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REVIEW

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Research Status, Critical Technologies, and Development Trends of Hydraulic Pressure Pulsation Attenuator

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Abstract

Hydraulic pumps are a positive displacement pump whose working principle causes inherent output flow pulsation. Flow pulsation produces pressure pulsation when encountering liquid resistance. Pressure pulsation spreads in the pipeline and causes vibration, noise, damage, and even pipeline rupture and major safety accidents. With the development of airborne hydraulic systems with high pressure, power, and flow rate, the hazards of vibration and noise caused by pressure pulsation are also amplified, severely restricting the application and development of hydraulic systems. In this review paper, the mechanism, harm, and suppression method of pressure pulsation in hydraulic systems are analyzed. Then, the classification and characteristics of pulsation attenuators according to different working principles are described. Furthermore, the critical technology of simulation design, matching method with airborne piston pumps, and preliminary design method of pulsation attenuators are proposed. Finally, the development trend of pulsation attenuators is prospected. This paper provides a reference for the research and application of pressure pulsation attenuators.

Keywords: Hydraulic pump, Pressure pulsation, Pulsation attenuator

1 Introduction

Hydraulic systems have the advantages of small volume, light weight, large rigidity, fast response, large carrying capacity, and easy safety protection [1–3]. These systems widely used in the aerospace field. The hydraulic system accounts for approximately 60% of the total weight of an airborne flight control system [4]. The hydraulic system is mainly used for retracting the landing gear, the flaps, and the speed brake; controlling the flat tail, the aileron, and the rudder; and adjusting the intake cone and the auxiliary intake valve. With the continuous improvement of aircraft requirements, hydraulic systems are developing in the direction of high pressure, high power, and variable pressure.

The hazards of vibration and noise caused by pressure pulsation are amplified with the high pressure of airborne hydraulic systems, seriously affecting the safety and reliability of aircrafts. The airborne hydraulic system has strict requirements on pressure pulsation. The pressure pulsation of military aircrafts below 21 MPa is within ±10%, and the pressure pulsation of 28 MPa and above does not exceed ±15% [5]. The pressure pulsation of civil aircrafts is within ±2.5%.

Pressure pulsation attenuators are widely used in civil aircraft hydraulic systems, which can greatly reduce pressure pulsation and pipeline fatigue and improve system reliability. For example, the Airbus A380 airborne hydraulic system applies the Eaton PV3-300-31 engine driven pump, which integrates a pressure pulsation attenuator, and its pressure pulsation is within ±1% [6].

At present, the Eaton Company and Linde Company abroad can integrate a pulsation attenuator into a hydraulic pump, directly generating a hydraulic power source with small pulsation. The Parker Company also

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has composite pulsation attenuator products with a good attenuation effect and a low maintenance cost. The pulsation attenuator products at home are mostly for the aviation field and not disclosed to the public.

For the pressure pulsation attenuator, this paper summarizes and analyzes the pulsation generation mechanism, characteristics, critical technologies, and development trends and provides reference for the research and design of pulsation attenuators.

2 Generation Mechanism, Hazards, and Suppression Methods of Pressure Pulsation

2.1 Generation Mechanism of Pressure Pulsation

In a hydraulic system, pressure pulsation mainly occurs in two aspects, namely, the hydraulic pump and the actuator, control element, and load.

As the power component of the hydraulic system, the hydraulic pump mainly includes the gear, impeller, and piston pumps [7], which are parts of the positive displacement pump whose working principle is shown in Figure 1. The piston reciprocates periodically in the pump cylinder to alternately increase and decrease the working volume in the cylinder to complete oil suction and pressure. The working principle of the positive displacement pump determines the flow pulsation of its output with periodic variation [8–10]. Figure 2 shows the flow rate characteristics of the nine pistons of the piston pump [11].

Hydraulic oil has certain compressibility, like liquid spring, to store and release energy. Thus, flow pulsation causes pressure pulsation after encountering system impedance in the pipeline, pressure pulsation in turn affects flow pulsation, and the two will transform each other. When the frequency of pressure pulsation overlaps with the natural frequency of the pipeline or the fluid frequency, resonance will occur and bring serious harm [12].

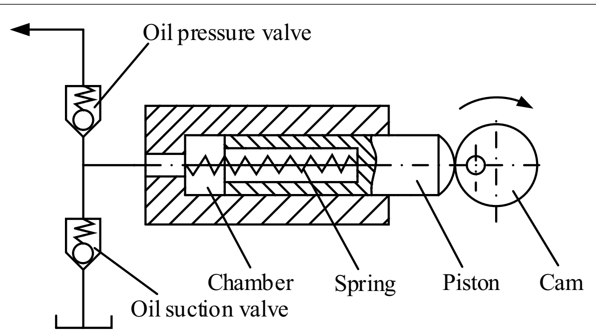

Figure 1 Working principle of positive displacement pump

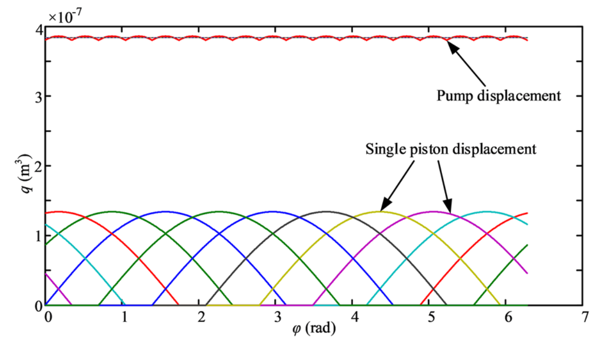

Figure 2 Flow rate characteristics of 9 pistons of the piston pump

2.2 Hazards of Pressure Pulsation

Pressure pulsation acts on the inner wall of the pipeline in the form of pulsating stress, causing mechanical vibration. If the frequency of pressure pulsation is near the natural frequency of the pipeline, then pipeline resonance will occur [13] and produce increased pressure pulsation, which not only directly causes pipeline rupture, but can also excite pipeline vibration to cause damage to the pipeline support structure and eventually make the pipeline system invalid [14, 15]. In light cases, leakage will occur, and in serious cases, explosion and other major accidents will occur. Twenty percent of civil aircraft failures are caused by hydraulic system failures [12]. In the early 1980s, a certain fighter plane broke the outlet pipe of a hydraulic pump due to fluid resonance and burned the entire plane after fire broke out. In the late 1990s, during a fighter plane's flight, the supporting structure was damaged due to fluid–solid coupling vibration, and the pipeline before oil filtration was broken, resulting in the failure of the hydraulic system. Russian SU-27 fighter planes have also experienced pipeline support structure failure and rupture accidents in the hydraulic energy system [14].

Pressure pulsation affects the service life of hydraulic components, reduces the working accuracy of actuators, affects the accuracy of parameter measurement, and increases maintenance costs. Many hydraulic components are damaged due to pressure pulsation [16].

Pressure pulsation generates noise, which affects the emotions of crew members and passengers and even brings harm to human health [17, 18]. The International Organization for Standardization proposed a noise standard that prohibits the noise in the hydraulic system from exceeding 70–80 dB [19].

The harm caused by pressure pulsation is amplified by the high pressure of the airborne hydraulic system [5, 14]. Therefore, controlling the pressure pulsation of the hydraulic system is of great significance for improving the reliability of the hydraulic system and reducing noise.

2.3 Suppression Methods of Pressure Pulsation

The suppression methods for pressure pulsation mainly include active and passive suppression methods.

2.3.1 Active Suppression Method of Pressure Pulsation

The active suppression of pressure pulsation uses active vibration eliminators to generate secondary pressure pulsation with an equal amplitude and opposite phase to the initial pressure pulsation, which are superimposed on each other to attenuate the pulsation. The active suppression methods of pressure pulsation mainly include the following.

(1) Staggered phase parallel piston pump. The oil suction and pressure ports of the two piston pumps are respectively connected and used as the oil suction and pressure ports of the parallel pump, as shown in Figure 3. Two pulsation curves of the same waveform and different phases are superimposed to reduce the pulsation amplitude and the double pump pressure pulsation. The parallel pump output flow rate is twice that of the single pump, but the pulsation rate is approximately one quarter that of the single pump [20–22].

(2) Distributed pulsation suppression method. The branch line is led out from main hydraulic pipeline 1, and an active vibration damping valve is installed, which is equivalent to an overflow valve in principle and generates overflow. Through a specific control method, the active vibration damping valve generates secondary pulsation sources with the same amplitude and opposite phase as the initial pulsation source. The two sources are superposed and offset to achieve the purpose of damping pressure pulsation, as shown in Figure 4 [23–26].

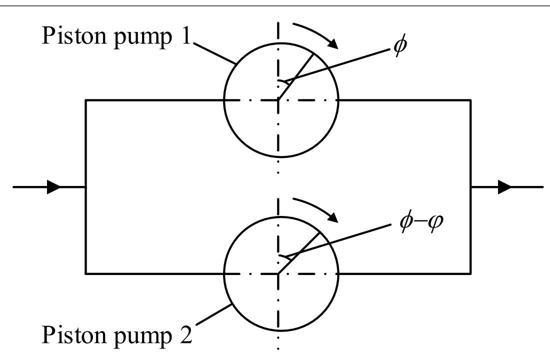

Figure 3 Principle of staggered phase parallel piston pump

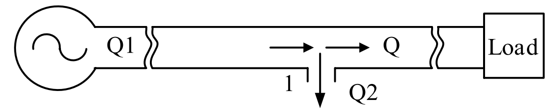

Figure 4 Principle of distributed pulsation active control

(3) Active suppression of pressure pulsation with inflow method. Through closed-loop control, the slippage pump injects a certain flow of oil into the main pipeline when the trough of the pressure pulsation wave comes to reduce the pressure pulsation of the system, as shown in Figure 5, indicating the advantage of strong adaptability [27].

In summary, the active suppression method of pressure pulsation has a strong adaptive capability, can be automatically adjusted according to pulsation characteristics, and can effectively suppress pulsation in a wider pulsation frequency band. However, the active suppression method has the following disadvantages: (1) increases the number of parts of the hydraulic system and reduces the reliability of the system, and (2) lags in the detection and analysis of signals, which affects the pulsation suppression. Therefore, the active suppression technology of pressure pulsation needs further research and improvement.

2.3.2 Passive Suppression Method of Pressure Pulsation

The passive suppression of pressure pulsation can suppress pressure pulsation from two aspects: the pulsation source and the load according to its generation mechanism.

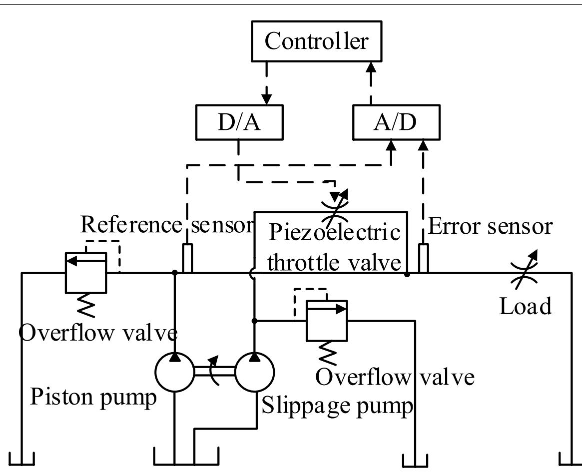

Figure 5 Principle of active suppression with inflow method

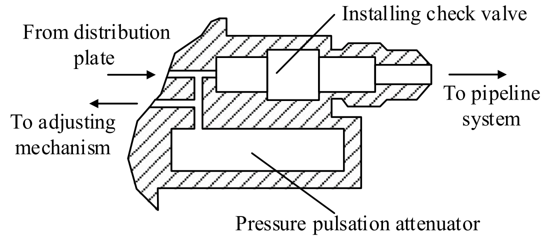

Figure 6 Partial sectional view of pressure pulsation attenuator in pump body

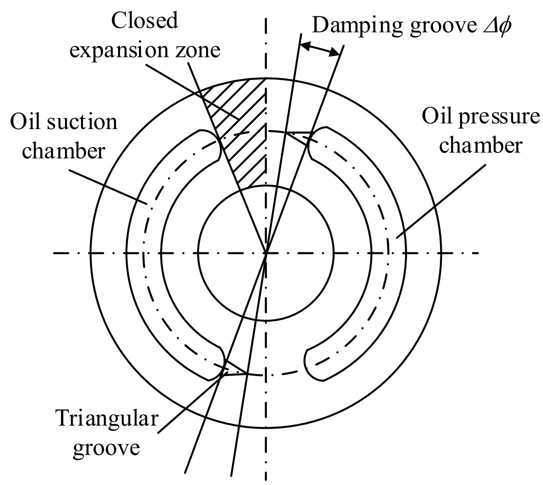

Figure 7 Plan view of distribution plate of piston pump

2.3.2.1 Suppression of Pulsation at Pulsation Source (Hydraulic Pump)

(1) Pressure pulsation attenuator. The principle of adding pressure pulsation attenuators to the pump body is the same as that of adding pressure pulsation attenuators to the pipeline [14, 28], as shown in Figure 6, which is described in detail later.
(2) Optimal size of triangular groove of distribution plate. The front end of the triangular groove on the distribution plate produces oil backflow, and the rear end produces oil impact, which are the main factors affecting the pulsation amplitude of the piston pump [29–31]. The geometric shape of the triangular groove can be optimized by establishing a mathematical model and simulation calculation to reduce the pulsation amplitude, as shown in Figure 7.
(3) Piston number of piston pump. The number and the odd and even numbers of the pistons are closely related to the pressure pulsation, and the odd number of pistons have a smaller pulsation than the adjacent even number of pistons. The larger the number of pistons is, the smaller the pulsation amplitude will be [32, 33]. However, 9 and 11 plungers are widely used in airborne piston pumps currently due to the volume and weight limitations of piston pumps.

At present, the hydraulic pump technology is relatively perfect, the pressure pulsation is difficult to reduce by improving the structure of the hydraulic pump, and the cost is relatively high.

2.3.2.2 Suppression of Pulsation at Load Reducing the pressure pulsation from the load, reducing load impedance, and increasing the attenuation and absorption of pressure pulsation have the advantages of short research and development period, economy, and reliability.

The main method for reducing system impedance is to install accumulators, hydraulic hoses, and pressure pulsation attenuators on pipelines. Accumulators can attenuate certain pressure pulsation, but their respective working frequency band and volume are narrow and large, requiring the adjustment of inflation pressure [34–37]. Hydraulic hoses can also attenuate pressure pulsation and isolate vibration, but the effect is limited. Pressure pulsation attenuators can attenuate pulsation and reduce vibration and noise and have the advantages of short development period, safety, and reliability [11, 12]. Pressure pulsation attenuators with different structures have different working frequency bands and attenuation effects and are suitable for different hydraulic systems.

2.4 Summary

Currently, design methods for efficient pressure pulsation attenuators are lacking. Furthermore, the research and development of integrated pulsation attenuators under high pressure, large flow rate, and high pulsation frequency conditions are in the exploratory stage. Therefore, studying the attenuation mechanism and design methods of pulsation attenuators is of great significance.

3 Type and Characteristics of Pressure Pulsation Attenuators

Pressure pulsation attenuators attenuate and absorb the pulsation generated by the hydraulic system according to the principle of an air silencer, which is also called a hydraulic silencer [38–40]. Active pressure pulsation attenuators are described above and in the research and development stage. Therefore, passive pressure pulsation attenuators are widely used in hydraulic systems.

Passive pressure pulsation attenuators can be divided into three categories according to their working

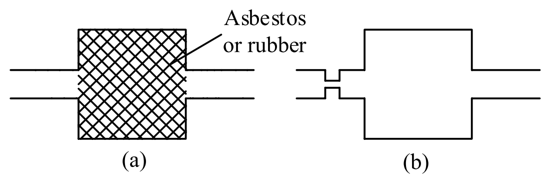

Figure 8 Resistive pulsation attenuator: **a** damping material type, **b** orifice plate type

principles: resistive, reactive, and composite pulsation attenuators [41].

3.1 Resistive Pulsation Attenuator

Resistive pulsation attenuators are also called damping pulsation attenuators. Following the characteristics of resistive silencers that use sound absorbing materials to reduce noise, resistive pulsation attenuators use similar means to reduce pressure pulsation [42]. Resistive pulsation attenuators can be divided into two types: the damping material and orifice plate types. The damping material type shown in Figure 8(a) uses damping materials with a large attenuation coefficient, such as asbestos and rubber. The orifice plate type shown in Figure 8(b) uses an orifice plate to form a large friction. When passing through the resistive pulsation attenuator, the flow and the pressure pulsation can be reduced by converting the pulsation energy into heat through friction [1]. The orifice plate type pulsation attenuator can also change the impedance characteristics of the system and avoid resonance points to realize the shock absorption effect [43]. Resistive pulsation attenuators can effectively suppress pressure pulsation, but cause a large energy loss. Meanwhile, the damping material generates serious heat and must be replaced frequently. Given these deficiencies, pure damping pulsation attenuators are rarely used in existing hydraulic systems.

3.2 Reactive Pulsation Attenuator

Hydraulic oil has a certain compressibility, and pressure pulsation has the same transmission characteristics as the acoustic wave. Thus, the hydraulic pressure wave is also called the oil acoustic wave. According to the principle of acoustic noise attenuation, reactive pulsation attenuators use various methods to make pressure waves interfere in pipelines or attenuators to reduce pressure pulsation. Therefore, reactive pulsation attenuators are also called hydraulic filters and can be divided into resonance, expansion chamber, and interference types of pulsation attenuators.

3.2.1 Resonance Type Pulsation Attenuator

The H-type pulsation attenuator is a basic resonance-type pulsation attenuator and also known as the Helmholtz pulsation attenuator. Most of the other resonance-type pulsation attenuators were developed based on the H-type pulsation attenuator.

As shown in Figure 9(a), the H-type pulsation attenuator is composed of a chamber and a neck connected with the chamber. The attenuator's structure is simple and reliable, and the pressure pulsation is efficiently attenuated in a narrow frequency band through its self-resonance action [44–46]. Given the attenuator's narrow attenuation frequency band, the natural frequency must be accurately calculated to ensure the effectiveness of the design [47, 48], but its volume is usually large, which limits the application range. Figure 9(b) shows the T-type pulsation attenuator, which is suitable for working environments with a specific working frequency, a high pulsation frequency, and a large fundamental frequency energy [11, 14].

The H-type pulsation attenuator has only one chamber and a single resonance frequency. Therefore, this attenuator has a good attenuation effect only on the pressure pulsation near the resonance frequency and has nearly no attenuation effect on the pressure pulsation at other frequencies [44]. Most hydraulic systems have high-order harmonic pulsations in addition to fundamental frequencies. Meanwhile, the rotation speed of the hydraulic pump also varies with the working conditions. Therefore, H-type pulsation attenuators with only a single resonance frequency cannot meet the requirements, and multi-degree-of-freedom H-type pulsation attenuators have emerged.

The parallel and composite H-type pulsation attenuators in Figure 10(a) and 10(b), respectively, are generally superior to H-type pulsation attenuators. Better attenuation effect can be achieved in a wider frequency band through the reasonable design of the installation location, the hole depth, and other parameters [42, 49, 50]. The series H-type pulsation attenuator in Figure 10(c) can stagger each resonance frequency by reasonably designing the size of each chamber and neck

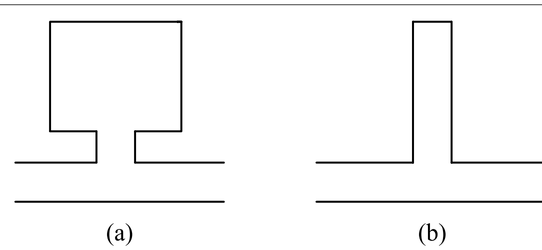

Figure 9 H-type and T-type pulsation attenuators: **a** H-type, **b** T-type

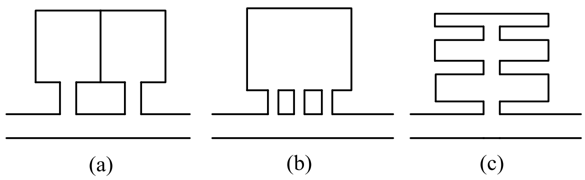

Figure 10 Multi-degree-of-freedom H-type pulsation attenuators: **a** parallel H-type, **b** composite H-type, **c** series H-type

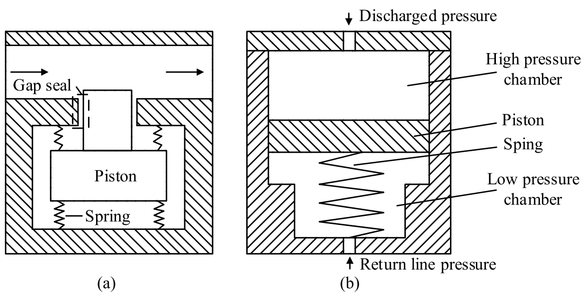

Figure 11 Pulsation attenuators based on mass-spring vibration system: **a** hydraulic spring, **b** mechanical spring

for its effective operation in multiple frequency bands, but multiple resonance frequencies are not easy to configure during the design and the processing [51, 52]. The attenuators in Figure 10 have a large volume and a small attenuation efficiency per unit volume.

The pipe and chamber walls are regarded as rigid bodies when designing and modeling resonance pulsation attenuators, but fluid-structure coupled vibration type pulsation attenuators mainly use the deformation of the structure under pressure pulsation to absorb energy. When the pulsation frequency is near the natural frequency of the structure, the structure resonates itself and has the best attenuation effect [44].

Figure 11(a) shows a pressure pulsation attenuator based on a mass-hydraulic spring vibration system. The spring supports the piston to leave a proper gap with the matching hole and balance the pressures inside and outside the chamber. By adjusting the volume of the chamber, the equivalent hydraulic spring stiffness can be adjusted to change its natural frequency and thus suitable for hydraulic systems with most levels of the pressure [53, 54]. Figure 11(b) also shows a pressure pulsation attenuator based on a mass-spring vibration system, but it uses a mechanical spring. When the parameters of the hydraulic pump change within a certain range, the pressure pulsation attenuator can be adaptively adjusted [55]. Both are compact in structure and can be integrated into a hydraulic pump.

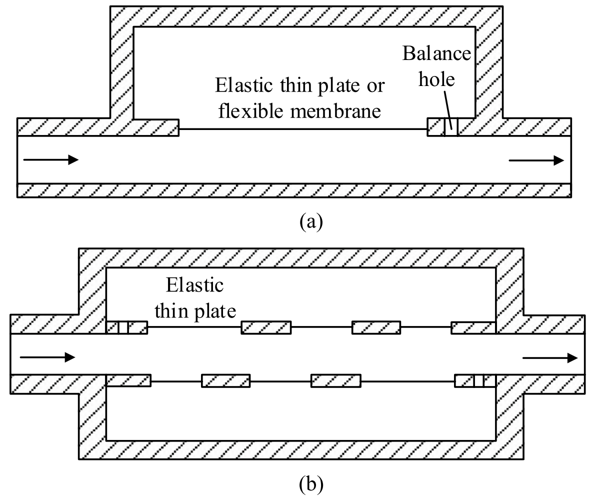

Figure 12 Thin plate vibration type pulsation attenuators: **a** Single thin plate, **b** multiple thin plates

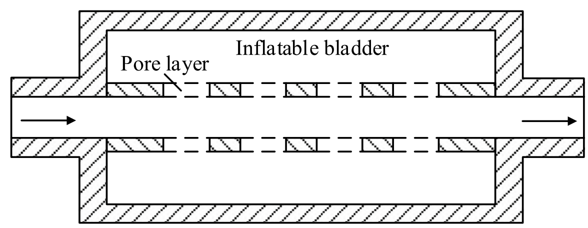

Figure 13 Inflatable bladder type pulsation attenuator

Figure 12(a) shows a thin plate vibration type pressure pulsation attenuator. The mass-spring vibration system is replaced by an elastic thin plate, so that the static balance hole and the fluid in the balance chamber form a Helmholtz resonance system, and the elastic thin plate and the fluid in the chamber form a forced vibration system, thus realizing the double filtering of structural resonance and fluid resonance, and the structure is compact [56, 57]. Figure 12(b) shows a multi-degree-of-freedom thin plate vibration type pulsation attenuator, which can achieve a better filtering effect in a wider frequency band [58].

Figure 13 shows an inflatable bladder type pulsation attenuator. The inflatable bladder has good compressibility, and its equivalent elastic modulus can be adjusted by pre-inflation pressure. The attenuation effect is good in the narrow frequency band below 1300 Hz, but the optimization of the bladder inflation pressure and the pore layer size has a great influence on the attenuation effect [59, 60].

Figure 14 shows a diaphragm vibration type pulsation attenuator with a bionic structure, which consists of a chamber, a support plate, and a bionic unit group. Elastic diaphragms with different lengths are sequentially fixed

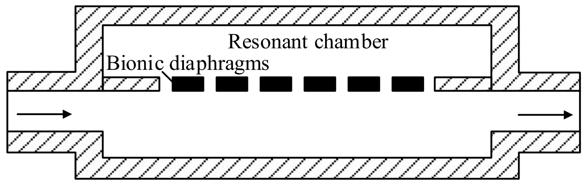

Figure 14 Bionic hydraulic pulsation attenuator

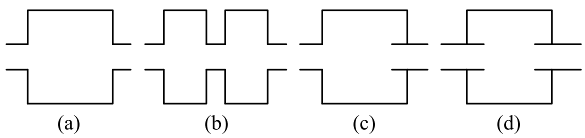

Figure 15 Several expansion chamber type pulsation attenuators: **a** C-type, **b** series type, **c** K-type, **d** insert-pipe type

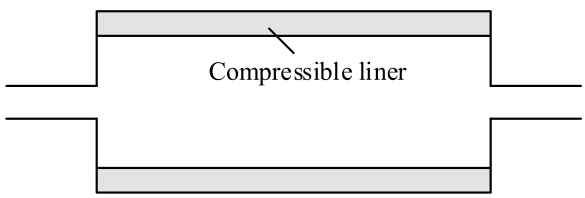

Figure 16 Expansion chamber pulsation attenuator based on compressible liner

on the support plate to simulate the "space-frequency domain" dynamic characteristics of the cochlear basilar membrane and realize the attenuation of pressure pulsation in multiple frequency bands [61].

3.2.2 Expansion Chamber Type Pulsation Attenuator

Figure 15 presents single expansion chamber type (also called C-type), series type, K-type, and insert-pipe type expansion chamber type pulsation attenuators. When hydraulic oil with pulsation characteristics enters the chamber, the fluid suddenly expands and then compresses due to the action of the sudden expansion pipe and the compressibility of hydraulic oil. This process releases the pulsation energy of the fluid and reduces the pressure pulsation [11]. The working principle of this type of attenuator indicates that its attenuation to low-frequency pulsation is not obvious, but it has a good attenuation effect to high-frequency pulsation [6] and is small and easy to install. K-type and other pulsation attenuators with an insert-pipe have a better attenuation effect than the other attenuator types [62, 63].

Figure 16 shows an expansion chamber pulsation attenuator based on the compressible liner approach, which increases the attenuation effect by adding a compressible liner (such as polyurethane) to the chamber wall to increase the capacitance of the attenuator. Under the same pulsation attenuation effect, adding a compressible liner can reduce the overall size and noise [64, 65]. The attenuation frequency band range can be expanded by changing the material composition of the compressible liner without changing the external dimension of the attenuator [66].

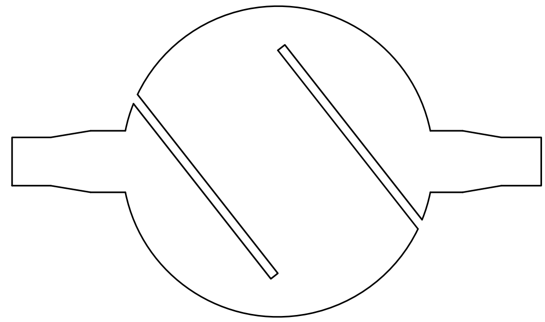

Figure 17 Expansion chamber pulsation attenuator with baffle plates

Figure 17 shows an expansion chamber pulsation attenuator with baffle plates. In addition to using resonance and the interference of waves to reduce pulsation, vortex is formed by the combined action of baffle plates and the chamber to further attenuate pulsation. The optimal attenuation effect and frequency bandwidth can be achieved by optimizing the length, angle, and spacing of baffle plates [67].

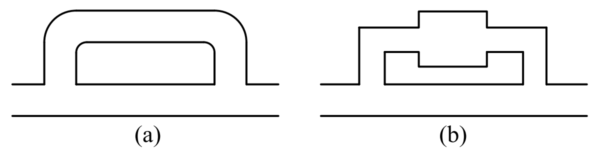

Figure 18 Interference type pulsation attenuators: **a** original type, **b** improved type

3.2.3 Interference Type Pulsation Attenuator

Figure 18(a) shows an interference type pulsation attenuator (also called a shunt tube filter, a Quincke tube, or a Herschel–Quincke tube, i.e., a HQ tube for short), which uses the principle of wave interference and superposition to achieve the purpose of pulsation attenuation [68, 69], but its volume is large and its working frequency band is narrow. The improved HQ tube shown in Figure 18(b) can significantly shorten the length of the parallel tube, and the vibration of each tube decreases due to the addition of a C-type pulsation attenuator in the parallel tube [43].

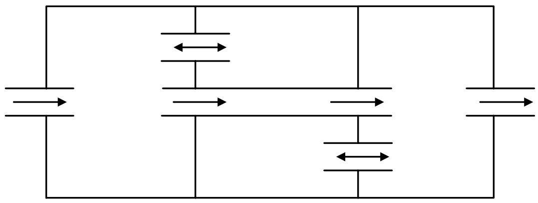

Figure 19 Pulsco pulsation attenuator

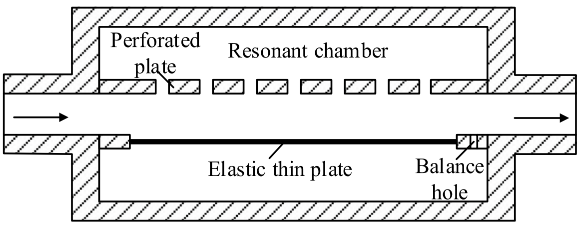

Figure 20 Composite perforated plate pulsation attenuator

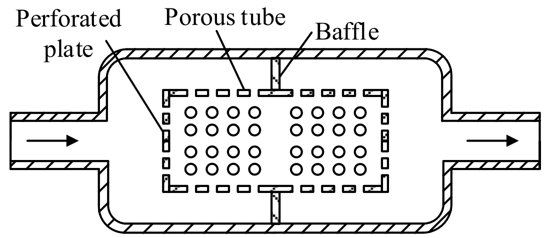

Figure 21 Composite porous tube pulsation attenuator

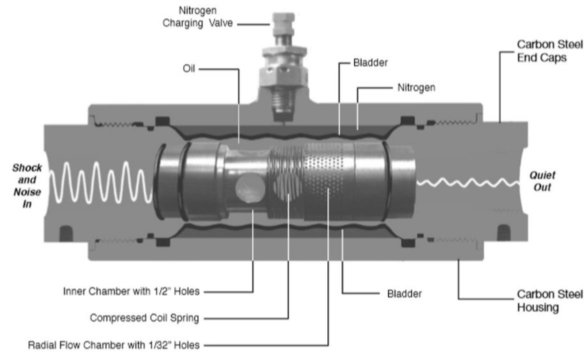

Figure 22 Pulse-Tone pulsation attenuator

Figure 19 shows a Pulsco pulsation attenuator, which includes three chambers and three short pipes and is connected in series with the hydraulic pipeline. Both interference and resonance filtering are used to attenuate the pulsation. The working frequency band is wide, but the volume is large, and it is mainly used to attenuate the pressure pulsation generated by the hydraulic pump [43].

Given their special structure and filtering effect, the two interference type pulsation attenuators above are seldom applied in engineering practice and need further study.

3.3 Composite Pulsation Attenuator

Figure 20 shows a composite perforated plate pulsation attenuator, which consists of a balance chamber, a perforated plate, an elastic thin plate, and a fixed plate. The perforated plate and the chamber behind the plate form a parallel Helmholtz resonance system. When the pressure pulsation flows through, the fluid in the small hole reciprocates like a piston to convert pulsation energy into heat energy through damping and friction. The pulsation attenuation effect on the medium and high frequency bands is good. In addition, the elastic thin plate and the chamber behind the plate have the same principle as the thin plate vibration type pulsation attenuator mentioned above, which has a good attenuation effect on the low frequency band. Therefore, the combination of the two can realize the effective pulsation attenuation of the wide frequency band. [70, 71].

Figure 21 shows a composite porous tube pulsation attenuator, which consists of a chamber, a baffle, and a porous tube. This attenuator combines the basic principles of resistive and reactive pulsation attenuators and attenuates pulsations through friction and resonance. The composite pulsation attenuator has a wide frequency band, a good attenuation effect, and a compact structure and is convenient for practical engineering applications [72].

Figure 22 shows the Pulse-Tone pulsation attenuator. The chamber has three different baffles or diffusers. Oil impacts the rubber bladder after passing through small holes, and the bladder will deflect after being hit by pulsation to reduce vibration and noise. The chamber's large area and ability to oscillate at a high frequency result in high attenuation efficiency under high pressure and rotation speed.

3.4 Summary

The working frequency, attenuation mechanism, advantages, and disadvantages of various pressure pulsation attenuators are shown in Table 1.

4 Critical Technology of Pressure Pulsation Attenuator

The requirements for pressure pulsation are continuously increasing with the pressure and safety requirements of airborne hydraulic systems. The development of high-performance pulsation attenuators is one of the popular topics in the research on airborne hydraulic systems.

Table 1 Characteristics of various pressure pulsation attenuators

Type	Working frequency	Attenuation mechanism	Advantages	Disadvantages
Resistive	Unlimited	Damping and friction	Obvious attenuation effect	Serious fever and large energy loss
Reactive	Narrow	Resonance and interference of pressure pulsation wave	Stable work	Larger volume
Composite	Wide	Friction and resonance	Wide working frequency band, good effect	Large volume and difficult design

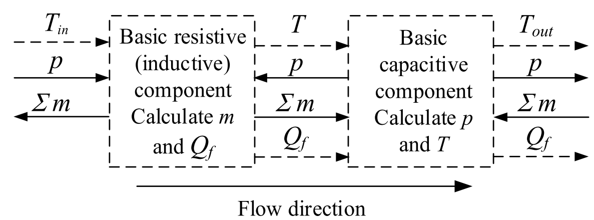

Figure 23 Connection rules of hydraulic component models

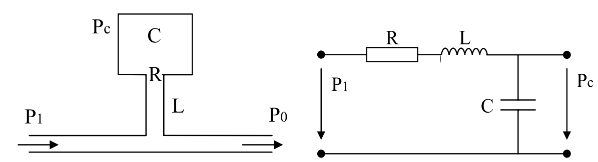

Figure 24 Electrical schematic diagram of H-type pulsation attenuator

Breaking through the critical technology of pressure pulsation attenuators is the basis of developing small-volume, light-weight, and high-performance pulsation attenuators.

4.1 Simulation Method of Pressure Pulsation Attenuator

The simulation of pressure pulsation attenuators is divided into one- and three-dimensional simulations.

4.1.1 One-Dimensional Simulation of Pressure Fluctuation Attenuator

4.1.1.1 Mathematical Modeling The hydraulic system has many kinds of hydraulic components, with capacitive, resistive, or/and inductive characteristics. To meet the one-dimensional modeling requirements of most hydraulic components, the centralized parameter method is adopted to define three basic hydraulic components, namely, the basic capacitive, resistive, and inductive components. Then, the three basic components are combined and connected to establish various complex hydraulic components. The input and output rules and the connection rules of the three basic components are shown in Figure 23 [73], where T is the temperature, p is the pressure, Σm is the mass flow, and Q_f is the heat generation rate due to pressure drop.

Hydraulic pipelines show capacitive, resistive, and inductive characteristics due to fluid compressibility, fluid viscosity, and fluid inertia, respectively. These characteristics are very similar to those of electrical elements, such as capacitance, resistance, and inductance. In most cases, one-dimensional theoretical analysis and the numerical calculation of hydraulic pipelines can be directly carried out by using electrical theoretical formulas. This effectively combines fluid mechanics and electrical transmission theory and is widely used in the analysis of information transmission characteristics, such as flow rate, pressure, and power in hydraulic pipelines. Therefore, the initial expression of fluid network theory is obtained by the electric-hydraulic analogy method [74].

By using the electric-hydraulic analogy method, some hydraulic components, such as pulsation attenuators, can be directly analyzed and calculated theoretically. According to the similarity theory of fluid network, the H-type attenuator is equivalent to an inductance-capacitance series circuit. This attenuator's structural principle is shown in Figure 24.

In practical application, fluid network theory extends modeling methods, such as the wave, frequency, and characteristic line methods, according to different engineering requirements and emphases [74].

4.1.1.2 Commercial Software AMESim was originally developed by IMAGINE, France as an engineering simulation software, which can realize engineering modeling and simulation in mechanical, hydraulic, and other multidisciplinary fields. AMESim's solver adopts an algorithm adaptive integrator and automatically judges and processes discontinuous points. The software can automatically select the best integration algorithm according to the dynamic characteristics of the system. AMESim is one of the most widely used hydraulic system simulation software [73, 75, 76].

Using the AMESim software, a one-dimensional model of the nine-piston pump is established, as shown in Figure 25. The pulsation attenuator and the load are simulated by the chamber and the orifice components, respectively. Different chamber volumes are set at the

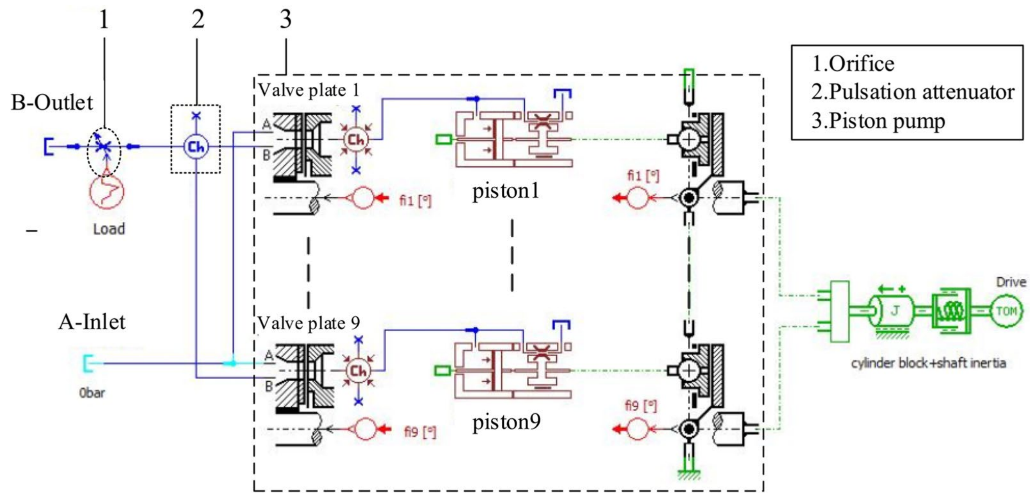

Figure 25 One-dimensional simulation model of nine-piston pump

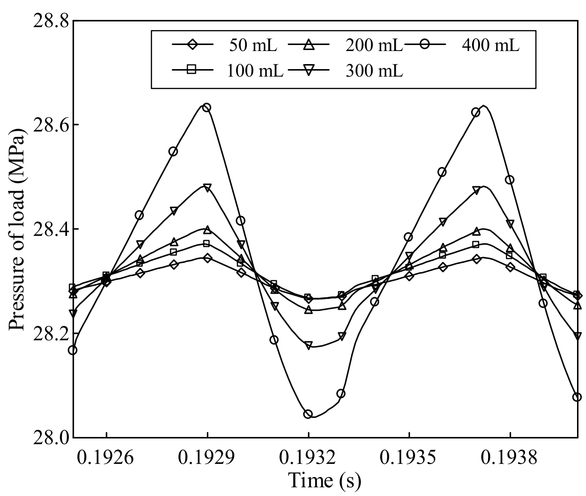

Figure 26 Pulsation characteristics of pump outlet under different volumes of chamber

pump outlet, and pulsation characteristic curves of the different volumes are obtained, as shown in Figure 26 [11].

Xu et al. designed a single-chamber insert-pipe water silencer and compared the calculation results of the one-dimensional theoretical model and the Sysnoise software, as shown in Figure 27(a). The reliability of the one-dimensional calculation model was verified [77, 78]. Zhang et al. established and simulated a one-dimensional acoustic model through the GT-POWER software and obtained the transmission loss (TL) of the expansion chamber type pulsation attenuator, as shown in Figure 27(b) [79]. Broatch et al. used a one-dimensional time domain method to analyze the frequency domain response of the pulsation attenuator and obtained different numerical transmission losses, as shown in Figure 28(c), providing a solution for one-dimensional unsteady flow [80].

One-dimensional simulation can accurately reveal the influence of the attenuator volume size on pressure pulsation characteristics, but cannot simulate the effect of shape and internal structure on pulsation attenuation. Therefore, three-dimensional numerical simulation is adopted to improve the simulation accuracy of pulsation attenuator.

4.1.2 Three-dimensional Simulation of Pressure Fluctuation Attenuator

Three-dimensional simulation is a computational fluid dynamics (CFD) analysis [81, 82]. On the basis of the finite element and volume methods, Fluent, CFX, STAR-CCM+, and other computational fluid dynamics software are applied to obtain the required physical quantities, and then transmission loss is calculated. First, a three-dimensional geometric model is established and meshed. Then, boundary conditions are set in the CFD software, and the solution model is selected. Finally, iterative calculation is carried out. After completing the calculation, the required data, such as contour and vector, are obtained through post-processing to calculate and evaluate the attenuation effect [6].

Mehdizadeh analyzed the transmission loss of two kinds of expansion chamber pulsation attenuators through the three-dimensional finite element method, which produced consistent results with the experimental results; the simulation accuracy can be improved by

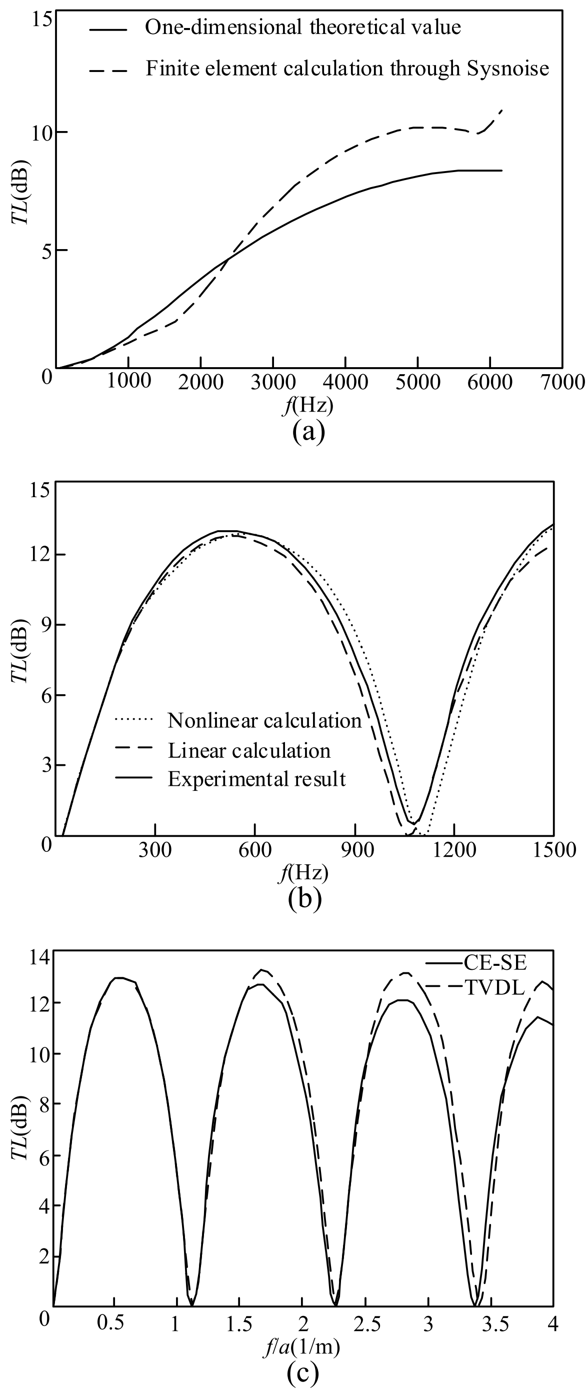

Figure 27 Results of one-dimensional simulation: **a** comparison of TL between calculated value and simulation result of insert-pipe water muffler, **b** comparison of simulation results and measured values of expansion chamber pulsation attenuator, **c** comparison of TL of different numerical schemes of expansion chamber pulsation attenuator

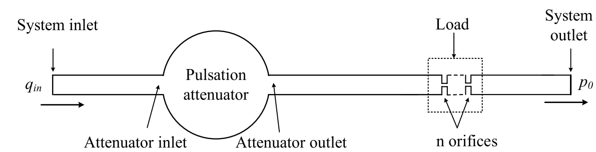

Figure 28 Three-dimensional simulation modeling strategy

improving the mathematical model [83, 84]. Torregrosa et al. simulated the pulsation attenuators of rectangular and elliptical chambers with the three-dimensional finite volume method and verified the effectiveness of the method through experiments [85]. Zhang et al. calculated the transmission loss through the coupling model based on the boundary and finite element methods and analyzed the influence of the length and diameter of the chamber on the attenuation effect [86]. Shan et al. obtained the inlet and outlet pressure pulsation curves of the pulsation attenuator through the CFD analysis method and compared them with the results obtained by the spectrum analysis method, verifying the feasibility of the CFD analysis method [87, 88]. Shen used the Fluent software to simulate the flow field of grid pulsation attenuator and optimized the gap width and number parameters [89]. Tian et al. used the CFX software to establish five muffler models, simulated their steady flow fields, and obtained the resistance loss of muffler under different flow rates [90]. Ma et al. proposed a design scheme of pulsation attenuator and verified its feasibility and accuracy by three-dimensional simulation [91].

Yin used n ($n \geq 2$) fixed orifices to simulate the load and proposed a three-dimensional modeling strategy for the pulsation attenuator, as shown in Figure 28 [11].

Solidworks was used to establish the pulsation attenuator and load model. Then, Hypermesh was used for meshing. Finally, CFX was used for simulation pre-processing, calculation, and post-processing. The pressure pulsation curve of the main monitoring points is shown in Figure 29 [11].

In general, three-dimensional simulation has higher accuracy and precision than one-dimensional simulation. Three-dimensional simulation can conveniently consider the nonlinear dissipation between the flow and sound fields [6], providing better technical support for the research on pulsation attenuators. However, three-dimensional simulation requires further improvement. For example, some of the current simulations do not consider the viscosity-temperature characteristics of oil, while some do not consider the influence of load on the result. The current research does not simulate the complex pulsation attenuator, and most studies are qualitative. Theoretical research and simulation methods

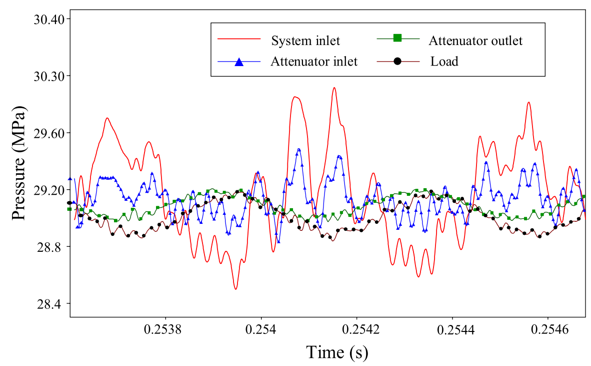

Figure 29 Pressure pulsation curve of the main monitoring points

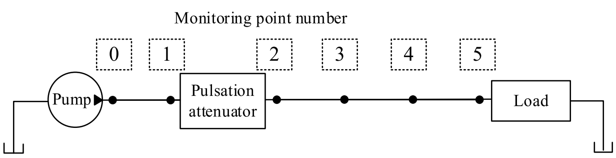

Figure 30 Schematic diagram of pulsation attenuator modeling

require improvement according to the comparison between the simulation and experimental results.

4.2 Influencing Factors of Pressure Pulsation Attenuator

The control variable method is usually used to study the influencing factors of pressure pulsation attenuators. CFD simulation is carried out for different values of a certain factor, and the above-mentioned appropriate evaluation methods are selected for comparative study to determine the relationship between this factor and the pulsation attenuation effect. Through the multi-factor study of pressure pulsation attenuators, the primary and secondary influences of each factor on the pulsation attenuation effect are analyzed. Common factors include the volume and geometric shape of the chamber, the inlet and outlet diameters, and the distribution angle.

Yin analyzed the effects of the volume and shape of the chamber on the pulsation attenuation effect through the CFX software. The modeling diagram is shown in Figure 30 [11], where monitoring point 0 is the outlet of the hydraulic pump, monitoring point 1 is the inlet of the pulsation attenuator, monitoring point 2 is the initial section of the outlet of the pulsation attenuator, monitoring point 3 is the middle section of the outlet of the pulsation attenuator, monitoring point 4 is the end section of the outlet of the pulsation attenuator, and monitoring point 5 is the load end. The pulsation amplitude of each monitoring point under different cavity volumes is shown in Figure 31(a); the pulsation amplitude of each monitoring point under different cavity shapes is shown in Figure 31(b); the pulsation amplitude of each monitoring point under the different inlet and outlet diameters of the sphere attenuators is shown in Figure 31(c); and the pulsation amplitude of each monitoring point under the different inlet and outlet distribution angles of the sphere attenuators is shown in Figure 31(d).

Selamet et al. used the numerical simulation method to study that effect of length on the attenuation performance of the cylindrical chamber pulsation attenuator under constant diameter [92]. Venkatesham et al. used Green's function analysis method to study the effect of the inlet and outlet positions on pulsation attenuation performance [93]. Nag et al. used the finite element method to simulate a cylindrical pulsation attenuator in the low frequency band, and their analysis shows that increasing the diameter of the chamber could enhance the attenuation effect [94]. Jang et al. used the acoustical and flow topology optimization methods to obtain the best baffle layout in the chamber under different transmission losses [95].

4.3 Matching Method of Pressure Pulsation Attenuator and Constant Pressure Variable Displacement Pump

4.3.1 Flow Rate Matching

Impedance is the main factor affecting the average output flow, while the cross-sectional area is the main factor affecting impedance. For most pulsation attenuators, the cross-sectional area in the chamber is generally larger than that of the connecting pipes on both sides, so the chamber has no effect on the average flow rate of the system in most cases, and the flow rate is only affected when the diameter of the inlet and outlet pipes is extremely small. Therefore, the diameters of the inlet and outlet pipes can be designed according to the flow-pipe diameter requirements of the hydraulic system when matching the flow rate.

4.3.2 Pulsation Amplitude Matching

When the pressure and flow rate of the constant pressure variable displacement pump change, the load and impedance of the system hardly change, so the change trends of the output pressure and flow pulsation amplitudes are consistent and approximately synchronous. Therefore, only one of the two pulsation matchings must be performed well. Given that the main evaluation indexes of the pulsation attenuator are all based on the pressure pulsation amplitude, the pulsation matching between the pulsation attenuator and the piston pump is mainly aimed at the pressure pulsation amplitude matching. The flow- and pulsation-pressure characteristics of constant pressure variable displacement pump shown in Figure 32 indicate that the maximum pulsation amplitude is obtained at point b regardless of the flow or pressure

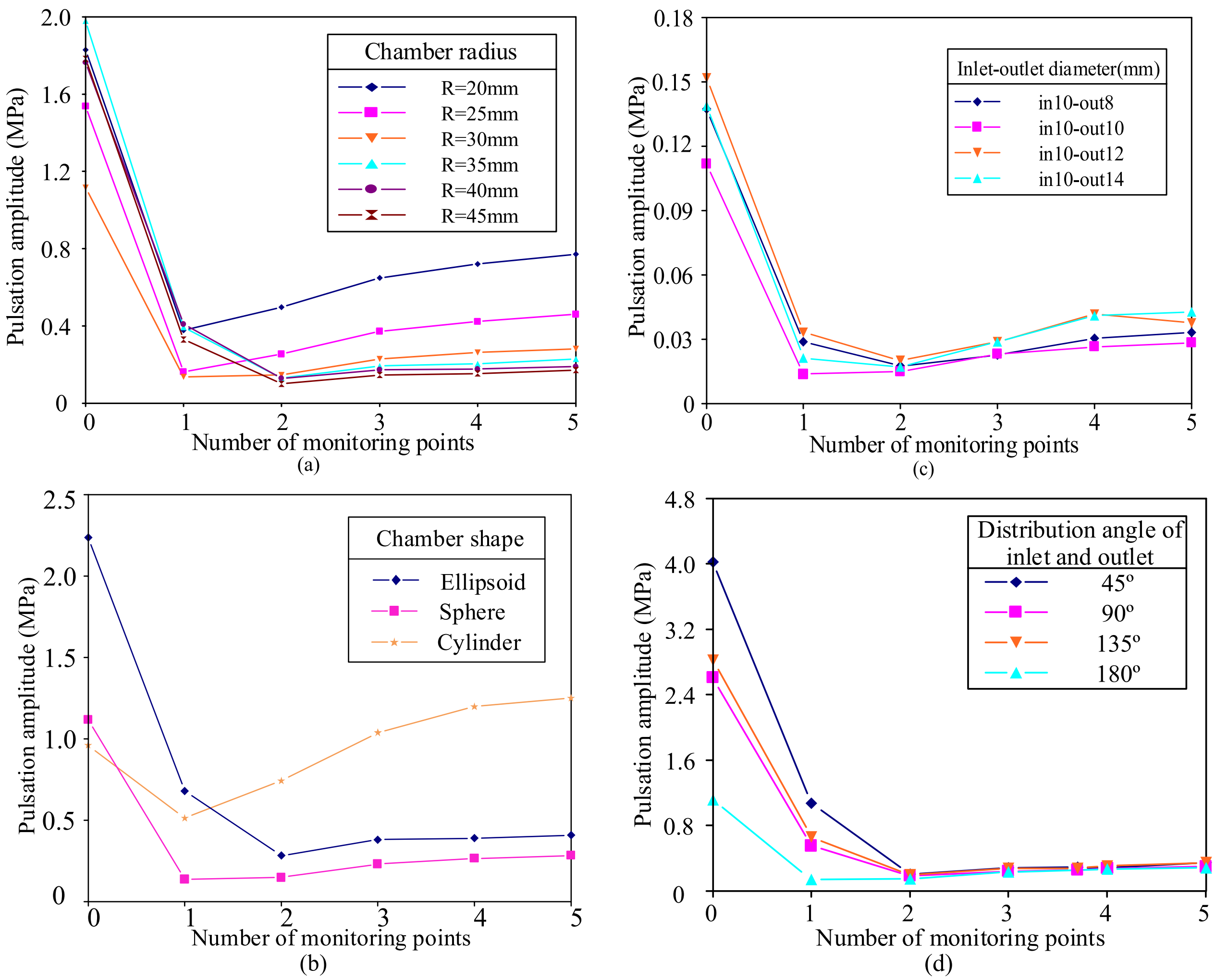

Figure 31 Effects of different factors on pulsation attenuator: **a** chamber radius, **b** chamber shape, **c** inlet-outlet diameter, **d** distribution angle of inlet and outlet

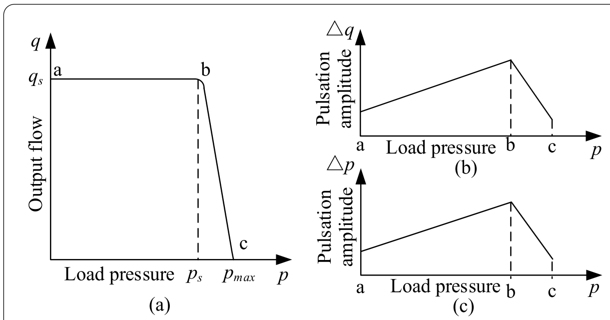

Figure 32 Flow-pressure and pulsation-pressure characteristics of constant pressure variable displacement pump: **a** flow-pressure characteristics, **b** flow pulsation-pressure characteristics, **c** pressure pulsation-pressure characteristics

pulsation, so the pulsation attenuator can complete the matching of the point b working conditions.

4.3.3 Pulsation Frequency Matching

The pulsation frequency of the constant pressure variable displacement pump is mainly determined by rotation speed n. The characteristics of the average pressure hardly change with the increase of the rotation speed, while the average flow rate gradually increases.

As shown in Figure 33, the flow-pressure characteristic curves at different rotation speeds are compared in the same coordinate system to find the "dangerous points" at each rotation speed. These points are connected to generate a line, and the pulsation amplitude of each point on the line is matched to complete the matching of the entire pulsation frequency. To clearly show the relationship between points 1, 2, 3, and 4 in Figure 33, the

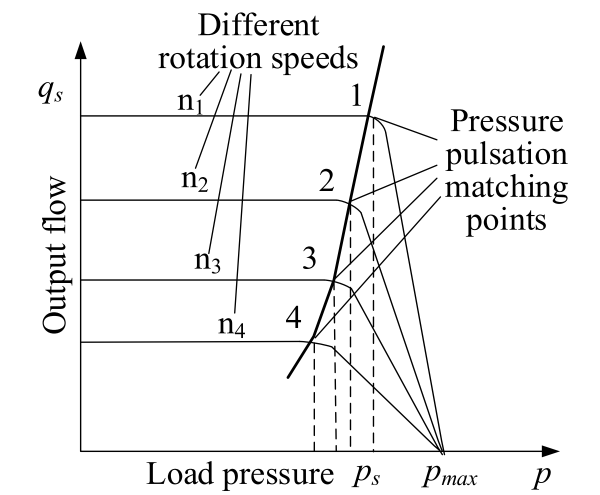

Figure 33 Flow-pressure characteristics at different rotation speeds

difference of the four points on the abscissa is enlarged. In fact, the pressure difference of these points is generally 0.1–0.5 MPa at different rotation speeds.

4.4 Design Method of Pressure Pulsation Attenuator

The general design principle of pressure pulsation attenuators is to optimize their parameters under the conditions of satisfying the volume constraint, the pressure loss limitation, and the actual installation space for the pressure pulsation to reach the maximum attenuation in the required frequency band [96]. The design flow of pressure pulsation attenuators is shown in Figure 34.

4.5 Optimization Method of Pressure Pulsation Attenuator

The optimization of pressure pulsation attenuators can be divided into research method and structure optimizations. Xuan et al. calculated the attenuation characteristics of the H-type pulsation attenuator by using the three-dimensional finite volume and improved pulse methods in the time domain, which produced results that are consistent with the numerical simulation results [97]. Shen optimized the parameters of a pulsation attenuator composed of an accumulator and a Helmholtz attenuator, such as chamber volume, inflation pressure, and damping holes, such that the pressure pulsation was attenuated by more than 5 dB [98]. Chu et al. proposed an external inserted pulsation attenuator, and their numerical simulation results show that the attenuation effect is better than that of the H-type pulsation attenuator with the same volume [99]. Shan proposed adding a composite damping layer on the inner wall of the chamber to improve the attenuation effect and proved that the composite damping layer meets the requirements in strength through boundary layer theory [96].

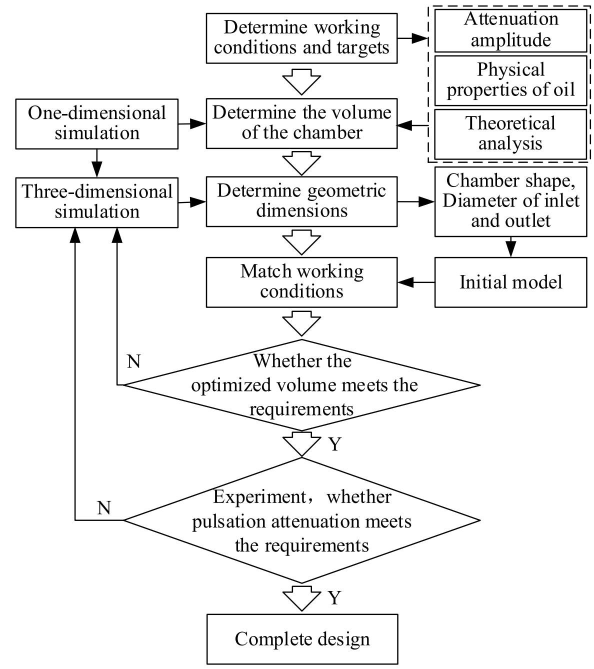

Figure 34 Design flow of pressure pulsation attenuator

5 Development Trend of Airborne Pressure Pulsation Attenuator

With the continuous development of airborne hydraulic systems, the research and design of pressure pulsation attenuators must also keep pace with the times. On the basis of the research results on pulsation attenuators at home and abroad, the following forecast on its development trend is made.

(1) Research and development of broadband pressure pulsation attenuator.

Aviation piston pump is the "heart" of the hydraulic system, which converts mechanical energy output by aviation engine, motor, or ram air turbine into hydraulic energy to provide energy for the flight control system, landing gear retraction, and so on. Under different flight conditions, the rotation speeds of the engine, motor, or ram air turbine vary, causing the pressure pulsation frequency of the aviation piston pump to change.

For the pressure pulsation of the airborne hydraulic system to meet the requirements at different rotation speeds of the aviation piston pump, a broadband pressure pulsation attenuator must be developed.

(2) Integrated design of piston pump and pressure pulsation attenuator.

The aviation piston pump is one of the main pulsation sources of airborne hydraulic systems. The integration of a pulsation attenuator on the aviation piston pump to reduce pressure pulsation from the source is an efficient method of suppressing pulsation. The integrated piston pump and pressure pulsation attenuator design is the key technology.

The integrated aviation pump and pressure pulsation attenuator design includes the inlet and outlet installation position, shape, number, and angle and the optimal matching of the pulsation attenuator.

(3) Simulation and verification method of pressure pulsation attenuator.

In the numerical simulation of pulsation attenuators with the CFD software, hydraulic oil is mostly considered as incompressible fluid, and the influence of temperature is ignored as well as the oil viscosity-temperature and viscosity-pressure characteristics [100]. In addition, the quality of mesh and the choice of turbulence model also have a great influence on the simulation results.

Accurate modeling, simulation, and verification methods for pressure pulsation attenuators are the basis for their accurate design and will be the focus of future research.

(4) Intelligent pressure pulsation attenuator.

Chamber volume has a great influence on pulsation attenuation. With the emergence of new materials, controllers, and algorithms, the design of a pressure pulsation attenuator that can adapt to the changing pulsation amplitude and frequency by changing the chamber volume is the future research direction.

The volume of the chamber can be actively changed by a new type of actuator or adaptively changed by intelligent materials.

6 Conclusions

Flow pulsation is the essential characteristic of a positive displacement hydraulic pump. Pressure pulsation occurs when flow pulsation encounters liquid resistance. Excessive pressure pulsation brings great challenges to the reliability and flight safety of aircrafts. The efficient and reliable reduction of the pressure pulsation of airborne hydraulic systems is currently a popular and difficult research topic.

This paper introduces the generation mechanism of pressure pulsation in hydraulic systems, compares the advantages and disadvantages of active and passive pressure pulsation suppression methods, analyzes the pulsation attenuation mechanism and characteristics of different types of pressure pulsation attenuators, proposes critical technologies of pressure pulsation attenuators, and looks forward to its future research emphasis, providing reference for the research of pressure pulsation attenuators.

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Authors' Contributions
YW, JF and SG were in charge of the whole trial; TS wrote the manuscript; CT assisted with sampling and laboratory analyses. All authors read and approved the final manuscript.

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Competing Interests
The authors declare no competing financial interests.

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References

[1] Z L Wang, B Chen, L H Qiu. Trends of future aircraft hydraulic system. *Hydraulics Pneumatics & Seals*, 2000, 1: 14-18. (in Chinese)
[2] J H Brahney. Hydraulic pumps: The key to power generation. *Aerospace Engineering*, 1991, 12: 9-13.
[3] J H Brahney. Will hydraulic systems meet tomorrow's aircraft power requirement. *Aerospace Engineering*, 1991, 11: 9-13.
[4] T L Chang. *Hydraulic control systems*. Beijing: Tsinghua University Press, 2014. (in Chinese)
[5] P G Xiu, J W He, L Hong. Study on 28 MPa hydraulic system pressure ripple of aircraft. *Aeronautic Standardization & Quality*, 2009, 6: 13-14. (in Chinese).
[6] Y Wang, F Q Hao, S R Guo. Research status and development trend of the expansion chamber pressure pulsation attenuator. *Machine Tool & Hydraulics*, 2015, 43(15): 180-186. (in Chinese)
[7] J A Zhao, Y L Fu, J M Ma, et al. Review of the cylinder block/valve plate interface in axial piston pumps: Theoretical models, experimental

investigations, and optimal design. *Chinese Journal of Aeronautics*, 2021, 34(1): 111-134.
[8] R J Qi. Mechanism research of pressure ripple for hydraulic system. *Journal of Tongji University*, 2001, 29(9): 1017-1022. (in Chinese)
[9] X G Zhang, L Quan, Y Yang, et al. Output characteristics of a series three-port axial piston pump. *Chinese Journal of Mechanical Engineering*, 2012, 25(3): 498-505.
[10] L Li, Y Y Chen. Working principle and design of axial piston pump. *The Fifth International Symposium on Fluid Power Transmission and Control (ISFP'2007)*, Qinhuangdao, China, June 6, 2007: 294-296.
[11] Y F Yin. *The attenuation mechanism and design method of pressure pulsation attenuator*. Beijing: Beihang University, 2018. (in Chinese)
[12] Y L Fu, H Q Jing. The development of fluctuation attenuation technology for aviation hydraulic system. *Chinese Hydraulics & Pneumatics*, 2012, 2: 3-7. (in Chinese)
[13] B Xu, Q K Feng, X L Yu. Study on pressure pulsation and piping vibration of complex piping of reciprocating compressor. *Nuclear Power Engineering*, 2008, 29(4): 79-83. (in Chinese)
[14] Z L Wang. *Aircraft high pressure hydraulic energy systems*. Beijing: Beihang University Press, 2004. (in Chinese)
[15] Y L Dou, T Zhang. Research on fluid pulsation attenuator used in the ship pipework. *Chinese Journal of Ship Research*, 2008, 3(4): 40-44. (in Chinese)
[16] H Ortwig. Experimental and analytical vibration analysis in fluid power systems. *International Journal of Solids and Structures*, 2005, 42(21-22): 5821-5830.
[17] E Kojima, M Shinada. Characteristics of fluidborne noise generated by a fluid power pump. *Bulletin of JSME*, 1986, 29(258): 4147-4155.
[18] E Kojima, T Ichiyanagi. Research on pulsation attenuation characteristics of silencers in practical fluid power systems. *International Journal of Fluid Power*, 2000, 1(2): 29-38.
[19] C Su. Noise control of hydraulic system. *Chinese Hydraulics & Pneumatics*, 2012, 2: 39-40. (in Chinese)
[20] L J Yu, X D Wang, X C Zhang. Control and simulation of flow pulsation in axial plunger pump. *Journal of Xi'an Jiaotong University*, 2013, 47(11): 43-47. (in Chinese)
[21] J M Bergada, S Kumar, D L Davies, et al. A complete analysis of axial piston pump leakage and output flow ripples. *Applied Mathematical Modelling*, 2012, 36(4): 1731-1751.
[22] N D Manring, Z L Dong. The impact of using a secondary swash-plate angle within an axial piston pump. *ASME Journal of Dynamic Systems, Measurement, and Control*, 2004, 126(1): 65-74.
[23] Z X Jiao, P Chen, Q Hua, et al. Theoretical study on vibration active control of power supply and pipeline systems. *Journal of Beijing University of Aeronautics and Astronautics*, 2002, 28(4): 465-469. (in Chinese)
[24] P C Ouyang, Z X Jiao, H M Liu. Study on distributed active control of fluid pulsation in hydraulic piping. *Journal of Beijing University of Aeronautics and Astronautics*, 2007, 33(9): 1060-1063. (in Chinese)
[25] P C Ouyang, H M Liu, Z X Jiao. Active control of fluid pulsation based on bypass overflow principle. *Acta Aeronautica Et Astronautica Sinica*, 2007, 28(6): 1302-1306. (in Chinese)
[26] P C Ouyang, Z X Jiao, H M Liu, et al. Active control on fluid borne pulsation using piezoelectric valve as absorber. *2006 IEEE Conference on Robotics, Automation and Mechatronics*, 2006, 69(33): 99-114.
[27] X W Ji, H L Liu, J Ke, et al. Active control on fluid pulsation with inflow method based on BP neural network. *Hydraulics Pneumatics & Seals*, 2016, 36(10): 17-21. (in Chinese)
[28] S R Guo, Y L Lu. Pressure fluctuation analysis and suppression of hydraulic energy system. *Chinese Hydraulics & Pneumatics*, 2011, 11: 49-51. (in Chinese)
[29] X Li, S P Wang, B C Huang. Simulation analysis of flow fluctuation in aviation piston pump and its structure optimization. *Journal of Lanzhou University of Technology*, 2010, 36(3): 60-64. (in Chinese)
[30] Z R Shi, G Parker, J Granstrom. Kinematic analysis of a swash-plate controlled variable displacement axial-piston pump with a conical barrel assembly. *Journal of Dynamic Systems, Measurement, and Control*, 2010, 132(1): 1-8.
[31] J E Ma, Y T Fang, B Xu, et al. Optimization of cross angle based on the pumping dynamics model. *Journal of Zhejiang University Science A*, 2010, 11(3): 181-190.
[32] C Q Liu, J H Jiang. Effect of plunger number on hydraulic transformer's flow pulsation rate. *High Technology Letters*, 2013, 19(1): 30-36.
[33] M Ye. Theoretical analysis of flow pulsation of axial piston pump. *Machine Tool & Hydraulics*, 1990, 2: 41-46. (in Chinese)
[34] K Ichiryu. Development of accumulator for high frequency ripple absorption. *Bulletin of JSME*, 1972, 15(88): 1215-1227.
[35] K Ichiryu. Vibration damping method of oil hydraulic system by accumulator. *Bulletin of JSME*, 1969, 12(53): 1110-1120.
[36] K L Xing, S H Ge, C S Ding, et al. Experimental study of the effect of new series connected bag accumulator on pressure pulsation of oil source. *Machine Tool & Hydraulics*, 1998, 1: 39-40. (in Chinese)
[37] P A Xie, Q Wang. Study on attenuation of fluid-borne pulsation using accumulator. *Noise and Vibration Control*, 2000, 4: 2-5. (in Chinese)
[38] J H Fang, Y Q Zhou, X D Hu, et al. Study on aerodynamic quality and fluid simulation of expansion mufflers. *Journal of System Simulation*, 2009, 21(20): 6399-6404. (in Chinese)
[39] S Ramamoorthy, K Grosh, J M Dodson. A theoretical study of structural acoustic silencers for hydraulic systems. *The Journal of the Acoustical Society of America*, 2002, 111(5 Pt 1): 2097-2108.
[40] J M Dodson, D R Dowling, K Grosh. Experimental investigation of quarter wavelength silencers in large-scale hydraulic systems. *Noise Control Engineering Journal*, 1998, 46(1): 15-22.
[41] Y Zhang. *The performances of the pressure pulsation attenuator in hydraulic systems*. Hangzhou: Zhejiang University, 2011. (in Chinese)
[42] R Du. *The characteristic analysis of pulsation dampers in hydraulic systems*. Chengdu: Southwest Jiaotong University, 2005. (in Chinese)
[43] Y G Cai. *Fluid transmission pipeline dynamics*. Hangzhou: Zhejiang University Press, 1990. (in Chinese)
[44] X P Ouyang, L Li, X Fang, et al. Research status and prospects of resonant-type hydraulic pulsation attenuators. *Journal of Mechanical Engineering*, 2015, 51(22): 168-175. (in Chinese)
[45] L Kela. Resonant frequency of an adjustable Helmholtz resonator in a hydraulic system. *Archive of Applied Mechanics*, 2009, 79(12): 1115-1125.
[46] N E Earnhart, K A Cunefare. Compact Helmholtz resonators for hydraulic systems. *International Journal of Fluid Power*, 2012, 13(1): 41-50.
[47] A Selamet, P M Radavich, N S Dickey, et al. Circular concentric Helmholtz resonators. *The Journal of the Acoustical Society of America*, 1997, 101(1): 41-51.
[48] A Selamet, Z L Ji. Circular asymmetric Helmholtz resonators. *The Journal of the Acoustical Society of America*, 2000, 107(5): 2360-2369.
[49] Z J Zhou, X Z Chen, Z B Dai, et al. Study on the modal and damping effect of the parallel type hydraulic pulsation attenuator. *Machinery & Electronics*, 2016, 34(7): 3-5. (in Chinese)
[50] X R Zeng, J C Zhang. Research on porous concentric hydraulic muffler. *Machine Tool & Hydraulics*, 1990, 1: 41-43. (in Chinese)
[51] E Kojima, T Ichiyanagi. Development research of new types of multiple volume resonators. *Bath Workshop on Power Transmission and Motion Control*, University of Bath, UK, 1998: 193-206.
[52] C B Guan, Z X Jiao. Modeling and optimal design of 3 degrees of freedom Helmholtz resonator in hydraulic system. *Chinese Journal of Aeronautics*, 2012, 25(5): 776-783.
[53] J Mikota, H Reiter. Development of a compact and tuneable vibration compensator for hydraulic systems. *International Journal of Fluid Power*, 2003, 4(1): 17-31.
[54] J Mikota. A novel, compact pulsation compensator to reduce pressure pulsations in hydraulic systems. *World Scientific*, 2001, 45: 69-78.
[55] F Gao, X P Ouyang, H Y Yang, et al. A novel pulsation attenuator for aircraft piston pump. *Mechatronics*, 2013, 23(6): 566-572.
[56] S H He, X Z Wang, Z Y He, et al. Filtering properties of thin plate hydraulic pulsation attenuator. *Journal of Mechanical Engineering*, 2013, 49(4): 148-153. (in Chinese)
[57] L X Huang. A theoretical study of passive control of duct noise using panels of varying compliance. *The Journal of the Acoustical Society of America*, 2001, 109(6): 2805-2814.
[58] S H He, Q Q Sang, H B He, et al. Filtering properties of fluid pulsation attenuator with multiple thin plate based on absorbing principle of mechanical vibration. *Journal of Central South University (Science and Technology)*, 2015, 5: 1457-1462.
[59] K A Marker, E R Gruber, K A Cunefare. Linear multimodal model for a pressurized gas bladder style hydraulic noise suppressor. *International Journal of Fluid Power*, 2013, 14(2): 5-16.

[60] E R Gruber, K A Cunefare, P W Danzl, et al. Optimization of single and dual suppressors under varying load and pressure conditions. *International Journal of Fluid Power*, 2013, 14(3): 27-34.
[61] S H He, Y W Xiong, W Wang. Research on filtering characteristics of hydraulic pulsation attenuator based on bionic principle of basilar membrane of cochlea. *Journal of Mechanical Engineering*, 2016, 52(4): 171-177. (in Chinese)
[62] J H Fang, Y Q Zhou, X D Hu, et al. Research on pressure loss and CFD simulation of the insert-pipe resistance mufflers. *Journal of Wuhan University of Technology (Transportation Science & Engineering)*, 2009, 33(4): 795-798. (in Chinese)
[63] F Yang, B Deng. Acoustic length correction of the extended inlet and outlet of a circular coaxial expansion chamber hydraulic suppressor. *The 25th International Congress on Sound and Vibration*, Hiroshima, Japan, July 8-12, 2018, 2884-2891.
[64] K A Marek, N E Earnhart, K A Cunefare. Model and analysis of a cylindrical in-line hydraulic suppressor with a solid compressible liner. *Journal of Sound and Vibration*, 2014, 333(24): 6312-6331.
[65] N E Earnhart, K A Cunefare. Compact Helmholtz resonators for hydraulic systems. *International Journal of Fluid Power*, 2012, 13(1): 1-50.
[66] F Yang, B Deng. Pulsation attenuation characteristics study of an expansion chamber hydraulic suppressor with a compressible liner using a lumped parameter mode. *Journal of Vibration and Shock*, 2018, 37(20): 221-226. (in Chinese)
[67] Y M Li, Z H Zhang, Y Zhang. Analysis of the flow field and optimal design for certain pressure pulsation attenuator based on FLUENT. *Machine Tool & Hydraulics*, 2010, 38(11): 86-88. (in Chinese)
[68] G W Stewart. The theory of the Herschel-Quincke tube. *Physical Review*, 1945, 17(2): 107-108.
[69] A Selamet, N S Dickey, J M Novak. The Herschel–Quincke tube: A theoretical, computational, and experimental investigation. *Journal of the Acoustical Society of America*, 1994, 96(5): 3177-3185.
[70] S H He, A M Ye, W Wang. Characteristics of compound pressure pulsation attenuator. *Journal of Changsha University of Science and Technology (Natural Science)*, 2015, 3: 91-97. (in Chinese)
[71] A M Ye. *Research on the characteristics and experiments of compound pressure pulsation attenuator*. Changsha: Changsha University of Science & Technology, 2015. (in Chinese)
[72] J Q Chen, S L Han, M Y Cui, et al. Numerical and experimental study on a new flow pulsation attenuator used in fluid-filled pipeline. *Fluid Machinery*, 2014, 1: 1-5. (in Chinese)
[73] C G Li, M Yan. *Modeling and simulation analysis of hydraulic systems*. Beijing: Aviation Industry Press, 2008. (in Chinese)
[74] Z C Luo. *Fluid network theory*. Beijing: China Machine Press, 1988. (in Chinese)
[75] Y G Yu, G F Gong, G L Hu. Simulation technique of AMESim and its application in hydraulic system. *Hydraulics Pneumatics & Seals*, 2005, 3: 28-31. (in Chinese)
[76] D H Su, J H Yu. New hydraulic simulation technology of AMESim and its application. *Machinery*, 2006, 33(11): 35-37. (in Chinese)
[77] W W Xu, D Z Wu, L Q Wang, et al. Analysis on characteristics of new muffler for marine pipelines. *Journal of Drainage and Irrigation Machinery Engineering*, 2013, 31(2): 142-145. (in Chinese)
[78] H A Tian, Z Y Xun, C F Zheng, et al. Simulative investigation into acoustic characteristics of water silencers. *Chinese Journal of Ship Research*, 2007, 2(6): 62-64. (in Chinese)
[79] D L Zhang, L F Zhao, H Y Zhang, et al. Comparative analysis of simulation accuracy for silencer transmission loss. *Journal of Vibration Engineering*, 2010, 23(6): 649-654. (in Chinese)
[80] A Broatch, J R Serrano, F J Arnau, et al. Time-domain computation of muffler frequency response: Comparison of different numerical schemes. *Journal of Sound and Vibration*, 2007, 305(1-2): 333-347.
[81] J M Middelberg, T J Barber, S S Leong, et al. CFD analysis of the acoustic and mean flow performance of simple expansion chamber mufflers. *ASME 2004 International Mechanical Engineering Congress and Exposition*, Anaheim, California, USA, November, 13-19, 2004: 151-156.
[82] J M Middelberg, T J Barber, S SLeong, et al. Computational fluid dynamics analysis of the acoustic performance of various simple expansion chamber mufflers. *Proceedings of Acoustics*, 2004: 123-127.
[83] O Z Mehdizadeh, M Paraschivoiu. A three-dimensional finite element approach for predicting the transmission loss in mufflers and silencers with no mean flow. *Applied Acoustics*, 2005, 66(8): 902-918.
[84] T W Wu, C Y R Cheng, P Zhang. A direct mixed-body boundary element method for packed silencers. *The Journal of the Acoustical Society of America*, 2002, 111(6): 2566-2572.
[85] A J Torregrosa, A Broatch, A Gil, et al. Analysis of acoustic networks including cavities by means of a linear finite volume method. *Journal of Sound and Vibration*, 2012, 331(20): 4575-4586.
[86] Z R Zhang, J L Li, C M Mak. Simulation analysis of acoustic attenuation performance for different shape of an expansion chamber silencer. *2009 Second International Conference on Information and Computing Science*, Manchester, England, May 21-22, 2009: 10-13.
[87] C J Shan, X H Liu, G Z Wang. Flow characteristics CFD resolution of pressure pulsation attenuator of high pressure piston pump. *Machine Tool & Hydraulics*, 2005, 7: 113-114. (in Chinese)
[88] C J Shan. *Pressure pulsation attenuator mathematic simulation and CFD flow resolution*. Chengdu: Southwest Jiaotong University, 2004. (in Chinese)
[89] Z F Shen. Design and numerical simulation of grating pressure pulsation attenuator. *Ship Engineering*, 2016, 38(9): 67-70. (in Chinese)
[90] H A Tian, G M Liu, Y Q Yang. Emulational investigation of resistance loss of silencer used in seawater pipeline. *Journal of Heilongjiang Institute of Science and Technology*, 2004, 14(1): 29-32. (in Chinese)
[91] J M Ma, F Wang, G W Yang, et al. Design and analysis of one type of pulse filter for hydraulic system. *Acta Aeronautica ET Astronautica Sinica*, 2009, 40(11): 285-293. (in Chinese)
[92] A Selamet, P M Radavich. The effect of length on the acoustic attenuation performance of concentric expansion chambers: An analytical, computational and experimental investigation. *Journal of Sound and Vibration*, 1997, 201(4): 407-426.
[93] B Venkatesham, M Tiwari, M L Munjal. Transmission loss analysis of rectangular expansion chamber with arbitrary location of inlet/outlet by means of Green's functions. *Journal of Sound and Vibration*, 2009, 323(3-5): 1032-1044.
[94] S Nag, A Gupta, A Dhar. Effect of geometric parameters on the acoustical performance of single inlet single outlet expansion chamber muffler. *International Conference on Electrical, Electronics, and Optimization Techniques*, Chennai, India, March, 3-5, 2016: 2522-2526.
[95] G W Jang, J W Lee. Optimal partition layout of expansion chamber muffler with offset inlet/outlet. *International Journal of Automotive Technology*, 2015, 16(5): 885-893.
[96] C J Shan. Structure design and attenuation method of hydraulic mufflers. *Journal of Jiamusi University (Natural Science Edition)*, 2013, 31(5): 749-751. (in Chinese)
[97] L K Xuan, Y Liu, J F Gong, et al. A time-domain finite volume method for the prediction of water muffler transmission loss considering elastic walls. *Advances in Mechanical Engineering*, 2017, 9(2):1-9.
[98] Z F Shen. The optimization of a centrifugal pump pressure pulsation attenuator design and experimental study. *Ship Science and Technology*, 2017, 39(3): 78-81. (in Chinese)
[99] L P Chu, L L Yang, B Zhao, et al. Research on attenuating pressure pulsation of reciprocating compressor using the external inserted volume-choke-volume acoustic filter. *Compressor Technology*, 2017, 2: 1-5. (in Chinese)
[100] J E Ma. *Study on flow ripple and valve plate optimization of axial piston pump*. Hangzhou: Zhejiang University, 2009. (in Chinese)