Topology Optimization Design of a 914 Mechanical Tee Based on SIMP

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Abstract. A topology optimization design was created for the key parts of the mechanical tee to find a solution to the challenge posed by its substantial weight and complicated local structure. In this paper, a SIMP algorithm is used as the theoretical basis for the topology optimization, and the numerical simulation was performed by Matlab. Furthermore, it was concluded that the filter radius and density penalty factor were the crucial criteria that affected the optimization result, allowing the relevant parameters of the optimization software to be adjusted. The topological optimization software was used to optimize the upper and lower shells of the mechanical tee, while the weight of the tee was reduced by about 20% without reducing in strength.

1. Background

Pipe tapping and plugging device is the major equipment for the pressure tapping operation with non-stop transportation of the pipes. Major auxiliaries of the tapping and plugging device include the header of the tapping machine, the clamp valve and the mechanical tee [1]. As an essential device that connects the pipes with the tapping and plugging equipment, the tee plays a crucial role during the maintenance and emergency repair process, serving as a critical component that can bear the full weight of the tapping machine and its auxiliary structure [2]. Higher demand for large-diameter pipeline tees is evident due to an increase in pipe diameter and transmission pressure requirements. A relatively comprehensive procedure has been established for the maintenance and emergency repair of oil and gas pipelines on land [3], utilizing welding to attach the tee to the pipeline. Furthermore, the mechanical tee is also widely employed during the maintenance, and emergency repair of offshore pipelines since the development of oil and gas resources has gradually transferred from land to sea.
Therefore, the maintenance and emergency repair process of submarine pipelines have become increasingly challenging, particularly regarding underwater welding [4]. In addition, utilizing the mechanical tee can eliminate underwater welding work, saving a considerable amount of operational costs. At present, the domestic development of underwater mechanical tees in China is still in the initial stages. Most mechanical tees currently available are heavy with a complicated structure, which not only poses significant challenges during the installation process but also generates higher manufacturing costs.

To solve these problems, key parts of the current mechanical tee are improved by optimization design of the mechanical structure to reduce the tee weight. Fig. 1 shows the three-dimensional structure of the 914 mechanical tee, which is presented as an example in this paper.

Fig. 1 914 Mechanical Tee.

Considering the types of design variables and problem-solving methods, the continuum structure optimization can be divided into shape optimization, size optimization, and topology optimization [4-5]. In topological optimization, the homogenization method, the variable density method (SIMP (Solid Isotropic Microstructures with Penalization)), the variable thickness method, and the topological function description method are the widely used methods for topological expressions and material interpolation [6-8]. In this paper, the SIMP algorithm is applied.

2. Topology Optimization Theory

2.1. Numerical Simulation of the SIMP Theory

According to the basic optimization theory, there are three key elements for optimization design, namely design variables, objective functions, and constraints. The design variable refers to a set of parameters that can improve performance through changes. The objective function requires optimal design performance since it is a function related to the design variables, while constraint conditions limit the design and is also a requirement in design variables and other performances [9].

As one of the most widely used methods in the continuum optimization design, the SIMP algorithm takes the density of the cell material as the design variable, whose cell density changes continuously from 0 to 1. When the cell density is 0, the cell is empty; when the cell density is 1, the cell is real, and when the cell density ranges between 0 and 1, the density value of the cell represents imaginary material [10]. Accordingly, the expression of the mathematical optimization model is as follows:
In this equation, \( X=(x_1, x_2, \ldots, x_i) \) is the design variable denoting the cell density; \( f(X) \) is the design objectives signifying the mechanical properties or weight; while \( g(X) \) and \( h(X) \) are the design responses requiring constraint. Accordingly, the finite element optimization model is as follows:

\[
\begin{align*}
\text{Minimize:} & \quad f(X) = f(x_1, x_2, \ldots, x_i) \\
\text{Subject to:} & \quad g(X) \leq 0 \quad j = 1, \ldots, m \\
& \quad h_i(X) \leq 0 \quad k = 1, \ldots, m_h \\
& \quad X_i^L \leq X_i \leq X_i^U \quad i = 1, \ldots, n
\end{align*}
\]

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Topological optimization can be streamlined to simplify the Matlab program. If the design area is a two-dimensional rectangular area, the number of elements and nodes can be easily characterized after discretization, and the aspect ratio of the structure can also be determined by the element ratio in the horizontal (nelx) and vertical (nely) directions. Moreover, it also aims to maximize the structural rigidity, that is, minimize the flexibility, based on the SIMP topology optimization, and the expression is as follows.

\[
\begin{align*}
\text{Minimize:} & \quad c(x) = \sum_{i=1}^{n} \rho \left( u_e^T k_e u_e \right) \\
\text{subject to:} & \quad V(x) = V_0 \\
& \quad KU = F \\
& \quad 0 < x_{\text{min}} \leq x \leq 1
\end{align*}
\]

In the formula, \( c(x) \) represents the objective function; \( U \) and \( F \) represent the global deformation and the force vector, respectively; \( K \) denotes the matrix of the global stiffness; \( u_e \) and \( k_e \) signify the displacement vector and the stiffness matrix of the elements, respectively; \( x \) represents the vector of the design variable; \( x_{\text{min}} \) is the minimum vector of the relative density, and the minimum vector is non-zero to avoid singularity; \( n \) is the number of the discretized elements in the design area; \( P \) is the penalty factor; \( V(x) \) and \( V_0 \) denote the material body and the design area volume; and \( f \) is the specified volume ratio. The optimization case is as follows:

Fig. 2 indicates that the plane structure exhibits a length of 500mm, a width of 200mm and a thickness of 5mm. Above the plane in the center, is a concentrated force of \( F=100N \) perpendicular to the length direction. The plane material is 45# (ASTM 1045) steel, the density \( \rho=7.8 \times 10^3kg/mm^3 \), the elastic modulus \( E=2 \times 10^5MPa \), and the allowable shear stress[\( r\)] = 60MPa. Furthermore, it is required that the plane structure is optimized under conditions involving constant length and strength, aiming to minimize the structure volume.

![Fig. 2 Plane Structure.](image-url)
radius, are used to obtain the optimization results and draw the iterative convergence curve. The optimization results and iteration curve are shown in the Figure 3, 4 and 5. Here, the iteration termination condition is displayed as change<0.01, and three groups with representative results are selected.

1) The density penalty factor is $p = 3$, the filter radius is $r_{\text{min}} = 1.5$, while the iteration is terminated after 45 times.

![Fig. 3 Iteration Result 1.](image)

2) The density penalty factor is $p = 1.5$, the filter radius is $r_{\text{min}} = 1.5$, while the iteration is terminated after 29 times.

![Fig. 4 Iteration Result 2.](image)

3) The density penalty factor is $p = 3$, the filter radius is $r_{\text{min}} = 1$, while the iteration is terminated after 34 times.

![Fig. 5 Iteration Result 3.](image)

These three representative simulation results are selected after performing several simulations by changing the density penalty factor or filter radius, respectively. As shown in Fig. 3, the standard iteration indicates that the beam structure is finally optimized into a truss structure. Fig. 4 shows that this structure displays significantly decreased volume and better convergence effects when the density penalty factor is changed to 1.5, while a serious grid dependence will be evident in the optimization results even though the iteration can rapidly converge. Fig. 5 indicates that when the filter radius is changed to 1, the checkerboard grid effect still occurs even though the iteration can also realize convergence. In addition, both the grid dependence effect and the checkerboard grid are undesirable results in the optimization process. As for the SIMP material interpolation model, the significant factors affecting the optimization results is related to the density penalty factor and the filter radius. By
comparing several simulation results, it is evident that the optimization effect is ideal when the density penalty factor is $p = 3$, and the filter radius is $r_{\text{min}} = 1.5$.

3. Topology Optimization of Key Parts of the 914 Mechanical Tee

Considering that the total weight of the current mechanical tee design is 30 tons, ways should be identified to reduce both the tee weight and the manufacturing cost. The optimal results are obtained by analyzing the key performance of the structure before and after optimization. This paper mainly focuses on the weight reduction of the upper and lower shell of the tee. This structure is shown in Fig. 6.

3.1. Loads and Boundary Conditions

The significant loads of the upper and the lower shell are denoted by 15MPa internal pressure, the force of the slip on the shell after anchoring, and the pressure of the sealing rubber on the shell. Since the positive pressure of the slip on the outer wall of the pipeline is $3.52 \times 10^6$N, the reaction force between the slip and the area where the shell is installed is $3.52 \times 10^6$N and calculated to be approximately 2.7MPa, which is in accordance with the principle of force balance. The sealing rubber of the tee can at least provide the extrusion pressure, which satisfies the working pressure of the tee working pressure, while the pressure of the sealing rubber on the shell is set as 17.7MPa. Therefore, there are three primary load sources for the upper and the lower shells, including the maximum tested pressure of 15MPa, the reaction force of the slip at 2.7MPa, and the pressure of sealing rubber on the shell at 17.7 MPa. According to the maximum working pressure of the tee, the load on the inner surfaces of the upper and lower shell slips and the sealing rubber installation is set as 17.7MPa, while other internal pressures are set as 15MPa. The boundary condition dictates that the two shells are connected and fixed via the bolt holes.

![Fig. 6 Upper Shell and Lower Shell.](image)

To analyze the grid independence of the optimization process, the 17mm, 20mm, 25mm, 30mm, 35mm, and 40mm grids are each divided. The relationship curve between the grid size and the remaining structure volume after optimization processing is shown in Fig. 7, and the data are summarized in Table 1.
The relationship between the Grid Size and Structure Volume.

**Table 1. Change of Grid Size and Structure Volume.**

| Grid Size Range | 17-20 | 20-25 | 25-30 | 30-35 | 35-40 |
|----------------|-------|-------|-------|-------|-------|
| Upper Shell Change | 3.6% | 5.3% | 3.3% | 9.0% | 17.0% |
| Lower Shell Change | 0.5% | 4.5% | 5.9% | 13.8% | 3.9% |

Combined with the structure density nephogram and the calculation efficiency after optimization, a volume change value below 6% is considered a reasonable result. Although reasonable results are present when the upper and the lower shells divide the grid into 17mm-30mm portions, only 0.5% of the volume change in the lower shell is evident when the grid size of the lower shell is 17mm-20mm. Therefore, the smaller shell mesh is divided into 20mm-30mm due to calculation efficiency. When the grid sizes of the two shells are 35mm and 40mm, a larger gully will appear on the surface of the structure, damaging the structure reconstruction. With optimization and weight reduction in mind, the upper shell grid size is divided into 20mm segments, and the smaller shell grid size is divided into 25mm sections during topology optimization. Then, a reasonable upper and lower shell optimization density nephogram with a density threshold of 0.2 is selected, as shown in Fig. 8.

**Fig. 8** Upper and Lower Shell Optimization Density Nephogram.

**4. Model Reconstruction and Performance Evaluation**
Remodeling and strength checks are performed on the optimized parts, while the optimized model scheme was subjected to evaluation and analysis.
4.1. Remodeling and Strength Checks

The optimized density nephogram allows for the intuitive examination of the weight reduction area of the parts, as well as the remodeling of the design. A significant reduction is evident in the wall thickness of the upper and lower shells, and the remodeled structure is shown in Fig. 9.

A strength check is performed on the remodeled shell. The structural stress and deformation nephograms are shown in Fig. 10, and Fig. 11.

![Remodeling of the Upper and Lower Shells](image)

The maximum stress and maximum deformation of the upper shell are 274.16MPa and 0.43mm, respectively, while the maximum stress and maximum displacement deformation of the lower shell are 277.49MPa and 0.52mm, respectively. Besides the concentration effect of the local stress, the maximum
stress of both shells is not only lower than the yield stress of Q345 material but also satisfies the safety criteria of being above 1.5, while the deformation value also meets the usage requirements.

4.2. Structural Modal Analysis

To avoid resonance while the optimized tee is in use, the changes in the first six modal frequencies of the two shells before and after optimization are calculated, as shown in Fig. 12.

![Structural Modal Frequency Change Curve Before and After Optimization](image)

As shown by the curve, there is almost no change in the stiffness and the order frequencies of the lower shell before and after the optimization, while a substantial decrease is apparent in the frequency order of the upper shell after optimization. However, the natural frequency of both shells exceeds the vibration frequency of the tee during operation. Therefore, the resonance phenomenon is not present.

4.3. Optimization Performance Evaluation

Subjecting the upper and lower shells of the tee to a series of software evaluations produces an optimized design that presents enhanced weight reduction. Furthermore, the optimized part structure meets the strength requirements, and the summary of the part performance before and after optimization is shown in Table 2 and Table 3.

| Table 2. Structural Performance Data before Optimization. |
| Mass/kg | Maximum Stress/MPa | Maximum Deformation/mm | First Order Frequency /Hz |
|----------|-------------------|------------------------|--------------------------|
| Upper Shell | 7844 | 265.75 | 0.42 | 930.12 |
| Lower Shell | 9474 | 176.28 | 0.26 | 569.38 |

| Table 3. Structural Performance Data after Optimization. |
| Mass/kg | Maximum Stress/MPa | Maximum Deformation/mm | First Order Frequency /Hz |
|----------|-------------------|------------------------|--------------------------|
| Upper Shell | 4753 | 274.16 | 0.43 | 573.6 |
| Lower Shell | 5544 | 277.49 | 0.52 | 546.9 |

Following structural remodeling, the weight of the upper shell is reduced by 3109 kg, with
amaximum stress of 274.16MPa; the weight of the lower shell is reduced by 3930 kg, with a maximum stress of 277.49MPa; the total weight of the two shells is reduced by 7860 kg, with an overall weight reduction of nearly 26%. The safety factor of the parts after remodeling is maintained within a range of 1.2~1.6, meeting the strength requirements. A decrease is apparent in the first-order natural frequency of the remodeled upper shell, while a reduction in its stiffness is evident due to a decline in the shell thickness. However, the natural frequencies of both shells exceed the resonance frequencies during operation, meeting the stiffness requirements.

5. Conclusion

This paper introduces the theory of variable density topological optimization, while the key parts of the 914 mechanical tee are subjected to optimization processing using topological optimization software. The following three conclusions are drawn:

(1) During the optimization process, the grid size of the upper shell is not affected when it remains within 17mm-30mm, while these values are 20 mm-30 mm for the lower shell.

(2) After optimization, a weight reduction of 26% is evident in both the upper and lower shells of the entire 914 mechanical tee, ensuring compliance with the strength and deformation requirements.

(3) After optimization, the natural frequency of the structure meets the usage requirements and can be improved via optimization software such as setting stiffeners.

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