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Application of Isogeometric Spline Finite Element Method to the Sealing Performance of High-Pressure Pipe Joint

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Abstract: *Aiming at the problem that the traditional gasket joint can not meet the sealing requirements when the pressure of the high-pressure air system of the industry is raised to 40MPa, an optimal design scheme of the ferrule fittings is proposed. In view of the complex structure, geometric nonlinearity and difficulty in theoretical calculation, a three-dimensional axisymmetric model of the ferrule fitting structure was established by using the nonlinear finite element simulation software, and the effect of the equivalent stress on the inner and outer contact surface of the ferrule front end and the ferrule deformation on the sealing was analyzed during the ferrule assembly and internal pressurization. Then the approximate range of the applied preload force and the approximate range of the maximum pressure resistance of the pipe are determined, and the measures to reduce the stress concentration of the ferrule are given. The results show that the optimized 22mm ferrule fittings have no leakage and obvious deformation in the test of 50MPa air pressure and 60MPa water pressure, which provides a useful reference for the further development of the ferrule fittings of high-pressure air system.*

Keywords: *high-pressure air system; ferrule fittings; seal; testing verification; finite element simulation*

1. Introduction

For the 40MPa high-pressure air system, the sealing performance is very important, which is directly related to the vitality of the industry, and the failure of the pipeline mostly occurs at the pipeline joint. At present, the industry's high-pressure air system pipe joints are still using the standard issued by Industry building Industry Corporation in 1995. The joints designed by this standard are all connected with threaded gaskets. The joints have high welding requirements, and the joints are bulky and inconvenient to install in the narrow space of the industry. Regular maintenance is required.

Metal-to-metal seal usually refers to the seal composed of two metal parts in contact with each other, between them do not use non-metallic materials as an intermediate layer, widely used in valves, pumps, pipe joints, compressors and other industrial sealing parts, the advantages of this way is vibration resistance, corrosion resistance, long service life. Ferrule joint is a typical metal-to-metal sealing structure, in engineering practice is generally the use of the same type of metal, their material mechanical properties are similar, so the connection is more solid, and strong corrosion resistance, ferrule joint because of its small, fast installation characteristics in a variety of pipe systems have been widely used.

Regarding the research on the sealing structure and performance analysis of the ferrule joint, reference

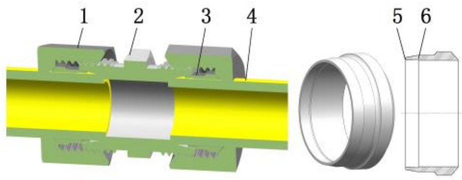
[1] describes the leak-proof performance mechanism of the ferrule joint, pointing out that the core of realizing the effective sealing of the ferrule is to always ensure that the clearance of the ferrule seal is zero. Reference [2] introduced the structure and sealing principle of single-ferrule pipe joint, and pointed out that the embedding of the ferrule on the pipe surface, the fitting of the outer circle of the ferrule with the inner cone of the joint, the fitting of the cone of the tail end of the ferrule with the inner cone of the compression nut, as well as the material, manufacturing quality and installation accuracy of the ferrule were all causes of the seal failure of the pipe joint. Reference[3] introduces the composition and sealing principle of the progressive single ferrule joint, and analyzes the possible reasons for the failure of the inner and outer sealing pairs formed by assembly. Ferrule eccentricity, deformation, foreign matter in the contact surface and manufacturing defects are the reasons for the leakage of ferrule pipe joints. Reference [4] analyzed the factors affecting the air tightness of the jacket-type pipe joint, including installation torque, pipe hardness, pipe wall thickness and vertical section of the pipe end, and proposed joint leakage prevention measures such as controlling tightening torque, selecting suitable pipe materials, improving installation pass rate and selecting qualified jacket-type joints.

Research on sealing structure simulation of pipe joint, Yan et al. studied the influence of internal fluid on the sealing characteristics of pipe fittings and the change of sealing characteristics caused by fluid pressure switch based on the establishment of a multi-scale model of the jacketed pipe joint, and finally found that the sealing reliability of the jacketed pipe joint can be effectively improved by improving the fluid pressure[5]. Based on the material constitutive model considering ratchet effect, Xiangyu Wang et al. conducted numerical simulation on the behavior of pipe joints in the plastic loosening stage, and concluded that large lateral load, initial preload and friction coefficient would accelerate plastic loosening and thus affect the sealing effect[6]. In reference [7], according to the non-penetrating contact rule, finite element simulation was conducted on the double-clamp ferrule joint of the vehicle high-pressure hydrogen supply system, and the normal force on the contact surface was determined to be the key factor affecting the sealing. The structure of the joint is analyzed and studied, and the mechanism of sealing, pressure resistance and vibration resistance of the joint is mainly studied. Based on the multi-scale model of pipe joints, reference [8] studied the influence of high-pressure pulsating fluid on the sealing characteristics of pipe joints, and obtained the law that high-pressure fluid could improve the sealing performance of pipe joints and that the influence of fluid pulsation on joint sealing was limited. Based on the multi-scale finite element model of actual roughness, Zhengxin Yang et al. analyzed the law of the influence of material parameters on the sealing state of aviation pipe joints, and concluded that the yield strength of ferrule materials should be close to the yield strength of pipeline materials, so as to obtain the best sealing state of pipe joints[9].

The above studies are based on low and medium pressure jacketed pipe fittings with a maximum pressure range of 10MPa to 35MPa(maximum outer diameter 10mm pipe), and the sealing performance of 40MPa high-pressure pipe fittings (minimum outer diameter 10mm pipe) is not analyzed. Based on the actual size parameters of the 22mm jacketed pipe joint, a 1/4 3D finite element simulation model was established in this study. The sealing state of the pipe joint was simulated and analyzed based on the finite element governing equation of the sealing process, and the process of fitting and internal pressurization of the ferrule was studied, The equivalent stress of the inner and outer contact surface of the front end of the ferrule, the influence law of the deformation of the ferrule in the assembly process on the structural seal, the approximate range of the preload force and the approximate range of the maximum pressure of the pipe are determined, and the measures to improve the stress concentration of the ferrule are put forward.

2. SEALING MECHANISM OF FERRULE TYPE PIPE JOINT

The structure of the ferrule type pipe joint of 40MPa high-pressure air system is shown in Figure 1, which is mainly composed of four parts: compression nut, midbody, ferrule and pipe. By pressing the threaded connection between the nut and the intermediate, the inner and outer surfaces of the front end of the ferrule are tightened with the inner cone of the intermediate body and the outer surface of the pipe to achieve a seal. This paper focuses on the sealing problem of the contact part between the ferrule and the midbody and the pipe.



1.compression nut 2.midbody 3. ferrule 4.pipe 5.cutting edge 6.stoppkante

Figure1. *Diagram of pipe fittings and ferrules*

2.1. The Leakage Formula of the Circular Pipe Seal Gap

According to the laminar flow equation of fluid along the radial annular gap, the formula of leakage through the circular pipe seal gap is：

$$Q = \frac{\Delta P \pi d_1 \delta^3}{12 \mu B} \tag{1}$$

where Q is the amount of leakage through the sealing gap, ΔP is the pressure difference between the two ends of the gap, d_1 is pipe outer diameter, δ is sealing clearance, μ is dynamic viscosity, B is sealing width, respectively.

It can be seen from the formula that when the pressure difference, dynamic viscosity, sealing width and pipe outside diameter are constant, the leakage is mainly determined by the sealing gap and sealing width. To achieve effective sealing, it is necessary to reduce the sealing gap and increase the sealing width[1].

2.2. Sealing Mechanism

In the metal-to-metal sealing structure, it can be divided into three kinds according to the different shapes of the sealing surface, cone/cone seal, cone/sphere seal and cylinder/sphere seal. Among them, cone/cone seal is the most widely used in the three sealing structures because of its advantages such as large equivalent contact width of sealing surface, long leakage length, large leakage resistance, strong anti-stress relaxation and stress corrosion leakage ability[10]。

During the assembly process of the ferrule type pipe joint, while tightening the compression nut, the inner cone of the nut constantly squeezes the back end of the ferrule, pushes the ferrule into the gap between the 24 degree inner cone hole of the intermediate and the pipe, so that the outer surface of the front end of the ferrule fits the inner cone of the intermediate to form the first sealing surface. With the further tightening of the nut, the front edge of the ferrule is squeezed into the pipe and the pipe is continuously fitted to form a second sealing surface, and the two groups of metal-to-metal sealing surfaces inside and outside are just effective in blocking the leakage channel of the fluid. With the further deepening of the ferrule, the back end of the ferrule is slightly raised, resulting in elastic deformation, which will store a certain pre-loading force, so that the ferrule can prevent the loosening of the compression nut while providing the contact surface contact stress, which is the sealing mechanism of the ferrule[4].

The relation industry between the maximum working pressure and the outside diameter of the traditional ferrule pipe joint is shown in Table 1. It can be seen from the table that with the increase of the outside diameter, the maximum working pressure decreases correspondingly. How to increase the working pressure of the large pipe diameter is one of the problems in the design of this study.

Table1. *Maximum working pressure of traditional pipe fittings*

pipe outside diameter/mm	maximum working pressure/MPa
6	63
8	
10	
14	32
18	
22	
28	25
34	
42	
50	16

3. Finite Element Model

3.1. Model Assumption

In order to simplify the calculation, the following hypothesis is proposed [11]:

（1）Ignoring the influence of surface hardening treatment on the ferrule, it is assumed that the ferrule has a definite elastic modulus and poisson ratio;

（2）Without considering the dynamic effect, the model is analyzed only in the static equilibrium state;

（3）Ignoring the body weight of the fluid, the fluid pressure is changed to the pressure on the internal surfaces;

（4）The analysis is based on the structure of the material, ignoring the influence of time factor;

（5）Ignoring the inertia and dynamic effects caused by instantaneous loading, the loading time is relatively long.

3.2. Finite Element Governing Equation of Sealing Process

In order to solve the complex nonlinear problem in the sealing process, it is necessary to establish the finite element governing equation of the sealing process。 Based on the principle of virtual work, the practical problem is transformed into the equilibrium problem under virtual displacement, and the finite element governing equation of sealing process is obtained:

$$\iiint_{v} \sigma_{ij} \delta_{\varepsilon ij} dv = \iiint_{v} x_i \delta_{ui} dv + \iint_{s} x_i \delta_{ui} ds \qquad (2)$$

where, σ_{ij} is the static allowable stress, $\delta_{\varepsilon ij}$ is virtual strain, υ is the given volume force boundary, s is the given surface force boundary, X_i is the Unit load vector, δ_{ui} is virtual displacement, respectively.

3.3. Buildup of Model

Taking the 22mm outer diameter carpeted pipe joint as an example, the model is modeled. Due to the quasi-static analysis, the welded sealing part of the pipe and the open connecting pressure pipe part are ignored, and the end of the pipe joint is simplified into a cylinder. This paper does not analyze the stress of the thread, and directly ignores the compression nut. The simulation model of the jacketed pipe joint of the 40MPa high-pressure air system is shown in Fig. 2, the structural parameters are shown in Table 2, and the material properties are set according to Table 3.

Table2. *Structure parameters of ferrule pipe joint*

parameter	value /mm	parameter	value /mm
pipe lengthL_1	50	midbody length L_2	35
pipe outer diameter d_1	11	midbody outer diameter d_3	11.1
pipe inner diameter d_2	8	midbody inner diameter d_4	9.5

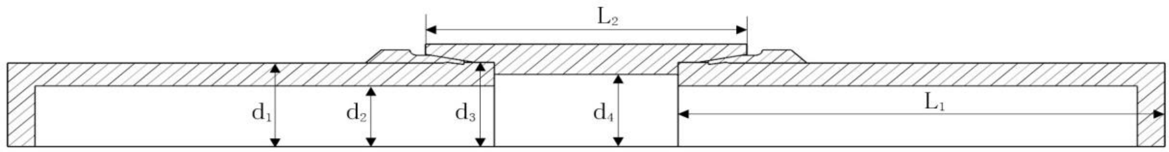


Figure2. *Simulation model of ferrule pipe joint*

Table3. *Material properties of pipe fittings*

material	poisson ratio	yield strength /MPa	strength of extension /MPa	elasticity modulus /GPa	density /kg·m^{-3}
316 stainless steel	0.3	300	600	193	7980
HDR duplex stainless steel	0.3	600	700	200	7800

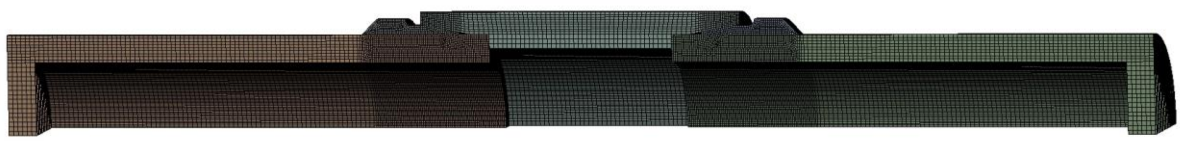


Figure3. *Grid setup of pipe joints*

In order to improve the computational efficiency of the computer simulation, the 1/4 simplified model was adopted, ignoring the role of the compression nut, establish the parts sub-models of the pipe, the ferrule and the intermediate respectively, and the assembly was carried out in the assembly module without setting contact during assembly.

The number of layers of the grid usually depends on the complexity of the problem and the accuracy required. In general, the denser the grid, the higher the accuracy of the calculation results, but the corresponding calculation time and memory consumption will also increase, and it is usually guaranteed that the smallest area must have at least two layers of the grid. The overall mesh size is set to 0.5mm, and the local mesh size of the pipe and intermediate mesh near the ferrule and contact area is set to 0.2mm. Slice the contact area and refine the local mesh. The mesh is made of hexahedral mesh and solid element, as shown in Fig.3. The total number of grid nodes is 420281 and the total number of cells is 373864.

3.4. Boundary Conditions and Load Settings

Pipe and ferrule, ferrule and midbody are standard contact, the friction coefficient is set to 0.1, and the augmented Lagrange multiplier algorithm is adopted. Both ends of the pipe and the external surface of the middle are provided with fixed constraints; Preload force is applied to the back end of the ferrule; The area enclosed by the pipe, ferrule, and intermediate applies hydrostatic pressure to the area through which the fluid can flow.

3.5. Grid Independence Test

When applying the preload force of 40KN to the back end of the ferrule, change the local size of the ferrule and the pipe near it and the intermediate mesh, set to 0.1mm, 0.2mm, 0.3mm, 0.4mm and 0.5mm respectively, and simulate the assembly process of the ferrule. The results are shown in Figure 4. When the local mesh size is reduced to 0.2mm, the calculation result of the maximum contact pressure of the ferrule pipe joint is independent of the mesh, indicating that the calculation accuracy can be satisfied when the local mesh size is set to 0.2mm.

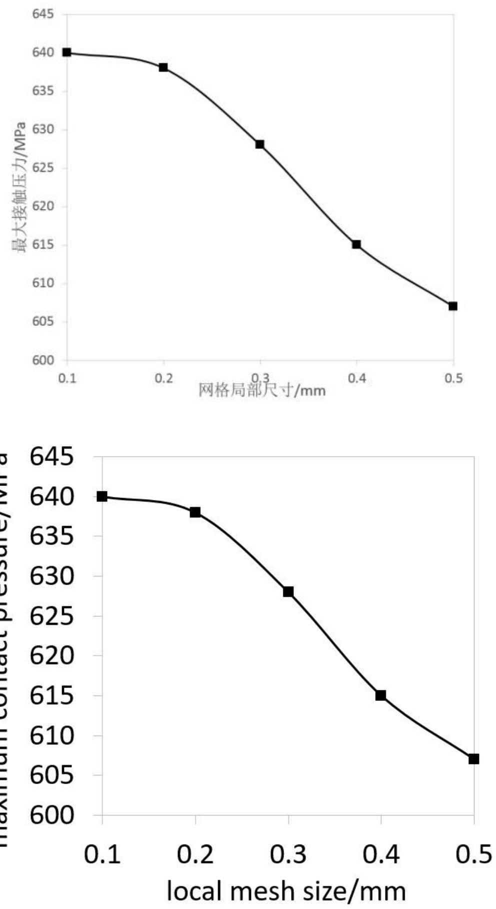


Figure4. *Grid independence test*

4. SEALING STATE ANALYSIS OF PIPE JOINTS

Based on the finite element contact algorithm, bilinear follow-up strengthening model is used to simulate the pipe elastoplastic constitutive model. Stress analysis was performed based on Von Mises yield criterion. The calculation formula of Von Mises equivalent stress is as follows：

$$\sigma_v = \sqrt{\sigma_1^{\ 2} + \sigma_2^{\ 2} - \sigma_1\sigma_2 + 3\tau^2} \qquad (3)$$

Where, σ_1 and σ_2 are the principal stresses， τ is shearing strength, respectively.

4.1. Sealing Performance Analysis of Pipe Joint During Assembly

In the actual assembly process, it is difficult to measure the size of the assembly preload, because the recommended number of installation rings of the compression nut is 1.25, the pitch is 2mm, the actual displacement at the back end of the ferrule is 2.5mm, we can obtain the approximate range of the preload force of the ferrule through displacement simulation. Set the ferrule axial displacement of 2.5mm, and the pipe joint clamping process is shown in the figure. After the displacement of 1mm, the outer surface of the ferrule begins to completely fit the inner cone of the middle body, and then the front end of the ferrule is gradually embedded into the outer surface of the pipe. With the continuous clamping of the ferrule, the stop edge begins to act, and then the middle part of the ferrule is arched, and the cutting edge is further pressed down to complete the assembly. The axial force at the back end of the ferrule was extracted, with a maximum value of 46.5KN.

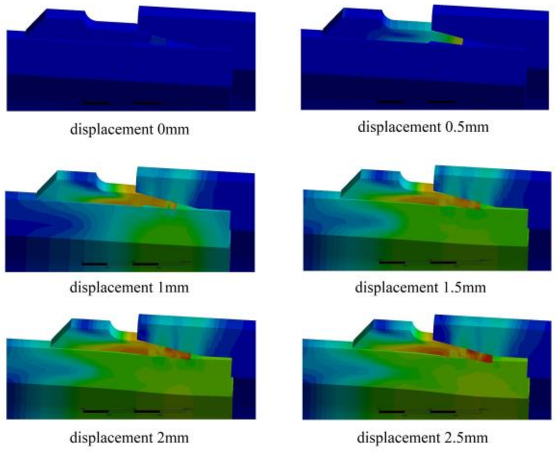
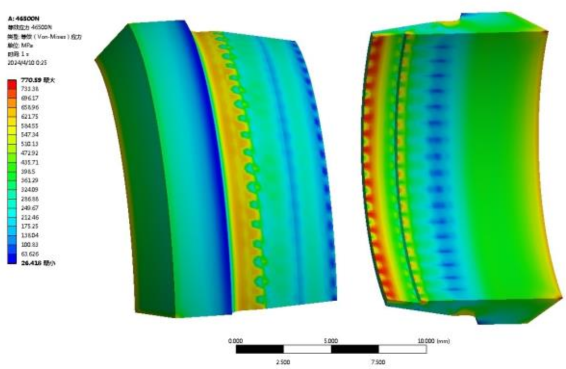


Figure5. *Stress distribution of joint under different ferrule displacements*

4.1.1 Analysis of stress concentration points

In the design of middle and low pressure, the edge stress has little influence and can not be calculated. For high-pressure air, the damage of the transition connection part will bring serious consequences, and simulation analysis must be carried out.

When the pre-load force of 46.5KN is applied to the back end of the ferrule, the equivalent stress distribution on the inner and outer surfaces of the ferrule is shown in Figure 6, and the maximum stress value in the figure has reached 770MPa, which has exceeded the tensile strength of the material. The following measures can be taken：（1）surface treatment of the ferrule to improve the surface strength；（2）appropriately increase the yield strength of the material of the ferrule. It can be seen from Figure 6 that the stress concentration danger point of the ferrule mainly has four areas: cutting edge, stopping edge, middle platform of the outer surface and tail end. From the structural point of view, the cutting edge is the main force part, which is difficult to correct, and the card ferrule is modified as follows：(1) Change the inner cone Angle of the stopping edge to a transition arc; (2) the width of the middle platform on the outer surface is increased by 1mm; (3) The chamfer Angle at the end of the ferrule is 0.3mm.



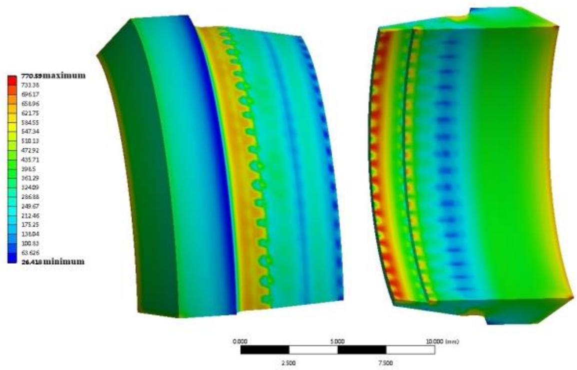


Figure6. *Stress concentration point of ferrule*

When the actual assembly is to apply a force at the back end of the ferrule, it is only the back end of the ferrule displacement of 2.5mm, in addition to the front end of the ferrule clamping and the middle arch, the actual displacement of the ferrule is less than 2.5mm, in summary, the preload force should be less than 46.5kN.

4.1.2 Analysis of front end contact surface of card cutting edge

The greater the contact pressure on the contact surface, the better the coordination between the two rough surfaces, and it is believed that the pressure on the contact surface is greater than three times the pressure of the sealing medium to form a reliable seal. When the internal pressure remained constant at 80MPa, the change of the minimum equivalent stress (Von-Mises) at the front end of the ferrule cutting edge with the preloading force was shown in FIG. 7. When the pre-loading force is 32KN, the minimum equivalent stress is 237.35MPa, considering the surplus, the pretightening force should be greater than 36KN。

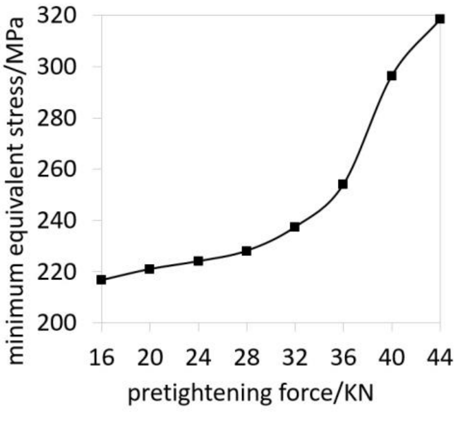


Figure7. *Minimum equivalent stress at the front end of the cutting edge of the ferrule*

4.1.3 Analysis of ferrule deformation

Good contact is a prerequisite for a metal-to-metal seal. The surface of the general object has a certain roughness, there will be a lot of micro-convex, when the contact stress is greater than a certain value, the plastic flow will occur on the surface of the ferrule, so that the number of micro-convex contact continues to increase, increasing the effective contact area between the contact surface, when a certain amount can ensure the effective sealing of the metal-to-metal contact surface.

In the assembly process, the ferrule occurs axial displacement, the inner and outer surfaces of the front end of the ferrule are subjected to the friction and extrusion pressure of the intermediate and the pipe respectively, and the surface is elastoplastic deformation, and the local material block is formed, and then the high stress area is formed in some locations, resulting in plastic deformation of the material, so that the micro-convex state of the contact surface is optimized to achieve effective sealing.

In the process of force deformation of an object, the elastic deformation zone, elastoplastic deformation zone and plastic deformation zone are interwoven. From a microscopic point of view, the fluctuation of atoms in a stable position is an elastic deformation, and the relative slip of atoms escaping from a stable position or atomic layer is a plastic deformation. The state of each atom is different. There is no strict distinction between elastic and plastic deformation, which can be approximated according to the yield strength and tensile strength obtained by quasi-static tensile test. This paper assumes that the equivalent stress (Von Mises) 0-250 Mpa is the elastic deformation zone, 250-500 Mpa is the elastoplastic deformation zone, and more than 500MPa is the plastic deformation zone. The equivalent stress information of all nodes of the two ferrules was extracted, and the preloading force was successively increased from 0MPa to 44KN every 4.4KN, and the proportion of nodes in the three regions was drawn according to Figure 8. It enters the elastoplastic deformation zone from 13.2KN to 24.6KN, and the elastic deformation zone is equal to the elastoplastic deformation zone. A plastic deformation zone appears from 30.8KN.

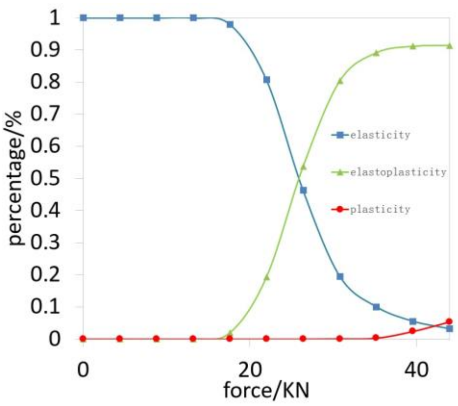
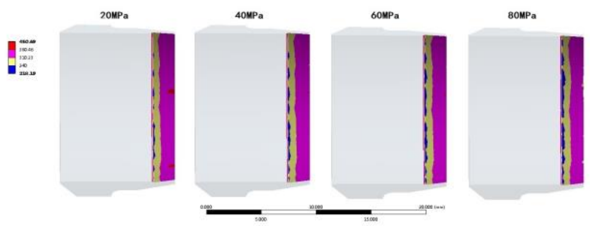


Figure8. *Change curve of ferrule deformation area*

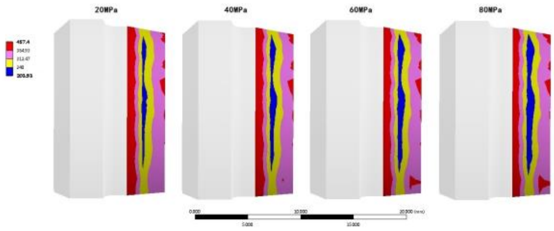
4.2. Sealing Performance Analysis of Pipe Joint During Pressurization

4.2.1 Analysis of the contact surface of the inner and outer surfaces of the ferrule

When the preload force is 40KN, the process of the internal pressure gradually increasing from 0MPa to 80MPa is simulated. The equivalent stress on the contact surface of the ferrule seal is shown in Figure 9. In order to facilitate the quantitative evaluation of the sealing performance, the concept of seal failure width ratio was proposed, which was defined as the ratio of the failure equivalent width inside the sealing surface to the total width. The curve was drawn as shown in Figure 10. It can be seen from the figure that with the increase of the internal pressure of the fluid, the seal failure width ratio of both the inner and outer surfaces of the ferrule increased proportionally, and the inner surface was always larger than the outer surface. The area where the inner surface is likely to leak first.



（a）ferrule inner surface



（b）ferrule outer surface

Figure9. *Equivalent stress on the contact surface of the ferrule seal*

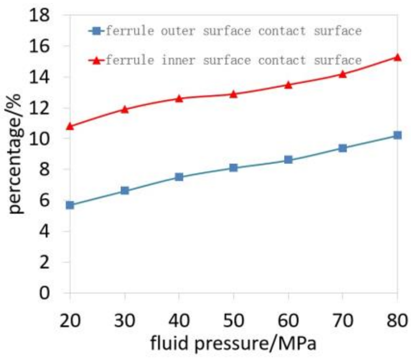


Figure10. *Seal failure width ratio*

4.2.2 Stress Analysis of Pipe Front Contact Area

Extract the equivalent stress value of the same position in the pipe contact area, as shown in Figure 11. With the increase of internal fluid pressure, the equivalent stress value of the pipe contact area also gradually decreased. When the internal fluid pressure reached 80MPa, the Von Mises equivalent stress value was 365.6MPa, greater than 240MPa, which met the sealing requirements.

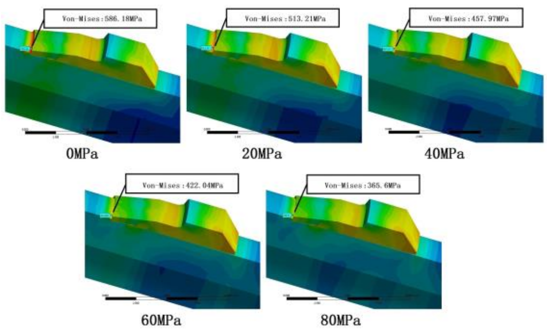


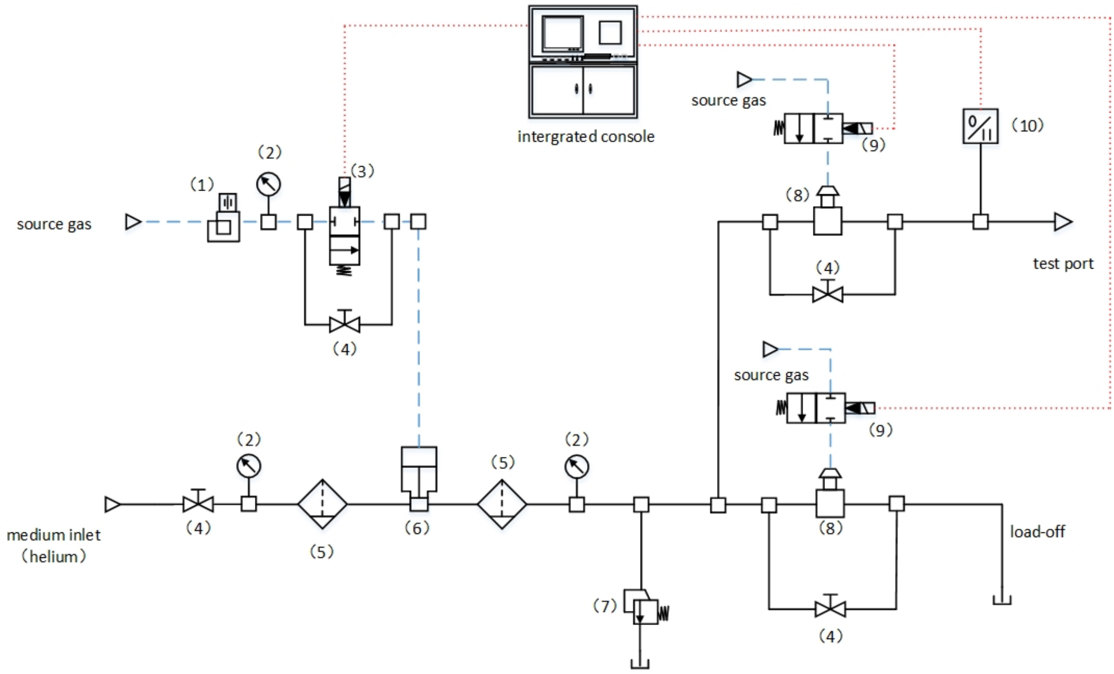
Figure11. *Stress distribution of joint under different ferrule displacements*

5. EXPERIMENTAL VERIFICATION

5.1. Test Bench and Pressurization Principle

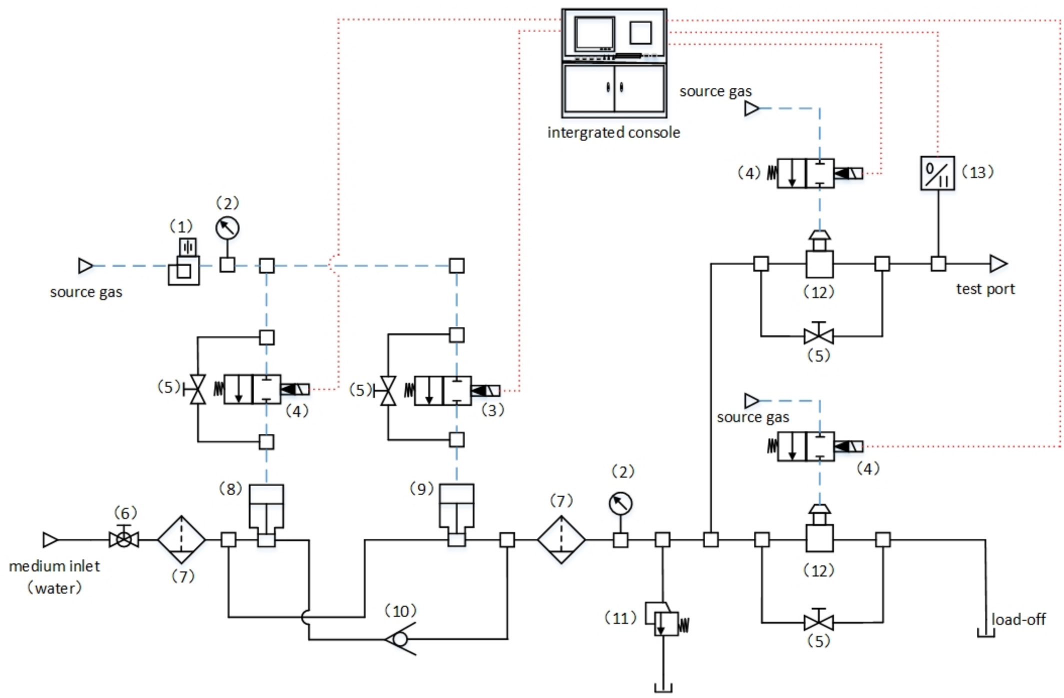
The comprehensive pressure test bench of the high-pressure joint designed in Fig. 12 was used to conduct water pressure and air pressure tests. The test bench was driven by compressed air, and the water pressure test range was 0-160MPa. The gas flushed by the test was helium, ranging from 0-90MPa.

Figure12. *Test bench*



1.pressure regulating value 2.pressure meter 3.electric proportional value 4.hand value 5.filtrator 6.pneumatic gas booster pump 7.safety value 8.pneumatic control value 9.solenoid directional value 10.transducer

（a）gas pressurized part



1.pressure regulating value 2.pressure meter 3.electric proportional value 4.solenoid directional value 5.hand value 6.manual ball value 7.filtrator 8.pneumatic diaphragm pump 9.pneumatic liquid booster pump 10.one-way value 11.safety value 12.pneumatic control value 13.transducer

（b）liquid pressurized part

Figure13. *Schematic diagram of the test bench*

5.2. Experimental Process

Helium gas was used in the air pressure test, and the test piece was connected to the air pressure test bench through the G1/4 adapter, and then transferred into the water box in the explosion-proof room for air pressure test. During the test, step up the pressure (as shown in Figure 14) to 50MPa and then hold the pressure for 15min, and no leakage (bubble) can be seen during the holding time.。

During the hydraulic test, the test part is connected to the hydraulic test pump through the G1/4 adapter, and then transferred to the explosion-proof room for hydraulic test. During the test, the step pressure was raised to 60MPa and held for 10min. No leakage or deformation was observed during the holding time.

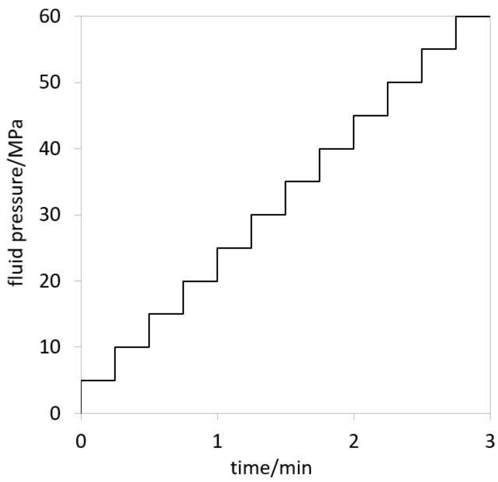


Figure14. *Step-up pressure curve*

6. Conclusions

Based on the finite element governing equation of sealing process, the finite element model of pipe joint with ferrule is established. The effect of the equivalent stress on the inner and outer contact surface of the front end of the ferrule and the deformation of the ferrule on the sealing is analyzed in the process of assembly and internal pressure of the ferrule：

（1）During the sealing performance analysis of the pipe joint assembly process, the pre-tightening force of the ferrule should be less than 46.5KN, and the stress concentration danger point of the ferrule mainly has four areas: cutting edge, stopping edge, middle platform of the outer surface and the tail end.

（2）For the pressurization process of the pipe joint, the sealing failure width ratio of the inner surface and the outer surface of the ferrule increases proportionally with the increase of the internal fluid pressure. The inner surface is always larger than the outer surface, and the contact area of the inner surface may leak first.

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