Research Article

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Experimental and Numerical Investigations of Pipeline with Resonator

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Abstract: This article presents the current state of the art regarding the use resonators in straight pipes. There is considerable need to control and reduce pressure pulsation in pipelines supplied with pulsating flows. The use of a Helmholz resonator introduces an additional degree of freedom to the analysed dynamic system. Building on previous experimental investigations by the authors, which identified the nonlinear properties of straight pipes supplied with pulsating flows, this study describes an experimental test rig, measurement methods and mechanical analogies for one (1DOF) and two (2DOF) degrees of freedom. The results are presented in the form of a 3D map of amplitude-frequency characteristics, as a function of the resonator volume determined by piston height. The dynamic properties of the described system are presented as amplitude-phase characteristics, based on a comparison of the numerical and experimental results.

Keywords: Helmholz resonator, pipe natural frequency, double degree of freedom mechanical oscillator, pipe pulsation damping

1 Introduction

Pulsations and resonance in pipelines cause problems in a wide range of industrial applications, from process engineering to automobiles, including the latest thermoacoustic engines [1] and new metamaterials [2, 3]. The difficulty of identifying resonance in pipes with pulsating flow is due to the nonstationary and periodic nature of the supply, which is provided by reciprocating compressors [4] or receivers such as pumps or periodically working units. The most common way of reducing pulsation in ducts is to use various kinds of nozzles and orifices [5], which can reduce pulsation significantly. This solution is sufficient in many cases when resonance does not take place. The situation can change dramatically when the system enters the resonant work mode. This causes a sudden increase in transient pressure amplitudes, which risks seriously damaging and perhaps destroying the pipeline along with the associated machinery. Sometimes, pulsation and resonance have more human contexts, such as in automotive applications. Generally, the inlets and outlets of internal combustion engines (ICEs) can be treated as a pipeline with many variations in cross-section and shape. During the normal operation of vehicle powertrains, especially ICEs with turbochargers, there are periodic interchanges with significant changes in pulsation frequency [7–9]. The tuning inlets and outlets of vehicle systems are also a consideration. These systems need to be tuned to provide good external acoustics, especially for limousines and sport cars.

Effective methods for avoiding pulsation and resonance include:

- **Active Duct Damping**. This method involves the mounting of additional units, which enable active damping of excited resonance. According to active damping theory, the additional unit should have one degree of freedom and work on the principle of a mass-damping spring system. Of course, the natural frequency of the resonator should be the same as that of the damped system. A basic pipeline (which can also be represented as the mass-damping spring system) thus obtains an additional degree of freedom, allowing active pulsation damping. The most frequently used resonators take the form of an additional volume connected to the pipeline, similar to the Helmoholz resonator [10–13].
- **Anti-Resonant Design**. This method is based on the use of an anti-resonant duct. It requires very precise and narrow working conditions, especially in the frequency domain. The API 618 [14] standard of pipeline design and operation is still under development. The main problem is that resonant phenomena are very sensitive to initial and boundary

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conditions. Moreover, industrial applications are often much more complicated than the simplifications produced by current pipeline design tools, especially when empirical data is unavailable.

Given these considerations, the authors decided to design and build an experimental rig for a tube resonator with an adjustable chamber volume mounted in the middle of a common straight pipe. The main aim was the experimental identification of the dynamics of a pipe with a tube resonator. In the authors' opinion, these experiments (numerical and real) significantly simplify the design and operation of such pipes, and also facilitate inspection by technical authorities. The experimental results complete and complement the proposed design method and should decrease the time needed for precise selection of the appropriate duct damping system significantly.

2 Experimental test rig

A test rig was prepared to investigate the transitional states of varying pulsation frequencies in a pipe fitted with a tube resonator. In what follows, the results of a series of measurements that were taken for various frequencies will be presented. The main flow parameters evaluated were pressure, temperature and mass flow, measured at two control sections. The test procedure also examined the frequency change domain, in terms of initial and final values [15]. The amplitude frequency characteristics were estimated using the curve fitting function in Matlab software. Estimates were based on second-order inertial elements and provided quite a good representation of the acquired data [16].

The following assumptions were made:

- Real measurements were taken for transitional states during pulsation frequency changes (from 20 to 180 Hz) in pipes. The main features of the test rig are presented in Figure 1, including the pulse generator (PG) and three control sections (1,3).
- LabView software was used for measurements and post-processing, as in previous research [17, 18]. The main function of this software is to enable the measurement process to be performed in repeated sequences, each identified by time stamps defined as a peak from a photoelectric transducer. Fast Fourier Trasform signal windowing is performed based on the acquired timestamps. Each series of results is stored, and the identified magnitudes in the frequency domain can be exported. Thanks to this automatization, any particular measurement of one amplitude or phase characteristic in the frequency domain takes about a quarter of an hour.
- Each measurement series was repeated three times to ensure reliability.
- Analysis of the influence of frequency change during pulsation changes was performed for a range of amplitudes, from 20 to 180 Hz, under air pressure at three control sections.
- Parameters for estimating second-order inertial elements (the damping coefficient and resonant frequency) were proposed to describe the observed phenomena.
- The impact of the volume of the tube resonator chamber on amplitude–frequency characteristics was assessed.

The main parameters were as follows:

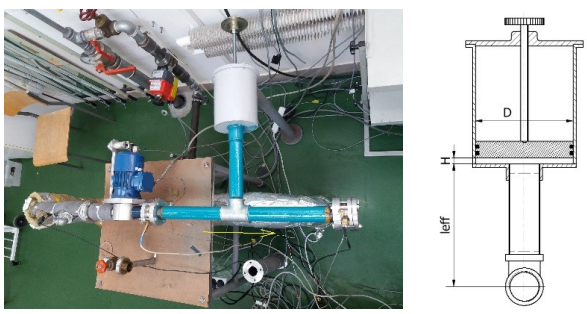
- Range of desired values for the frequency of the pulse generator, f=20–180 Hz.
- Pipe diameter, $D_p = 42 * 10^{-3} m$.
- Pipe length, $L_p = 0.544\ m$, determined for resonance at 114.7 Hz.
- Desired flow temperature, T=313.15 K.
- Mean Flow speed, u=20 m/s (mean value).
- Mean Pressure, p=115000 Pa.
- Tube resonator parameters as defined in Figure 2b: neck diameter same as pipe diameter, efficient neck length $l_{eff} = 250 * 10^{-3} m$, chamber diameter $D = 200 * 10^{-3} m$, tuneable tube high $h = 0 - 150 * 10^{-3} m$. Piston position defined as H parameter was changed using a metric screw with resolution $\Delta h = 1 * 10^{-3} m$.

Transient and mean values for pressure, temperature and specific mass flow rate were measured at control sections (1) and (3), shown in Figure 1, where the yellow arrows mark the direction of flow. Transient pressure was also measured in section (K), located in the middle of the pipe length.

Analysis of the dynamic properties of the pipes was performed using the second-order oscillating element as the reference, parametrizing objectively the differences between particular cases. It was thereby possible to approximate probes with a coefficient of determination greater than 95% ($R^2 > 0.95$). Second-order oscillating elements were estimated using equation (1) and the Curve Fitting Tool, with custom equation settings and default 95% confidence bounds:

$$M(f) = \left\{ \left[1 - \left(\frac{f}{f_n} \right)^2 \right]^2 + \left[\frac{2\zeta f}{f_n} \right]^2 \right\}^{-1/2} \tag{1}$$

Figure 1: General view of the experimental test stand.PG – pulse generator, CS 1,2 cross section 1,2, TR – tube resonator.



(a) (b)

Figure 2: (a) Zoomed view of tested pipeline with resonator. (b) Cross-section view of the tested resonator

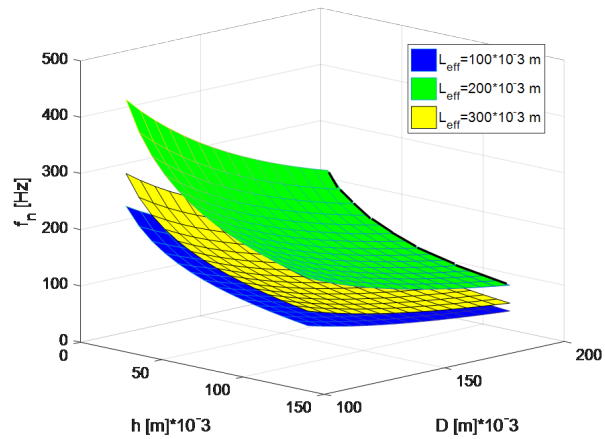
$$\phi = -tan^{-1}\left[\frac{\frac{2\zeta f}{f_n}}{1-\left(\frac{f}{f_n}\right)^2}\right] \tag{2}$$

$$f_r = f_n\sqrt{1-2\zeta^2} \tag{3}$$

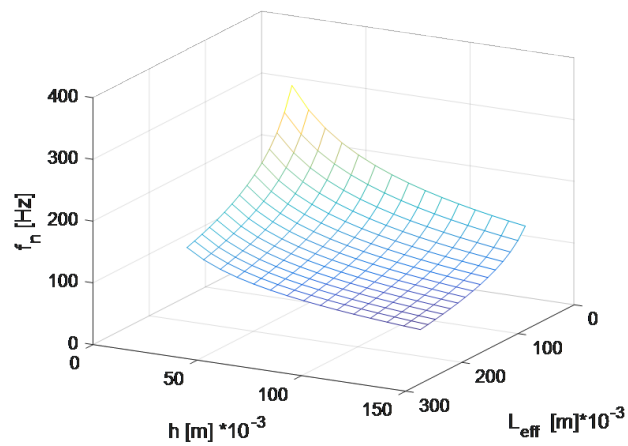
Where:
$M(f)$ – magnitude of oscillations [-]
f_n – resonance frequency [Hz]
ζ – relative damping coefficient [-]
f_r – resonance frequency under conditions of forced oscillations [Hz].

2.1 Resonator selection method

According to the well-known basic equation (4) describing the natural frequency of a Helmholz resonator, three-dimensional maps were processed, as shown in Figures 3a, 3b and 4.



(a)



(b)

Figure 3: (a) Influence of tube diameter and height on natural frequency of pipe; (b) Influence of tube height and neck length on natural frequency of pipe (D = 200 * 10 – 3m)

$$f_n = \frac{a}{2\pi}\sqrt{\frac{A}{V_0 * l_{eff}}} \tag{4}$$

Where:
a – speed of sound $[m/s]$
A – neck cross section area $[m^2]$
V_0 – chamber static volume $[m^3]$
l_{eff} – effective neck length $[m]$.

The selection process should first consider the choice of tube diameter, since this determines the whole design of the resonator. Next, the widest range of adjustable chamber heights for a fixed tube diameter should be found (dark line in Figure 3a). This enables a wide range of available resonant frequencies. Finally, the effective neck length, which includes the neck length and pipe radius, should

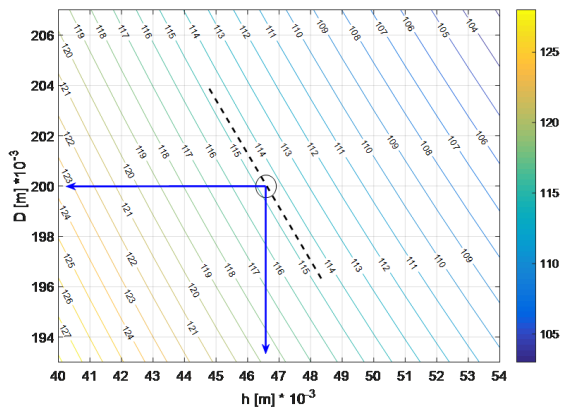


Figure 4: Resonator tube diameter and high exact selection based on real resonant frequency of tested pipe $f_r = 114.7$ Hz for $l_{eff} = 230\ m * 10^{-3}$

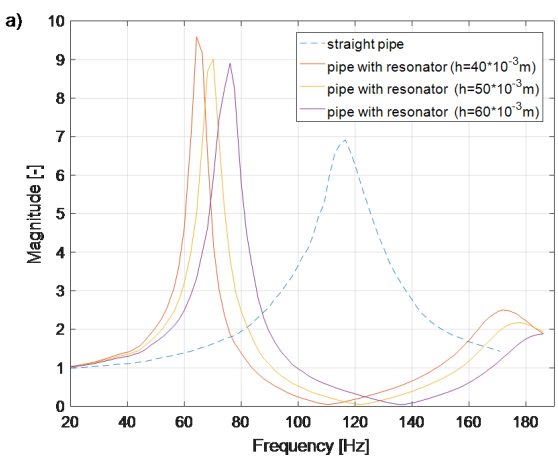


Figure 5: Magnitude of p_3/p_1 frequency characteristics for straight pipe (dashed line) and pipe with resonator for three piston positions h = 40; 50; 60*10−3 m

be chosen in such a way as to move the range of the adjustable resonator to the required region. This enables use of the same adjustable resonator in different frequency domains, simply by tuning the neck length, as presented in Figure 3b.

Figure 4 shows the precise theoretical selection of the required tube height for a fixed tube diameter. The dashed line represents the resonant frequency of the tested pipe (without the proposed resonator) $f_r = 114.7$ Hz and also for a fixed neck length $l_{eff} = 230 * 10^{-3}$m.

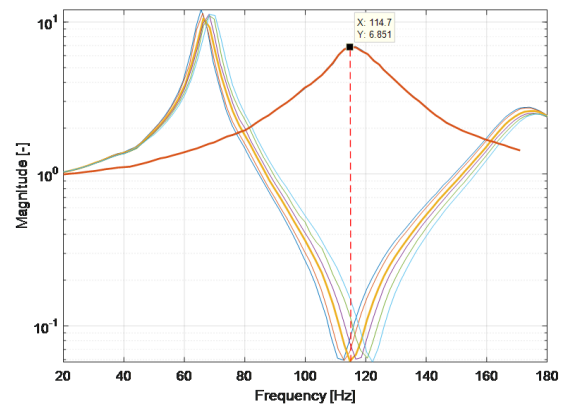


Figure 6: Precise tuning of the resonator position to the resonance of a straight pipe, h = 45; 47; **49**; 51; 53; 55*10−3 m

3 Experimental results and discussion

Experiments were performed according to the procedure for selecting the main resonator parameters presented in Sec. 2.1. According to the theory of active damping, the resonator should be designed to have exactly the same natural resonant frequency as the dynamic system. Because of the obvious discrepancies between the theoretical system and its realization, preliminary (Figure 5) and accurate (Figure 6) tuning of the resonator height was performed to reach exactly maximum damping at the original system resonance, identified for a straight pipe $f_r = 114.7$ Hz. It is worth mentioning that the logarithmic scale of magnitude enables more adequate graphical representation and identification of maximal damping effects (see Figure 6). Exact tuning was achieved for an adjustable resonator with $h = 49 * 10^{-3} m$, shown by the bold line in Figure 6.

The presented experimental results prove that the proposed method enables correct selection of resonator parameters, provided the measurement system is of sufficient quality. The reliable characteristics presented in Figures 5 and 6 were produced using a wide range of repeated probes. Each presented characteristic is a mean value from five series of probes.

Finally, Figure 7 presents a three-dimensional map of the influence of the piston position on variations in the magnitude of a pipe with a resonator in the frequency domain. At low piston positions, the variation in magnitude is rather similar, but from $h = 50 * 10^{-3}$m, this parameter has a significant influence. All double-resonance characteristics move in the direction of higher frequencies.

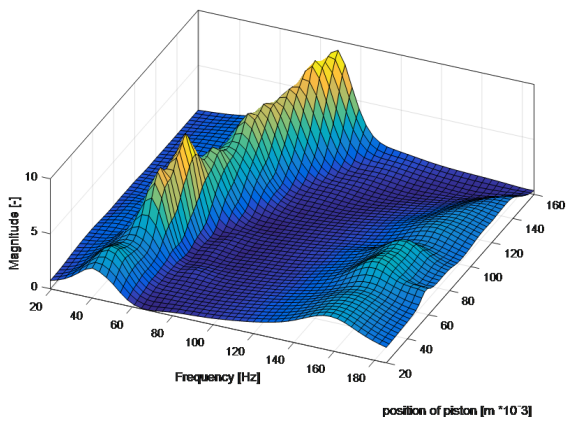


Figure 7: Influence of piston position on resonance modes – 3D graph

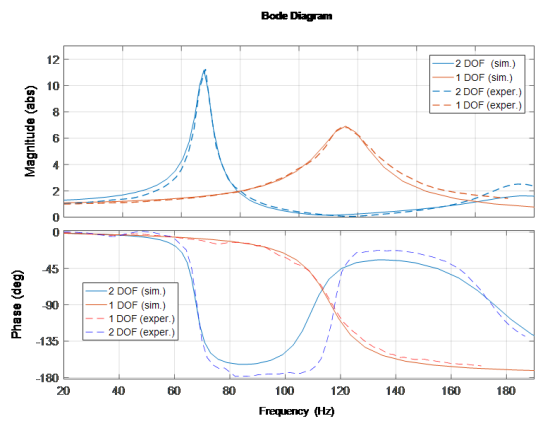


Figure 9: Magnitude and phase characteristics of tested system. Comparison of simulations and experimental research with one and two degrees of freedom.

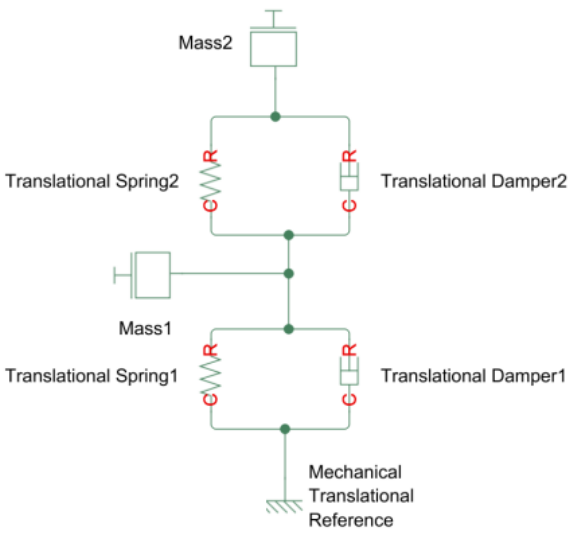


Figure 8: Model with two degrees of freedom (2DOF) in linear translation – Simulink model

4 Two DOF Simulnik models of the tested system

In this section, a mechanical system with two degrees of freedom (2DOF) will be presented, which reflects quite well the dynamic properties of the analysed system. This mechanical analogy, consisting of two mass-damping spring systems, is very often used in such cases. A single tube filled with air excited by harmonic pressure pulsation works analogously to a mass-damping spring system. The propagating longitudinal wave in a straight pipe is caused by compression and rarefaction of the air in the pipe. A Helmholz resonator can also be identified dynamically in this manner. A mechanical analogy of the analysed system can thus be designed, using the Matlab/Simulink environment (Figure 8). The bottom mass-damping spring system represents only a straight pipe with its resonant frequency defined as the relation between "Mass 1," "Translational Spring 1" and "Translational Damper 1" according to equation (3). The upper mass-damping spring system represents a Helmholz resonator with the same proportions between "Mass 1," "Translational Spring 1" and "Translational Damper 1" to ensure the same resonant frequency as the pipe.

It is worth mentioning that the variations in absolute pressure at cross-sections 1 and 3 are very untypical for dynamic 2DOF systems, because the mean value is relatively high $p_{mean} = 115\ 000\ Pa$. The amplitudes of the excited system are also relatively small, e.g., $p = 115\ 000\ Pa$. The proposed model was optimized for maximum adequacy with the experimental data, where the searched parameters were: $m_1; k_1; c_1; \alpha$.

The αcoefficient defines the conjugated values of $m_2 = \alpha * m_1;\ k_2 = \alpha * k_1;\ c_2 = \alpha * c_1$.

4.1 Comparison of Simulink model with experimental results

This subsection compares the simulation with experimental results. All results are summarised in Figure 9. Both the upper and bottom graphs present magnitude (upper) and phase (bottom) in the frequency domain in four cases: for simulations and experiments with one degree of freedom and with two degrees of freedom (with a resonator). Quite good similarity was reached between the simula-

tion and experiment, especially for the magnitude graphs. The differences in the phase characteristics between the modelled and experimental data were caused by inadequate correspondence between the analysed system and the 2DOF mechanical analogy.

The differences observed are caused by the fact that a very complicated dynamic flow system was approximated, with transient states of thermodynamically conjugated parameters including pressure, temperature and mass flow rates. Despite this, it is possible to use the proposed simulation model to support the design process, especially in combination with experimental research. The authors recommend that preliminary experimental data be used to tune the simulation model. Then, it is possible to refer to the simulation to shorten the process of designing pipelines with resonators. Finally, all parameters estimated based on the simulation should be verified experimentally.

Further research should examine:

- a wide range of mass flows;
- how the presented system works with a sine sweep constraint; and
- the application of a fast opening valve connecting the tested pipe with the proposed resonator.

5 Conclusions

This paper has presented a numerical and experimental study of a pipeline with resonator that could facilitate the first steps in the design and operation of such systems. A method was proposed for preliminary selection of the main parameters of the Helmholz resonator, according to the existing pipeline boundaries. A semi-experimental method was then presented for tuning the adjustable resonator with the support of simulation. A low-cost method for moving the adjustable range of the resonator by changing the length of the neck was also discussed. Finally, original experimental results were provided, which enable easier understanding of pulsation and resonance in pipelines with resonators.

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