The Research of Solution Methods on the Marine Exhaust Ejector Device with Swerve

Abstract. The research of this paper is about solution methods on the marine exhaust ejector device with swerve. The whole system of the device is divided into two parts, including the filtration stage and the channel stage. The porous media model is established for the damping of filtration stage. The scaled model experiments are achieved through the similar conversion under the principle of keeping the same Mach numbers of the main air jet. The experimental results and numerical simulation results agree well. It can provide the theoretical basis for the research of the exhaust ejector with swerve and the large-size model.

Streszczenie Przedstawiona praca dotyczy metod rozwiązywania problemów związanych z okrągowym eżektor wyciągowym z odchyleniem. Urządzenie składa się ze stopnia filtrującego i stopnia kanału. W stopniu filtrującym do tłumienia przyjęto model ośrodka porowatego. Skalowanie modeli eksperymentalnych osiągnięto przez przekształcenie według zasady utrzymania tej samej liczby Macha w głównym strumieniu powietrza. Wyniki eksperymentalne i symulacji numerycznych są zgodne. Praca może stanowić bazę teoretyczną do badań wyciągowego eżektora z odchyleniem oraz modelu w większej skali. Badania metod rozwiązania urządzenia okrągowego eżektor wyciągowego z odchyleniem

Keywords: gas turbine, exhaust ejector, porous media, similar conversion
Słowa kluczowe: turbina gazowa, eżektor wyciągowy, ośrodek porowaty

Introduction
The changes of the pressure around the high speed fluid can inject the circumjacent fluid. The device depending on this kind of principle is called the ejector [1].

In the work course of the marine gas turbine, the engine room maintains a high temperature. If the necessary technology measure is not adopted, the operating temperature of engine room will be too high to operate the instrument and equipment normally, and the fire can occur easily [2]. Therefore the important mission of lowering cabin temperature has been undertaken by the application of exhaust ejector [3].

The gas turbine is covered with the shell, and a fixed interval is remained between the cover shell and the gas turbine outside surface [4]. The air inlet is founded in the corresponding position of cover shell, and the air inside the engine room can flow through the hot surface of the unit [4]. The exhaust ejector can take the part of heat quantity from the hot surface with the injected cold air, and it also can reduce the temperature of engine room [5, 6]. The injected cold air is always called the second air [7].

Fig 1 shows the general arrangement plan of the exhaust ejector on the ship.

Fig.1. Arrangement plan of the exhaust ejector

The front end of the device locates inside the engine room, and the back end exposes outside. The hot gas exhausts from the gas turbine is called the main air. The waste gas of the gas turbine exhausts into the atmosphere through the bent pipe of 90 degrees. The smoke gas possesses high temperature and speed, and it has the effect of drawing the cold air of engine room. The second air can cool the surface parts of unit and mix with smoke gas. The second air passes the air filtration which locates in the front of the engine room, in order to enter the engine room from the ocean atmosphere.

Solution tactics
Considering the general arrangement plan, the whole system is separated into two parts, namely the filtration stage and the channel stage. The filtration stage produces the negative pressure of the second air entrance near the exhaust ejector. Along with the difference of work condition, the flow rate of the second air and the pressure loss of the air filtration are also dissimilar. In order to research the whole system by the way of numerical simulation, the filtration stage should be changed into the boundary condition. The work emphasis is the numerical simulation of channel stage. The whole solution course is shown as the Fig2.

Fig.2. The solution course of whole system

Aerodynamic Characteristic of Filtration Stage
The second air entrance of the exhaust ejector locates inside the engine room. The second cold air will be drawn into the exhaust ejector by the exhaust gas through the air filter that locates in front of the engine room. (The structure of air filter is shown as the Fig 3.) When the air flows through the filter at a definite velocity, there will be a fixed pressure loss.
In order to get the real performance of exhaust ejector, the definite negative pressure should be given to the second air entrance as a boundary condition, when computing the different work conditions. The maximum pressure loss of the air filtration of the engine room is limited to less than 400Pa. During the simulation of the original model, the different negative pressure will be set to the second air entrance in dissimilar work conditions.

Fig.4 shows the relationship between negative pressure and flow rate of the second air under different work conditions, and the picture also includes the resistance of engine room air filtration. The ejection performance of a certain model under different work conditions can be got in this way, but at least three kinds of negative pressure should be set to the second air entrance. In order to get the flow rate of injected air, the result must be combined with the resistance of engine room air filtration. This method brings a lot of unnecessary troubles to the computational work; therefore, it is necessary to take some feasible measures to simplify the work of computing. The application of the porous medium model can convert the filtration stage into a kind of boundary condition. It can give the polynomial curve through the relationship of velocity and total pressure into a kind of boundary condition. It can give the polynomial curve through the relationship of velocity and total pressure brings a lot of unnecessary troubles to the computational work; therefore, it is necessary to take some feasible measures to simplify the work of computing. The application of the porous medium model can convert the filtration stage into a kind of boundary condition. It can give the polynomial curve through the relationship of velocity and total pressure.

**Porous Media Conditions**

Porous media are modeled by the addition of a momentum source term to the standard fluid flow equations. The source term is composed of two parts: a viscous loss term (Darcy, the first term on the right-hand side of Equation 1), and an inertial loss term (the second term on the right-hand side of Equation 1).

\[
S_i = - \left( \sum_{j=1}^{3} D_{ij} \mu v_j + \sum_{j=1}^{3} C_{ij} \mu \frac{1}{2} \rho |v|^2 v_j \right)
\]

Where \( S_i \) is the source term for the \( i \) th (x, y, or z) momentum equation, \( |v| \) is the magnitude of the velocity and \( D \) and \( C \) are prescribed matrices. This momentum sink contributes to the pressure gradient in the porous cell, creating a pressure drop that is proportional to the fluid velocity (or velocity squared) in the cell. To recover the case of simple homogeneous porous media

\[
S_i = - \left( \frac{\mu}{\alpha} v_i + C_{i2} \frac{1}{2} \rho |v|^2 v_i \right)
\]

Where \( \alpha \) is the permeability and \( C_{i2} \) is the inertial resistance factor, simply specify \( D \) and \( C \) as diagonal matrices with \( 1/\alpha \) and \( C_{i2} \), respectively, on the diagonals (and zero for the other elements).

It also allows the source term to be modeled as a power law of the velocity magnitude

\[
S_i = -C_0 v_i^{C_1 - 1} v_i
\]

Where \( C_0 \) and \( C_1 \) are user-defined empirical coefficients. In the power-law model, the pressure drop is isotropic and the unit of \( C_0 \) is SI.

When we define the porous medium boundary condition, the resistance performance of the filtration can be used according to the data in Table 1. The relationship of total pressure loss and velocity is used to draw up the synthetic quadratic polynomial with the resistance performance. The synthetic formula expression is shown as the follow form.

\[
\Delta P = 1.922 v^2 + 0.591 v
\]

The permeability \( k \) and the inertial resistance factor \( C_2 \) can be got form the Equation 4, and they should be attached to the second air inlet boundary.

**Mathematical model and boundary condition**

By using the Boussinesq vortex viscosity supposition, the Navier-Stokes equations of the compressible glutinous gas, neglecting mass force, may be described as follows:

**Continuity equation:**

\[
\frac{\partial \rho}{\partial t} + \mu \frac{\partial}{\partial x_i} (\rho u_i) = 0
\]

**Momentum equation:**

\[
\frac{\partial \rho (u_i u_j)}{\partial t} + \frac{\partial}{\partial x_j} \left\{ \rho \left( u_i + \frac{u_j u_i}{2} \right) \right\} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \tau_{ij}
\]

**Energy equation:**

\[
\frac{\partial}{\partial x_i} \left[ \rho \left( e + \frac{u_j u_i}{2} \right) \right] = \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \frac{\partial}{\partial x_i} \left[ \lambda \left( \frac{\partial T}{\partial x_i} + u_j \frac{\partial T}{\partial x_j} \right) \right]
\]

**Equation of state:**

\[
\frac{L}{\rho} = RT
\]

In the formula:

\[
\tau_{ij} = (\mu + \mu_t) \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_t \left( \mu + \mu_t \right) \frac{\partial u_i}{\partial x_j}
\]

\( \tau_{ij} \) is viscous stress tensor; \( \rho \) is density; \( \mu \) is component of velocity; \( P \) is pressure; \( e \) is unit mass internal energy; \( \lambda \) is thermal conductivity; \( T \) is temperature; \( \mu_t \) is dynamically viscous coefficient; \( \mu \) is turbulent velocity coefficient.

When neglecting the gravity effect, by using a \( k-\varepsilon \) turbulence model to simulate vortex viscosity coefficient, the turbulence kinetic energy \( k \) and its rate of dissipation \( \varepsilon \) are obtained from the following transport equations:

\[
\rho \left( \frac{\partial k}{\partial t} + u_j \frac{\partial k}{\partial x_j} \right) = \frac{\partial}{\partial x_j} \left[ \mu + \mu_t \frac{\partial k}{\partial x_j} \right] + G_k - \rho \varepsilon - Y
\]
(11) \[ \rho \left( \frac{\partial \varepsilon}{\partial t} + \mathbf{u} \cdot \nabla \varepsilon \right) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_e} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{\mu} \frac{\varepsilon}{k} G_1 - C_{\mu} \rho \frac{\varepsilon^2}{k} \]

In these equations, \( G_1 \) represents the generation of turbulence kinetic energy due to the mean velocity gradients. To evaluate \( G_1 \) in a manner consistent with the Boussinesq hypothesis,
\[ G_1 = \mu_t S^2 \]
Where \( S \) is the modulus of the mean rate-of-strain tensor, defined as:
\[ S = \sqrt{2S_{ij}S_{ij}} \]

The average strain rate tensor is given by the equation as follows:
\[ S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \]

\( Y_M \) represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, \( Y_M = \rho c_2 M^2 \).

Where \( M_t \) is the turbulent Mach number, defined as:
\[ M_t = \sqrt{\frac{k}{\alpha}} \]

The turbulent viscosity, \( \mu_t \), is computed by combining \( k \) and \( \varepsilon \) as follows:
\[ \mu_t = \rho C_p \frac{k^2}{\varepsilon} \]

The value about constant \( C_{1t}, C_{2t}, C_v \) and the turbulence Prandtl number \( \sigma_t \) and \( \sigma_v \) of \( k \) and \( \varepsilon \) given by B.E.Launer and D.B.Spalding[8] are respectively: \( C_{1t}=1.44, C_{2t}=1.67, C_v=0.08, \sigma_t=1.0, \sigma_v=1.5 \).

Table 2 Main air parameters in each work conditions

<table>
<thead>
<tr>
<th>Work condition</th>
<th>Temperature (K)</th>
<th>Flow rate (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>893</td>
<td>27.8</td>
</tr>
<tr>
<td>0.85</td>
<td>883</td>
<td>26.3</td>
</tr>
<tr>
<td>0.5</td>
<td>813</td>
<td>21.7</td>
</tr>
<tr>
<td>0.28</td>
<td>810</td>
<td>16.7</td>
</tr>
<tr>
<td>MC</td>
<td>833</td>
<td>9.2</td>
</tr>
</tbody>
</table>

Table 2 gives the gas parameters of the main air under different work conditions. The 1.0 work condition is Full-load condition, and the MC work condition is the Low-speed cruise condition.

The boundary conditions are as follows:
Reference pressure is 101325Pa.
Main air gas inlet: mass flow import, gas parameter under different work conditions can refer to the data in Table 2, airflow direction is normal.
The second air inlet: It is confirmed combining the formula (4) by using the porous media condition, the temperature is 328K.
Exportation: pressure outlet, the pressure is 101325Pa, and the temperature is 300K.
Solid wall: adiabatic, and do not have the slipping.
Turbulence model: standard k-\( \varepsilon \) model.

Result of numerical simulation

Fig.5 shows total pressure contours on longitudinal section of different work conditions using the porous media conditions. It shows the total pressure distribution circumstances of 1.0 and 0.85 operating modes of two pictures from left to right. The total pressure distributes on the vertical section is consistent, but the extremum is different.

Table 3 Numerical simulation result under porous media condition

<table>
<thead>
<tr>
<th>Work condition</th>
<th>Inlet velocity (m/s)</th>
<th>Pressure loss under porous media condition (Pa)</th>
<th>Calculation result from formula (Pa)</th>
<th>Relative error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>12.46</td>
<td>321.00</td>
<td>305.99</td>
<td>4.68</td>
</tr>
<tr>
<td>0.85</td>
<td>11.62</td>
<td>279.73</td>
<td>266.61</td>
<td>4.69</td>
</tr>
<tr>
<td>0.5</td>
<td>9.22</td>
<td>177.19</td>
<td>168.76</td>
<td>4.76</td>
</tr>
<tr>
<td>0.28</td>
<td>6.93</td>
<td>101.51</td>
<td>96.55</td>
<td>4.89</td>
</tr>
<tr>
<td>MC</td>
<td>3.69</td>
<td>29.98</td>
<td>28.35</td>
<td>5.46</td>
</tr>
</tbody>
</table>

The negative pressure value of the second air inlet which comes from the numerical simulation accords with value of the intersection in Fig4 under the several work conditions. This appearance proves the correct porous media condition used in numerical simulation. The porous media model can be used correctly in numerical simulation to replace the component which has the specific total pressure characteristic. The whole system can be simulated using this method, and the physics model is changed to numerical model under the porous media condition.

Similar conversion

The geometry size of marine gas turbine exhausts injection is very large, thus the experiment of full ruder model is hard to accomplish under the condition of laboratory. The experiment of reduced rate model is used to do the research in laboratory. The similar norm must be observed in the experimental research, but the request is only satisfied on the special section in the engineering research.

The similar criterion which should be observed in model experiment of marine gas turbine exhaust ejector contains several criterion numbers. They are Geometry similar, Mach number similar and the Reynolds number similar. The Prandtl number, Froude Number, Strouhal number and Euler number also should be obeyed in research. Generally speaking, the Geometry similar is easy to achieve by the way of designing the experiment model under the fixed proportion.

It is difficult to satisfy the Mach number and Reynolds number at the same time in the experiment course. When
the Reynolds number is bigger than the critical Reynolds number, the change of loss efficiency $\zeta$ will be very little. This kind of area is called self-modeling area. The effect of viscosity frictional force is very small in this area. The value of Reynolds number will not influence the flow in the self-modeling area. The Reynolds can be express as the formula:

$$Re = \frac{uD}{v}$$

where $v$ is coefficient of kinematic viscosity, $u$ is the velocity, and $D$ is characteristic length.

The Reynolds numbers on the surface of main air jet is $10^7 > 1\times10^5$ through computing result and dimensional analysis, and the flow is in the self-modeling area. The Mach numbers on the surface of main air jet in experimental model is identical with situation of the real model.

In this article the conversion between the different states and geometric dimension models should be under the principle of keeping the same Mach numbers. The parameter of the model with 1:1 rate and hot state should be confirmed in first step. The parameter of the model with 1:1 rate and cold state should be changed under the principle of same Mach numbers on the main air jet. The parameter of the model with 1:4.15 and cold model will be getting in the same way.

The rate of the experimental model is 1:4.15. The result of numerical simulation can be compared with the experimental result, in order to validate the correctness of numerical simulation. And the result of model with 1:1 rate and hot state also can be validated.

The different states models will be changed as the follow relationship, when the rate is 1:1 and neglecting the influence of viscosity is neglected. In the formula, the $h$ means the hot state, and the $c$ means the cold state.

$$Q_h = \rho_hAV_h, \quad Q_c = \rho_cAV_c$$

$$\frac{Q_h}{Q_c} = \frac{P_h}{P_c} \frac{V_c}{V_h} = \frac{P_h}{P_c} \frac{T_c}{T_h}$$

When the change of the pressure is not much, it can be neglected. The flow ratio in different states in the same rate is the inverse ratio of square root about the temperature. It can get the main air flow value of the same rate about the cold model by using this formula, in order to replace the hot model. The different states models in the same ratio can be converted by using this formula.

In cold models, the flow rate of different scale models is decided by the ratio of area. We can get the relationship between the model with 1:1 rate with hot state and the model with 1:4.15 rate with cold state, through the conversion by the model with 1:1 rate with cold state. In other words, the big scale and hot model can be replaced with the little scale and cold model in experiment research. This can reduce the processing costs of experimental model. The experiment requirement of practical model on the ship also can be satisfied under the laboratory condition.

Table 4 gives the statistic results of different rates and state models. When the main air temperature is 300K, the inject coefficient of the ratio with 1:1 is bigger than the 1:4.15’s. The difference of hot state inject coefficient is 1.18% when the models are the same scale.

Generally speaking, the performance of the model with 1:1 rate in hot state can be replaced with ones of the model with 1:4.15 rate in cold state. The experiment requirement of practical model on the ship also can be satisfied under the laboratory condition.

**Experiment research**

The air blower put the air with the fixed pressure into the main air pipeline, which is linked with the air blower, in order to simulate the main air with cold state. The different work conditions can be controlled by changing the flow rate of the air blower. The second air inlet of the exhaust ejector lies inside of the pressurize tank, and there is an inlet pipeline for the second air on the top of the tank. The second air is pumped into the pressurize tank through the inlet pipeline which has several flow measuring instruments. The inlet of the exhaust ejector lies inside of the pressurize tank, and the outlet lies outside. The flow rate should be regulated under the fixed work condition during the experiment course. The parameter of the main air including the flow rate and temperature should be recorded. Under the same work condition, several group experiments with different negative pressure of the second air inlet should be done. The flow rate, temperature and the negative pressure of the second air should be recorded.

![Fig.6. The arrangement of experimental facilities](image)

**Fig 6 shows the arrangement of experimental facilities**

And the Fig 7 shows the local arrangement of the experiment equipments and the around devices.

![Fig.7. The local arrangement of experimental facilities](image)

The flow of main air should be adjusted by using the conversion relationship between the model of 1:1 rate with hot state and the model with 1:4.15 rate with cold state, in order to keep the Mach number coincident of the two kinds of state, under the laboratory condition.

The numerical calculation will be done by using the several experimental data, in order to verify the accuracy of the numerical computation. If the trend between the numerical simulation result and experimental data is consistent, the inject capability of the exhaust ejector under the 1.0 work condition can be replaced by the result of numerical simulation.

<table>
<thead>
<tr>
<th>Ratio</th>
<th>Main air temperature (K)</th>
<th>Main air flows (kg/s)</th>
<th>Second air Flows (kg/s)</th>
<th>Inject coefficient (%)</th>
<th>Hot inject coefficient (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1:1</td>
<td>893</td>
<td>27.80</td>
<td>5.21</td>
<td>18.75</td>
<td>18.75</td>
</tr>
<tr>
<td>1:1</td>
<td>450</td>
<td>39.16</td>
<td>5.51</td>
<td>14.07</td>
<td>19.82</td>
</tr>
<tr>
<td>1:1</td>
<td>300</td>
<td>47.96</td>
<td>5.54th</td>
<td>11.55</td>
<td>19.93</td>
</tr>
<tr>
<td>1:2</td>
<td>300</td>
<td>11.99</td>
<td>1.38</td>
<td>11.47</td>
<td>19.79</td>
</tr>
<tr>
<td>1:4.15</td>
<td>300</td>
<td>2.79</td>
<td>0.32</td>
<td>11.39</td>
<td>19.65</td>
</tr>
</tbody>
</table>
Table 5 shows the computation data comparing with the experimental results under the several work conditions. The trend between the computation data and experiment result is consistent. The contrast situation is shown as the Fig 8.

Table 5. The compare of computation and experiment result

<table>
<thead>
<tr>
<th>Work condition</th>
<th>Inlet resistance (Pa)</th>
<th>MF (kg/s)</th>
<th>SFNS (kg/s)</th>
<th>ICNS (%)</th>
<th>SFE (kg/s)</th>
<th>ICE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MC</td>
<td>-56</td>
<td>0.898</td>
<td>0.099</td>
<td>10.97</td>
<td>0.083</td>
<td>9.20</td>
</tr>
<tr>
<td>0.28</td>
<td>-120</td>
<td>1.616</td>
<td>0.224</td>
<td>13.84</td>
<td>0.172</td>
<td>10.67</td>
</tr>
<tr>
<td>0.5</td>
<td>-250</td>
<td>2.012</td>
<td>0.251</td>
<td>12.45</td>
<td>0.214</td>
<td>10.62</td>
</tr>
<tr>
<td>0.85</td>
<td>-360</td>
<td>2.587</td>
<td>0.334</td>
<td>12.91</td>
<td>0.285</td>
<td>11.00</td>
</tr>
</tbody>
</table>

In table 5, MF means main air flows, SFNS means second air flows under numerical simulation, ICNS means inject coefficient under numerical simulation, SFE means second air flows under experiment, ICE means inject coefficient under experiment.

Conclusions

The performance of the marine gas turbine exhaust ejector has been researched by combining the method of numerical simulation and experimental research, and the conclusions are summarized as follows:

1. The whole system is divided into two parts, including the filtration stage and the channel stage. The porous media model is established for the filtration stage damping, and the whole system can be simulated successfully.

2. The similar conversion relationship between the exhaust ejector device models at different scales and working conditions is established, under the principle of keeping the same Mach number on the special section.

3. The performance of exhaust ejector device can be tested under the laboratory conditions, and the trend of experimental results with numerical simulation is consistent.

The ejector performance data under different exhaust ejector device structure parameters should be accumulated in the future. The appropriate database of design should be established, and it can provide the design reference for the exhaust ejector device. The measurement problem about the flow field velocity and pressure distribution in experimental research should be solved in the further work.

REFERENCES


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