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From Pressure Pulses to Noise Sources – NVH-Oriented Design of Hydraulic Systems

Benedikt Müller[1], Heiko Baum[1]

[1] *FLUIDON GmbH, benedikt.mueller@fluidon.com, heiko.baum@fluidon.com*

Abstract.

This paper shows how fluid sound propagation in complex fluidic systems can be described accurately and in a numerically efficient manner using 1D CFD. Key modeling components are presented, and examples illustrate how these can be used to create a complete model for analyzing the vibration and resonance behavior of the fluid.

Based on this 1D CFD, a simplified, one-sided fluid-structure coupling is presented. Mechanical alternating forces on pipes and components are derived from transient pressure and velocity profiles, whereby the influence of the structural response on the fluid is neglected. This assumption significantly reduces the complexity of the model and enables highly efficient, fast-computing simulations.

The approach allows early identification of NVH-relevant loads and specifically supports the design of robust and structure-friendly piping systems.

Keywords: 1D CFD, unilateral fluid-structure interaction, simulation, vibration fingerprint, resonance behavior, NVH-relevant loads, NVH optimization

1. Introduction

Despite technical advances in automation, the machine operator remains a central figure in agricultural and construction machinery. Complex operating environments, legal frameworks, and the need for situational decisions make complete autonomy unrealistic at this point. In many cases, semi-automated systems take over the execution, while humans retain overall control and responsibility. This is precisely why acoustic comfort in the workplace cabin is becoming increasingly important: noise and vibrations have a direct impact on concentration, fatigue, and long-term health – and are becoming a decisive quality factor in machine development. NVH optimization is thus not only a technical necessity, but also an ergonomic and economic one. At the same time, reducing noise emissions into the environment is becoming increasingly important. This is particularly relevant when construction machinery is used in sensitive areas such as inner-city areas or during night work, where low noise pollution is essential.

Improvements in noise reduction on the drive side – for example, through quieter engines or electrification – means that hydraulic noises, such as those caused by pressure ripple from piston pumps, are no longer masked as they used to be. These noises are now clearly audible. This development is exacerbated by ever-higher power densities and increasing system dynamics, which make resonances in the branched piping system more likely. The result is structural excitation and airborne noise emissions, which significantly influence the overall NVH impression of the vehicles.

2. Hydraulic Pressure Ripple as a Source of NVH

Hydraulic pressure pulsations are primarily caused by time-varying volume flows. Typical excitation sources are displacement pumps with kinematically induced flow ripple, dynamic valve operations, and abrupt changes in flow boundary conditions, for example as a result of load changes or emergency shutdowns.

These excitations lead to the formation of pressure waves that propagate at the speed of sound in the fluid along the pipe system and are partially reflected at cross-sectional changes, branches, and open or closed boundary conditions. The physical principles of this wave propagation and the associated acoustic effects are described in detail in *technical acoustics* (see [1]).

Within this context, this article deliberately focuses on fluid acoustics rather than on the classic, component-oriented, concentrated parametric system simulation of fluid power systems. **Figure 2.1** illustrates the propagation and decay characteristics of pressure waves in a fluid power system with sufficient damping.

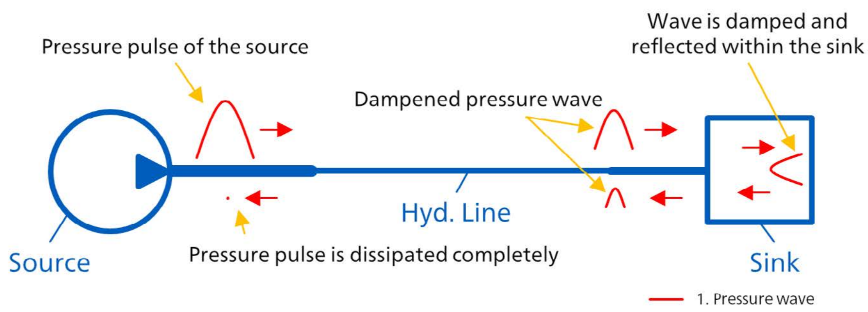

Figure 2.1: Pressure wave propagation in a system with high damping

In practice, the pressure ripple of the source – especially the pump – is often so pronounced that the introduced pressure wave is not completely dissipated. In addition, the effective system damping can be reduced, for example as a result high operating temperatures. Consequently, a relevant portion of the pressure wave remains in the pipe system and, after multiple reflections, reaches the excitation source again.

In the event of a resonance, this backward-travelling pressure wave hits a newly generated wave in-phase, causing the pressure pulsations to superimpose and reinforce each other (**Figure 2.2**).

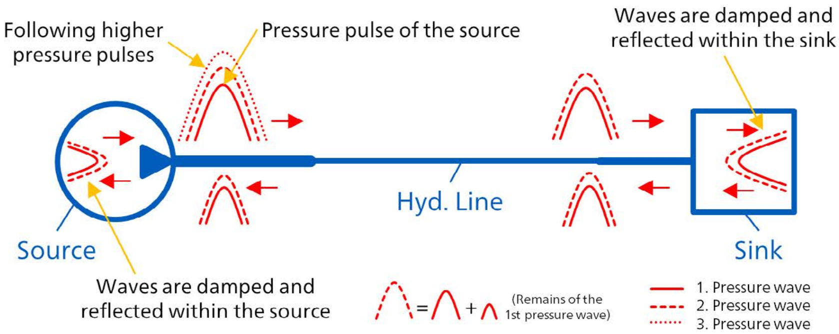

Figure 2.2: Pressure wave propagation in a system with insufficient damping

When excitation frequencies encounter the natural frequencies of the line system, resonant states arise with locally highly increased pressure amplitudes. From an NVH perspective, it is not only individual pressure peaks that are critical, but in particular, permanently acting, pulsation-induced alternating forces.

Two mechanisms of action are particularly important here:

- **Structural excitation** due to time-varying forces on pipe bends, elbows, branches, and fastening points
- **Sound emission**, either directly from vibrating pipes or indirectly via excited machine parts

While short-term pressure surges are often evaluated in terms of their maximum mechanical stress, pulsation-induced alternating forces lead to cyclic loads. These can cause material fatigue, loosening of connectors, or leaks in the long term, and have a significant impact on the overall NVH impression of modern machines. However, these effects are often underestimated in early development phases.

3. Why 1D CFD?

A transient, one-dimensional simulation approach is used to analyze the fluid-acoustic effects described above. This approach enables a time-resolved simulation of the pressure and volume flow profiles along complex, branched pipe systems. A prerequisite for the validity of the one-dimensional analysis is that the pressure waves propagate as plane waves at the speed of sound in the relevant frequency range – an assumption that is well fulfilled for typical hydraulic pipe systems in mobile machines.

Compared to three-dimensional CFD approaches, 1D CFD offers significant advantages:

- Significantly reduced modeling effort
- Very short computing times
- High numerical stability for highly dynamic processes
- Possibility of extensive parameter and variant studies

Compared to concentrated parametric (0D) methods, 1D CFD enables a more realistic representation of wave propagation and resonance effects, while offering high numerical robustness and often lower computing costs.

Another significant advantage of the time domain approach is that it takes into account non-linear effects such as those caused by real valve characteristics, pressure-dependent material properties, or transient boundary conditions.

4. COMPONENTS IN 1D CFD

With the *Piping Systems Library* [2], Fluidon has developed a model library that enables consistent mapping of hydraulic systems for fluidoacoustic analysis and NVH-oriented optimization. In addition to detailed pipe models, the library includes components for describing displacement units, valves, cylinders, and accumulators, thus enabling the consistent setup of acoustically relevant system models.

4.1. Pipes and Hoses

A central element in the acoustic simulation of hydraulic piping systems are pipes and hoses, whose behavior is decisive for the propagation of fluid sound in the system. The continuity equation and the Navier-Stokes equation are the basic fluid mechanics equations on which the calculation is based. A proven method for solving this partial differential equation system is the method of characteristics (MoC), which is characterized by high accuracy and low computational effort.

In the simulation of acoustic processes in pipes, it is necessary to take into account not only stationary friction but also dynamic friction, often referred to as frequency-dependent friction. The method developed by *Trikha* [3] implements this efficiently. *Theissen* [4] uses the method of characteristics with frequency-dependent friction to calculate oil hydraulic systems.

Figure 4.1 shows in the upper part that, in the case of slow oscillation, the flow profile is similar to that in the stationary case. However, if the frequency increases (lower part), history plays a major role in the shape of the flow profile, with higher shear velocities and thus increased friction occurring in the boundary layer.

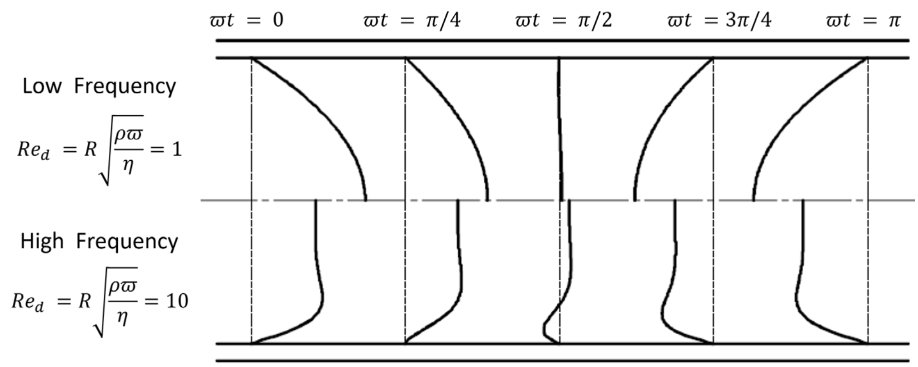

Figure 4.1: Flow profiles of oscillating pipe flows [5]

An important extension is the consideration of the viscoelastic behavior of hoses [6], [5], which is parameterized as a complex modulus of elasticity in the current implementation in DSHplus [7]. The viscoelastic component of the material behavior of hose material is represented by a Kelvin-Voigt model, which, connected in series with the linear elastic component, represents a so-called Zener body (**Figure 4.2**).

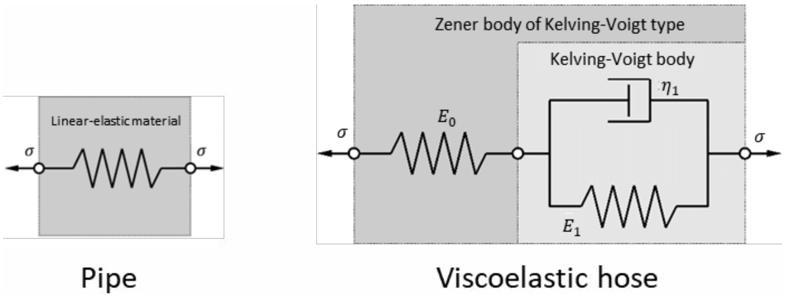

Figure 4.2: The viscoelastic component of the material behavior of hose material is represented by a Kelvin-Voigt model.

While the parameterization of a linear elastic steel pipe can be easily traced back to material and geometry values, measurements are still required for the exact parameterization of viscoelastic hose lines. The method for metrologically determining the four-pole transfer matrices in the frequency domain according to ISO 15086-3, which goes back to [8], has been used and further developed by Fluidon in numerous projects. Automated curve fitting methods then allow the models to be compared with the measurement data as a function of the operating parameters. **Figure 4.3** shows an example test setup for four-pole measurement. A comprehensive description of the measurement method, which originally comes from acoustics of exhaust systems for combustion engines, and the possible measurement setups, including a discussion of their specific advantages and disadvantages, can be found in *Tao et al.* [9].

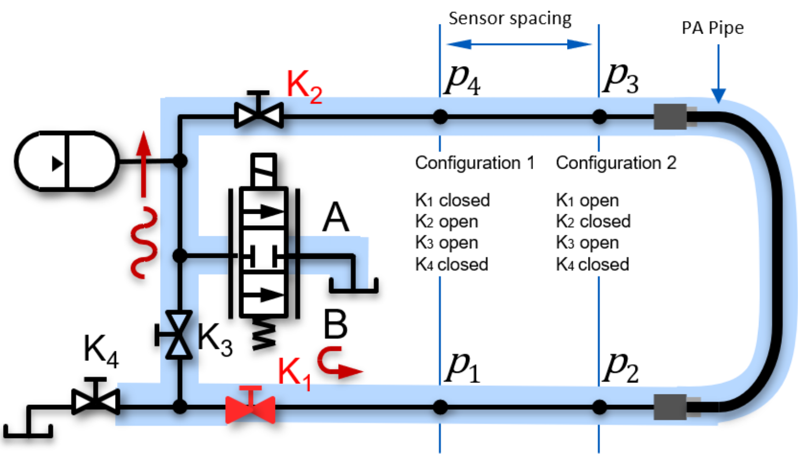

Figure 4.3: Test setup for four-pole measurement according to the secondary source method [10, p. 467]

Variable material data, thermohydraulic effects, cavitation, and gas transport are explicitly taken into account in the 1D CFD calculation and included in the solution of the conservation equations as time-dependent variables.

The time interpolation method proposed by *Savar et al.* [11] achieves greater flexibility in discretization with lower numerical damping, which is particularly advantageous when modeling complex systems. Further efficiency gains in the calculation of large models are achieved through parallel calculation of pipe sections.

4.2. Open / Closed End

In addition to the pipes, other components are required to model complete systems. The simplest boundary conditions are open and closed ends, at which either a pressure or a volume or mass flow is applied.

A pressure boundary condition can be used, for example, to represent a pressure source or a tank ($p \approx 0$). An applied volume flow can already represent a simple pump and forms the basis for more advanced models of displacement units.

If 1D CFD is coupled with concentrated parametric (0D) models, open or closed ends serve as defined interfaces to the 0D components.

4.3. Connectors

The algorithms of the method of characteristic and the resulting model structure do not allow two line elements to be connected directly. Instead, two or more lines are interconnected via connecting elements. With the help of the 2- to n-way connectors shown in **Figure 4.4**, up to 8 lines can be interconnected.

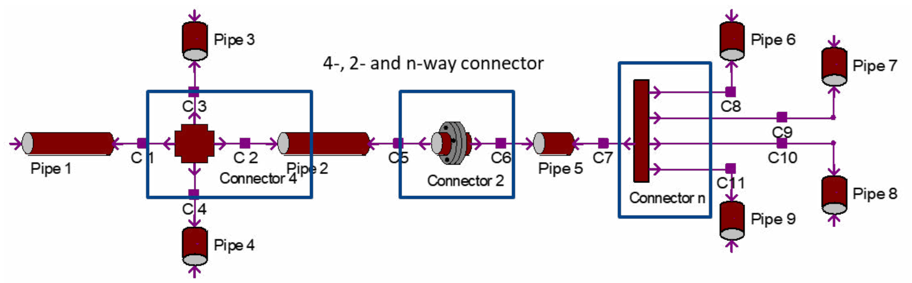

Figure 4.4: Connector types

4.4. Pumps and Motors

With the closed end, a volume flow can already be imposed on a system model, which can also be defined as variable over time, for example, according to a measured flow ripple or a pump start-up. In the case of a straight pipe closed at both ends, the pressure oscillation situation shown in **Figure 4.5** is obtained.

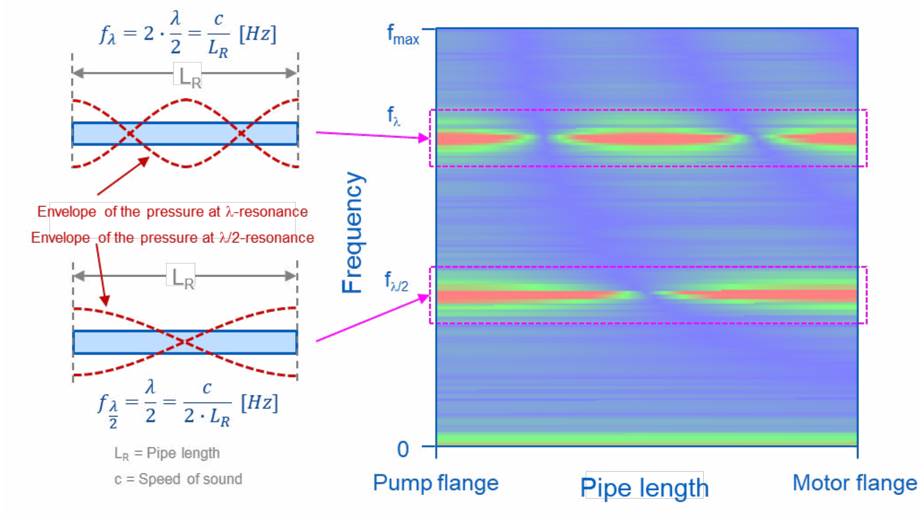

Figure 4.5: Resonances of a pipe with closed ends

A color-coded top view of the stored pressure values of the data matrix is displayed as a *pressure vector plot* [12]. The pipeline (abscissa) starts at the pump and ends at the consumer. The ordinate does not show the simulation time, but rather the excitation frequency. This is possible because the excitation frequency was changed linearly during the simulation, allowing a corresponding frequency to be assigned to each simulation time point. The horizontal lines indicate the first natural frequency ($\lambda/2$ resonance) and the second natural frequency (λ resonance). The pressure vector plot in Figure 4.5 clearly shows when the excitation frequency "meets" the natural frequencies of the pipeline at $f_{\lambda/2}$ and at f_λ.

Since the excitation frequency was continuously increased in the simulation, the resonances initially build up slowly, reach their maxima, and then decay again. In the top view, the shape of the envelope resembles the pressure waveform visualized in the left part of Figure 4.5 (but should not be confused with it), which makes it very easy to identify the resonances in the pressure vector plot.

However, this is often insufficient for the acoustic evaluation of hydraulic systems, as the internal structures of a displacement unit also contribute to the oscillation behavior. Acoustically, they act as a resonator in extension of the pressure line and thus change the resonance behavior of the overall system. **Figure 4.6** shows the pressure vibration situation in the pipeline plus displacement unit on both sides.

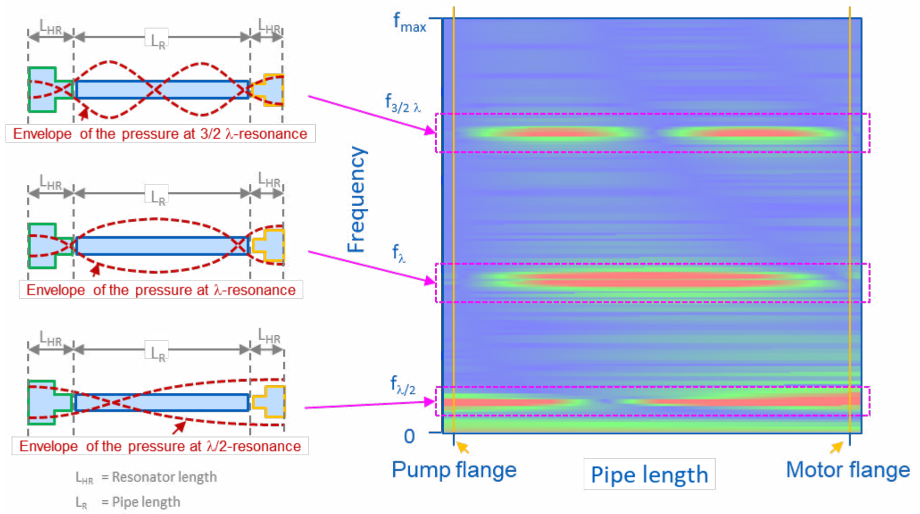

Figure 4.6: Resonances of the pump–pipeline–motor system

The resonator represents an equivalent model for the impedance of the displacement device. Until now, this had to be determined by measurement and is then represented in the system model by an equivalent model that is adapted as closely as possible to the measurement results using an optimization process.

In order to simultaneously impose a measured volume flow pulsation on the system model, Fluidon has developed the *hybrid pump model* [13], **Figure 4.7**. A modified Helmholtz resonator, which contains an additional throttle to increase damping, serves as the impedance replacement model. Experience has shown that it better suits for modeling the impedance of conventional hydraulic displacement units than the frequently used Norton model.

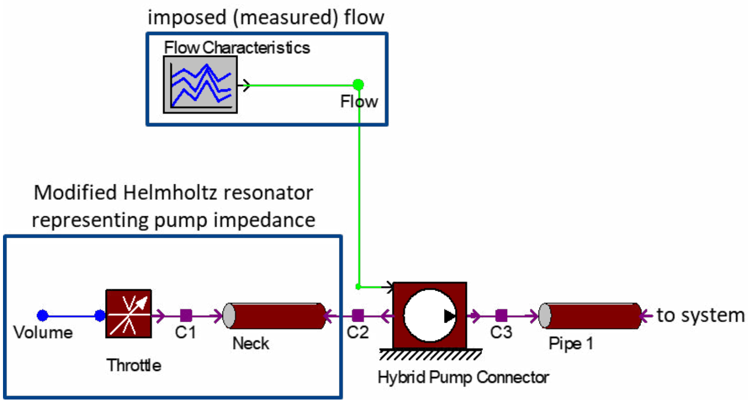

Figure 4.7: Hybrid pump model

Fluidon uses the secondary source method, which goes back to *Edge* and *Johnston* [14], to measure impedance. By using a frequency-variable excitation, the method was further developed in terms of results quality. It can be used on existing functional test benches with relatively little effort [15]. If the pump impedance is known, the flow ripple can be calculated from a measurement on the same test bench setup with the secondary source switched off [16]. However, this requires good quality impedance measurement or the use of a good quality replacement model. A method that is independent of the pump impedance is measurement using a low-reflection line termination (RaLa) [4], [17].

4.5. Resistors and Valves

The simple resistance component (**Figure 4.8** left) is based on a calculation comparable to that for the double connector, which solves the characteristic equations and the continuity equation. While $p_1 = p_2$ applies to the connector

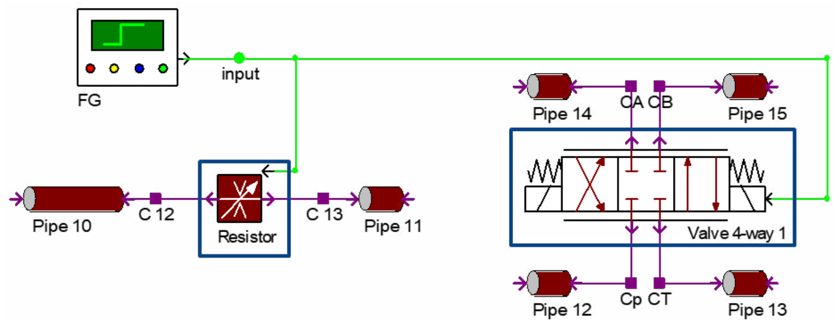

Figure 4.8: Resistance and valve component

In principle, valves can be modeled in 1D CFD as a combination of single hydraulic resistances with intermediate pipes and branches. Since a resistance cannot be directly coupled to a branch, but always requires an intermediate pipe, this approach leads to very extensive and confusing models, especially in the case of complex valve structures.

This is particularly true for valves with multiple effective control edges that are connected to each other via short channels, whose acoustic behavior is often negligible. The "valve" component was therefore introduced to efficiently map such components. It allows its connections to be flexibly linked to each other depending on a control signal. Optional characteristic maps for the effective opening cross-sections allow the mapping of almost any valve characteristics.

5. MODELING AND ANALYSIS OF COMPLEX SYSTEMS

The following section first describes the modeling of complex hydraulic pipe systems before presenting methods for analyzing vibration and resonance behavior based on this.

5.1. Model Creation of Complex Hydraulic Systems

Complex hydraulic systems are usually modeled on the basis of the circuit diagram and the associated CAD data. Steel pipes, valves, and manifold blocks can be mapped with sufficient accuracy in 1D CFD using CAD geometry and the corresponding material data. **Figure 5.1** shows an example of how the internal bores of a manifold block are taken into account in the model.

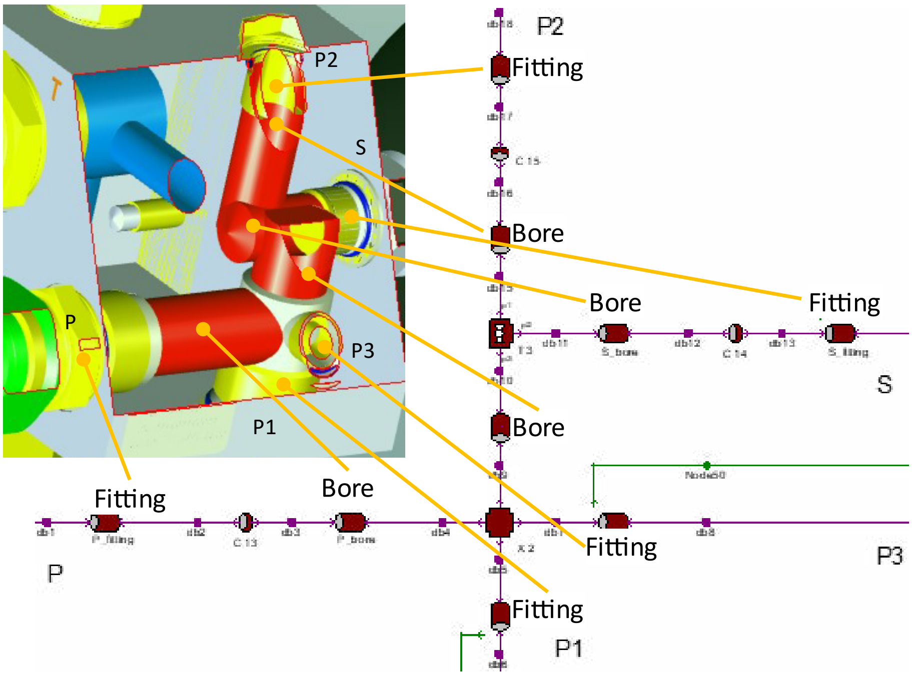

Figure 5.1: Depiction of a manifold block in the model

Hose crimp fittings are modeled as separate pipe sections. In many applications, return lines are not relevant to noise from an NVH perspective and can therefore often be neglected without significantly affecting the validity of the acoustic analysis.

As an example, the pressure-side model of the hydraulic system of a large mobile machine requires around 100 pipe components. The computing time for such a model on the *Fluidon Cube* platform [18] is around 9 hours (without parallelization).

5.2. Analysis of Vibration and Resonance Behavior

In many applications, the actual pulsation of the displacement units as excitation of the system is initially unknown, or it is useful to determine the system response independently of the excitation amplitude in order to obtain an amplitude-independent *vibration fingerprint* of the line system. For this purpose, a sine sweep with constant volume flow amplitude is applied to the system instead of the actual pulsation. The resulting system response is a direct measure of the admittance of the system. Admittance describes the frequency-dependent ratio of the system response to the applied excitation and is therefore a measure of the dynamic compliance of the fluidic system. It allows the identification of resonances and NVH-critical frequency ranges.

However, to analyze the vibration modes of a complex, branched piping system, it is not sufficient to evaluate the pressure curve at individual measuring points in the time or frequency domain. To systematically identify resonances and spatial vibration modes, Fluidon again uses the representation in the form of so-called *pressure vector plots* (**Figure 5.2**).

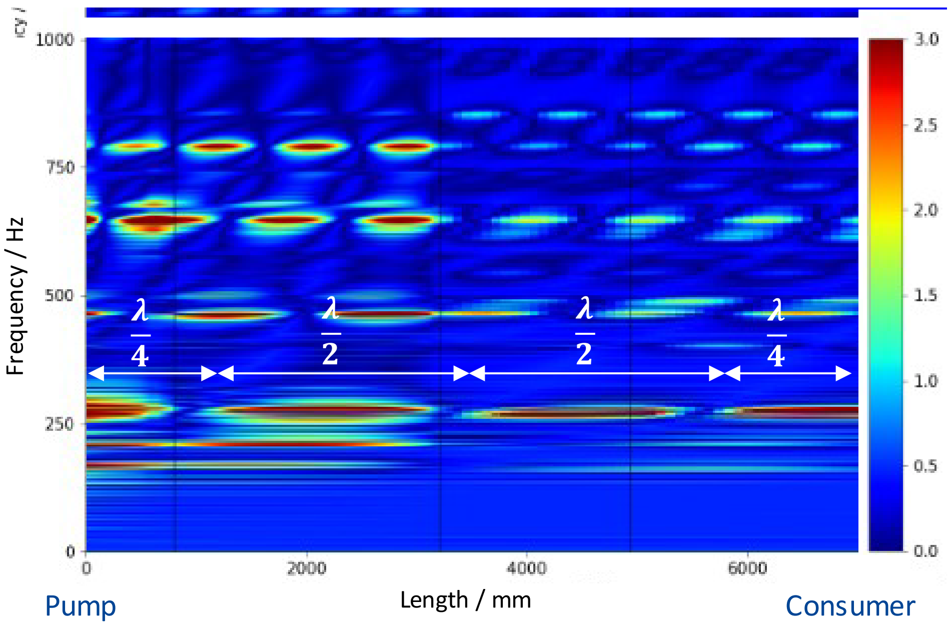

Figure 5.2: Pressure vector plot for analyzing resonances and vibration modes

As already introduced in Figure 4.5 and Figure 4.6, the x-axis describes the pipe length, for example from the pump outlet to a critical consumer. Vertical lines mark the transition between line sections, like a hose-pipe combination or a manifold block. The frequency is plotted on the y-axis, while the pressure admittance is represented as a color gradient. This visualization allows frequencies to be directly assigned to spatial vibration modes in the piping system.

The example shown clearly illustrates that a critical resonance vibration occurs at approximately 260 Hz, which is pronounced in the pipe branch shown over 3/2 wavelengths λ . To evaluate the overall system, such an analysis is performed for all relevant branches. The spatial characteristics of the critical vibration modes can be used to derive specific optimization measures, which are then evaluated using variant simulations.

Knowledge of the spatial vibration modes and dominant resonances in the pipe system forms the basis for the subsequent consideration of fluid-structure interaction. The pressure amplitudes can be used to directly calculate the dynamic forces acting and their transmission to pipes, fastenings, and adjacent structures.

6. Fluid-Structure Interaction

"A true, two-way fluid-structure interaction (FSI) exists when the flow is noticeably influenced by the structural change. However, the effect of the structural change on the flow is often so weak that it can be neglected. In this case, we refer to a one-way fluid-structure interaction." [19]

The article *Pressure Oscillation Analysis in Fuel Supply Systems* [20, pp. 337–358] describes how DSHplus takes bilateral FSI into account. However, the system of equations can be reduced so that only unilateral FSI is calculated. Based on the transient pressure and flow curves determined in the vibration fingerprint, the resulting hydrodynamic forces are calculated at selected points in the hydraulic system. This simplification significantly reduces the modeling effort and enables numerically very efficient simulations without significantly limiting the information content relevant to NVH issues. The aim is not to calculate deformations or stresses in detail, but to determine the time-dependent excitation forces that serve as input data for further structural dynamic or acoustic analyses.

The approach is particularly suitable for:

- Identifying NVH-critical line sections or system components
- Evaluating different plumbing concepts
- Early estimation of pulsation-induced continuous loads

To demonstrate the methodology, a practical pipe geometry is considered below.

6.1. Alternating Forces on a Double Pipe Bend

In addition to the acoustic pipe model, the spatial geometry of the piping system, as typically available in CAD planning, is required to calculate the fluid-dynamically induced forces. Based on the three-dimensional geometry, the piping system is divided into individual segments whose geometry is defined in the global coordinate system. **Figure 6.1** illustrates this transformation step from the CAD representation to the simulation model structure and shows an example of an assembly with several pipe bends and a connecting pipe section. Defining a boundary around this assembly makes it possible to specifically calculate the forces acting on the assembly.

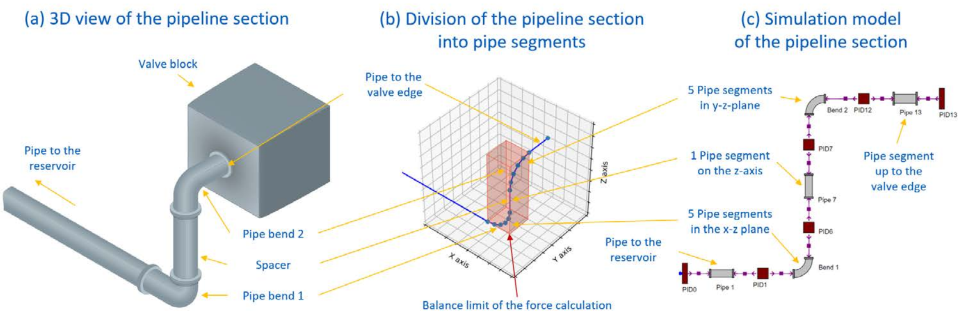

Figure 6.1: Preparation of the 3D representation for simulation

The forces are calculated by summing the time-dependent pressure and momentum components of all pipe segments within the balance boundary. Depending on the pipe routing and orientation in space, this results in force components in different spatial directions. The geometric arrangement of the example pipe already makes it clear in which directions forces can act preferentially, while other directions are in equilibrium in the static state. Changes in the operating state, for example due to pressure build-up or the start of flow, lead to corresponding changes in the distribution of forces on the assembly under consideration.

Figure 6.2 shows an example of such a simulation model and illustrates how the forces acting on a subassembly change over time. While pressure-related forces predominate in the static state, additional impulse forces occur when fluid flows through the pipe. Transient processes, such as the opening or closing of valves, lead to short-term changes in force, which are particularly noticeable in line bends and are often accompanied by visible or perceptible relative movements in real systems.

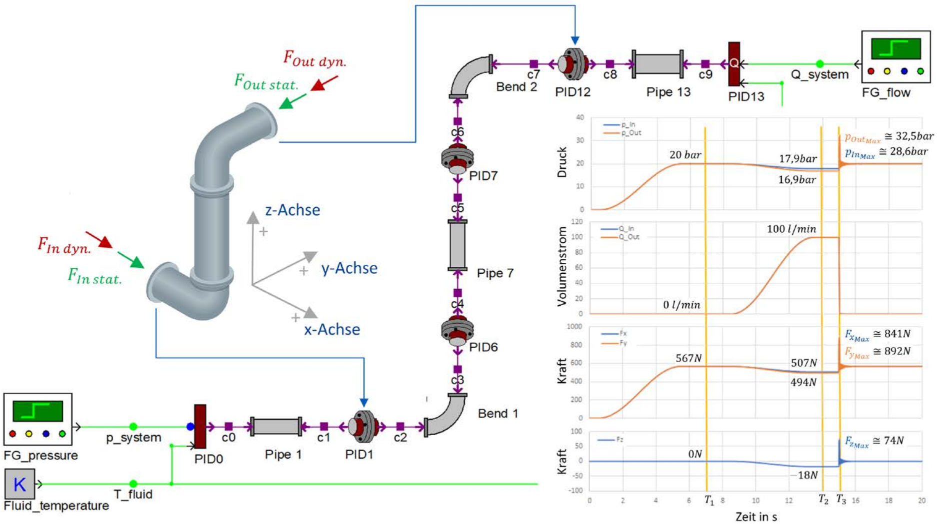

Figure 6.2: Simulation model of the pipe system

Similar to these temporary effects, permanent pressure and flow ripples induce dynamic forces, where resonance situations as described above lead to increased amplitudes.

Figure 6.3 shows the evaluation of a frequency-dependent excitation of the pipe system. With appropriate preparation of the simulation model, the force components shown can be calculated directly from the results of the vibration fingerprint in post-processing.

The illustration clearly shows that when certain excitation frequencies are reached, pronounced vibration modes of the fluid arise, which are directly reflected in increased dynamic forces on the assembly under consideration. The time- and frequency-dependent evaluation of the force components allows such resonance states to be clearly identified and spatially assigned.

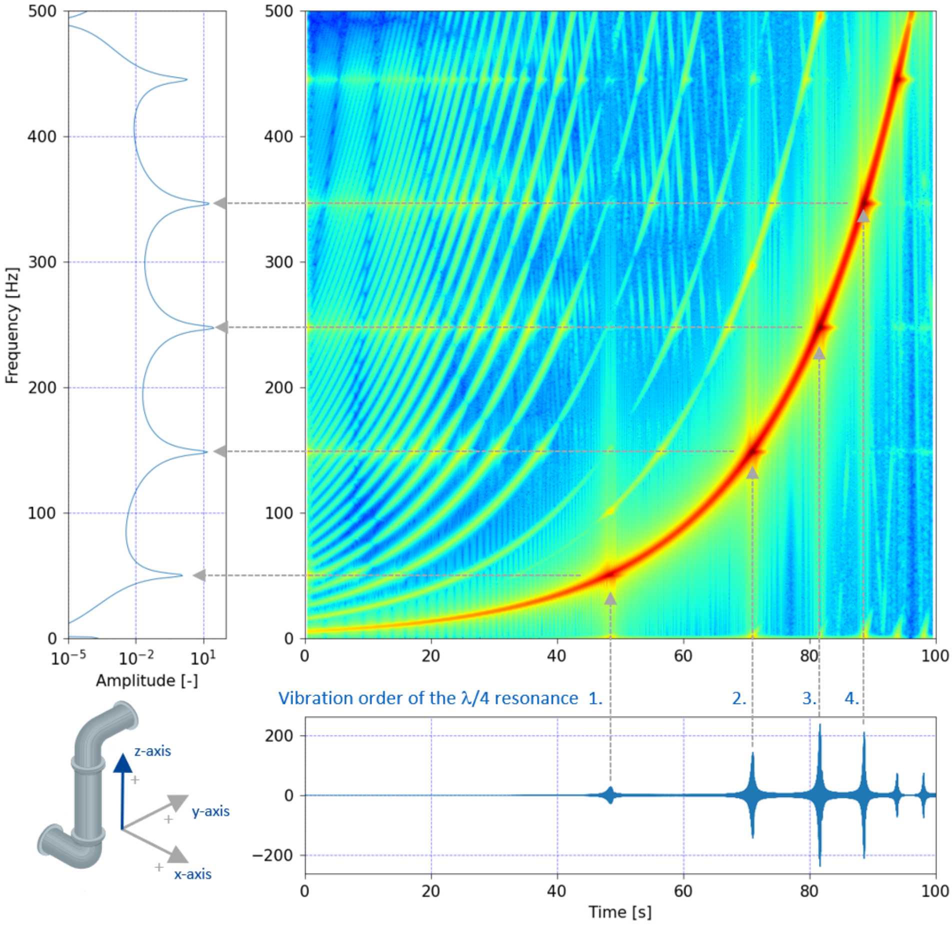

Figure 6.3: Analysis of the force component in the z-axis direction

The combination of pressure oscillation analysis and force calculation thus provides direct access to the structural mechanical evaluation of hydraulic systems. It enables dynamic alternating forces in relevant assemblies to be identified at an early stage and specifically incorporated into the design of line routing, mounting points, and structural measures. This approach is therefore a key tool for NVH-oriented system design.

7. Summary and NVH Implications

The results show that NVH problems in hydraulic systems are not exclusively attributable to rare extreme events such as pressure surges. Rather, flow ripple-induced continuous loads are a key cause of structural excitation and acoustic emissions and have a significant impact on the overall NVH impression of modern machines.

The numerically efficient simulation approach presented in this paper, based on transient 1D CFD in combination with simplified, unilateral fluid-structure coupling, makes it possible to identify and evaluate these effects early in the development process. NVH-critical excitation forces can be specifically localized and assigned to their cause in the hydraulic system.

Critical conditions can be avoided as early as the concept phase by making targeted adjustments to the line geometry, mounting concepts, or hydraulic boundary conditions. The approach thus represents a practical alternative to complex fully coupled FSI simulations and is particularly suitable for variant studies and the early design of structure-friendly and low-vibration line systems.

Overall, the methodology presented makes a significant contribution to the development of quiet, durable, and NVH-optimized hydraulic systems.

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Biographies

Benedikt Müller studied mechanical engineering at RWTH Aachen University and earned his doctorate in 2002 with a thesis on NVH topics in fluid technology. He has many years of experience in mobile hydraulics, fluid acoustics, and the development and simulation of hydraulic systems.

Heiko Baum studied mechanical engineering at RWTH Aachen University and completed his doctorate in 2001. Since 2001, he has been the managing director of FLUIDON GmbH, where he focuses on model-based engineering, digital twins, and automated simulation workflows for hydraulic and pneumatic systems. His long-standing experience in the simulation and development of fluid power systems bridges industrial applications and research projects in the context of Industry 4.0 and the Asset Administration Shell.