Parametric Analysis of the I-Stage Structure for a Dual-Stage Gas Pressure Reducing Regulator

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ABSTRACT Gas pressure regulator is an essential component using for the pressurized system in the aircraft. In our paper, we aim to analyze the impact of structural parameters on output pressure for the I-stage structure of a dual-stage gas pressure reducing regulator. Initially, a numerical simulation of the regulator was established and verified by a comparison of dynamic response from the deflation start of the vessel to the deflation complete. Moreover, parametric analysis of the I-stage structure for the regulator was examined to determine the primary and secondary variables and interdependencies with the Box-Behnken design method applied. Furthermore, a multi-objective optimization based on regression analysis was adopted by using the MOGA-II algorithm method, and a Pareto frontier was obtained. Results indicate that the spool mass, the leakage area of spool seal, and their interaction are the significant factors on overshoot. The overshoot presents a trend of decrease first and then increase with the mass and leakage area increase. The spool mass, the mainspring stiffness, the leakage area of spool seal, and their interactions are influential factors on stability. The stability improves with the spool mass decrease, and other factors increase. Besides, the feedback hole area has a small effect on stability. Moreover, the Pareto of the optimization indicates that the performance of the I-stage structure would be optimal when the spool mass is 23.94 g, the feedback hole area is 89.94 mm², the mainspring stiffness is 166.76N/mm, and the leakage area of spool seal is 0.06 mm².

INDEX TERMS Dual-stage gas pressure reducing regulator, overshoot and stability, box-Behnken design, parametric analysis, multi-objective optimization.

I. INTRODUCTION
Extensive application in aircraft propulsion systems, the pressurized system is a crucial part to guarantee the propellant supplying to the engine pump [1], [2]. As essential components of the pressurized system, gas pressure regulators are used to delivering the pressurized gas into the propellant tank with designed pressure. However, the large number of variables and nonlinearities burdens associated with the regulator is reflected in the fact that instability problems are challenging to solve [3]. For a long time, researchers have been engaged in studies related to improving the stability of the regulator [4]–[6].

Conventionally, plunger structure is applied in pressure reducing regulator. High-pressure gas passes through the variable orifice formed by the valve rod and the plunger cavity. In this way, pressure energy is converted into kinetic energy and loss in friction, making output gas satisfying required pressure. Initially, a single-stage structure was widely applied in the regulator. Stability studies typically employed both failure investigation and redesign. For example, Hurlbert and Abe [7] investigated and resolved an outlet pressure instability problem that had occurred in the space shuttle helium pressure regulator. Barbarits [8] illustrated the continuous improvements and spin-offs of Moog Inc. Xenon Pressure Regulator.

Although the single-stage regulator is acceptable, it is likely to result in suboptimal performance once intensive vibration occurred. Piloted regulators are chosen to address dynamic instability issues. A test facility for the MSL regulators was modified to conduct random vibration and shock environment tests, confirming that the slam start and shock
did not negatively impact regulator performance [9]. Sekita and his team [10] traced and solved the pressure oscillation problem generated by the natural frequency of the pressure regulator and the GHe line existence in the pressurized system.

Numerical simulation was adopted to shorten research time and reduce experimental costs as another research method besides experiment [11], [12]. Exemplification and comparison of four categories on sampling methods for dynamic simulation of pressure regulators were given, and the probability distributions and the sensitivity analysis under different methods were obtained [13]. A generalized dynamic model was developed to describe the behavior of a dome-loaded pressure regulator [14]. Zafer and Luecke [15] developed a comprehensive dynamical model for a gas pressure regulator to understand its behavior, which used a linearized version of the model to investigate the effects of parameter variations by using classical root-locus techniques.

Since the vast application fields and similar functions, the researches in various fields can be referred for the study of the regulators in air vehicles. The factors which had the most significant impact on stability were shown with the modeling of a pressure regulator [16]. Three typical faults of gas regulators were classified with a fault diagnosis method adopted combining a complete ensemble empirical mode decomposition with adaptive noise (CEEMDAN) and fuzzy c-means (FCM) clustering [17]. The gas temperature drop was estimated through a mathematical model by using a thermodynamic approach for a direct-acting pressure regulator [18]. Pressure field, velocity field, and energy consumption were studied on a new high multi-stage pressure reducing valve (HMSPRV). The pressure reducing and velocity increasing gradients were reflected mainly at throttling components for valve opening [19].

The majority of researchers have focused on improving the performance of the pressurized system in aerospace flight vehicles with high-performance aircraft theory proposed in recent years [20]–[22]. Matsumoto et al. [23] proposed a lighter and simpler than traditional gas pressure feed systems using in liquid rocket engines. The concept and mathematical models were established and verified with experiment results. Zaberchik et al. [24] designed, qualified, and integrated the cold gas propulsion system, which applied to the Adelis-SAMSON nano-satellite. Marotta Ireland’s Mechanical Pressure Regulator (MPR) was developed for the Xenon Feed System on the Alphabus Platform, which could be easily modified for use in both chemical propulsion systems and cold gas propulsion systems [25].

A dual-Stage gas pressure reducing regulator (DSGPRR) was considered to meet ambitious performance requirements for a next-generation pressurized system. The study demonstrated that the stability of the DSGPRR was directly related to the II-stage structural parameters [26]. Therefore, studies on the I-stage structural parameters are often ignored by most researchers. However, the I-stage structure plays a crucial role in the DSGPRR. The performance of the I-stage structure is vital to a steep pressure reduction in the DSGPRR. Meanwhile, improving the I-stage structure performance could effectively decrease the burden of the II-stage structure, which can enhance the general performance and extend the service life of the DSGPRR. Furthermore, the numerical simulation of the DSGPRR becomes necessary limited by the small volume and complex structure.

Therefore, this study aims to improve the performance of the I-stage structure in the DSGPRR. The numerical simulation model is established and verified by the test results. Then, the influence of structural parameters on output pressure characteristics of the I-stage structure in the DSGPRR is investigated by using the response surface method (RSM). Finally, the optimal structural parameters are obtained to reduce the overshoot and improve the stability of output pressure through the MOSA-II method.

II. SIMULATION MODEL AND METHOD

A. ESTABLISHMENT SIMULATION MODEL AND VERIFICATION

The DSGPRR consists of two plunger structure valves in series connection, and the schematic cross-section of the DSGPRR and its model is shown in Fig. 1. As an unloading pressure valve with a feedback cavity, the I-stage structure is shown in Fig. 1(a). C1 is the I-stage valve spool, C2 is the I-stage feedback hole, C3 is the I-stage spool seal, C4 is the I-stage mainspring, V1 is the I-stage high-pressure cavity, V2 is the I-stage low-pressure (the II-stage high-pressure) cavity, and V3 is the I-stage feedback cavity. The left part shows the I-stage structure, and a physical sketch is shown in this part. The right part of the figure is part of a simulation model built according to the structural sketch. The valve spool C1 is the moving element of the I-stage structure, and the output pressure is controlled by the position of the valve spool, which is adjusted by the mainspring C4 and the feedback cavity V3. Extra auxiliary spring, deputy cavity, and feedback diaphragm are added for superior stability in the II-stage structure. In Fig. 1(b), C5 is the II-stage auxiliary spring, C6 is the II-stage valve spool, C7 is the II-stage seal, C8 is the II-stage feedback hole, C9 is the II-stage feedback diaphragm, C10 is the II-stage auxiliary cavity, V4 is the II-stage low-pressure cavity, V5 is the II-stage feedback cavity, and V6 is the II-stage deputy feedback cavity.

The valve spools in the I-stage and II-stage structures have adjusted to the reserved position before the DSGPRR working. Affected by high-pressure gas, the I-stage valve closed and immediately opened by the spring force. Pneumatic pressure was decreased sharply via the I-stage structure, and the II-stage structure had excellent stabilization with the pressure further reducing. The dynamic response of output pressure was controlled by springs, volumes, orifices, feedback cavities, and feedback diaphragm.

A gas vessel was used as the gas source in the model connecting with the inlet of the I-stage structure. Considering the actual working condition of the DSGPRR, the air was used as the gas medium, the volume was 12 L, and the internal
pressure was set to 40 MPa. Moreover, a pneumatic orifice was connected with the outlet of the regulator to simulate the gas exhaust process. The ambient pressure at the outlet of the regulator was set to 0.1 MPa and a normal temperature. In summary, a complete sketch of the simulation model in AMESim (SIEMENS, Inc, Berlin, Germany) is shown in Fig. 2.

The dynamic response from the deflation start of the vessel to the deflation complete is investigated to verify the accuracy of the AMESim model. Fig. 3 shows the comparison results of simulation and experiment. Fig. 3(a) is a comparison diagram of the vessel output pressure obtained by the simulation and test. Fig. 3(b) is a comparison of the DSGPRR output pressure. The results show intuitively that little differences exist between simulation and test results, which verify the accuracy of the numerical simulation.

Moreover, the results show that the output pressure of the vessel decreases exponentially with time. The output pressure can be approximately regarded as the step response of a second-order system with feedback, and the adjustment time Δt is within 0.2 s. The output pressure is stabilized at 0.72 MPa after starting adjustment. Owing to the output pressure of the vessel is lower than the experimental output pressure of the regulator after 420 s, the actual output pressure of the regulator will quickly drop to near atmospheric pressure.

Fig. 4 shows the simulation results of the output pressure in the I-stage structure. The change in output pressure is consistent with the design. With the output pressure of the vessel gradually decreases, the output pressure rises from 2.3 MPa to 3.0 MPa, which caused by the presence of supply pressure effect (SPE) in a regulator [27].
The output pressure of the I-stage structure is rapidly attenuated because the vessel output pressure is lower than the normal working pressure at around 380 s. The results are in agreement with Sun et al. [28] and his coworkers, who studied for a typical pressurized system with a novel gas pressure regulator.

B. TEST DESIGN METHOD

In this study, a Box-Behnken design (BBD) was used in AMESim simulation as a statistical tool. In order to improve the performance of the I-stage structure, the influence of structural parameters on outlet pressure was analyzed to find the best parameters with small overshoot and superior stability.

Two parameters of the I-stage structure were selected to calculate the overshoot of the output pressure by equation (1). The peak value was the maximum within 0.2 s, and the steady value was the average value between 2 s to 8 s. The standard deviation of the output pressure between 2 s to 8 s was selected to quantify the stability as a stability response. The larger the standard deviation was, the worse the stability became.

\[
\text{Overshoot} = \frac{\text{Peak Value - Steady Value}}{\text{Steady Value}} \times 100\% \quad (1)
\]

The structure of the DSGPRR selects the investigation parameters. Three horizontal levels of the spool mass, the low-pressure cavity volume, the feedback hole area, the mainspring stiffness, and the leakage area of spool seal are separately set within the allowable range of the production capacity in the I-stage structure.

Consequently, the five-factor three-level response surface design scheme is used for the simulation test. The investigated factors and levels of the test are shown in Table 1.

The run orders of factors and their levels were designed by Design-Expert 8 software (StatEase, Inc, Minnesota, USA). The parameters in the design table were input into the AMESim model, and the simulation results were analyzed to determine the main effects and their interaction. The design orders and results are shown in Table 2.

### TABLE 1. The level of investigated factors.

| Factors                        | Low level (-) | Central level (0) | High level (+) |
|--------------------------------|---------------|-------------------|----------------|
| Spool mass \((X_1)\)          | 20 g          | 40 g              | 60 g           |
| Low-pressure cavity volume \((X_2)\) | 100 mm\(^3\) | 500 mm\(^3\)     | 900 mm\(^3\)  |
| Feedback hole area \((X_3)\)  | 10.27 mm\(^2\) | 50.27 mm\(^2\)  | 90.27 mm\(^2\) |
| Mainspring stiffness \((X_4)\) | 50 N/mm       | 125 N/mm          | 200 N/mm       |
| Leakage area of spool seal \((X_5)\) | 0.01 mm\(^2\) | 0.08 mm\(^2\)   | 0.15 mm\(^2\)  |

III. STUDY ON THE INFLUENCE OF STRUCTURAL PARAMETERS

A. REGRESSION EQUATION

The ANOVA function was adopted to fit the target, and the dynamic range data obtained from the simulation with a second-order model shown in equation (2). The maximum and minimum values of the 95% confidence interval parameter estimation values were averaged as the coefficient of the final regression equation parameter estimation. Finally, the second-order mathematical model was obtained and illustrated in equation (3) to (4).

\[
Y = \beta_0 + \sum_{i=1}^{k} \beta_i X_i + \sum_{i=1}^{k} \beta_{ii} X_i^2 + \sum_{i=1}^{k-1} \sum_{j=i+1}^{k} \beta_{ij} X_i X_j + \varepsilon \quad (2)
\]

where, \(Y\) is the responses (specific surface area); \(X_i\) and \(X_j\) are the independent factors. \(\beta_0\), \(\beta_i\) \((i = 1, 2, \ldots, k)\), \(\beta_{ij} \,(i = 1, 2, \ldots, k; \, j = 1, 2, \ldots, k)\) are unknown parameters and \(\varepsilon\) is an experimental error.

\[
Y_1 = 0.031 - 0.0020X_1 - 0.052X_5 + 0.0024X_1X_5 + 1.37X_5^2 \quad (3)
\]

\[
Y_2 = 0.038 + 0.0015X_1 + 0.00019X_3 - 0.00023X_4 - 0.19X_5 - 0.0000043X_1X_4 - 0.0058X_1X_5 + 0.0016X_4X_5 + 0.52X_5^2 \quad (4)
\]

where, \(Y_1\) and \(Y_2\) represent for the overshoot and the stability response. \(X_1\), \(X_3\), \(X_4\), and \(X_5\) show as actual values of the spool mass, the feedback hole area, the mainspring stiffness, and the leakage area of spool seal, respectively. \(X_1X_4\) is the interaction between the spool mass and the feedback hole area, the mainspring stiffness, and the leakage area of spool seal. \(X_1X_5\) represents the interaction between the spool mass and the leakage area of spool seal, and \(X_4X_5\) is the interaction between the mainspring stiffness and the leakage area of spool seal. Besides, \(X_5^2\) is the square terms of the leakage area of spool seal, respectively.

B. REGRESSION ANALYSIS

In Table 3, the P-value of the overshoot and stability functions are less than 0.0001, which indicates the models are significant. The values of R-Squared, Adj R-Squared, and Pred R-Squared are 0.9450, 0.9009, and 0.7798 in the overshoot model and 0.9641, 0.9354, and 0.8565 in the stability model, respectively, influenced the model to be acceptable. The value of Pred R-Squared is in reasonable agreement with the value...
of Adj R-Squared, showing that the results of the models are of satisfaction. Besides, Fig. 5 shows the results intuitively to verify the predicted values and actual values are reliable in this study.

The significance of each variable ($X_1, X_2, X_3, X_4$, and $X_5$) at 95% confidence is shown in Table 4. The sensitive order of each variable on overshoot is $X_5 > X_1 > X_4 > X_3 > X_2$. The results indicate that the primary items of the spool mass and leakage area of spool seal have a significant effect on overshoot. Furthermore, the interaction item of the spool mass and leakage area of spool seal also shows a significant effect, and the influence order is $X_5 > X_1 > X_1X_5$ (interaction item). Moreover, the sensitive order of each variable on stability response is $X_5 > X_1 > X_4 > X_3 > X_2$. The results show that the primary items of the leakage area of spool seal, spool mass, mainspring stiffness, and feedback hole area are the significant effect on stability, and the interaction items of the spool mass, mainspring stiffness, and leakage area of spool seal also are in the significant effect. The influence order is $X_5 > X_1 > X_4 > X_3 > X_2$. 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FIGURE 5. Comparison of predicted values with actual values.

FIGURE 6. The overshoot affected by the primary item.

Fig. 6 presents the overshoot affected by the primary item in the I-stage structure. The spool mass and leakage area of spool seal are the significant factors to the overshoot. As the spool mass increases, the overshoot presents a trend of decreasing first and then increasing. Deficient mass will cause the movement of the spool unstable, and excessive mass will make the movement of the spool unstable, and excessive mass will make over-inertia. As a result, the overshoot increases. When the spool mass is 38.5 g, the overshoot is the smallest.

Furthermore, the overshoot increases with the leakage area gradually increase. When the leakage area is less than 0.06 mm$^2$, the friction between the plunger rod and the spool seat will explode, resulting in a negative overshoot. Excessive sealing will prolong the adjustment time, aggravate the wear of the spool, and decrease the performance of the DSGPRR. Therefore, the overshoot of the output pressure should be greater than 0, and as close to 0 as possible. When the leakage area is 0.064 mm$^2$, the overshoot is the most suitable.

Moreover, Fig. 7 shows the stability response in the I-stage structure affected by the primary items. In Fig. 7(a), the increase of the mainspring stiffness and leakage area of spool seal will gradually reduce the stability response, which represents an improvement of stabilization. In Fig. 7(b), the increase of the spool mass will reduce the stability of the output pressure, and the change in the feedback hole area has a small effect on stability.

FIGURE 7. The response of stability effected by the primary item.

Fig. 8 shows the response surface plots of the overshoot with the interaction between the spool mass and leakage area of spool seal based on equation (3). The density of the
contour line moving toward the peak along the leakage area of spool seal is greater than the spool mass, indicating that the leakage area of spool seal has a significant effect on the overshoot than the spool mass. When the spool mass and leakage area of spool seal are more than 38.5 g and 0.06 mm$^2$, a strong interaction exists between two variables. When the leakage area of spool seal is of 0.01 mm$^2$ to 0.05 mm$^2$, the overshoot corresponding to the spool mass of 23 g to 53 g is decreasing negative value, indicating a small interaction exists. The values of the spool mass and leakage area of spool seal should be in the deep blue area to ensure small overshoot.

Fig. 9 presents the response surface plots of the stability response based on equation (4) with the interaction between each parameter of the spool mass, mainspring stiffness, and leakage area of spool seal. Fig. 9(a) demonstrates the interaction between the mainspring stiffness and leakage area of spool seal. The leakage area of spool seal has a greater effect on stability than the spring stiffness from the density of the contour line moving. When the spring stiffness decreases from 160 N/mm to 60 N/mm, the stability response corresponding to the leakage area of spool seal of 0.02 mm$^2$, 0.05 mm$^2$, 0.08 mm$^2$, 0.11 mm$^2$, and 0.14 mm$^2$ increases by 30.84%, 27.17%, 22.02%, 15.01%, and 5.77%, respectively. Once the leakage area of spool seal less than 0.11 mm$^2$ and the spring stiffness less than 170 N/mm, the stability response increases sharply with both the decreasing of the spring stiffness and leakage area of spool seal. The effect trend produces obvious changes, indicating that a strong interaction exists between two variables. The values of the spring stiffness and leakage area of spool seal could be appropriately increased to improve the stability of the output pressure, in which the interaction between the two variables can be reduced to ensure the stability response is at a low level.

Moreover, Fig. 9(b) presents the interaction between the spool mass and leakage area of spool seal. The spool mass is opposite to the leakage area of spool seal in the interaction of stability. The stability response increases either the spool
mass increases or the leakage area of spool seal decreases. The increase will be evident once the leakage area of spool seal is less than 0.11 mm$^2$, and the spool mass is more than 30 g. When the leakage area of spool seal decreases from 0.14 mm$^2$ to 0.02 mm$^2$, the stability response corresponding to the spool mass of 60 g, 50 g, 40 g, 30 g, and 20 g increases by 40.44%, 34.47%, 26.89%, 16.75%, and 2.23%, respectively.

Fig. 9(c) presents the same tendency that the interaction between the spool mass and mainspring stiffness. The stability response increases as the spool mass increase until it reached the maximum or the mainspring stiffness decreases until it reached the minimum. When the spool mass increases from 20 g to 60 g, the stability response corresponding to the mainspring stiffness of 60 N/mm, 90 N/mm, 120 N/mm, 150 N/mm, and 180 N/mm increases by 42.99%, 40.13%, 35.53%, 28.91%, and 20.20%, respectively.

The ANOVA results indicate that the spool mass, the feedback hole area, the mainspring stiffness, and the leakage area of spool seal are significant factors to improve the performance of the I-stage structure and the factor of the low-pressure cavity volume is insignificant to both the overshoot and stability response. Therefore, a reasonable choice for structural parameters is essential.

### IV. MULTI-OBJECTIVE OPTIMIZATION BASED ON MOGA-II ALGORITHM

#### A. OPTIMIZATION MODEL

A multi-objective optimization function was defined to decrease the overshoot and improve the stability of the
TABLE 5. Calculation settings for the MOGA-II algorithm.

| Item                                | Value               |
|-------------------------------------|---------------------|
| Number of initial designs           | 50                  |
| DOE scheme                          | RANDOM              |
| Optimization algorithm type         | MOGA-Adaptive Evolution |
| Number of Generations               | 60                  |
| Probability of Directional Cross-Over | 0.5                |
| Probability of Selection            | 0.05                |
| Probability of Mutation             | 0.1                 |
| DNA String Mutation Ratio           | 0.5                 |
| Total number of analyses            | 3000                |

DSGPRR as equation (5) according to equation (3) and equation (4), including the minimum of the overshoot \( f_1(X) \) and the minimum of the stability response \( f_2(X) \) as the optimization target. The spool mass \( X_1 \), the feedback hole area \( X_3 \), the mainspring stiffness \( X_4 \), and the leakage area of spool seal \( X_5 \) as the optimization parameters. Since there is a negative value of the overshoot, the function of the overshoot greater than zero is taken in the optimization process. According to equation (5), the optimization model was built by modeFRONTIER (ESTECO, Inc, Trieste, ITALY), as shown in Fig. 10.

\[
\begin{align*}
\min F(X) &= [f_1(X), f_2(X)] \\
\text{s.t. } f_1(X) &\geq 0 \\
20 \leq X_1 &\leq 60 \\
10.27 \leq X_3 &\leq 90.27 \\
50 \leq X_4 &\leq 200 \\
0.01 \leq X_5 &\leq 0.15
\end{align*}
\]

FIGURE 10. Multi-objective optimization model.

FIGURE 11. Pareto citizens and the selected designs of the optimization.

FIGURE 12. The parallel coordinates chart of the optimization.

random DOE designs were each coordinated with 60 generations runs. Thus, a total number of 3000 runs were carried out. Theoretically, the larger the numbers are, the closer the optimal designs to the real Pareto frontier become.

C. OPTIMIZATION RESULTS

Fig. 11 is the optimization results for the overshoot and stability response. The Pareto citizens are marked with a solid green circle, and the Pareto Frontier are marked with red. We can see most designs are located on the Pareto Frontier, indicating that the MOGA-II algorithm worked as expected and a clear trade-off between the overshoot and stability.

The parallel coordinates chart, as shown in Fig. 12, illustrates that all chosen items as vertical bars with each design represented by a polyline cutting across the bars in the proper values. Most of the best designs in the Pareto frontier have low values for \( X_1 \) between 20 g and 25 g, \( X_5 \) between 0.01 mm² and 0.03 mm², high values for \( X_3 \) between 79.6 mm² and 89.6 mm², \( X_4 \) between 190 N/mm and 200 N/mm. Theoretically, as the number of iterations increases, the results obtained become more and more accurate. We choose the design with the highest number
of iterations in the Pareto frontier as the optimal solution for this optimization.

Eventually, the selected optimal design is given in Table 6. The spool mass is 23.94 g, and the feedback hole area is 89.94 mm², the mainspring stiffness is 166.76 N/mm, and the leakage area of spool seal is 0.06 mm². Compared with the original design, the overshoot and stability response have a significant reduction specifically. The overshoot decreases to 1.7206 × 10⁻³ and the stability response decreases to 0.037265, which 116.13% and 27.27% for the overshoot and stability response decreasing achieved respectively.

### V. CONCLUSION

In this article, a simulation model for analyzing the dynamic characteristics of a dual-stage gas pressure reducing regulator was presented and verified. The output pressure of the I-stage structure in the regulator was deeply investigated from its two factors: overshoot and stability. Determination of the primary and secondary variables and their interdependencies was discovered. The main conclusions are as follows:

1. The structure parameters of the spool mass, the leakage area of spool seal, and their interaction are the significant factors to the overshoot. When the spool mass is 38.5 g, and the leakage area of spool seal is 0.064 mm², the overshoot is the smallest. Furthermore, the interaction will increase once the spool mass and leakage area of spool seal increase or decrease at the same time.

2. The spool mass, the mainspring stiffness, the leakage area of spool seal, and their interaction are the significant factors to the stability. The spring stiffness and the leakage area of spool seal should be appropriately increased with the spool mass decreasing to improve the stability of the output pressure.

3. The optimal design achieves a massive reduction in the overshoot and stability response for the I-stage structure output pressure of the DSGPRR. More specifically, when the spool mass is 23.94 g, the feedback hole area is 89.94 mm², the mainspring stiffness is 166.76 N/mm, and the leakage area of spool seal is 0.06 mm², the overshoot and stability response can reduce by 116.13%, and 27.27%, respectively.

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