INTRODUCTION

NGH is an ice-like crystalline solid formed from a mixture of water and natural gas that have been subjected to high-pressure and suitable low-temperature conditions.\(^1\) NGH is a new kind of clean energy that was discovered in the last 20 years and is widely distributed in the oceans and frozen belt. It is considered the most suitable potential energy alternative to oil and coal.\(^2-4\) Global resources of NGH are estimated to be approximately twice those of coal, oil, and natural gas.\(^5-7\)

Natural gas hydrate can remain stable at low temperatures \((T < 10^\circ C)\) and high pressures \((P > 10\, MPa)\).\(^8,9\) When the pressure decreases or the temperature increases, all or most of the NGH decomposes.\(^10-12\) Pressure cores recover gas hydrate preserved within the sedimentary matrix, without a disruption caused by gas hydrate dissociation or dissolved gas exsolution.\(^13\) They provide the least-disturbed samples of gas hydrate formations and the best samples for laboratory tests, including geophysical measurements, geomechanical tests, and gas hydrate quantification. The analysis of pressure cores in gas hydrate environments...
provides unique data sets that cannot be obtained by other techniques. The previous method for quantifying the total concentration of natural gas in a core was to conduct mass balance analyses based on the careful depressurization of pressure cores. However, this method cannot accurately measure the concentration of natural gas owing to the partial decomposition of NGH during the decompression process. Furthermore, depressurized cores do not enable further laboratory experiments. Although the current pressure-retaining sampling technology is relatively mature, the core samples do not provide sufficient important information.

To obtain more information on the NGH, the cores must be cut into small sections and sent to the laboratory for further analyses at in situ pressure. The current mature NGH pressure-retaining transfer devices are the Pressure Core Analysis and Transfer System developed by Geotek Ltd. in the UK, Pressure Core Characterization Tools developed by Santamarina et al., and Development of HYACE Tools in New Tests on Hydrates.

In this paper, a ship-borne pressure-retaining transfer device for NGH is presented, which has the following abilities: core gripping, pushing, cutting, and subsampling. This study mainly focuses on the performance of the pressure-stabilizing system of the transfer device and the influences of the accumulator's precharge pressure, volume, and other parameters on the system pressure. In addition, AMESim software was used to simulate the pressure-stabilizing system. The simulation results are consistent with the theoretical analysis results. Finally, the prototype of the pressure-stabilizing system and the transfer device were integrated, and the theoretical analysis and simulation results were verified through experiments.

2 | NGH TRANSFER DEVICE

As shown in Figure 1, the NGH transfer device consists of a catching and pushing unit, cutting unit, sonic detection device, and two ball valves. The device can transfer the core from the long gravity-piston pressure-retaining corer at in situ pressure and temperature. A catching and pushing unit is used to grip and drag the core. To grip and release the core, the catcher is opened and closed by ejector pins. The cutting unit, which consists of an oil-filled motor, gear set, core clamp, and blades, can smoothly and quickly cut the core without disturbances at an in situ pressure of 20 MPa.

To maintain the pressure of each cavity stable and the pressure fluctuations below 20% during the operation of the transfer system, a pressure-stabilizing system for the NGH transfer system must be designed.

3 | STUDY ON PRESSURE VARIATIONS OF NGH TRANSFER DEVICE

3.1 | Pressure variations due to back-and-forth movement of catcher

The movement of the catcher in the pressure-retaining cylinder can cause pressure variations. Thus, it is necessary to calculate whether the pressure variations cause pressure fluctuations in the transfer device. The schematic diagram of the catcher movement is shown in Figure 2.

The flow rate for an annular gap flow is expressed as follows:

\[ Q = \pi d_2 \delta \frac{\Delta P}{12 \mu L} \]  

(1)

When the catcher is propelled, the flow rate of the medium in the pressure-retaining cylinder can be calculated as follows:

\[ Q = \frac{\pi d_0 \nu}{4} \]  

(2)

where \( P_1 \) is the inlet pressure, \( P_2 \) the outlet pressure, \( d_0 \) the catcher diameter (73 mm), \( d_1 \) the hole diameter (75 mm), \( L \) the catcher length (1730 mm), \( \mu \) the dynamic viscosity of the medium \( (1 \times 10^{-3} \text{ Pa s}) \), \( \nu \) the catcher speed \( (5.3 \text{ mm/s}) \), and \( \delta \) the gap width \( (1 \text{ mm}) \). Thus, the flow rate of the medium is \( Q = 2.22 \times 10^{-5} \text{ m}^3/\text{s} \). The differential pressure can
be calculated by using the flow rate equation for the annular gap flow and the flow rate of the medium. The result is $\Delta P = 1.96 \times 10^{-2}$ bar.

According to the previous calculation results, it can be concluded that the pressure difference between the front and rear generated by the catcher movement is $1.96 \times 10^{-2}$ bar, which is approximately negligible compared to 200 bar, which the system must maintain. Nevertheless, it should be considered that the sediment in the sampler can block the annular gap between the catcher and cavity channel, thereby increasing the pressure fluctuations when the catcher moves forward. Because half the annular gap between the catcher and cylinder is blocked, and the width of the other half is 0.1 mm, $\Delta P' = 39.2$ bar, and the pressure increases by 19.6%. Hence, when the gap is greater than or equal to 1 mm, the movement of the catcher does not cause significant pressure variations in the cavity. When the gap is below 0.1 mm and half the gap is blocked, the pressure fluctuations increase by more than 20%.

### 3.2 Pressure variations due to leakage

If the sealing surface between the end cover of the NGH transfer device and ball valve or other solids is not well sealed, the gap causes leakage. It is assumed that the leakage exhibits a laminar flow at this time.

When a fluid with a density of $\rho$, dynamic viscosity of $\mu$, and kinematic viscosity of $\nu$ flows through the clearance seal at a height of $h$ and an average velocity of $\bar{u}$, its Reynolds number is defined as follows:

$$ R_e = \frac{2\bar{u}h\rho}{\mu} = \frac{2\bar{u}h}{\nu}. $$

(3)

When the fluid flow is laminar, according to the differential equation for fluid dynamics, the continuity equation of an incompressible fluid with density $\rho$ can be expressed as follows:

$$ \frac{w_r}{r} + \frac{\partial w_r}{\partial r} + \frac{1}{r} \frac{\partial w_\theta}{\partial \theta} + \frac{\partial w_z}{\partial z} = 0, $$

(4)

where $w_r$, $w_\theta$, and $w_z$ are the fluid velocities in the $r$, $\theta$, and $z$ directions (cylindrical coordinates), respectively.

When the fluid is incompressible, and the fluid viscosity is constant, the Navier–Stokes equation of the fluid with density $\rho$ becomes:

$$ \rho \frac{dw_r}{dt} = \rho X - \frac{\partial}{\partial x} + \eta \left( \frac{\partial^2 w_r}{\partial x^2} + \frac{\partial^2 w_r}{\partial y^2} + \frac{\partial^2 w_r}{\partial z^2} \right) $$

(5)

$$ \rho \frac{dw_\theta}{dt} = \rho Y - \frac{\partial}{\partial y} + \eta \left( \frac{\partial^2 w_\theta}{\partial x^2} + \frac{\partial^2 w_\theta}{\partial y^2} + \frac{\partial^2 w_\theta}{\partial z^2} \right) $$

$$ \rho \frac{dw_z}{dt} = \rho Z - \frac{\partial}{\partial z} + \eta \left( \frac{\partial^2 w_z}{\partial x^2} + \frac{\partial^2 w_z}{\partial y^2} + \frac{\partial^2 w_z}{\partial z^2} \right) $$

The leakage of fluid through the annular crack of the sealing ring can be considered a steady laminar flow of an incompressible fluid between two fixed, two-dimensional, parallel plates. As shown in Figure 3, the $x$-axis is the centerline of the flow channel, the pressure at the inlet of the flow channel is $P_m$, the pressure at the outlet is $P_n$, and the length, width, and height of the flow channel are $l$, $b$, and $h$, respectively.

According to the Continuity Equation (4) and the Navier–Stokes Equation (5) for an incompressible flow, the volume leakage rate $Lv$ can be obtained as follows:

$$ Lv = \frac{b \left( P_m - P_n \right)}{\mu} \int_0^l \left( \frac{h^2}{4} - y^2 \right) dy = \frac{bh^3 \left( P_m - P_n \right)}{12\mu}. $$

(6)

For the pressure-retaining transfer device designed in this study, $b = \pi d = 628$ mm, $\mu = 1 \times 10^{-3}$ Pa·s, $P_m = 200$ bar, $P_n = 1$ bar, $l = 5$ mm, and $h = 0.01$ mm. By substituting these parameters into Equation (6), the following is obtained: When the pressure inside the pressure-retaining transfer device is 200 bar, the leakage amount of a single end cover is $Lv = 12$ L/min. Owing to the high pressure in the device, serious leakage occurs as long as there is a gap of 0.01 mm owing to insufficient sealing.

The elastic modulus of a liquid volume can be obtained as follows:

$$ K = -\frac{\Delta PV_{m0}}{\Delta V}, $$

(7)

where $\Delta V$ is the leakage amount of the seawater ($\Delta V = 2 \int_0^l Lv dt$), $\Delta P$ the variations in the seawater pressure ($\Delta P = P_m - P_{ml}$), and $V_{m0}$ the initial volume of the seawater. 

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**FIGURE 2** Schematic diagram of catcher movement

**FIGURE 3** Laminar flow between two parallel plates
The initial pressure inside the pressure-retaining transfer device, $P_{m0}$, the initial volume of the seawater inside the pressure-retaining transfer device, $V_{m0}$, and $K$ the volume elastic modulus of seawater at 20°C. Substituting Equation (6) into Equation (7) leads to:

$$P_m = \frac{6l \mu V_{m0} P_{n0} + Kbh^3 P_{nt} t}{6l \mu V_{m0} + Kbh^3 t}.$$  \hspace{1cm} (8)

The pressure decrease caused by the device leakage was plotted with MATLAB and is shown in Figure 4.

As shown in Figure 4, at a pressure of 200 bar and seal clearance of only 0.01 mm, the pressure in the pressure-retaining transfer device decreases rapidly. In addition, the pressure relief speed gradually decelerates with decreasing pressure in the device. After 100 seconds, the pressure in the cavity decreases to 55 bar. When the clearance is 0.02 mm or 0.03 mm, the pressure leakage speed is faster. When leakage occurs in the device, the pressure inside the device decreases rapidly, and the sample decomposes if the in situ pressure is lost. Therefore, when designing a pressure-stabilizing system, the system must have a sufficient flow compensation capacity for possible leakages.

### 3.3 Pressure variations due to opening and closing of ball valve

The influence of the ball valve opening and closing on the pressure in the transfer device has two aspects. First, during the opening and closing process, the structural characteristics of the ball valve change the volume of the accommodated liquid. Because the liquid has a certain compressibility, the change in the liquid volume leads to pressure fluctuations. Second, a certain pressure difference occurs at both ends when the ball valve is closed. When the ball valve is suddenly reopened, the pressure differences at both ends lead to pressure fluctuations in the device.

The selected inner diameter of the ball valve is $d_b = 75$ mm, and the ball length is $l_b = 106$ mm. Thus, the variations in the liquid volume in the opening and closing process of the ball valve can be expressed as follows:

$$\Delta V_b = \frac{\pi d_b^2 l_b}{4} = 0.47L.$$  \hspace{1cm} (9)

Substituting these parameters into Equation (7) leads to $P_b = 78.3$ bar.

The variation in the pressure is 39.2%, which is well above the design target of 20%. Thus, the design of the pressure-stabilizing system must be adjusted to reduce pressure fluctuations caused by the opening and closing of the ball valve.

### 4 Design of Pressure-Stabilizing System

The design of the pressure-stabilizing system is a hydraulic system with a hydraulic pump and an accumulator for the inner-pressure control. The accumulator reduces the response time of the pressure fluctuations, while the hydraulic pump compensates for the leakage. The pressure-stabilizing system is shown in Figure 5. A manual pump is used to pressurize the top of the pressure-retaining sampler. The exhaust outlets 1 and 2 are used to discharge the air inside the transfer device. Ball valve 1 connects the sampler top to the cavity. Ball valve 2 is used to control the on–off of pipeline between the high-pressure pump and transfer device. Ball valve 3 is used to control the on–off of pipeline connected to the cavity between ball valves 5 and 6. When ball valve 4 is closed, the pressure in the pressure-retaining sampler can be maintained, when ball valve 4 is open, it connects the sampler chamber to the other pipes of the pressure-stabilizing system to maintain a stable pressure. Ball valves 5 and 6 are used to control the on–off between the sampler and...
transfer device. The high-pressure pump is used to provide a dynamic source for the pressure-stabilizing system.

5 | ESTABLISHMENT OF MATHEMATICAL MODEL FOR ACCUMULATOR

When establishing the mathematical model, the accumulator can be simplified to a system consisting of three parts: air cavity, liquid cavity, and connection pipeline. First, the force acting on each part is analyzed to establish the mathematical model. Subsequently, these models are combined to obtain the complete accumulator model, as shown in Figure 6.

The pressure, volume, and temperature are important parameters of an ideal gas and describe the gas state completely. In general, air is considered an ideal gas. To simplify the calculation, the nitrogen in the accumulator was considered an ideal gas in this study. The state equation of an ideal gas is as follows:

\[ \frac{PV^k}{T} = R, \]

where \( P \) is the absolute pressure of the gas, \( V \) the gas volume, \( k \) the adiabatic exponent (1.4), \( T \) the thermodynamic temperature of the gas, and \( R \) the gas constant.

The nitrogen in the accumulator is subjected to the pressure of the liquid in the liquid cavity and mainly moves in the axial direction. This behavior can be simplified to a spring-damping system, as shown in Figure 7. According to the force of the air cavity, the force equation is as follows:

\[ (P_b - P_a) A_{ac} = k_a V_a + c_a \frac{1}{A_{ac}} \frac{dV_a}{dt}, \]

where \( P_b \) is the oil pressure in the liquid cavity of the accumulator and \( P_a, k_a, \) and \( C_a \) the pressure, stiffness coefficient, and damping coefficient of the gas in the bladder.

The equation of the gas damping coefficient is as follows:

\[ C_a = 8\pi \mu_a l_a, \]

where \( \mu_a \) the gas viscosity coefficient and \( l_a \) the gas cavity length.

Based on Equation (10), the following equation can be obtained:

\[ P_{a0} V_{a0}^k = P_a V_a^k, \]

where \( P_{a0} \) is the initial pressure, \( V_{a0} \) the initial gas cavity volume of the accumulator, \( P_a \) the gas cavity pressure of the accumulator at any moment, and \( V_a \) the volume of the gas in the bladder at any moment.

\[ k_a = -\frac{\Delta F}{\Delta x} = \frac{\Delta P}{\Delta V} = -\frac{1}{V_a} \frac{dP}{dV} = A_{ac}^{-1} \frac{kP_{a0} V_{a0}^k}{V_a^{k+1}}. \]

Taking the differential of the right end of Equation (13) with respect to \((P_{a0}, V_{a0})\) leads to:

\[ \begin{cases} \frac{dP_a}{dt} V_{a0}^k + \frac{dV_a}{dt} kP_{a0} V_{a0}^{k-1} = 0 \\ \frac{dP_a}{dt} V_{a0}^k + \frac{dV_a}{dt} kP_{a0} V_{a0}^{k-1} = 0 \end{cases} \]

By not considering the compressibility of the oil and assuming that the oil inlet flow rate of the accumulator is \( q \), the relationship between the oil inlet flow rate and volume variation of the gas cavity becomes \( q_a = -\frac{dV_a}{dt} \). The negative sign indicates that the volume variation of the gas cavity is contrary to the oil flow change.

\[ q_a = -\frac{dV_a}{dt}. \]

Substituting Equation (16) into Equation (15) and using the Laplace transform lead to:

\[ P_a(s) = \frac{kP_{a0}}{V_{a0}} V_a(s). \]
Furthermore, substituting Equation (16) into Equation (11) and using the Laplace transform lead to:

$$[P_b(s)-P_a(s)]A_{ac} = -\left(\frac{k_a}{A_{ac}s} + \frac{c_a}{A_{ac}}\right)Q_a(s).$$  \hspace{1cm} (18)

Equation (18) reflects the relationship between the gas cavity pressure of the accumulator, gas cavity volume, and system flow rate.

Because the compressibility of a liquid is much lower than that of a gas, the compressibility of the liquid can be neglected in the accumulator model. Thus, it is considered an incompressible liquid. The force model of the liquid cavity is shown in Figure 8. The following equation can be derived without considering the elasticity of the accumulator tube wall or the circumferential motion of the fluid.

The force equation of the liquid cavity is as follows:

$$\left( P_1 - P_b \right)A_{ac} = m_b \frac{d^2V_a}{dt^2} + B_b \frac{dV_a}{dt},$$ \hspace{1cm} (19)

where $P_1$ is the accumulator inlet pressure, $m_b$ the fluid mass of the liquid cavity ($m_b = \rho V_b$), $\rho$ the liquid density, $V_b$ the gas volume in the bladder at any time, $B_b$ the viscous damping of the oil in the liquid cavity ($B_b = 8\pi\mu_b l_b$), $\mu_b$ the dynamic viscosity coefficient of the oil, and $l_b$ the length of the liquid cavity.

A schematic of the accumulator inlet pipeline is shown in Figure 9.

By analyzing section BC of the pipeline, its force balance equation is obtained:

$$P_{1m} - P_{1n} = L_1 \frac{dQ_a}{dt} + R_1Q_a,$$ \hspace{1cm} (20)

where $L_1$ is the fluid sensitivity ($L_1 = \frac{m_1}{A_1^2} \frac{dV_a}{dt}$), $R_1$ the fluidic resistance ($R_1 = \frac{128\mu_1}{\pi d_1^3}$), $l_1$ the length, and $d_1$ the inner diameter of the pipeline in section BC.

Similarly, the force balance equation of section AB can be obtained as follows:

$$P_2 - P_{2m} = L_2 \frac{dQ_a}{dt} + R_2Q_a,$$ \hspace{1cm} (21)

where $L_2$ and $R_2$ are the fluid sensitivity and fluidic resistance of the pipeline in section AB, respectively. They can be calculated like $L_1$ and $R_1$.

For the cross-sectional CD of the pipeline, the following equation is used:

$$P_{1n} - P_1 = \xi_m \nu_1^2/2,$$ \hspace{1cm} (22)

where $\xi_m$ is the local resistance coefficient and $\nu_1$ the flow velocity of the oil in cross-sectional CD.

Similarly, the following equation is used for cross-sectional BE:

$$P_{2m} - P_{1m} = \xi_m \nu_2^2/2,$$ \hspace{1cm} (23)

where $\xi_m$ is the local resistance coefficient and $\nu_2$ the flow velocity of the oil in cross-sectional BE.

The following equation can be obtained based on Equation (11), (19), (20), (21), (22), and (23):

$$P_2 - P_a = L_2 \frac{dQ_a}{dt} + R_2Q_a + L_1 \frac{dQ_a}{dt} + R_1Q_a + L_b \frac{dQ_a}{dt} + R_bQ_a + k_a \frac{dV_a}{dt} + R_aQ_a + \xi \nu_2^2/2 + \xi \nu_1^2/2.$$ \hspace{1cm} (24)

The mathematical model of the accumulator can be obtained with a Laplace transform of Equation (24):

$$G_a(s) = \frac{Q_a(s)}{P_a(s)} = \frac{A_{ac}}{k_a + \frac{k_p a^2}{V_a} \frac{s^2}{\omega_{n2}^2} + 2\xi \omega_{n2} s + 1},$$ \hspace{1cm} (25)

where $\omega_{n2}$ is the undamped natural frequency ($\omega_{n2} = \sqrt{\frac{k}{m}}$), $k_x$ the equivalent spring stiffness ($k_x = \frac{k_p a^2}{V_a} A_{ac}^2 + k_b$), $m$ the
equivalent mass \( m_e = m_b + m_1 \left( \frac{A_e}{A_1} \right)^2 + m_2 \left( \frac{A_e}{A_2} \right)^2 \), \( \zeta_2 \) the equivalent damping ratio \( \left( \frac{C_e}{2m_e\omega_n} \right) \), and \( C_e \) the equivalent damping coefficient of the accumulator \( \left( C_e = 8\pi \mu_b \left( l_b + l_1 \left( \frac{A_e}{A_1} \right)^2 + l_2 \left( \frac{A_e}{A_2} \right)^2 \right) + 8\pi \mu_d f_b \) \).

According to Equation (25), the mathematical model of the accumulator is a second-order oscillation system. The transfer function is mainly related to the accumulator prefilling pressure, volume, and steady-state working pressure of the system.

The previous content mainly covers only the modeling of the accumulator, whereas the pressure-stabilizing system models the loop with accumulator and its performance. A common accumulator circuit, which absorbs pressure fluctuations, is shown in Figure 10.

In this study, the pressure variation process was considered an adiabatic process, and the adiabatic exponent \( k \) was 1.4.

The throttle flow equation is expressed as follows:

\[
q_0 = q - q_R, \quad (26)
\]

\[
q_R = K_R \sqrt{P_2 - P_s}, \quad (27)
\]

where \( q_a \) is the flow into the accumulator, \( q_R \) the flow through the throttle valve, \( K_R \) the flow coefficient of the throttle valve, \( P_2 \) the pressure at the accumulator inlet (ie, the system pressure), and \( P_s \) the outlet pressure of the throttle valve.

When the system is in a steady state, there are no flow pulsation and pressure pulsation in the pipeline:

\[
P_{20} = P_{a0}, \quad (28)
\]

\[
q_0 = q_{R0}, \quad (29)
\]

where \( P_{20} \) is the steady-state pressure at the accumulator inlet (ie, the steady-state pressure of the system), \( P_{a0} \) the steady-state pressure of the accumulator bladder, \( q_0 \) the steady-state flow of the system, and \( q_{R0} \) the steady-state flow through the throttle valve.

Taking the derivative of Equation (27) with respect to \( P_{20}, q_{R0} \) and substituting Equation (29) into the new equation lead to:

\[
\begin{align*}
\frac{dq_a}{dt} &= \frac{K_a}{2(P_{20} - P_s)} \frac{dP_{20}}{dt} - \frac{d\omega^2}{dt} \\
\frac{dq_R}{dt} &= \frac{1}{2(P_{20} - P_s)} \frac{dP_{20}}{dt} 
\end{align*}
\]

The following equation can be obtained by applying the Laplace transform to Equations (30) and (24):

\[
G_3(s) = \frac{P_2(s)}{Q(s)} = \frac{2(P_{20} - P_s)}{g_0} \frac{s^2 + 2\zeta_2\omega_n s + \omega_n^2}{s^2 + \left(2\zeta_2\omega_n + K\frac{A_e^2}{A_0}\right)s + \omega_n^2},
\]

where \( \omega_n \) and \( K \) are intermediate coefficients: \( K = \frac{\Delta_2^2}{k_e + \frac{\Delta_2^2}{\gamma_{m0}}} \).

By substituting \( s = j\omega \) into Equation (31), the frequency characteristic of the system can be obtained:

\[
G_3(j\omega) = \frac{P_2(j\omega)}{Q(j\omega)} = \frac{2(P_{20} - P_s)}{g_0} \frac{-\omega^2 + 2\zeta_2\omega_n(j\omega) + \omega_n^2}{-\omega^2 + \left(2\zeta_2\omega_n + K\frac{\omega_n^2}{\gamma_{m0}}\right)(j\omega) + \omega_n^2}.
\]

6 | STUDY ON CHARACTERISTICS OF PRESSURE-RETAINING SYSTEM BASED ON AMESIM

6.1 | Modeling of pressure-retaining system

The scheme of the previously mentioned pressure-stabilizing system is shown in Figure 11. The sampler and NGH
transfer device are represented by a structure connected to a spring, mass block, piston, and hydraulic cavity in the simulation model. The hydraulic components are connected according to the pipeline design of the pressure-stabilizing system, and the on–off of pipeline is controlled through variable current limiter and piecewise linear signal source components.

The specific system parameters are listed in Table 1.

## 6.2 Study of effect of accumulator on system pressure

### 6.2.1 Influence of accumulator precharge pressure on system pressure

By using the batch processing function of the AMESim software, the precharge pressure of the accumulator was set as batch processing parameter. Pressures of 80, 120, 160 (base pressure), 200, and 240 bar were applied. Next, the precharge pressure was increased from 80 to 240 bar. The simulation results are shown in Figure 12.

According to Figure 12, the accumulator can effectively reduce the peak pressure of the system under the pressure impact. When the precharge pressure is 200 bar, 23% of the added value of the peak pressure can be effectively reduced. When the accumulator precharge pressure is below the stable working pressure of 200 bar of the system, the ability of reducing the pressure fluctuation amplitude enhances with increasing accumulator precharge pressure. Moreover, the response speed of the accumulator becomes faster. When the precharge pressure of the accumulator is greater than the stable working pressure of 200 bar of the system, the accumulator's ability to reduce the amplitude of the pressure fluctuations degrades because the accumulator only absorbs the peak part of the pressure fluctuations. The frequency of the pressure fluctuation is similar to that without an accumulator. The optimal precharge pressure of the accumulator is 0.8-0.9 times the working pressure of the system.

### 6.2.2 Influence of accumulator volume on system pressure

According to the previously presented analysis of the mathematical model for the accumulator, the natural frequency of the accumulator decreases when the accumulator volume \( V_0 \) is increased, which weakens the ability of the accumulator to absorb pulsation. In the AMESim simulations, the accumulator volumes were set to 2.5, 6.3, 10, and 16 L. The simulation results are shown in Figure 13.

According to the curve, when the pressure fluctuation source and accumulator precharge pressure are constant, the pressure peak in the system decreases gradually with increasing accumulator volume. When the accumulator volume is 16 L, the

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### TABLE 1 Simulation parameters

| Parameter                        | Value      |
|----------------------------------|------------|
| Motor speed                      | 168 g      |
| Opening pressure of overflow valve | 200 bar    |
| Mass block                       | 20 kg      |
| Spring coefficient of elasticity | 1000 N/m   |
| Piston diameter of cylinder      | 100 mm     |
| Diameter of piston rod           | 60 mm      |
| Chamber volume at upper end of sampler | 1 L    |

---

**FIGURE 12** Pressure curves of chamber 2 for different precharge pressures of accumulator (chamber 2 represents the NGH transfer device)

**FIGURE 13** Pressure curves of chamber 2 for different accumulator volumes
added value of the peak pressure decreases by 35.9%. However, when the volume increases to 6.3 L, the volume continues to increase, which has no significant impact on the pressure reduction of the loop. Moreover, the larger the accumulator volume, the slower the response speed of the loop. Therefore, the accumulator volume should be sufficiently large (but not too much) to compensate for the fluctuations of the loop flow.

6.2.3 Influence of accumulator inlet diameter on system pressure

The inlet diameter of the accumulator can be regarded as a current-limiting hole. Thus, the size of the inlet diameter also affects the absorption effect of the accumulator on the pressure fluctuations, thereby affecting the loop pressure. The specific curve is shown in Figure 14.

The inlet diameter of the accumulator has no effect on the pressure of the NGH transfer device at the injection stage (0 bar). However, the smaller the inlet diameter of the accumulator at the charging stage, the faster the charging speed. When the inlet diameter of the accumulator increases to 6 mm, the charging speed of the NGH transfer device is slightly reduced. This is because when the accumulator inlet area increases to a certain extent, the change in the inlet area does not significantly change the flow rate in the NGH transfer device.

6.3 Study of effect of pipeline on system pressure

6.3.1 Selection of hydraulic pipeline model

The AMESim software includes three hydraulic pipeline models: HL000, HL03, and HL09. The HL000 model only considers the compressibility of the pipelines. Thus, the deformations of the pipelines at high pressure are calculated with the Young modulus of the pipe wall material and thickness of the pipe wall; the pressures at the inlet and outlet of pipelines are equal. The HL03 model considers the friction in the pipeline and calculates the pressure loss of the pipeline through the relative roughness of the pipe wall and Reynolds number. Hence, the pressures at the inlet and outlet are different. The HL09 model considers the inertia of the fluid to simulate the pressure wave in the pipeline. When the parameter settings are equal, the HL000, HL03, and HL09 models are selected to simulate the pressure change in the loop. The results are shown in Figure 15.

With HL03 and HL09, the added value of the system peak pressure is 26.5% smaller than with HL000. Moreover, because HL03 and HL09 consider the friction factor of the pipelines, the decrease in the speed of the pressure fluctuation amplitude is significantly accelerated. In this example, the curves simulated with HL03 and HL09 basically coincide. However, the pressure wave in the pipeline is not the key factor of this study. Moreover, the simulation results prove that the pressure wave is not significant. Therefore, HL03 was selected for the subsequent simulations.

6.3.2 Influence of pipeline inner diameter on system pressure

The pressure curve in cavity 2 for specific pipelines with different inner diameters is shown in Figure 16. For a large inner pipeline diameter, the pressure reduction effect is improved. However, when the inner diameter of the pipeline increases to 8 mm or 10 mm, the pressure fluctuation curve
remains approximately constant. Hence, when the inner diameter of the pipeline increases to a certain extent, it has little impact on the pressure in the loop. In addition, when the inner diameters of the pipelines are 8 and 10 mm, the added values of the peak pressure of the system are 28% and 30% higher than those for an inner diameter of 6 mm, respectively. Consequently, the best inner diameter for the pipeline is the inlet diameter of the accumulator.

6.3.3 Influence of pipeline length on system pressure

The pressure curve in chamber 2 for pipelines with different lengths is shown in Figure 17. According to the curve, when the pipeline lengths are 3 and 5 m, the pressure change curve in chamber 2 remains approximately equal. Hence, the pipeline length has little influence on the pressure fluctuation. When the pipeline lengths are 3 and 5 m, the peak pressures increase by 7.7% and 9.6% compared with that of a pipeline length of 1 m, respectively. The increase is approximately negligible. Although the pipeline length has little effect on the pressure reduction, the longer the pipeline, the higher the peak pressure. To maintain a stable pressure in the NGH transfer device, the length of the pipeline should be shortened as much as possible in the actual design.

6.4 Influence of NGH transfer device volume on system pressure

The volume of the NGH transfer device provides space for core cutting and pushing. In addition, it can affect the pressure of the system. In this study, the dead volume of chamber 2 was modified to represent different NGH transfer devices with different volumes. The simulation results are shown in Figure 18. With decreasing volume transfer device, less time is required to obtain a pressurized system. This is because the ratio of the pressure variation and bulk modulus is equal to the ratio of the changed volume and the ratio of the original volume (according to the equation of the liquid bulk modulus). Thus, when the volume is small, a pressurization to 200 bar requires less time. The four pressure curves of chamber 2 are similar, and the final stable pressure is 200 bar. Hence, the change in the volume of the transfer device does not change the final stable pressure value. Therefore, the volume can be chosen according to the requirements of the transfer structure in the actual design.

7 Prototype integration and experiments

To verify the effectiveness of the pressure-stabilizing system, a principle prototype was integrated, as shown in Figure 19.
The pressure-stabilizing system includes a working platform for the installation of various hydraulic components, a manual pressure pump is used to pressurize slowly the NGH transfer device and to cause the sample tube to move, the manual pump is produced by Unipac Technology Company, the model is u-jb-3, and the maximum working pressure is 800 bar; a three-plunger pump with a manual overflow valve is used to create rapidly a high-pressure environment inside the transfer device, and the model of high-pressure pump is WJ15.08-2.2-6, with a flow rate of 4 L/min and a maximum working pressure of 250 bar; an accumulator controls the high-pressure environment in the NGH transfer device and reduces the pressure fluctuations in the system; two gauges are used to display the pressure in the transfer device in real time, monitor the pressure change, and prevent too high pressures, and the model of pressure gauge is Y-101A, with a measuring range of 0-250 bar; a pressure sensor is used to record the pressure changes in the system, the pressure sensor is a 522 series sensor manufactured by Huba Control company, which model is 522.9 K3S0, the measuring range is 0-250 bar, the output signal is 4-20 mA current, the working voltage is 7-33 V, and the response time is <2 ms; moreover, three corrosion-resistant ball valves are used to control the on–off of pipelines.

7.1 Test on pressurization ability of pressure maintenance system

The primary function of the pressure-stabilizing system is to create a stable high-pressure environment for the NGH transfer device. Therefore, after the integration of the prototype, the pressurization capacity is tested first, including whether the plunger pump can be pressurized to 200 bar and whether the manual overflow valve can effectively control the pressure in the loop. The loop pressure data collected during the tests are shown in Figure 20.

Based on the test data, the pressure-stabilizing system can be pressurized to 200 bar, and the system pressure can be maintained at the set value. In the actual operation, the noise and vibration are negligible, which satisfies the work requirements.

7.2 Experiments on pressure-stabilizing system for pressure variation reduction

The impact of the catcher movement on the system pressure is shown in Figure 21. According to the pressure curve, in the grasping process, the chamber pressure of the NGH transfer device remains at 201.2 bar. When the catcher starts moving, the system pressure is reduced by approximately 1 bar. When the catcher stops moving, the system pressure returns to the

FIGURE 18 Pressure curves of chamber 2 for different NGH transfer device volumes

FIGURE 19 NGH transfer device and pressure-stabilizing system

FIGURE 20 Test curves of pressurization ability of pressure-stabilizing system
original value. This is because the flow of the liquid medium in the device slightly affects the reading ability of the pressure sensor.

In the transfer process of the NGH sample, the opening and closing of the ball valve change the system volume, which alters the pressure in the transfer device, thereby affecting the NGH stability. During the test, the precharge pressure of the accumulator was set to 120 and 180 bar, respectively. The pressure changes in the transfer device caused by the opening or closing of the ball valve are shown in Figure 22. When the ball valve is open, there is a gap between the valve seat and ball, and the space is filled with air. When the ball valve is closed, the gap between the ball and valve seat is filled with liquid. Thus, the liquid volume expands, and the pressure is reduced. Without the accumulator, the pressure decreases from 215 to 190 bar. With the accumulator, the pressure decreases rapidly, then increases rapidly, and finally reaches a stable state. Compared to the accumulator with a precharge pressure of 120 bar, the accumulator with a precharge pressure of 180 bar can compensate more quickly for the pressure impact caused by the opening or closing of the ball valve. In addition, the pressure in the stable state is similar to the original value, which verifies the simulation results.

Moreover, the accumulator with a precharge pressure of 180 bar was connected to the transfer device through pipelines of 0.5 and 5 m, respectively, and ball valve opening and closing tests were conducted. The pressure change in the transfer device is shown in Figure 23. When the length of the inlet pipeline of the accumulator increases from 0.5 to 5 m, the minimal pressure in the transfer device decreases from 162 to 143 bar, and the pressure decreases by more than 20%. The increase in the pipeline length also leads to a slower pressure recovery. For pipeline lengths of 0.5 and 5 m, it takes 2.5 and 5 seconds to restore the pressure to 175 bar, respectively. The final pressures are similar, thereby indicating that the length of the inlet pipeline does not affect the pressure in the final transfer device. The experiments prove that a shorter inlet pipeline of the accumulator leads to a faster response to and adsorption of the pressure fluctuations. These results agree well with the previously presented simulation results.

8 | CONCLUSIONS

In this paper, a pressure-stabilizing system for the transfer of NGH was presented. This hydraulic system controls the internal pressure of the transfer device through a hydraulic pump and an accumulator. The hydraulic system can respond quickly and sensitively to pressure changes in the transfer device. The accumulator compensates for leakage in the system with its
stored energy, therefore maintaining a high-pressure environment in the transfer device for a long time by reducing the pressure impact and pulsation. Moreover, a hydraulic pump improves the ability of the loop to compensate for leakage.

The performance of the pressure-stabilizing system was analyzed theoretically, and its working state was simulated with AMESim software. The theoretical analysis and simulation results show that the precharge pressure of the accumulator should be 0.8-0.9 times the working pressure. Moreover, the accumulator volume should be selected reasonably for the accumulator design because an increased accumulator volume decelerates the response of the system. Once the integration of the pressure-stabilizing system prototype was completed, the pressurization capacity of the prototype was tested, and experiments on how to reduce effectively pressure fluctuations in the NGH transfer device were conducted. The experimental results verify the theoretical analysis and simulation results. The operating pressure can reach 20 MPa when the fluctuations of the inner pressure are below 20% during the transfer process of the NGH sample.

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