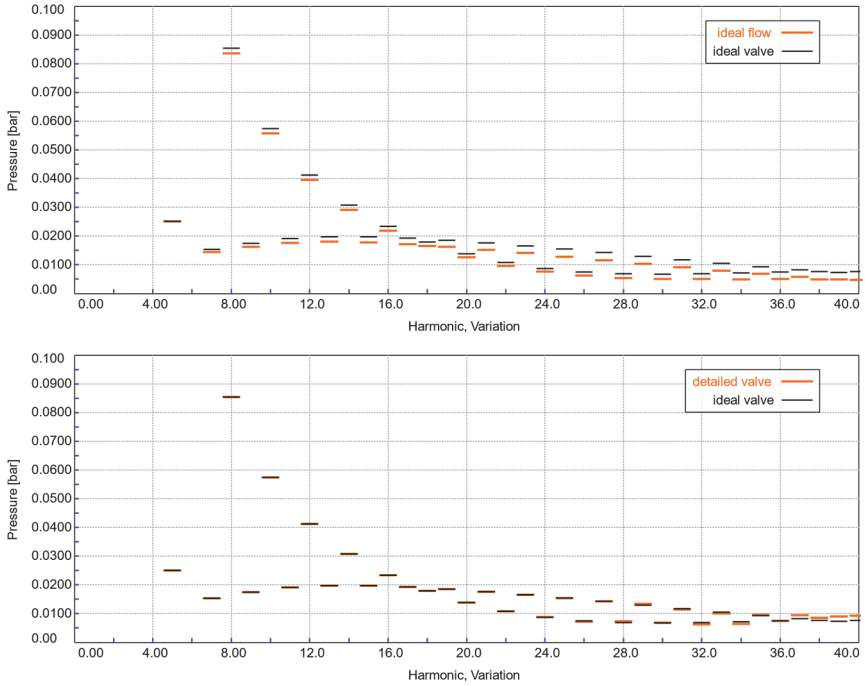


Fig 5 | Zoom in on higher harmonics. Predicted (peak-to-peak) pressure pulsations in the anechoic system comparing ideal valve model with the fixed flow model (top) and the detailed valve model (bottom).

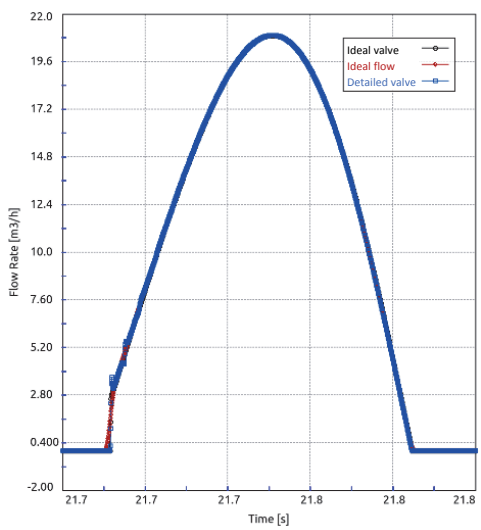


Fig 6 | Flow profile at the discharge side of the pump model using an anechoic system. Comparison between different modelling methods.

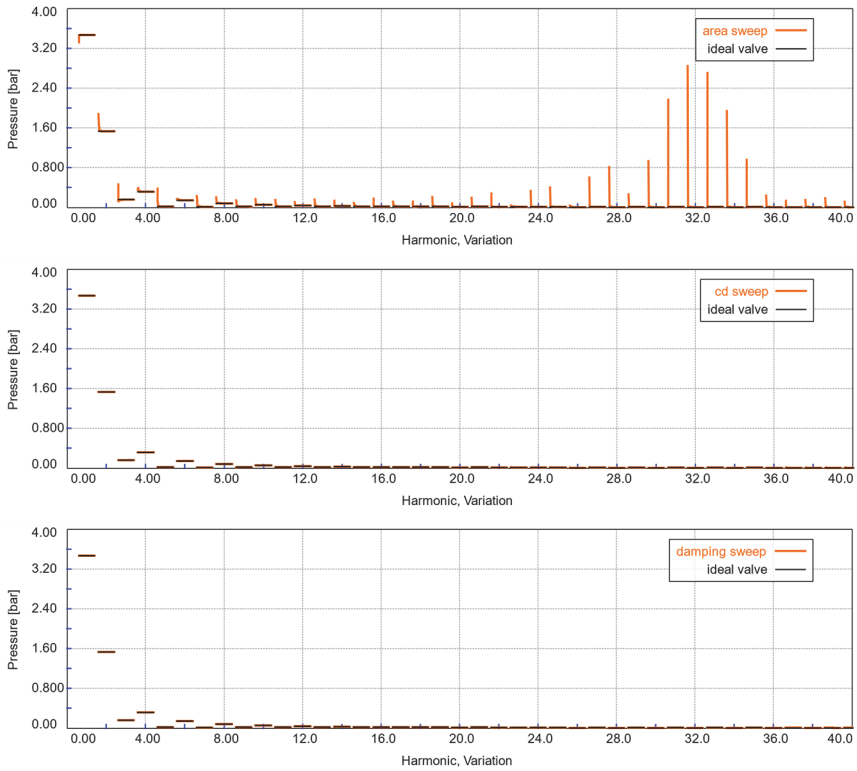
Parameter sensitivity

Fig 7 shows the result of the effect of changing parameter values on the predicted pulsations. From this figure it can be noted that there is a limited influence of spring stiffness, discharge coefficient and damping coefficient on pulsations. Furthermore, the values of high mass and small valve area and small preload can significantly influence the predicted pressure pulsations.

The effects of mass, area and preload are further illustrated in Fig 8 and Fig 9. From Fig 8 it can be noted that a very small valve area causes the valve to repeatedly open and close during the discharge phase. This unstable valve motion has been identified before in [2] for cases where the valve area becomes smaller than the port area.

Fig 9a shows that the preload is necessary for a detailed valve to close at the correct moment and prevent reverse flow through the valve. Fig 9b shows that if the mass of the valve increases, the valve starts to lag relative to the ideal valve motion. This can be explained as the relative magnitude of the force pushing the valve open/close is much smaller than for the case of lower valve mass.

Thus far, the results of the model using an anechoic boundary condition show us that the pulsations mainly depend on the moment of valve opening/closing. Especially valve lag during which flow reversal occurs can significantly influence the pulsations. The eigen-frequency of the valve does not seem to be an important factor in predicting pulsations. It is noted that if realistic valve parameters are selected, difference between the ideal valve, detailed valve, and ideal flow model are neglectable.



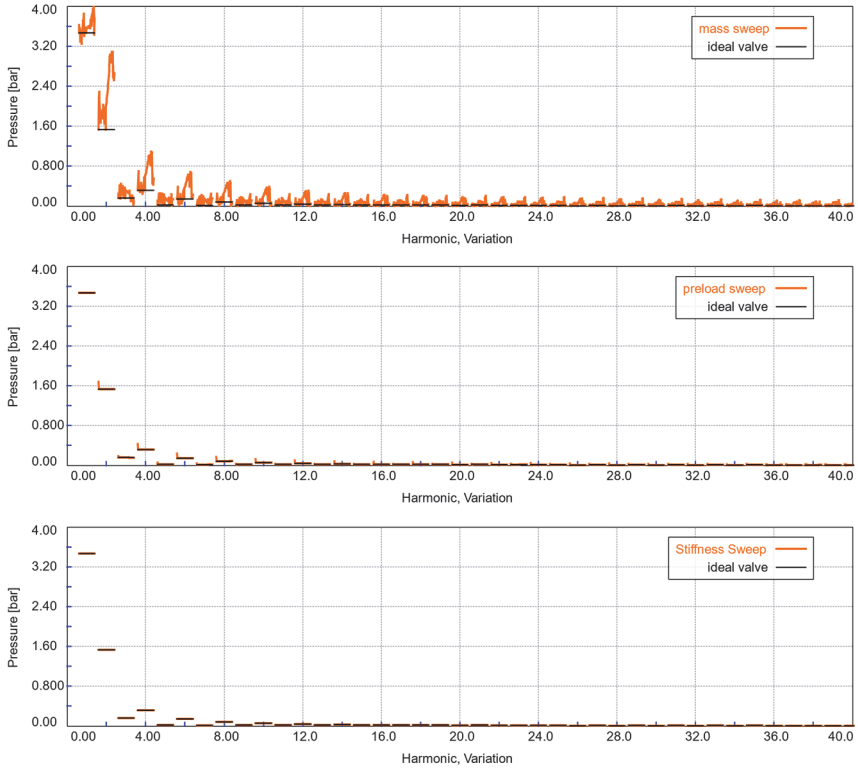


Fig 7 | Predicted pressure (peak-to-peak) pulsations results including comparison with the ideal valve model. x-axis contains harmonic number where the variation of the parameter is plotted around the harmonic in the range between ±0.4.

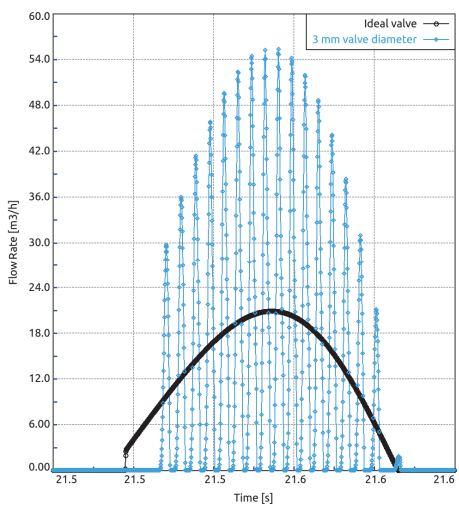
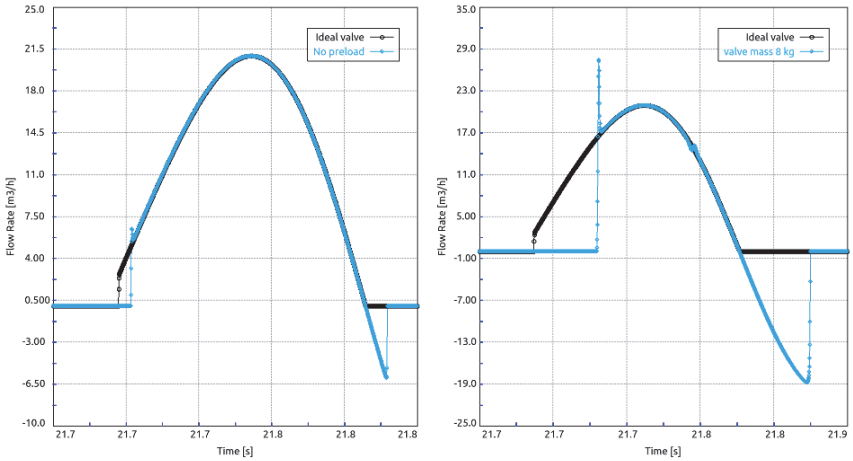


Fig 8 | Discharge flow ideal valve model vs detailed valve model with 3 mm valve diameter.



(a) Ideal valve vs no preload (b) Ideal valve vs 8 kg valve

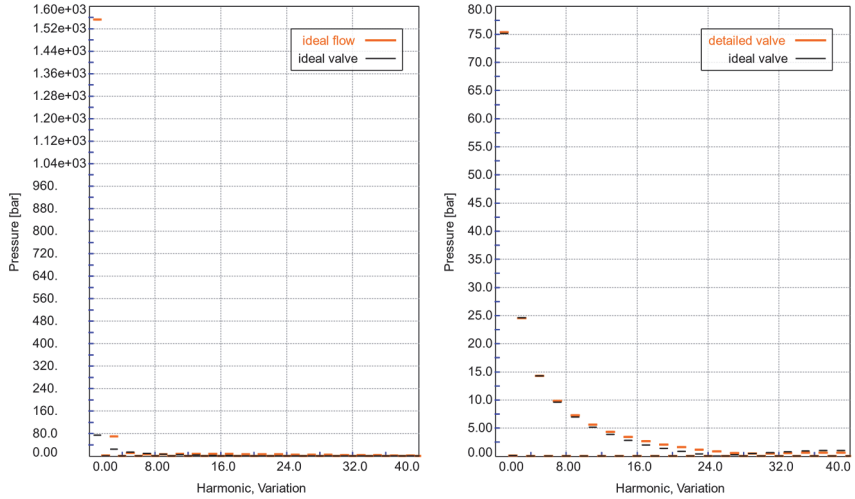
Fig 9 | Discharge flow ideal valve model vs detailed valve model with no preload (a) and base valve with 8 kg valve mass (b).

Fixed pressure system with acoustic resonance

In this section the results of the system with acoustic resonance are discussed. The goal of this system is to identify any additional effects of including the valve model and the effect of changing valve parameters on the predicted pressure pulsations while resonance occurs in the connecting piping system.

Ideal flow model vs ideal/detailed valve model

Fig 10 and Fig 11 show a comparison of the predicted pulsations for the ideal valve model with the ideal flow model and the detailed valve model. From this figure it can be noted that the ideal flow model predicts significantly higher pulsations. Only some small difference in predicted pulsations can be noted between the ideal and detailed valve model. Fig 12 shows the discharge flow at the pump. The systems with valve show that even during the suction phase, the discharge flow remains positive. Due to acoustic resonance, the pressure at the discharge side drops significantly and forces the discharge valve to open during the suction phase.



(a) Ideal valve vs fixed flow model (b) Ideal valve vs typical valve

Fig 10 | Predicted (peak-to-peak) pressure pulsations in the system with resonance comparing ideal valve model with the ideal flow model (a) and the detailed valve model (b).

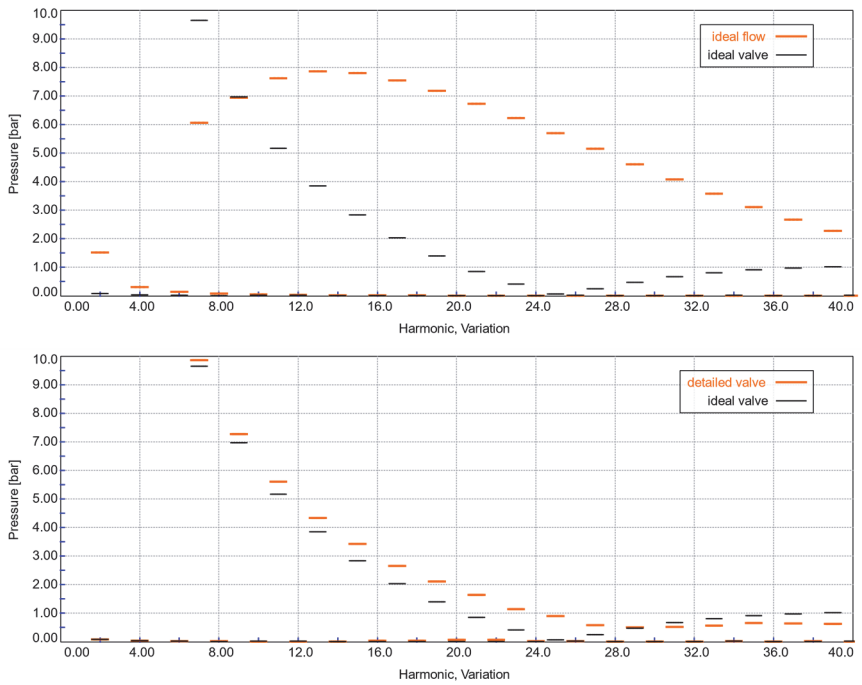


Fig 11 | Zoom in on higher harmonics. Predicted (peak-to-peak) pressure pulsations in the anechoic system comparing ideal valve model with the fixed flow model (top) and the detailed valve model (bottom).

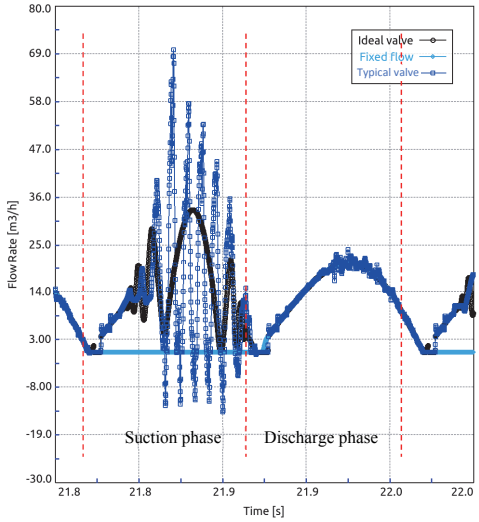
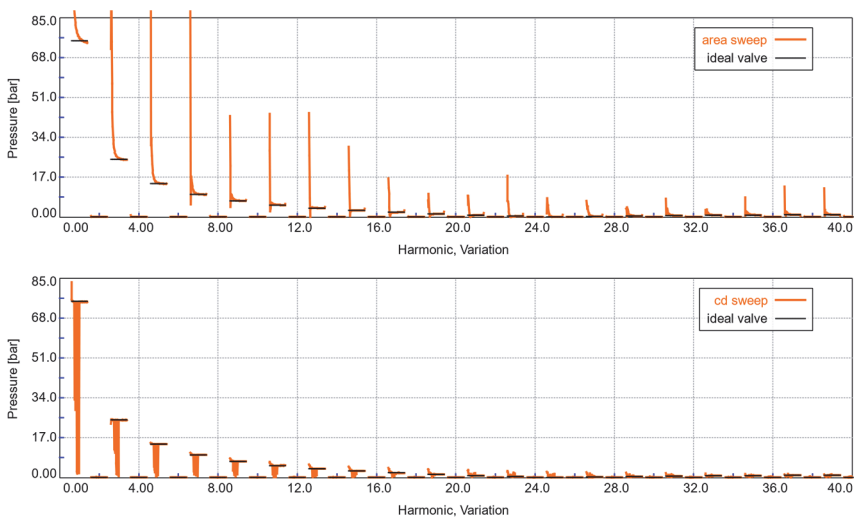


Fig 12 | Discharge flow comparing ideal/detailed valve model and ideal flow model.

Parameter Sensitivity

Fig 13 shows the result of the effect of changing parameter values on the predicted pulsations with acoustic resonance. From this figure it can be noted that there is a limited influence of spring stiffness, damping coefficient, and preload on the predicted pulsations. Also, a somewhat more significant influence of discharge coefficient and valve mass can be observed. However, regardless of the value selected for these parameters, the predicted pressure pulsations remain several orders of magnitudes lower than those predicted by the ideal flow model. Lastly, the valve area can significantly influence the predicted pressure pulsations especially for smaller values of valve area. This is due to similar effects as were observed for the system with an anechoic boundary condition.



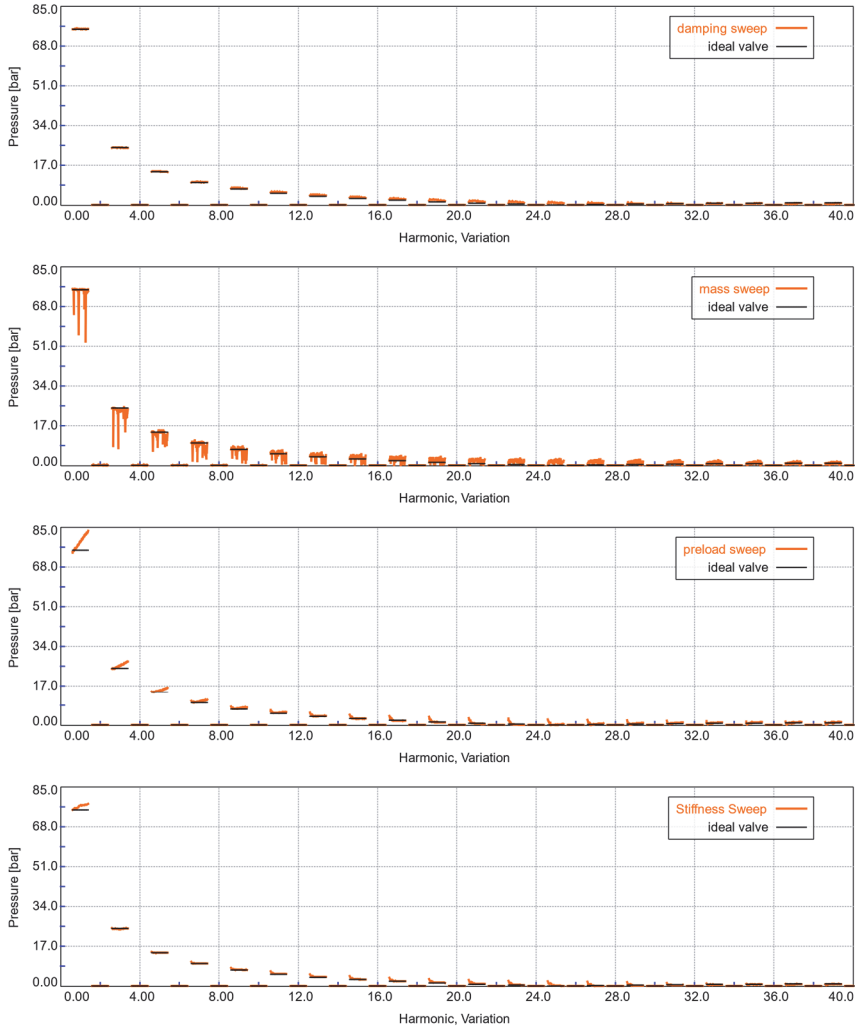


Fig 13 | Predicted pressure (peak-to-peak) pulsations results including comparison with the ideal/detailed valve model with acoustic resonance. x-axis contains harmonic number where the variation of the parameter is plotted around the harmonic in the range between ±0.4.

Case study

For a final comparison between the different modelling methods, a numerical case study is presented. The system is based off of a real-life industrial system with several modifications in length and orientation. Furthermore, due to these modifications, the system is no longer guaranteed to conform to the API 674. However, care has been taken to ensure that the general conclusions of the comparison are not significantly influenced.

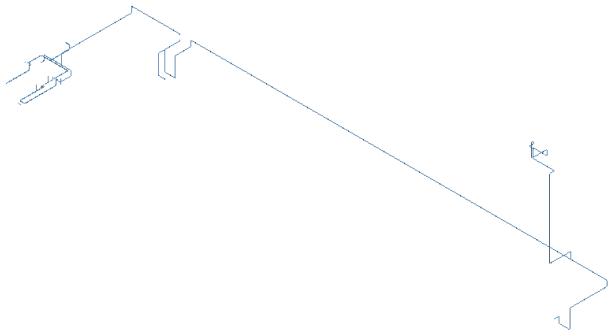


Fig 14 | Overview system based on real-life industrial system.

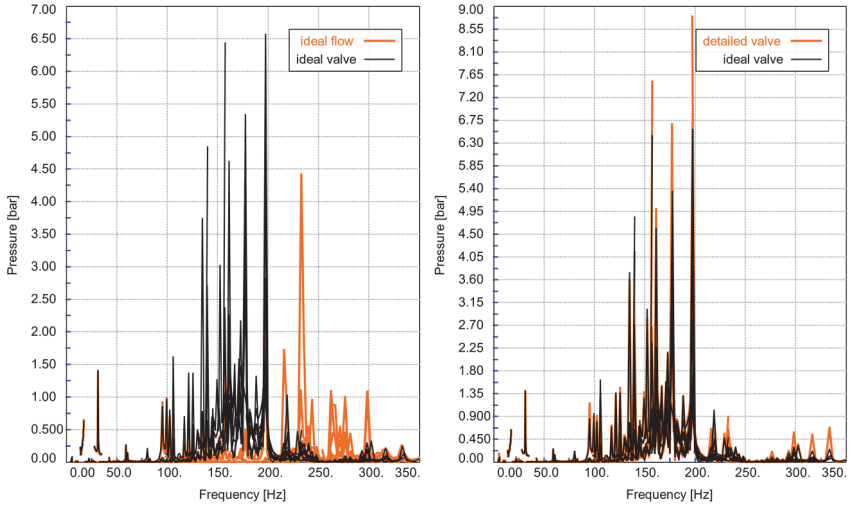
An overview of the system is shown in Fig 14. The system includes a triplex pump with a pressure differential of around 170 bar. The suction side is connected to a constant pressure located at around 20 m upstream of the pump. The discharge side connects to several discharge streams with constant pressure at distances between 40 and 200 meters.

In order to better identify any potential difference between the modelling methods with regards to any acoustical resonance in the system, the frequency of the pump is varied between 4.7 and 5.8 Hz with a total of 20 steps.

Comparison modelling methods

As the data in terms of pulsation amplitude per harmonic at the different locations is too large for this paper to discuss, the pulsations at a single point in the system is selected and discussed in this report. This location is selected based on a pre-processing assessment and the writer believes that the results at this location make for a good comparison for the complete system. Fig 15 shows the comparison between the ideal flow model, ideal valve model, and detailed valve model. The results show that the including the ideal valve model predicts several peaks in pressure pulsations between the 130-200 Hz. While the ideal flow model predicts a significant increase in pulsations at around 220 Hz. The comparison between the ideal valve and detailed valve show there are only minor difference in the predicted pulsations. Differences in predicted peak pulsations between these two can very well be contributed to the resolution in frequency variations.

Part of the difference between the results of the ideal flow model and the systems including a valve can be explained using Fig 16. Here it can be noted that the models including the valve allow for higher frequency flow fluctuations causing the additional pulsations peaks between 130-200 Hz. The increase around the 220 Hz as predicted by the fixed flow model is a direct result between acoustic resonance between the damper and the pump manifold. As was seen in the simplified model, the addition of the valve model can reduce the predicted pulsations around this frequency due to acoustic resonance.



(a) Ideal valve vs fixed flow model (b) Ideal valve vs typical valve

Fig 15 | Comparison maximum pulsations case study. Location several meters downstream of discharge side pump.

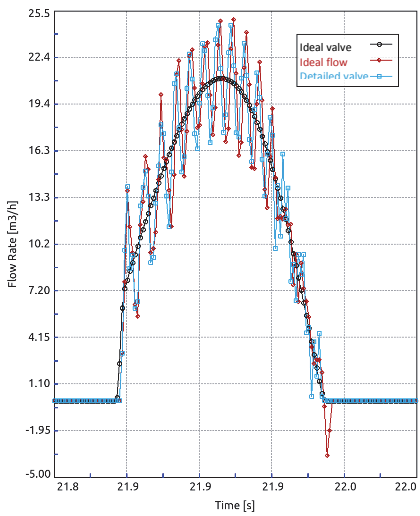


Fig 16 | Discharge flow rate single cylinder.

DISCUSSION

The study as presented in this paper includes a large amount of permutations resulting in the computation of over 600 different cases. Regarding these permutations, several important remarks can be made to provide context to the interpretations and conclusions.

It is important to note that the valve motion is significantly influenced by the valve parameter, even if pulsations are not. Large valve motion and high impact speeds at valve stops might lead to excessive wear of the valve. For these types of studies, a detailed valve model might be interesting to determine or optimize the performance of the valve for a particular piping system / pump configuration.

Furthermore, the study is limited as only variations for a single (valve) parameter are studied per permutation. Furthermore, a large number of additional, possibly interesting, configuration can be studied such as more resonance at different harmonics, interaction between cylinders, advanced valve properties and cross-correlation between parameters.

Lastly, during the current study, mostly idealized conditions are studied. This was done in order to magnify any potential effects of changing parameter values. Especially the predicted opening of the discharge valve during the suction phase when acoustic resonance occurs could very well be contributed to the idealized conditions of a straight pipe at perfect resonance conditions. The occurrence of this phenomena, in the calculated severity might be rare in real-life applications.

CONCLUSIONS

Based on the results as presented in this paper, several modelling aspects should be considered when conducting a pulsation analysis from an engineering/practical perspective.

When including the valve dynamics during an analysis and the valve is well-designed / proven for the type of pump, no significant difference between the results of an ideal valve and a detailed valve model are observed.

The ideal flow model can predict excessive and unrealistic pulsations if acoustic resonance creates large pressure fluctuations at the suction or discharge nodes of the pump cylinder. For this case including an ideal/detailed valve model in the system can result in better accuracy.

If improper design of a valve or if valve-life is of concern, a more detailed model of the valve including all parameters can be used. However, the research as presented in this paper it does not cover the analysis of these phenomena in detail and is not clear to what extend a detailed valve model is able to capture possible detrimental valve behaviors.

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APPENDIX A – Values used for reciprocating equipment

Parameter	unit	Value
Piston Diameter	[mm]	63.53
Crank Arm Radius	[mm]	57.2
Con Rod Length	[mm]	286
Clearance Volume	[%]	280
Fluid Bulk Modulus	[Mpa]	2200
Running Speed	[Hz]	5