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ORIGINAL PAPER

Reduction of fluid noise in modern aircraft hydraulics by integrated broadband attenuators

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Abstract

New axial piston pumps with fixed displacement volumes, such as the high efficient power package (HEPP) pumps operate at variable rotational speeds and thus variable flow rates in a wide range. Using intelligent pump control systems, small and light packages are possible. In contrast, the fluid pulsations vary due to the rotational speed and have to be controlled avoiding stress or wear in the hydraulic system components (e.g. seals of tubing) and potentially impacting cabin noise. Due to the excitation frequencies, a broadband silencer for high pressure applications is required. This is realised by a multi-Helmholtz-resonator (MHR) concept within one cylindrical volume. The design will be adapted for pump-specific frequency characteristics in aircraft hydraulic conditions and fine adjusted during test verification by means of an adjustable tube-inside a pressure shell. For cost, weight and complexity reason a common broadband attenuator is integrated in the hydraulic circuit connecting the two redundant HEPP motorpumps. The attenuator position is optimised to achieve maximal acoustic performance and minimize pulsation stress on the pump, especially at the main operating point. The proof of concept has been achieved experimentally comparing a HEPP piston pump with and without the MHR.

Keywords Axial piston pump · Fluid borne noise · Cabin noise · Helmholtz-resonator · Broadband attenuator

1 Introduction

The electrification of aircraft systems is an essential contribution to green air transport. Numerous solutions are therefore currently being developed at both component and system level. The required hydraulic components and systems are analysed, particularly in terms of their operational reliability, safety and causes of failure of high-pressure flexible hydraulic hoses, e.g. [1]. Modern pump technologies are tending towards highly efficient electrohydraulic power packs (HEPPs) with special electro motor pumps (EMPs) at their core [2]. The HEPP shows continuous operations for empennage primary flight controls as well as intermittent operation for high flow consumers such as the high lift system and main landing gear. It also includes an improvement of reliability and maintainability using a smart prognostic health monitoring (PHM) [3] and final assembly line (FAL) advantages, such as pre-testing and quick installation, too.

As a result of the design process, a duplex EMP configuration is required to ensure redundancy meeting safety and reliability requirements with high energy efficiency [4]. Significant hydraulic pulsations within the volume flow can cause noise emissions throughout the aircraft. This noise caused by the hydraulic fluid (FBN) can cause discomfort via various transmission paths. First, the hydraulic lines transmit the structure-borne noise (SBN) throughout the aircraft and at each point of contact within the body (Fig. 1). Second, the FBN can cause serious problems if resonances occur in other components [1]. This can lead to very high pressure amplitudes and thus shorter lifetimes with lower maintenance intervals and higher costs. Decoupling by flexible hoses and shock

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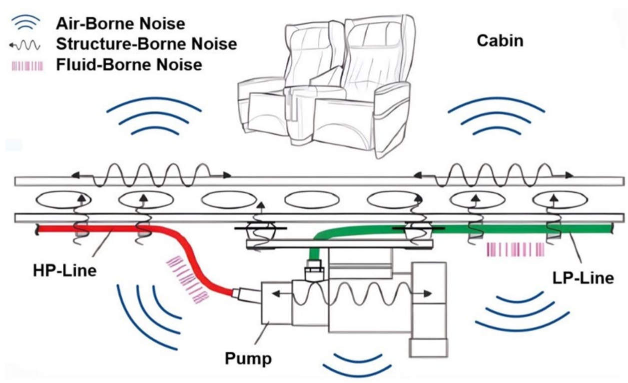

Fig. 1 Different transmission paths of cabin noise through pumps [7]

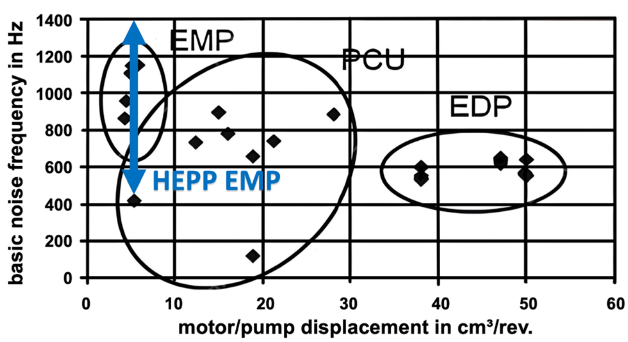

Fig. 2 Nominal frequency compared to the displacement range of typical hydraulic pumps and motors for aircraft in contrast to the wide frequency range of the new HEPP EMP [8]

absorbers is state of the art, but still lacks satisfactory cabin comfort at such high pressure amplitudes.

Due to progressive engine development with higher bypass ratios, these are becoming quieter and quieter and are no longer the main source of noise. Other continuously operated systems are becoming more noticeable acoustically and are therefore the focus of NVH technology [5]. Vibration sources, effects in hydraulic systems as well as transmission reduction methods are presented e.g. in [6].

The assessment of cabin noise is the responsibility of the airframer. No precise quantified data is available from this side for component development. Usual pump specifications are pressure pulsations of less than 1% of the nominal pressure.

Figure 2 illustrates the existing frequency spectrum and hydraulic displacement of various hydraulic motor or pump applications [1]. Due to the variable speed technology and the number of pistons, the novel HEPP EMP excites a wide frequency range of 450–1500 Hz and therefore requires an effective attenuator.

2 Passive reduction of fluid noise

Passive noise reduction technologies are preferred in aircraft and many other applications due to their low component and system complexity. The absence of moving parts and active controls reduces the risk of failures and lowers costs and weight.

In acoustics, there are two different operating principles: expansion chambers and resonators, with resonators chosen for this application because of their high damping properties. Due to resonance, the frequency bandwidth of the transmission loss is limited (see Eq. (2), [9]).

2.1 Physics of Helmholtz resonators

As early as the 1850s, Hermann von Helmholtz developed air resonators and used them to identify different frequencies in music or other complex sounds. These resonators consist of a known volume inside a rigid container, which at the time had an almost spherical shape, a thin and short neck at one end and a larger hole at the other end that emits the sound.

Well-known examples of Helmholtz resonance include open car windows causing the side windows to buffer, or sounds made when blowing across the top of an open empty bottle. They are also found in internal combustion engines, exhaust diffusers, subwoofers, various musical instruments and in architecture, either to amplify or reduce narrowband sounds.

A Helmholtz resonator can be thought of as a single mass-spring system, with the volume (in various forms) acting as the elastic spring and the short, thin, cylindrical neck acting as the point mass. Like any system with one degree of freedom (SDOF), it has a unique resonant frequency, which is given by:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S_0}{l_e V}} \tag{1}$$

where c is the speed of sound, V the volume of the resonator, S_0 the area of the neck and l_e the effective neck length.

For the given application, the transmission loss is

$$\mathrm{TL} = \log_{10}\left(1 + \frac{1 + 2R_s}{R_s^2 + \left(Q\left(\Omega - \frac{1}{\Omega}\right)\right)^2}\right) \tag{2}$$

with the reduced resonance resistance R_s, the relative frequency Omega Ω as well as the quality factor Q [9].

2.2 Variable temperatures

Due to the different operating conditions, a wide temperature range from − 55 to 90 °C has to be considered, with the focus on − 15 to 90 °C to achieve full performance. For the given FBN, this results in a decreasing bulk modulus and density with increasing temperatures. Thus, the sonic velocity varies in a range of approx. 30% (Fig. 3).

2.3 Design parameters of Helmholtz resonators

The given performance requirements demand only ± 1% residual pulsations after the attenuator within the full range of the pump operating speed, the different fluid specifications and the wide fluid temperature range.

Therefore, parametric studies of resonator volume, neck diameter and neck length were performed to determine the range of values for the final geometry. Using one neck and the wide range of excitation frequencies, the lower frequency limit requires a volume of about 0.80 L, whilst only 0.06 L is needed for the upper frequencies. This already shows the value range of the required design.

Using the lower and upper frequency bands and neck diameters of 2 and 5 mm, a study of the required neck length leads to a maximum length of 10 mm for volumes below 1.0 L. Figure 4 shows the trends for low frequencies and all temperatures. Higher frequencies and smaller neck diameters lead to significantly smaller resonator volumes.

Similarly, the neck diameter was also investigated. Figure 5 shows the tendency of increasing volume with increasing diameters at low frequencies and all temperatures. Higher frequencies and/or longer neck lengths lead to significantly smaller resonator volumes.

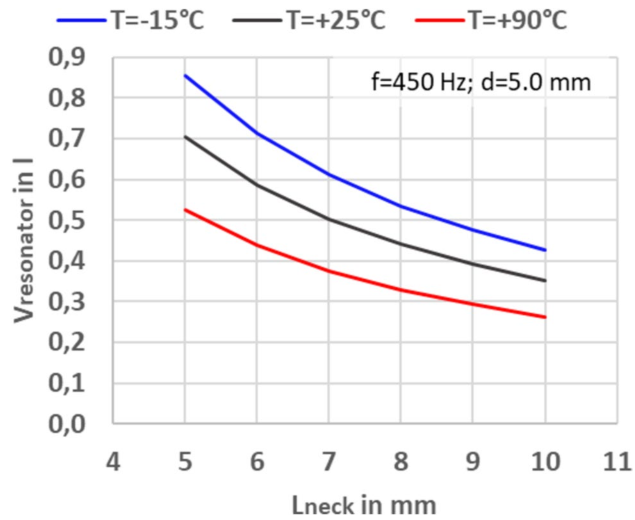

Fig. 4 Resonator design study with varying neck length

In summary, the lowest frequencies determine the resonator volume. It is possible to find a workable design for small neck diameters and longer neck lengths.

2.4 Resistance and transmission loss

In a next step, the transmission loss of a single resonator is evaluated. Figure 6 illustrates the influence of the dimensionless resistance for 1–50%, which is due to the energy loss of the fluid pulsation during its interaction with the resonator. Low resistance leads to low damping in the resonator and typically results in a high transmission loss at resonance, whilst high damping results in a greater effect on the effective frequency range [9].

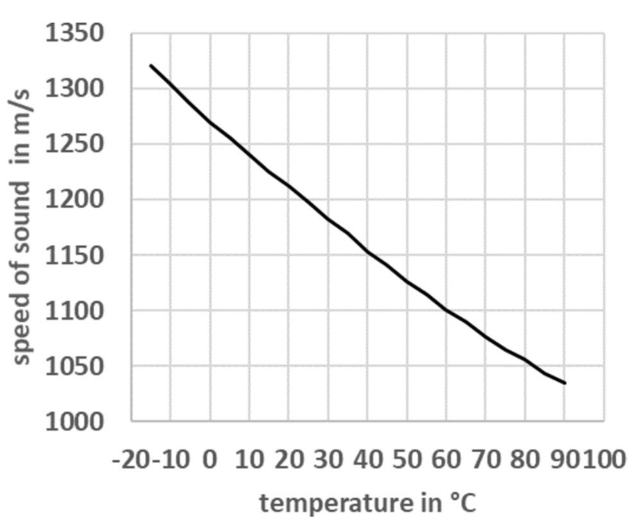

Fig. 3 Speed of sound of the given fluid at different temperatures

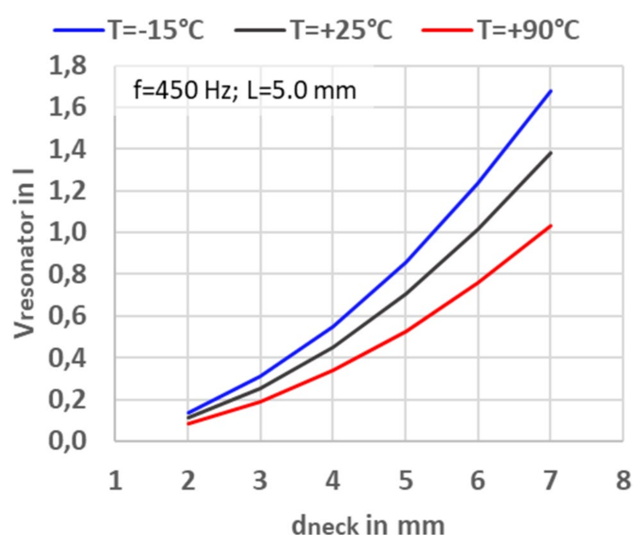

Fig. 5 Resonator design study with varying neck diameter

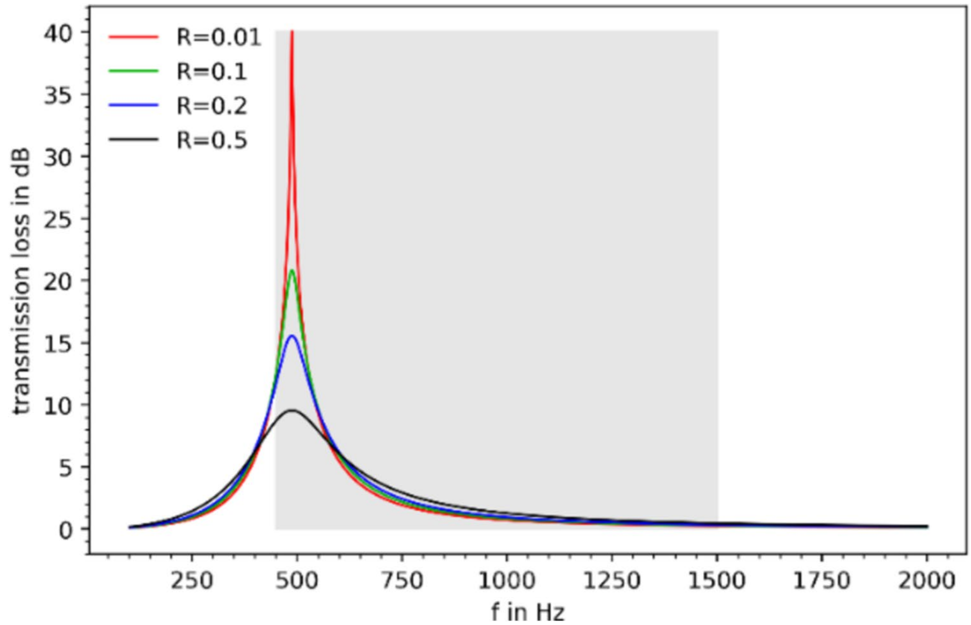

Fig. 6 Transmission loss of a single resonator for different dimensionless resistances

3 Multi-resonator for broadband applications

Due to the wide frequency range, different resonators have to be used simultaneously to achieve broadband operation. One possibility is to use a single volume in combination with several necks. Figure 7 shows an example of the interaction of three resonators in the frequency range mentioned.

A design with numerous necks has already been developed for other industrial applications and frequency ranges. For aircraft applications, the specific design was developed by scaling the geometry to the respective frequency.

In general, the frequency bandwidth is responsible for the number of necks required. In this application, the bandwidth is much larger than in industrial applications. Therefore, more necks were chosen. The inner tube was designed according to the dimensions of the system or neighbouring applications to avoid turbulence and pressure losses.

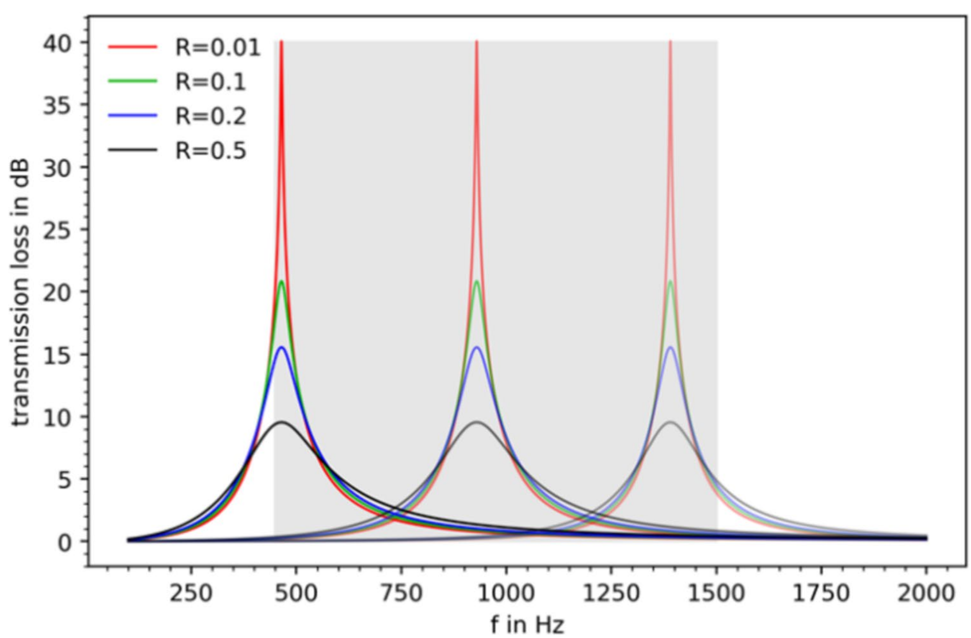

Fig. 7 Transmission loss of a triple resonator for different dimensionless resistances

The number and position of the holes depends on the frequency range and the expected energy of the vibration as well as the required damping factor within this frequency range. This influences the frequency position of the respective holes and their frequency spacing (and therefore number). Inner tube symmetry prevents incorrect installation. The final design of a cylindrical resonator volume with an integrated tube is shown in Fig. 8.

4 Validation in hydraulic test bench

The proof of concept was carried out with the novel high-efficiency motor-pump unit (MPU) of Liebherr Aerospace Lindenberg GmbH at technology readiness level 6 [1], as shown in Fig. 9.

The highly efficient permanent magnet synchronous machine (PMSM) drives an axial piston pump (APP) with 9 pistons in a speed range of 3000–10,000 rpm.

The MPU was integrated into the Fluid Born Noise test rig of the TUH/FST for fluid pulsation tests, which was built in Fig. 10. This test stand consists of an ISO 10767-1 confirmed setup with a pulsation measuring tube equipped with five different highly dynamic pressure sensors sampled at 25 kHz, as described in detail in [8]. The diameter of the measuring tube deviates slightly from the standard. There are two different load valves to adjust the flow and a 20-m long flexible hose between the measuring tube and an additional servo valve at the end.

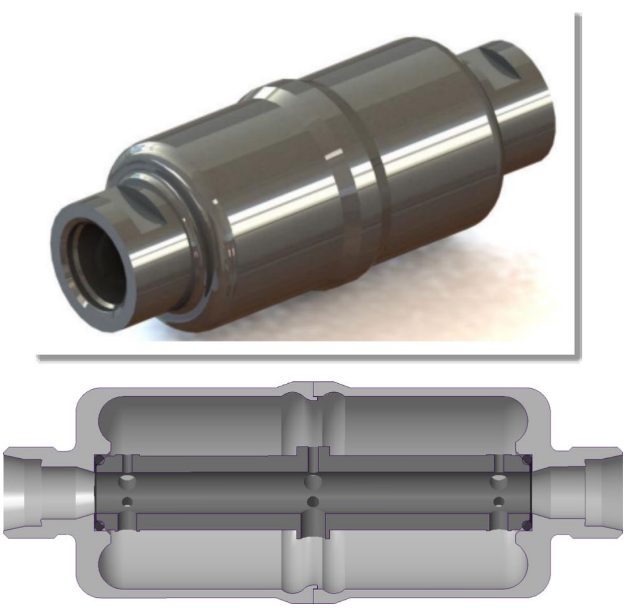

Fig. 8 MHR cylindrical attenuator design with outer resonator and multiple necks in the inner tube (Hydac)

Fig. 9 HEPP motor pump unit

The 20-m long flexible hose is used to dampen the back reflection of the downstream needle valve (NV1) to measure the so-called low reflection line ripple measurement (LR^2) with the pulsation tube.

Two different test campaigns were conducted, one to characterise the fluid pulsation of the MPU without a multi-Helmholtz resonator (MHR) and the second test campaign to characterise the fluid pulsation with an integrated MHR, as described in Fig. 11.

For each test campaign, both the ISO 10767-1 method and the LR^2 method were tested.

The result of the first test campaign is the flow impedance characteristic of the pump source in the frequency domain according to the ISO 10767-1 method, which enables a model-based simulation to be carried out for further development, which is essential for optimising the MHR (Fig. 12).

The results of the LR^2 method for both test setups, with and without MHR, are shown in Fig. 13 which prove that the MHR dampens sufficiently below the threshold from the system specifications in the tested operating range (corresponds to a pump speed range of 1000–7000 rpm) except for a small peak. The pump speed was controlled at the control cabinet and the load was adjusted with a NV1 to reach the desired pressure value. There were no servo valve in the test rig. SV stands for switching valve here.

The operating principle of this novel HEPP concept is constant pressure regulation with a variable speed and fixed displacement pump. Thus, the effective flow/pump load changes according to the actual speed whilst regulating the output-pressure constant.

Finally, POC testing of this novel technology for aerospace has been successfully conducted, providing the basis for further development with integration of the MHR into the system.

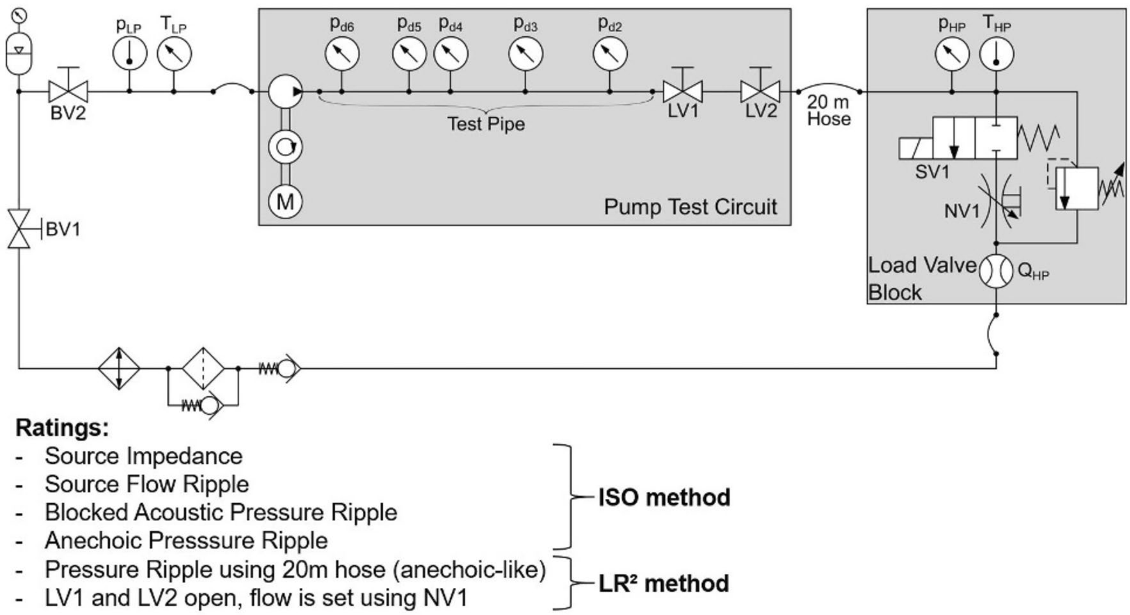

Fig. 10 Fluid born noise test rig at TUH/FST

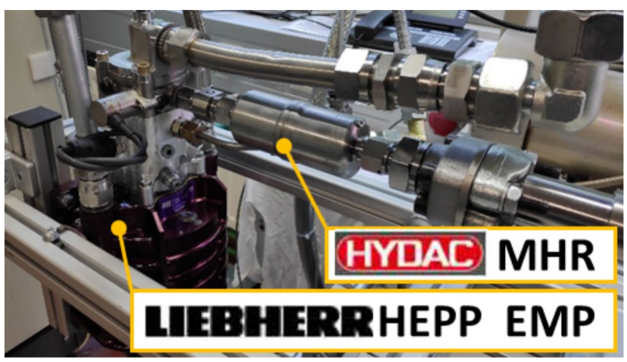

Fig. 11 Test setup at fluid borne sound test rig from TUH/FST: HEPP MPU with MHR

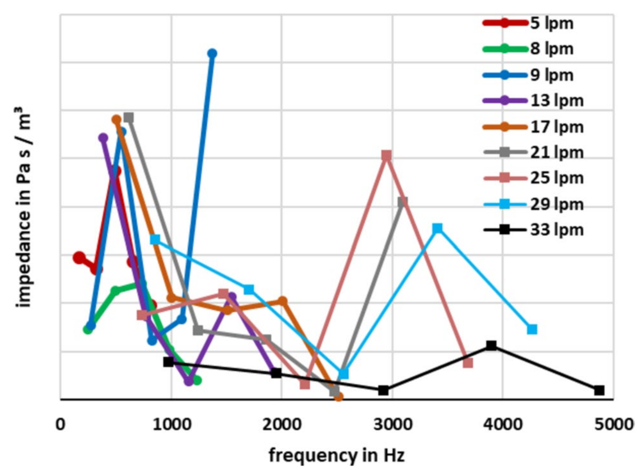

Fig. 12 Pump source flow impedance according to ISO10767-1

5 Application in aircraft hydraulics

For cost, weight and complexity reasons, a common MHR is integrated into the HEPP circuit connecting the two redundant MPUs with special flexible hoses and special check valves connected to the MHR with a T-branch and a rigid pipe, resulting in the basic architecture shown in the Fig. 14.

The final optimisation was performed regarding the length of the rigid pipe between the T-branch and the MHR to minimise the pulsating load on the pump from the back-reflecting shaft, especially at the main operating point. Length has been evaluated according to the wavelength requirements in the following chapter assuming rigid walls of the pipe.

The digital mockup (DMU) model of the redundant MPU configuration with the flexible hoses, the T-branch, the rigid pipe and the common MHR is shown in Fig. 15.

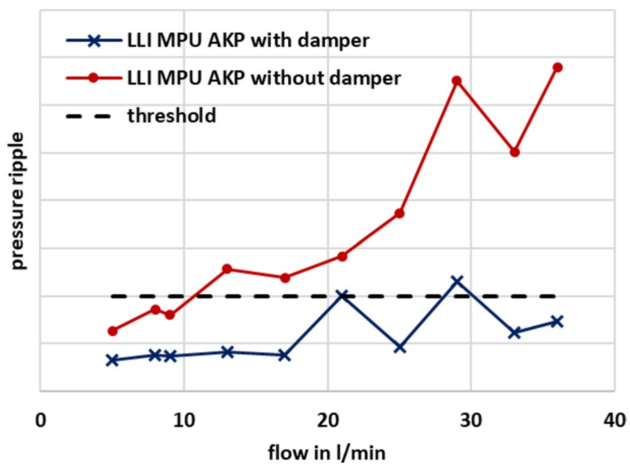

Fig. 13 Test results: pressure ripple of the LR² method with/without MHR

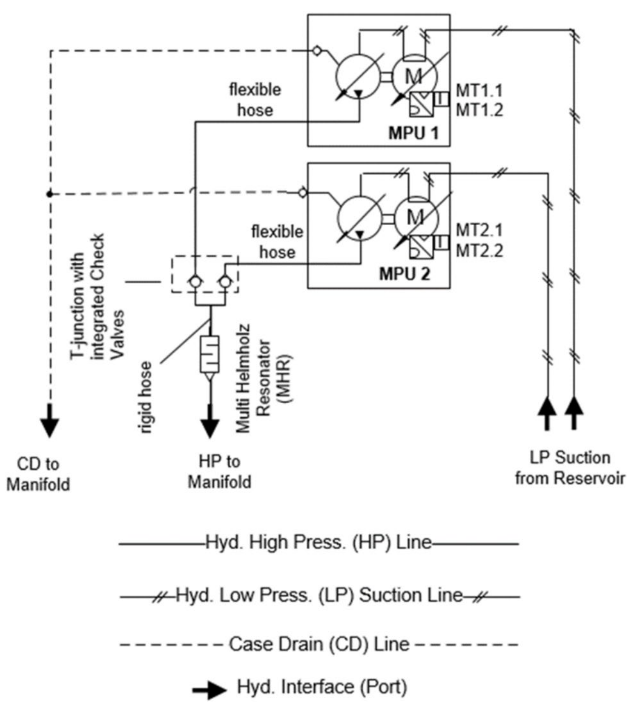

Fig. 14 Basic architecture: two MPUs with joint MHR

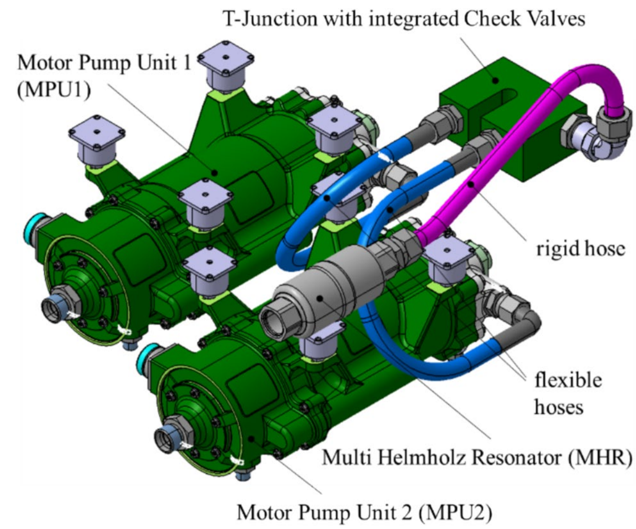

Fig. 15 Digital mockup design of two MPUs with joint attenuator (MHR)

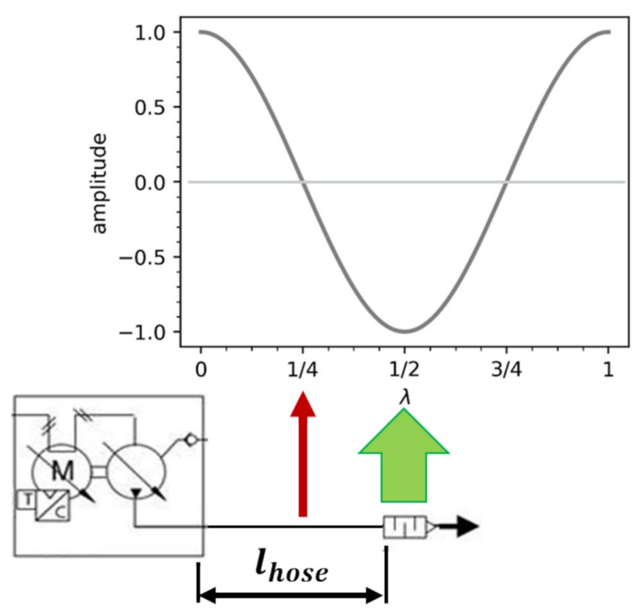

Fig. 16 Pulsation of the pressure wave in the hose: pressure amplitude over wavelength

6 Placing the attenuator: hose length determination

When adding an attenuator to the pump, the position in the hydraulic circuit is essential for the efficiency of the pump. Therefore, the relation to the wavelength must be taken into account, similar to e.g. the positioning of vibration dampers for structural vibrations [10, 11].

Assuming a maximum pressure due to the compression at the exchange piston in the pump, the wave continues to travel through the hose with different amplitudes (Fig. 16).

The ideal position of the damper would therefore be directly behind the pump or at least as close as possible to the pump.

Furthermore, the pulsation node at ¼ of the wavelength should be avoided. In expansion chambers, a drop in damping due to resonances was observed at this distance [12, 13]. The efficiency of the attenuator thus becomes negligible. This has been demonstrated for changing cross-sections, e.g. of expansion dampers.

Another position for a very good efficiency of the sound attenuator is thus ½ wavelength, where the amplitude is again maximum.

With the given package, a resonator position near the pumps could not be realised. First the use of two pumps would require two resonators, which would increase the weight and cost of the DMU. Secondly, the installation space did not allow for such a configuration. Therefore, the T-branch connects the two pumps to an MHR with two flexible and one rigid hose. The compliance of the flexible hose was taken into account by applying a reduced sound velocity. The "ideal" half wavelength thus depends on both the type of hose and the excitation frequency of the variable speed technique pump (Fig. 17).

Since the catalogue of requirements specifies a use of the pump with a nominal speed of mostly 2500 rpm, the shaft length was determined for this speed. However, since other speeds may require other lengths, hoses with adjustable lengths are not feasible at all.

The total length of the hoses is therefore between 1.07 and 1.49 m. This also includes the length of the T-junction (Fig. 18).

Finally, the length of both the flexible and rigid hose is determined and iteratively changed as part of the design process, taking into account the weight and cost of the packaging and accessibility during operation.

The interaction with the hose length has been solved by a Levenberg–Marquard least square algorithm in python. Boundaries have been implemented regarding minimum length of pipe and hose for placement.

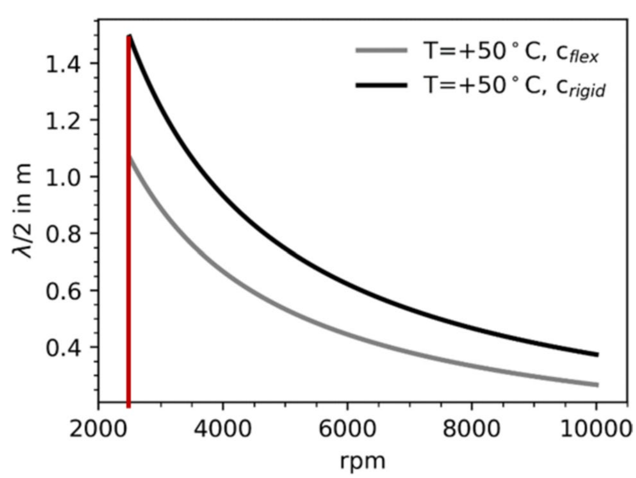

Fig. 17 Half wavelength of pressure pulsation in a flexible or rigid hose as a function of the pump delivery speed

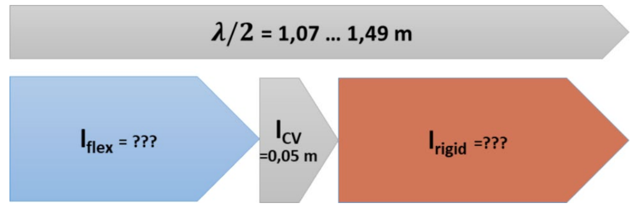

Fig. 18 Half wavelength and its distribution in the different hose length

7 Conclusions

Fluid pulsations in hydraulic pumps vary due to the rotational speed and may cause stress or wear in the hydraulic system components as well as significant cabin noise. Due to the range of excitation frequencies, a broadband attenuator for high pressure applications is realised by a multi-Helmholtz-resonator (MHR) concept within one cylindrical volume and an application specific design. The attenuator position is optimised to achieve maximal acoustic performance and minimise pulsation stress on the pump, especially at the main operating point. The proof of concept has been achieved experimentally comparing a HEPP piston pump with and without the MHR meeting the performance requirements over a wide frequency range.

Final development on system level is underway. Testing with optimisation will be conducted in the near future.

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Author Contributions M.K. was responsible for calculations, parameter studies and simulations in this project. D.M., R.E. and L.Z. were doing the package design, the integration of the silencer and the pump development in gerneral. P.K. and T.K. did transmission loss simulations, the industrial design and prototyping of the silencer. M.S. and F.T. were testing at their lab with participation of D.M., P.K. and T.K. L.K., D.M., P.K. and F.T. were supervising the project at each affliation. The manuscript has been generally edited by D.M. and M.K. based on the given input, picutres and data of all Co-authors, who all were reviewing before submission.

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Data availability No datasets were generated or analysed during the current study.

Declarations

Conflict of interest The authors declare no competing interests.

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