Prediction approach of compression loss in bent-axis piston pump with cavitation

Lv-jun Qing, Li-chen Gu, Bin Yang and Zhu-feng Lei

Abstract

Compression loss is one of the volumetric losses of bent-axis piston pumps, and its accurate evaluation is the basis for predicting the effective flow rate and volumetric efficiency of bent-axis piston pumps. The effect of changes in gas content on the effective bulk modulus rarely is considered in the existing compression loss models, which does not apply to the compression loss of bent-axis piston pumps under cavitation. However, cavitation exists in the actual work, especially in increasing the speed of the bent-axis piston pump. To solve this problem, a compression loss model is proposed. Firstly, the modified Henry’s law and the simplified transport equations of gas are used to describe the oil gas content under cavitation, and the effective bulk modulus equation is established. Then, combined with the definition of compressibility, the compression loss model of the bent-axis piston pump is proposed. Finally, the compression loss of the bent-axis piston pump under different operating conditions is analyzed through simulation, and the results are verified by experiment. The results show that the model prediction results are a good agreement with the experimental results under different operating conditions. The conclusions provide a new method for evaluating the effective flow rate and volumetric efficiency in the bent-axis piston pump.

Keywords

Cavitation, effective bulk modulus, bent-axis piston pump, volumetric losses, status assessment

Date received: 13 July 2022; accepted: 4 October 2022

Handling Editor: Chenhui Liang

Introduction

Compression loss is one of the volumetric losses of bent-axis piston pumps, it occurs when the piston chamber goes from the suction to the delivery. Compression loss decreases the effective flow rate and volumetric efficiency of piston pumps, and consequently, the system’s stability and safety. Therefore, it is crucial for the performance evaluation of bent-axis piston pumps to calculate the compression loss accurately.

Compression loss is caused by the compressibility of hydraulic fluid in the piston chamber of the bent-axis piston pump, which is related to the structural parameters, operating conditions, and the compressibility of fluid.1–3 In terms of the structural parameters of the piston pump, Zaluski analyzed the influence of the oil compressibility in dead space volume on the volumetric losses and established a model to study the effect of the position of the swash plate axis of rotation on the volumetric efficiency of the piston pump considering the

1National Joint Engineering Research Center for Special Pump Technology, Xi’an Aeronautical Institute, Xi’an, China

2Key Laboratory of Education Ministry for Modern Design and Rotor-Bearing System, Xi’an Jiaotong University, Xi’an, China

Corresponding author:
Lv-jun Qing, National Joint Engineering Research Center for Special Pump Technology, Xi’an Aeronautical Institute, West Two Ring, Xi’an 710077, China.
Email: qinglvjun@xaau.edu.cn

Creative Commons CC BY: This article is distributed under the terms of the Creative Commons Attribution 4.0 License (https://creativecommons.org/licenses/by/4.0/) which permits any use, reproduction and distribution of the work without further permission provided the original work is attributed as specified on the SAGE and Open Access pages (https://us.sagepub.com/en-us/nam/open-access-at-sage).
compressibility of oil.4 Wilhelm and Van de Ven proposed a variable displacement piston pump based on an adjustable six-bar-crank-rocker-slider mechanism to reduce energy loss.5 According to the relationship between leakage loss, compression loss, and volumetric efficiency, Wang established a volumetric efficiency model based on the concept of pressure carryover to study the influence of structural parameters of the valve plate on volumetric efficiency in the piston pump.6 Zhang et al. established a two-dimensional piston pump leakage loss and compression loss model and verified the models through simulation and experiment.7 Frosina et al. first established a 3D-CFD model to reserve for the flow losses caused by cavitation and then established a fast-lumped parameter model to analyze the influence of the valve plate geometry.8

As to the effect of operating conditions on compression loss, Xu et al. developed an energy loss model of the piston pump and analyzed the influence of compression loss, leakage loss, and mechanical loss on the volumetric efficiency of the piston pump under different displacement conditions through simulation and experiment.9 Zhang et al. used the CFD method to build a piston pump model to study the effective flow rate of the piston pump, which showed that compression loss is the main factor affecting flow pulsation.10 Paszota systematically analyzed the influence of system pressure, displacement, gas content, and viscosity on the effective bulk modulus of oil, and established a compression loss model of piston pumps to investigate the influence of compressibility of oil on energy loss of variable piston pumps.11,12 Koukouvinis et al. used numerical calculation methods to analyze the evolution law of cavitation of positive displacement pumps. The results showed that the cavitation can be more pronounced as the pump’s speed increases, and the thermodynamic effect can be significant during compression.13 Shah et al. proposed the hybrid Rayleigh–Plesset equation model based on the implicit relationship between the radius of the bubble and the Energy of the system to predict the dynamics of external gear machines.14 Wang used the controlled volume method to optimize the influence of the valve plate geometry to prevent the piston chamber pressure from below the vapor pressure.15

In the compressibility of the oil, Berta et al. presented a numerical model for axial piston pump considering cavitation and verified it experimentally.16 Van de Ven developed a numerical calculation model describing the relationship between the compressibility of the oil and the energy losses and studied the influence of oil gas content, the effective bulk modulus of oil, and system pressure on its volumetric efficiency through numerical simulation and experiment.17,18 Suo et al. proposed a full cavitation model and compressibility model to simulate the influence of cavitation on the flow rate of the piston pump.19 Gullapalli et al. used an experiment to study the influence of oil gas content on flow loss and noise of the piston pump, the results showed that the volume efficiency of the piston pump decreased, the mechanical efficiency increased slightly, and the overall efficiency decreased with the increase of gas content.20 Vacca et al. analyzed the influence of cavitation on the effective flow rate of swash-plate piston pumps based on compressible flow models.21 Patrosz used the CFD analysis and mathematical equations to build a model of aerated fluid compression and leakage model and studied the influence of oil characteristics on the working chamber pressure peaks of the piston pump.22 Koralewski defined the modulus of non-aerated and aerated oil and analyzed the influence of oil compressibility on the volumetric loss of the variable piston pump.23

The works mentioned above show that the fluid compressibility effect has a significant impact on the effective flow rate and volumetric efficiency of the piston pump. Most works about the compression loss of the piston pump rarely consider the effect of changes in gas content on the effective bulk modulus. Due to the change of the rotational speed, the minimum pressure in the working chamber and suction pipeline of the bent-axis piston pump changes dynamically, and the oil has the cavitation phenomenon, especially in the process of the increase of the speed, which leads to the increase of the gas content of the oil and the decrease of the effective bulk modulus. The phenomenon is also influenced by the oil temperature. When cavitation occurs, it is also accompanied by the phenomenon, such as the decrease in output flow rate and efficiency, the increase in pressure pulsations and vibrations, and the increase in oil temperature and wear, which influences the performance and service life of the bent-axis piston pump. However, the existing compression loss model is not suitable for compression loss characteristics of the bent-axis piston pump under the dynamic change of oil gas content caused by the cavitation.

In this study, the gas content of the working chamber is analyzed, and the effective bulk modulus equation is established. Combined with the definition of compressibility of the oil, the compression loss model of the bent-axis piston pump is proposed to calculate the influence of the cavitation on compression loss for the bent-axis piston pump. A test is then performed to validate the simulated results. This paper is organized as follows. In Section 2, the compression loss model of the bent-axis piston pump is proposed. In Section 3, the effects of cavitation on the effective bulk modulus of oil and the compression loss are studied through the simulation. In Section 4, the analysis and discussion of the experimental results are described. The conclusions of this work are drawn in Section 5.
Mathematical model of the compression loss

Principle of the compression loss in the bent-axis piston pump

Figure 1 shows the schematic of a bent-axis piston pump, which consists of a valve plate, cylinder block, pistons, driving shaft, bevel gear, pump case, etc. Due to the angle between the rotation axis of the cylinder block and the axis of the driving shaft, the piston assembly reciprocates in the cylinder block when the driving shaft and bevel gear drive the piston assembly and the cylinder block to rotate.

When the piston is in the suction region of the valve plate, the piston chamber is communicated with the inlet port. As the piston rotates, the volume of the piston chamber gradually increases, and the oil in the inlet port is sucked into the piston chamber due to the pressure difference between the inlet port and piston chamber. When the piston is in the oil discharge region of the valve plate, the piston chamber is communicated with the oil outlet port. As the piston rotates, the volume of the piston chamber gradually decreases, and the oil in the chamber continuously is discharged into the outlet port.

Due to the compressibility of oil, the high-pressure oil in the outlet port flows back to the piston chamber when the piston chamber is connected with the oil discharge region of the valve plate, which leads to reduce the effective flow rate of the bent-axis piston pump. This flow loss is called compression loss, which is related to the compressibility of the oil and the structural parameters of the piston chamber.

Dynamic effective bulk modulus of the aerated oil

Analysis of the aerated oil in the piston chamber. The oil in the bent-axis piston pump inevitably contains a certain amount of gas under the actual operating conditions. The component and content of the gas contained in the oil in the piston chamber change dynamically with the pressure of the piston chamber. According to the pressure, the oil in the piston chamber can be divided into four stages, as shown in Figure 2.

When the piston chamber pressure \( p \) is higher than saturation pressure \( p_s \), the oil in the piston chamber consists of liquid and dissolved gas. When the piston chamber pressure \( p \) is between high vapor pressure \( p_{vh} \) and saturation pressure \( p_s \), the oil is composed of free gas, dissolved gas, and liquid. When the piston chamber pressure \( p \) is between low vapor pressure \( p_{vl} \) and high vapor pressure \( p_{vh} \), the oil is composed of free air, vapor, and liquid. When the piston chamber pressure \( p \) is lower than low vapor pressure \( p_{vl} \), all liquid is vaporized, and the piston chamber is composed of free air and vapor. Therefore, the aerated oil in the piston...
chamber may contain three components that affect the effective bulk modulus of the aerated oil, which are liquid, free gas, and vapor at the same time.

**Dissolve air and vapor transport equations.** According to the modified Henry’s law, the mass fraction of vapor in oil is related to pressure under the equilibrium state. When the piston chamber pressure is lower than low vapor pressure, all the liquid oil vaporizes, forming vapor. When the piston chamber pressure is higher than the low vapor pressure and lower than the high vapor pressure, the vapor content decreases with the increase of the pressure. When the piston chamber pressure is higher than the high vapor pressure, all the vapor condenses and the content of vapor is 0. Therefore, the target value \( f_{v-th} \) of mass fraction of vapor under different pressure is governed as

\[
f_{v-th} = \begin{cases} 
0 & p > p_{th} \\
(1 - f_0)(1 - 10k_f + 15k_f^2 - 6k_f^3) & p < p_{th} \\
1 - f_0 & p = p_{th} \\
1 & p < p_{th} 
\end{cases}
\]

Where, \( k_f = \frac{p - p_{th}}{p_{th} - p_{vl}} \) (2)

When the piston chamber pressure is lower than the high vapor pressure of the oil, the dissolved air is released to form free air. When the piston chamber pressure is higher than the high vapor pressure, the condensation of vapor occurs, which is related to the mass gas content of vapor generated. The simplified transport equation of vapor is defined as

\[
df_v = \begin{cases} 
\frac{k_f}{\tau} (1 - f_0 - f_v) \sqrt{p_{ch} - p} & p < p_{th} \\
\frac{k_f}{\tau} f_v \sqrt{p_{ch} - p} & p > p_{th}
\end{cases}
\]

Where, \( f_v \) is the mass fraction of vapor in aerated oil. \( k_{v1}, k_{v2} \) are the coefficient of calculation for the mass fraction of vapor. \( \tau \) is the characteristic time, and its value is the time for one revolution of the pump.

In terms of free air content in aerated oil, when the piston chamber pressure is lower than the high vapor pressure of the oil, the dissolved air is released to form free air. When the piston chamber pressure is between the high vapor pressure and saturation pressure, the free air content decreases with the pressure. When the piston chamber pressure is higher than saturation pressure, all free air is dissolved, and free air gas content is 0. Thus, the target value \( f_{f-th} \) of mass fraction of free air in aerated oil is described as

\[
f_{f-th} = \begin{cases} 
f_0 & p > p_{th} \\
(1 - 10k_f + 15k_f^2 - 6k_f^3) & p_{vl} < p < p_{th} \\
0 & p < p_{th}
\end{cases}
\]

Where, \( k_f = \frac{p - p_{th}}{p_{th} - p_{ps}} \) (5)

When the mass fraction of free air \( f_f \) is lower than the target value \( f_{f-th} \), the dissolved air in oil is released, and the mass fraction of free air increases. On the contrary, the free air is dissolved in oil, and the mass fraction of free air decreases. The simplified transport equation of free air is expressed as

\[
\frac{df_f}{dt} = \begin{cases} 
\frac{k_f}{\tau} (f_f - f_{f-th}) \sqrt{p_{s} - p} & f_f < f_{f-th} \\
\frac{k_f}{\tau} f_f \sqrt{p_{s} - p} & f_f > f_{f-th}
\end{cases}
\]

Where, \( k_{f1}, k_{f2} \) are the coefficient of calculation for the mass fraction of free air.

Assuming that the time required for the piston chamber pressure to increase linearly from absolute zero pressure to saturation pressure is \( t \), then the relationship between the piston chamber pressure and time is as follows

\[
p = k_p t \quad 0 < p < p_s
\]

Where, \( k_p \) is the pressure gradient.

**Effective bulk modulus of the aerated oil.** The effective bulk modulus of aerated oil is based on the following assumptions: (a). the oil compression or expansion process is an adiabatic process. (b). the gas is ideal and varies with pressure in a way determined by a constant polytropic index. (c). the oil volume is a unit volume and the influence of dissolved air on the oil volume is not considered. (d). the effect of dissolved gas on effective bulk modulus is ignored.

According to the relationship between the volume, density, and quality of each component of aerated oil, the volume of liquid, vapor, and free air under a certain pressure is described as

\[
\begin{align*}
V_l &= m_{l0} \left( \frac{p_v}{p_{vl}} \right) \frac{m_{l0}}{\rho_0} \frac{p_0}{p} \frac{p_{vl}}{p} \\
V_v &= m_{l0} \left( \frac{p_v}{p_{vl}} \right) \frac{m_{l0}}{\rho_0} \frac{p_0}{p} \frac{p_{vl}}{p} \\
V_f &= m_{l0} \left( \frac{p_v}{p_{vl}} \right) \frac{m_{l0}}{\rho_0} \frac{p_0}{p} \frac{p_{vl}}{p}
\end{align*}
\]

Where, \( V_l \) is the volume of the pure liquid in aerated oil. \( V_v \) is the volume of the free air in aerated oil. \( V_f \) is the volume of the vapor in aerated oil. \( f_v \) is the mass fraction of the vapor in aerated oil. \( f_f \) is the mass fraction of the free air in aerated oil. \( m_{l0} \) is the quality of aerated oil. \( m_f \) is the quality of the free air in aerated oil. \( m_v \) is the quality of the vapor in aerated oil. \( m_f \) is the quality of the liquid in aerated oil. \( E_f \) is the effective bulk modulus of pure oil. \( \rho_0 \) is the density of pure oil at the standard atmospheric pressure and 0°C. \( \rho_0 \) is the standard atmospheric pressure. \( \rho_f \) is the density of vapor under a certain pressure. \( \rho_{of} \) is the density of
vapor at saturated vapor pressure. \( p_v \) is the saturated vapor pressure of the oil. \( p_f \) is the density of free air under a certain pressure. \( \rho_{f0} \) is the density of free air at standard atmospheric pressure. \( \lambda \) is the polytropic index.

The change rate of the volume of liquid, vapor, and free air with pressure is expressed as

\[
\begin{align*}
\frac{dV_l}{dp} &= -\frac{\rho_{f0}}{\rho_{f0}} \left( E_f \frac{df}{dp} + E_l \frac{dl}{dp} \right) + 1 - f_v - f_f \\
\frac{dV_v}{dp} &= \frac{\rho_{f0}}{\rho_{v0}} \left( \frac{df}{dp} - \frac{f_f}{\rho_f} \right) / \lambda \\
\frac{dV_f}{dp} &= \frac{\rho_{f0}}{\rho_{f0}} \left( \frac{df}{dp} - \frac{f_f}{\rho_f} \right) / \lambda
\end{align*}
\]

\[(9)\]

Aerated oil may have three components at the same time, which include liquid, free air, and vapor. Under normal operating conditions, there is very little free air and vapor in the oil, the volume interaction between the components is ignored, then the effective bulk modulus of aerated oil is described as

\[
E = \frac{V_l}{\frac{dV_l}{dp} + \frac{dV_v}{dp} + \frac{dV_f}{dp}}
\]

\[(10)\]

By substituting equations (9) and (10) into equation (12), the effective bulk modulus of aerated oil can be obtained

\[
E_e = \frac{c_1 p^{1/\lambda}(1 - f_v - f_f) + c_2 f_f + c_3 f_v}{c_4 p^{1/\lambda} \left( E_f \frac{df}{dp} + E_l \frac{dl}{dp} \right) + 1 - f_v - f_f} + c_2 \left( \frac{df}{dp} - \frac{f_f}{\rho_f} \right) + c_3 \left( \frac{df}{dp} - \frac{f_f}{\rho_f} \right)
\]

\[(11)\]

\[
\begin{align*}
c_1 &= e^{-\frac{p_{f0}}{\rho_{f0}}} / \rho_{f0} \\
c_2 &= p_{v0}^{1/\lambda} / \rho_f \\
c_3 &= p_{v0}^{1/\lambda} / \rho_f \\
c_4 &= -c_1 / E_l
\end{align*}
\]

\[(12)\]

Dynamic pressure of the piston chamber. During one revolution of the bent-axis piston pump, the piston chamber completes the oil suction and oil discharge respectively, the pressure of the piston chamber in the suction stroke is the key factor for cavitation. According to the working principle of the bent-axis piston pump, the volume change rate of the piston chamber is expressed as

\[
\frac{dV_{pc}}{dt} = q_s
\]

\[(13)\]

By substituting equations (13) and (14) into equation (15), the dynamic pressure of the piston chamber can be obtained

\[
p = \frac{p_A - \frac{p}{2} \left( \frac{\pi n A_p}{30 C_A} R_s \sin \gamma \sin \frac{\pi n}{30} t \right)^2}{30 C_A}
\]

\[(16)\]

Compression loss of the bent-axis piston pump

The working chamber is composed of the volume \( V_2 \) between the piston and the cylinder block and the volume \( V_1 \) corresponding to the cylinder block, as shown in Figure 3. \( V_1 \) is generally a fixed value and is determined by the design, while \( V_2 \) changes dynamically with the angular position of the piston. \( V_2 \) is minimum when the piston moves to the bottom dead center.
(BDC), as shown in Figure 3(a). $V_2$ is maximum when the piston moves to the top dead center (TDC), as shown in Figure 3(b). During the movement of the piston from the suction region to the discharge region, the reduced oil volume is defined as the compressive displacement loss $V_c$, as shown in Figure 3(c). According to the definition of the bulk modulus of oil, the compressive displacement loss can be expressed as

$$V_c = N \int \frac{V_{th} + NV_d}{NE} dp$$  \hspace{1cm} (17)\n
Where, $V_c$ is the compressive displacement loss of the pump, $E_e$ is the effective bulk modulus of oil, $V_d$ is the volume of the piston chamber at BDC, $V_{th}$ is the theoretical displacement, $N$ is the number of the piston.

The compressive flow rate loss of the bent-axis piston pump is expressed as

$$Q_c = nV_c$$  \hspace{1cm} (18)\n
The percentage of compressive displacement loss to the theoretical displacement (compressive loss percentage) can be obtained

$$\eta_c = \frac{V_c}{V_{th}} \times 100$$  \hspace{1cm} (19)\n
**Numerical analysis**

In this paper, the numerical integration method based on the trapezoidal rule is used to deal with the differential equations in the dynamic model. The simulation tool is Lenovo Workstation, and its basic parameters are Inter(R) Xeon(R) CPU E5-1680 v4 @ 3.4 GHz, Windows 10. The subroutine module in Matlab is used to build up the calculation model of the effective bulk elastic modulus of the oil and the calculation model of the compression displacement loss of the single-piston chamber, and then build up the calculation model of compression flow rate loss. The variation law of the effective bulk elastic modulus of hydraulic oil and compression loss characteristics of the bent-axis piston pump under different operating conditions are analyzed through the numerical calculation results. In the simulation, L-HM46 hydraulic oil is used as the bent-axis piston pump’s working medium, and its main parameters are shown in Table 1. The oil pressure first increases linearly from 0.01 to 30 MPa, then decreases linearly from 30 to 0.01 MPa, and the whole process takes 5 s.

**Effective bulk modulus of the oil**

The simulated effective bulk modulus of the oil increases with the pressure of the oil in the compression process, and decreases with the pressure in the expansion process at the initial gas content of 12.29%, as shown in Figure 4. The simulated effective bulk modulus of oil in the compression process is smaller than that in the expansion process at the same pressure. The reason is that the pressure gradually decreases from 30 MPa to saturation pressure in the expansion process, the gas content in the oil is higher than the theoretical dissolution rate, and the gas content decreases with gas dissolution. Therefore, the simulated effective bulk modulus of oil in the expansion is higher than that in the compression process. When the pressure is lower than the saturated vapor pressure, the oil vaporizes, the gas content increases significantly, and the simulated effective bulk modulus of the oil decreases significantly.

The simulated effective bulk modulus of the oil at the same pressure point increases with the pressure change period at the initial gas content of 12.29%, as presented in Figure 5. When the pressure change period increases from 0.06 to 5 s, the effective bulk modulus at 1 MPa is 52 MPa, which is unchanged, while the effective bulk modulus of oil at 30 MPa increases from 1556 to 1666 MPa, which increases by 110 MPa. The reason

| Parameters | Values | Parameters | Values |
|------------|--------|------------|--------|
| $E_e$/MPa  | 1700   | $X_0$/%    | 12.29  |
| $\rho$/kg.m$^{-3}$ | 850     | $p_0$/MPa  | 0.12   |
| $\rho_0$/kg.m$^{-3}$ | 1.2      | $P_d$/MPa  | 0.05   |
| $\rho_0$/kg.m$^{-3}$ | 10.62    | $P_e$/MPa  | 0.04   |
| $T$/°C     | 25     | $t$/s      | 1      |
| $R_f$/mm   | 44.45  | $\gamma$   | 45     |
| $C_s$      | 0.62   | $P_s$/MPa  | 0.1    |
| $V_{th}$/ml/r | 80.4  | $N$        | 7      |

**Table 1. Main parameters.**

![Simulated effective bulk modulus versus pressure: comparison between compression process and the expansion process.](image)
is that the longer the pressure change period and the higher the pressure, the more free air is dissolved in the oil, and the gas content in the oil reduces, so the effective bulk modulus is larger.

Figure 6 shows that the simulated effective bulk modulus of the oil decreases with the gas content at the low-pressure stage. With the increase of the pressure, the effect of the gas content on the effective bulk modulus is gradually weakened at the high-pressure stage, and the simulated effective bulk modulus is gradually consistent. When the initial gas content increases from 0.1% to 12.29%, the simulated effective bulk modulus at 1 MPa decreases from 1342 to 52 MPa, with the decrease of 1290 MPa, and the simulated effective bulk modulus at 30 MPa decreases from 1699 to 1556 MPa, with the decrease of 143 MPa. The reason is that the free air in the oil continues to dissolve as the pressure increases, resulting in the actual air content in the high-pressure stage being the same, and the difference in the simulated effective bulk modulus at the high-pressure stage gradually decreases.

Compression loss of the bent-axis piston pump

The simulated compressive loss percentage in the bent-axis piston pump increases rapidly with the increase of the discharge pressure in the pressure range 0.1–2 MPa, and increases linearly with the pressure in the pressure range 2–30 MPa, as illustrated in Figure 7. As the initial gas content is greater, the compressive loss percentage is greater, and the nonlinear characteristics are more pronounced.

In Figure 8, it is seen that the compression loss percentage in the bent-axis piston pump is positively proportional to the initial gas content at the same discharge pressure. When the gas content increases from 0.1% to 12.29%, compression loss percentage increases from 1.5% to 4.8% at the discharge pressure of 25 MPa.

From Figure 9, the simulated compression loss percentage nonlinearly increases with the discharge pressure increases from 0.05 to 0.8 MPa at the initial gas content of 0.1%, and the suction pressure of 0.05 MPa, increases linearly with the discharge pressure in the pressure range 0.8–30 MPa. When the suction pressure increases, the simulated compression loss percentage decreases at the same discharge pressure, and its
nonlinear characteristics weaken in the discharge pressure range 0.05–0.8 MPa. The reason is that its nonlinear characteristics depend on the effective bulk modulus. When the suction pressure is smaller, the effective bulk modulus of the oil is smaller at the low-pressure stage, and the nonlinearity of compression loss is more obvious.

As can be seen from Figure 10, the influence of the suction pressure on the compression loss percentage is divided into three stages under initial gas content of 0.1% and discharge pressure of 5 MPa. When the suction pressure gradually decreases from 0.05 to 0.048 MPa, the simulated compression loss percentage increases from 0.6% to 29.1%, and the average increase rate of 14,250%/MPa, which belongs to strong influence stage. The compression loss percentage increases from 0.3% to 0.6% as the suction pressure decreases gradually from 0.12 to 0.05 MPa, and the average increase rate of 4%/MPa, which belongs to the weak influence stage. The compression loss percentage is maintained at 0.3% in the suction pressure range of 0.12–1 MPa, which is basically not affected by the suction pressure. The greater the discharge pressure, the greater the overall compression loss.

The reason is that the initial free air is completely dissolved in the oil, and does not affect its effective bulk modulus as the suction pressure is higher than 0.12 MPa. In the pressure range of 0.05–0.12 MPa, as the pressure decreases, the free air content increases with the suction pressure decreasing from 0.12 to 0.05 MPa, and the effective bulk modulus decreases. When the suction pressure is lower than 0.05 MPa, the oil vaporizes, the gas content increases rapidly, and the effective bulk modulus decreases rapidly, resulting in a significant increase in the simulated compression loss in the bent-axis piston pump.

Experimental validation

To verify the proposed model, First, the improved Nykanen model, the IFAS model, the Zhou model, and Zhejiang University experimental data25–28 are used to validate the dynamic effective bulk modulus model. Then, a hydraulic pump test rig is used to validate the compression loss model by measuring the

---

**Figure 8.** Simulated compression loss percentage versus volume fraction of air under different discharge pressures.

**Figure 9.** Simulated compression loss percentage versus discharge pressure under different suction pressures.

**Figure 10.** Simulated compression loss percentage versus suction pressure under different discharge pressures.
effective bulk modulus of the bent-axis piston pump under different operating conditions.

Validation of the effective bulk modulus model

The simulated effective bulk modulus based on different theoretical models is shown in Figure 11. The comparison shows the agreement between the proposed model of effective bulk modulus (PEBM) and other models, and both increase with the increase of pressure. When the pressure is higher than the saturation pressure, the calculated results of the PEBM are higher than those of the Nykanen model and the IFAS model, but slightly lower than those of the Zhou model. Because the gas content in the Nykanen model and the IFAS model remains constant, while the Zhou model and the proposed model consider the gas content decreases with the pressure. The effect of the time change rate of gas content is not taken into account in the denominators of the Zhou model, so the calculated results are slightly larger than those of the proposed model. When the pressure is lower than the high saturated vapor pressure, the results calculated by the Zhou model and proposed model are obviously smaller than those calculated by the Nykanen model and the IFAS model, because the former consider the oil vaporization effect.

Feng used the test rig to study the effective bulk modulus of L-HM46 hydraulic oil in the range of 0.2–20 MPa at 25°C. The experimental results and the calculation results of the proposed model are shown in Figure 12. The calculation results of the proposed model are consistent with the experimental results in the 0.2–20 MPa range. In the range of 0.2–10 MPa, the effective bulk modulus increases rapidly with the pressure. In the 10–20 MPa range, the effective bulk modulus changes gently with the pressure, and finally basically maintains at about 1700 MPa, close to the effective bulk modulus of pure oil, which further verifies the correctness of the dynamic effective bulk modulus model.

Validation of the compression loss model

Test rig Description. As shown in Figure 13, the test rig consists of a subsystem for driving the test pump, the test pump, control valves, sensors, and data acquisition system, and so on. A typical bent-axis piston pump widely used in the industry is chosen as the test pump. The speed of the test pump is adjusted by the hydraulic motor, and its load is adjusted by the load valve. Flow sensors are installed on the inlet pipeline, outlet pipeline, and house case leakage pipeline of the test pump. A speed sensor and a torque sensor are mounted on the drive shaft. Pressure sensors are mounted on the inlet pipeline and outlet pipeline of the test pump. The parameters of the test pump and the main sensor are shown in Table 2.

First, experiments are carried out at a speed of 600 r/min and intervals of 200 r/min under the no-load and self-absorption conditions. According to the test data, the critical speed \( n_s \) corresponding to the cavitation phenomenon of the bent-axis piston pump is determined. Then, experiments are completed under the speed varying from 600 r/min to \( n_s \) in increments of 200 r/min, as well as discharge pressure from 1 to 25 MPa in increments of 2 MPa. The inlet flow rate, outlet flow rate,
and leakage of house case are calculated by equations (20)–(22).

\[ Q_{in} = N_{in} V_{in} / \Delta t \]  
\[ Q_{out} = N_{out} V_{out} / \Delta t \]  
\[ Q_{d} = N_{d} V_{d} / \Delta t \]

Where, \( N_{in} \) is the number of pulse signals from the inlet flow sensor during the sampling time, \( V_{in} \) is the volume corresponding to each pulse of the inlet flow sensor, \( \Delta t \) is the sampling time, \( N_{out} \) is the number of pulse signals from the outlet flow sensor during the sampling time, \( V_{out} \) is the volume corresponding to each pulse of the outlet flow sensor, \( N_{d} \) is the number of pulse signals from the flow sensor mounted on house case leakage pipeline during the sampling time, \( V_{d} \) is the volume corresponding to each pulse of the leakage flow sensor.

The compressive flow rate loss \( Q_{c} \), and compressive displacement loss \( V_{c} \) can be derived by equations (23)–(24).

\[ Q_{c} = Q_{in} - Q_{out} - Q_{d} \]  
\[ V_{c} = Q_{c} / n_e \]

Where, \( n_e \) is the actual speed of the test pump.

**Experimental results.** Figure 14 shows the variations of flow rate and volumetric efficiency with the speed are mainly divided into two stages under the self-priming and the no-load conditions. In the range of 600–1400 r/min, the flow rates of the inlet and outlet increase linearly with the speed, and are basically consistent with the theoretical flow rate. The volumetric efficiency remains the same. In the range of 1400–1800 r/min, the growth rate of the inlet flow rate and outlet flow rate gradually decreases with the increase of the speed, and the inlet flow rate and outlet flow rate are significantly less than the theoretical flow, and the volumetric efficiency drops significantly. The reason is that cavitation is observed in the bent-axis piston pump if the speed is higher than 1400 r/min under the self-priming condition.

In Figure 15(a), it is shown that the comparison between the calculated results of the different compression loss models and the measured results under no-
load conditions. In the range of 600–1400 r/min, the compression loss percentage calculated by different models is basically consistent with measured results, and both slightly increase with the rotational speed. In the range of 1400–2800 r/min, the compression loss model (PEBM) and the measured results are basically consistent, and both increase rapidly with the rotation speed, but there is a large difference between the compression loss model (IFAS) and the measured results. The reason is that the dynamic pressure of the piston chamber in the bent-axis piston pump decreases with speed, as shown in Figure 15(b). During the suction stroke, the pressure of the piston chamber is divided into three stages. It increases greatly with the increase of the angle position in the range of 0°–30°, it decreases with the increase of the angular position in the range of 150°–180°. Due to the influence of the damping structure of the valve plate, the pressure of the piston chamber is lower than the critical pressure (0.04 MPa) in the above angle position range. In the range of 30°–150°, the pressure first decreases and then increases with the increase of the angle position, and decreases as a whole with the increase of the rotation speed. The piston chamber is located in the angle position of 90°, and the pressure is lower than 0.4 MPa, and the oil cavitation occurs under the rotation speed exceeding 1400 r/min. Cavitation will intensify as the rotation speed increases further.

When the speed is in the 600–1400 r/min range, the cavitation is weak, and it has little effect on the gas content of the oil. When speed is higher than 1400 r/min, the overall dynamic pressure of the piston chamber decreases, and the cavitation is significant, which has a significant impact on the gas content of the oil. The existing compression loss model, which uses the effective bulk modulus (such as the IFAS model) no consider change of the gas content caused by cavitation, and is not suitable for the calculation of the compression flow rate loss under cavitation conditions. A new compression loss model established in this paper adopts the bulk elastic modulus to consider the dynamic change of gas caused by cavitation, and can be used to calculate the compression loss of the bent-axis piston pump under the cavitation.

Figure 16 shows the real characteristic curve of flow rate in bent-axis piston pump under different operating conditions. The volumetric efficiency of the bent-axis piston pump slightly decreases with the discharge pressure. In the range of 1000–1400 r/min, the rotational speed has little effect on volumetric efficiency, as shown in Figure 16(a). The ratio of compression flow rate loss to the total volumetric losses (compression loss ratio) in the bent-axis piston pump decreases with the increase of discharge pressure. The compression loss ratio increases with the rotation speed, and it exceeds 50% under the condition of speed 1000–1400 r/min and discharge pressure 6–25 MPa, as shown in Figure 16(b). The compression loss is the main source of volumetric losses in the bent-axis piston pump.

Figure 17 shows measured results and calculated results of the compression loss percentage under the load conditions. It can be observed that the measured results are basically consistent with the calculated results, and increase with the discharge pressure. The discharge pressure is the main impact factor of the compression loss percentage. The result further verifies the correctness of the compression loss model.
Conclusions

In this paper, the effect of cavitation is considered in the compression loss of bent-axis piston pumps, which is studied through a mathematical model. The mathematical model of the compression loss is based on the dynamic effective bulk modulus equation considering cavitation. The compression loss of the bent-axis piston pump under different working conditions is analyzed by simulation and experiment, and the following conclusions are as follows:

(1) The distribution law of the effective bulk elastic modulus of the oil in the low-pressure stage is the key factor affecting the compression loss. The smaller the initial value of the effective bulk elastic modulus, the larger the nonlinear range of the low-pressure stage and the larger the compression loss.

(2) Under the no-load conditions, the compression flow rate loss of the bent-axis piston pump remains unchanged in the range of 600–1400 r/min, while increases rapidly in the range of 1400–2800 r/min. The reason is that the rotational speed of the bent-axis piston pump exceeds 1400 r/min in the experiment, and the oil in the piston chamber appears the cavitation.

(3) The compression flow rate loss is positively correlated with the speed and discharge pressure of the bent-axis piston pump. The compression loss ratio exceeds 50% under the condition of speed 1000–1400 r/min and discharge pressure 6–25 MPa, which is the main source of volumetric losses in the bent-axis piston pump.

Although the compression loss model is established to analyze the influence of cavitation on the compression loss characteristics of the bent-axis piston pump, the cavitation is closely related to the dynamic pressure of the piston chamber during one revolution. In order to reduce volumetric losses due to cavitation, the influence of pump rotational speed, valve plate structure, and oil supply pressure on the dynamic pressure of the piston chamber and volumetric losses in bent-axis piston pump need to be further studied.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.
Funding
The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work was supported in part by the National Natural Science Foundation of China [No. 52105086], China Postdoctoral Science Found [No. 2021M692509], Natural Science Basic Research Program of Shanxi Province [No. 2021JQ-860], Special Research Project of Education Department of Shanxi Province [No. 21JK0702], Science Foundation of Xi’an Aeronautical Institute [No. 2020KY0212, No. 2020KY0223].

ORCID iD
Lv-jun Qing https://orcid.org/0000-0003-4731-6101

References
1. Geng B-L, Gu LC, Liu J-M, et al. Dynamic modeling of fluid nonlinear compression loss and flow loss oriented to fault diagnosis of axial piston pump. Proc IMechE, Part C: J Mechanical Engineering Science 2021; 235: 3236–3251.
2. Marinaro G, Frosina E, Senatore A, et al. A fast and effective method for the optimization of the valve plate of swashplate axial piston pumps. J Fluid Eng 2021; 143: 091203–091213.
3. Williamson C and Manring N. A more accurate definition of mechanical and volumetric efficiencies for digital displacement® pumps. In: Proceedings of the ASME/BATH 2019 symposium on fluid power and motion control. Longboat Key, FL, 7–9 October 2019, p.V001T01A031. New York: ASME.
4. Zaluski P. Influence of the position of the swash plate rotation axis on the volumetric efficiency of the axial piston pumps. Machines Technologies Materials 2014; 8: 12–15.
5. Wilhelm SR and Van de Ven JD. Design and testing of an adjustable linkage for a variable displacement pump. J Mech Robot 2013; 5: 041008.
6. Wang S. Improving the volumetric efficiency of the axial piston pump, J Mech Des 2012; 134: 111001.
7. Zhang C, Ruan J, Xing T, et al. Research on the volumetric efficiency of a novel stacked roller 2D piston pump. Machines 2021; 9: 128.
8. Frosina E, Marinaro G and Senatore A. Experimental and numerical analysis of an axial piston pump: A comparison between lumped parameter and 3D CFD approaches. In: Proceedings of the ASME-JSME-KSME 2019 8th joint fluids engineering conference. San Francisco, CA, 28 July–1 August 2019, p.V001T01A042. New York: ASME.
9. Xu B, Hu M, Zhang J-H, et al. Characteristics of volumetric losses and efficiency of axial piston pump with respect to displacement conditions. J Zhejiang Univ Sci A 2016; 17: 186–201.
10. Zhang B, Ma J, Hong H, et al. Analysis of the flow dynamics characteristics of an axial piston pump based on the computational fluid dynamics method. Eng Appl Comput Fluid Mech 2017; 11: 86–95.
11. Paszota Z. Mathematical model defining volumetric losses of hydraulic oil compression in a variable capacity displacement pump. Pol Marit Res 2015; 21: 90–99.
12. Paszota Z. Comparison of the powers of energy losses in a variable capacity displacement pump determined with or without taking into account the power of hydraulic oil compression. Pol Marit Res 2015; 22: 32–43.
13. Koukouvinis P, Murali-Girija M, Karathanassis IK, et al. Chapter 10 - cavitation in positive displacement pumps. In: Koukouvinis P and Gavaises M (eds) Cavitation and bubbling dynamics. London: Academic Press, 2021, 303–329.
14. Shah Y, Vacca A and Dabiri S. Air release and cavitation modeling with a lumped parameter approach based on the Rayleigh–Plesset equation: the case of an external gear pump. Energies 2018; 11: 3472.
15. Wang S. The analysis of cavitation problems in the axial piston pump. J Fluid Eng 2010; 132: 1–6.
16. Berta GL, Casoli P, Vacca A, et al. Simulation model of an axial piston pump inclusive of cavitation. In: Proceedings of the ASME 2002 international mechanical engineering congress and exposition, New Orleans, LA, 17–22 November 2002, pp.29–37. New York: ASME.
17. Van de Ven JD. On fluid compressibility in switch-mode hydraulic circuits—Part I: modeling and analysis. J Dyn Syst Meas Control 2013; 135: 021013.
18. Van de Ven JD. On fluid compressibility in switch-mode hydraulic circuits—Part II: experimental results. J Dyn Syst Meas Control 2013; 135: 021014.
19. Suo X, Jiang Y and Wang W. Hydraulic axial plunger pump: gaseous and vapor cavitation characteristics and optimization method. Eng Appl Comput Fluid Mech 2021; 15: 712–726.
20. Gullapalli S, Michael P, Kensler J, et al. An investigation of hydraulic fluid composition and aeration in an axial piston pump. In: Proceedings of the ASME/BATH 2017 symposium on fluid power and motion control, Sarasota, FL, 16–19 October 2017, p.V001T01A028. New York: ASME.
21. Vacca A, Klop R and Ivantysynova M. A numerical approach for the evaluation of the effects of air release and vapour cavitation on effective flow rate of axial piston machines. Int J Fluid Power 2010; 11: 33–45.
22. Patrosz P. Influence of properties of hydraulic fluid on pressure peaks in axial piston pumps' chambers. Energies 2021; 14: 3764.
23. Koralewski J. Influence of viscosity and compressibility of aerated oil on determination of volumetric losses in a variable capacity piston pump. Pol Marit Res 2013; 20: 4–17.
24. Xiaoming Y, Chu W, Xuan Z, et al. Research on theoretical model of dynamic bulk modulus of elasticity of gas-liquid mixed fluid. Mech Eng J 2020; 56: 209–217.
25. Zhou J, Vacca A and Manhartgrabner B. A novel approach for the prediction of dynamic features of air release and absorption in hydraulic oils. J Fluid Eng 2013; 135: 091305.
26. Gholizadeh H, Burton R and Schoenau G. Fluid bulk modulus: Comparison of low pressure models. Int J Fluid Power 2012; 13: 7–16.
27. Kim S and Murrenoff H. Measurement of effective bulk modulus for hydraulic oil at low pressure. J Fluid Eng 2012; 134: 021201.
28. Feng B. Study on effective fluid bulk modulus and measurement in hydraulic systems. Zhejiang: Zhejiang University, 2011.