Optimization of thin-walled sealing structure based on mechanical simulation

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Abstract. In the hydraulic and gas pipelines, even thin-walled cylinder device, the built-in elastomeric O-rings are widely used to ensure their sealing. Unfortunately, a case of excessive radial contact pressure will lead to leakage risk. At the same time, the aging of elastomeric O-ring will result in decrease of radial contact pressure, which can also bring about leakage risk. To increase the life of the seal structure, the external shape memory alloy (SMA) sealing ring is subsequently developed for the actual product. However, excessive contact pressure will cause the leakage risk with the lateral deformation at the outside of the sealing interface of the thin-walled cylinder. Based on mechanical simulation, a proposal for the optimization design of seal structure with thin-walled cylinder is given. That is, the radial contact seal on the inner and outer side of the thin-walled cylinder can simultaneously made by SMA. Results gained show that the theoretical value of the resultant force from the SMA at the inside and outside of the sealing interface of the thin-walled cylinder is zero. The bilateral seal structure with SMA ring can avoid the leakage risk caused by the unilateral force and deformation on the thin-walled cylinder, and effectively prolong the service life of the sealing device.

1. Introduction
At present, many researches have been done on the sealing performance of pipelines included liquid or pneumatic medium [1-3]. The common seal forms include the built-in elastomeric O-rings and the external SMA ring seal [4, 5]. For the built-in elastomeric O-ring seal, a case of excessive radial contact pressure will lead to leakage risk, which is caused by the internal deformation at the inside of the sealing interface of the thin-walled cylinder [6]. Meanwhile, the aging of elastic rubber ring will result in the decrease of radial contact pressure, which can also bring about leakage risk. These situations could be due to elastomeric O-ring degradation caused by chemical attack, overheating and loss of resilience from compression set [7, 8]. To prolong the seal life, many researchers developed external SMA sealing ring. When it turns into heating up phase transition, the overall shape of it will shrink. So then, the radial contact seal is achieved on the outer cylindrical surface of the thin-walled cylinder [9, 10]. Unfortunately, excessive contact pressure will cause the leakage risk with the lateral deformation at the outside of the sealing interface of the thin-walled cylinder [6, 9]. In order to make use of the advantages of SMA seal for long life, based on mechanical simulation, a suggestion of the optimization design for seal structure with thin-walled cylinder is given. Herein, the radial contact seal on the inner and outer side of the thin-walled cylinder can simultaneously implemented by SMA. Although the sealing performances of the elastomeric O-ring and the external SMA ring have been investigated in depth, few information of it is known about in thin-walled cylinder device. In this study, theoretical formulae and experimental observation and finite element method (FEM) simulation are
proposed to analyze the bilateral seal structure with SMA ring. Moreover, comparison of sealing capabilities between the bilateral seal structure and the unilateral seal structure can be gained. The results obtained can be used to improve sealing performance and prolong life of the thin-walled cylinder.

2. Stress state between thin-walled cylinder and SMA seal ring

The mechanical model of thin-walled cylinder under radial loading produced by SMA seal ring can be shown in figure 1. The sealing interface is existent between the thin-walled cylinder A and B. In figure 1(a), the outer radial fastening force \( F_1 \) from the SMA seal ring at the outside of the sealing interface is larger than zero. Excessive contact pressure caused by the outer radial fastening force \( F_1 \) will cause the leakage risk with the lateral deformation at the outside of the sealing interface of the thin-walled cylinder. In figure 1(b), the radial contact seal on the inner and outer side of the thin-walled cylinder is simultaneously made by SMA seal ring. And then, the theoretical value of the resultant force resulted from the outer radial fastening force \( F_1 \) and the inner radial fastening force \( F_2 \) at the inside and outside of the sealing interface of the thin-walled cylinder is zero. The bilateral seal structure with SMA seal ring could avoid the leakage risk caused by the unilateral force and deformation on the thin-walled cylinder.

![Figure 1. Schematic of thin-walled cylinder under different types of radial loading: (a) unilateral seal structure; (b) bilateral seal structure.](image)

By adopting elastic-plastic theory, most of the previous work dedicated to investigate the stress state between pipeline and SMA joint use the approximate formula [11]. The stress state between pipeline and SMA joint can be used to solve the thin-walled cylinder problem and axisymmetric problem. As result of the austenitic phase transformation with SMA, the SMA seal ring can produce restore stress. The hoop stress will be provided by the recovery stress with SMA in the pipeline joint system. Subsequently, the radial compressive stress produced in the pipeline joint system. The radial compressive stress will produce radial fastening force \( F_1 \) and \( F_2 \). Under radial fastening forces, the thin-walled cylinders and SMA seal rings connect together tightly. The constitutive relation of SMA can be expressed as:

\[
\sigma - \sigma_0 = D(\xi)\varepsilon - D(\xi_0)\varepsilon_0 + \Omega(\xi)\varepsilon - \Omega(\xi_0)\varepsilon_0 + \alpha(T - T_0)
\]  

Where \( \sigma \) is the second P-K stress, \( \xi \) is martensite fraction of SMA, \( \varepsilon \) is the Green strain, \( D \) represent modulus of SMA, \( \Omega \) is transformation tensor, \( \alpha \) is thermal coefficient of expansion for SMA, \( T \) is phase transition temperature. Especially, \( \sigma_0, \xi_0, \varepsilon_0 \) and \( T_0 \) show the initial state of SMA. In addition, the martensite fraction \( \xi \) includes \( \xi_T \) and \( \xi_s \). \( \xi_T \) is purely temperature-induced martensite with multiple variants. \( \xi_s \) denotes the fraction of the material that has been transformed by being stressed into a single martensite variant.

In this elastic-plastic theory, the relationship between hoop stress, radial compressive stress and radial fastening force in the connected pipelines are given as:
\[ \sigma_{r1} = -\frac{R_{out1}^2 F_1}{(R_{out1}^2 - R_{in1}^2) S_{out}} \left( 1 - \frac{R_{in1}^2}{r^2} \right) \]  
(2)

\[ \sigma_{\theta1} = -\frac{R_{out1}^2 F_1}{(R_{out1}^2 - R_{in1}^2) S_{out}} \left( 1 + \frac{R_{in1}^2}{r^2} \right) \]  
(3)

\[ \sigma_{r2} = \frac{R_{out2}^2 F_2}{(R_{out2}^2 - R_{in2}^2) S_{in}} \left( 1 - \frac{R_{in2}^2}{r^2} \right) \]  
(4)

\[ \sigma_{\theta2} = \frac{R_{out2}^2 F_2}{(R_{out2}^2 - R_{in2}^2) S_{in}} \left( 1 + \frac{R_{in2}^2}{r^2} \right) \]  
(5)

Where \( \sigma_{r1} \) and \( \sigma_{\theta1} \) are the radial compressive stress and the hoop stress respectively under the outer radial fastening force \( F_1 \), \( \sigma_{r2} \) and \( \sigma_{\theta2} \) are the radial compressive stress and the hoop stress respectively under the inner radial fastening force \( F_2 \), \( R_{in1} \) and \( R_{out1} \) are the outer radius and the inner radius respectively under the outer radial fastening force \( F_1 \), \( R_{in2} \) and \( R_{out2} \) are the outer radius and the inner radius respectively under the inner radial fastening force \( F_2 \), \( S_{out} \) is the exterior surface area under the outer radial fastening force \( F_1 \), \( S_{in} \) is the interior surface area under the inner radial fastening force \( F_1 \), \( r \) is constant radius lain between the exterior surface and the interior surface of the thin-walled cylinder.

3. Finite element analysis
When the thin-walled cylinder is compressed or expanded by the SMA seal ring, the contact pressure and lateral deformation on the sealing interface of the thin-walled cylinder will increased. In the initial phase, the inner surface diameter of the outer SMA seal ring is smaller than the outer surface diameter of the connected thin-walled cylinder. In the same way, the outer surface diameter of the inner SMA seal ring is larger than the inner surface diameter of the connected thin-walled cylinder. In the first place, the above diameters of the SMA seal ring are expanded or shrinked under given temperature. During this process, the twin martensite can be transformed into detwinned martensite by external load. As the SMA seal ring is heated, the recovery of it is limited by the connected thin-walled cylinder. Thus, the SMA seal ring will produces large recovery forces on the thin-walled cylinder.

To simulate the contact pressure and lateral deformation of SMA seal ring coupling with the thin-walled cylinder, 2-D axisymmetric finite element models of the unilateral seal structure and the bilateral seal structure are established using ANSYS software as showed for two cases in figure 2. Non-uniform grid is introduced for the contact regions and the rest of the thin-walled cylinder. It can be used to reduce the amount of meshing as well as receive the contact pressure and lateral deformation distributions in the concerned region. Quadrilateral grids are introduced for enhancing the accuracy of simulation. As the diameters of the SMA seal ring are expanded or shrinked, the radial loading distribution on the thin-walled cylinder can be shown in the figure 3.

**Figure 2.** Finite element model of each thin-walled cylinder assembly: (a) unilateral seal structure; (b) bilateral seal structure.
Figure 3. The load distribution on the thin-walled cylinder of each assembly: (a) unilateral seal structure; (b) bilateral seal structure.

In figure 2, the material kinds of the thin-walled cylinder A and B are all stainless steel. The material kinds of the SMA seal ring are all TiNi alloy. The geometric and mechanic characteristics of the thin-walled cylinder and the SMA seal ring are summarized in table 1. The condition of first row is for the thin-walled cylinder and second row is for the SMA seal ring. Thereinto, \( L \) is the contact length between the SMA seal rings and the thin-walled cylinders, \( E \) is elastic modulus, \( v \) is Poisson’s ratio, \( \rho \) is density, \( r \) is constant radius lain between the exterior surface and the interior surface of the thin-walled cylinder, \( T \) is wall thickness of the thin-walled cylinder and the SMA seal ring located on the contact regions. For the outer SMA seal ring, \( r \) is 11.5mm. For the inner SMA seal ring, \( r \) is 8.5mm.

In figure 3, \( F_1 \) is the outer radial fastening force; \( F_2 \) is the inner radial fastening force. Based on the data from table 1, contact pressure generated by radial fastening force are in the range of 15MPa to 90MPa.

Table 1. Geometric and mechanic characteristics of the thin-walled cylinder assembly.

| \( L \) (mm) | \( E \) (MPa) | \( v \)  | \( \rho \) (T/mm\(^3\)) | \( r \) (mm) | \( T \) (mm) | \( F_1 \) (N) | \( F_2 \) (N) |
|-------------|-------------|--------|----------------|-------------|-------------|-------------|-------------|
| 15          | 2E5         | 0.3    | 7.8E-9         | 10          | 1           | 1.5E4–9E4   | 1.5E4–9E4   |
| 15          | 7.5E4       | 0.33   | 7.8E-9         | 11.5/8.5    | 1.5         | /           | /           |

The assumptions for computational region of the thin-walled cylinder are given as follows: 1) The edge AB is located on the central axis. 2) The pipe internal pressure is 0.2MPa. 3) The radial fastening forces are derived from experimental maximum pull forces between the thin-walled cylinder and the SMA seal ring.

4. Results and discussions
Plastic deformation ratio is an important parameter used for evaluating leakage risk of the bilateral seal structure and the unilateral seal structure. In other words, the security criterion to ensure sealing necessary of the thin-walled cylinder is that the maximum plastic deformation can not result in leakage. When the plastic deformation ratio on sealing interface is gradually increasing, the leakage rate will rapidly increase. It can be seen from the experimental setup based on helium mass spectrum leak detecting technology in figure 4. Under the actual seal conditions in table 1, the relationship between plastic deformation ratio and leakage rate is shown in figure 5, and the relationship between contact pressure and plastic deformation ratio for different seal structure is displayed in figure 6. Plastic deformation ratio is the value of the deformation of the thin-walled cylinder located on the contact regions to the original thickness of it.

In figure 5, it can be seen that the leakage rate will rapidly increase when the plastic deformation ratio on sealing interface goes beyond 1%. As the plastic deformation ratio does not exceed 1%, the leakage rate is steadily below 5E-11Pa·m\(^3\)/s. These experimental results can be used to explain deformation outcomes of the bilateral seal structure and the unilateral seal structure, which are shown in figure 6.

Figure 6 depicts plastic deformation ratio for the unilateral seal structure and the bilateral seal structure under different contact pressure. While the wall thickness of the unilateral seal structure is less than 1mm, the contact pressure must be lower than 20MPa. Otherwise, plastic deformation ratio will go beyond 1%, which will make the leakage rate increase and exceed 5E-11Pa·m\(^3\)/s. These results agree well with reference [11]. Nevertheless, when the same wall thickness 1mm is applied for the
bilateral seal structure, the contact pressure can easily greater than 20MPa, as far as it reaches 300MPa. Under the contact pressure 300MPa, the plastic deformation ratio just gets to 1%. On account of the theoretical value of the resultant force from the SMA at the inside and outside of the sealing interface of the thin-walled cylinder is zero, the bilateral seal structure with SMA ring can efficiently avoid the leakage risk.

Figure 4. Experimental setup based on helium mass spectrum leak detecting technology.

Figure 5. Relationship between plastic deformation ratio and leakage rate.

Figure 6. Relationship between contact pressure and plastic deformation ratio for unilateral seal structure and bilateral seal structure.

5. Conclusions
In this paper, leakage rate and plastic deformation ratio for above two kinds of the thin-walled seal structure are investigated by experimental observation and FEM simulation. Based on theoretical analysis and mechanical simulation, an optimization proposal design of seal structure with thin-walled cylinder is given. Namely, the radial contact seal on the inner and outer side of the thin-walled cylinder should simultaneously made by SMA. Results gained show that the theoretical value of the resultant force from the SMA at the inside and outside of the sealing interface of the thin-walled cylinder is zero. These results make the plastic deformation ratio on sealing interface less than 1%. Furthermore, the leakage rate will steadily below 5E-11Pa·m³/s. Thus, the bilateral seal structure with SMA ring can avoid the leakage risk caused by the unilateral force and deformation on the thin-walled cylinder, and effectively prolong the service life of the sealing device.

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