A novel hydrogen turbo-expander with active magnetic bearings and an eddy current brake

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Abstract. A cryogenic expander for hydrogen liquefier with active magnetic bearings (AMB) has been designed and will be made and tested. The shaft design and strength review are significant factors for the turbine since it affects the performance and safety of the expander significantly. In order to produce lower temperature hydrogen, the liquefier needs a very small and high-speed turbo-expander. However, there are few studies on expander with both active magnetic bearings and eddy current brakes. In this study, the speed of the turbine is up to 100000 rpm. Its cooling power is about 4 kW. The inlet pressure is 1MPa, and the outlet pressure is 0.7MPa. Under this condition, the turbine is designed, and the simulation is done. An eddy current brake which can govern the rotational speed quickly is designed and studied as well.

1. Introduction

Hydrogen is a clean and non-toxic secondary energy source. Liquid hydrogen is the most effective way to store and transport hydrogen energy [1]. It has the characteristics of large storage density, high specific impulse, and low boiling point. It has been widely used in transportation and aerospace propulsion and other fields [2]. With the vigorous development of the liquid hydrogen market, research on large-scale hydrogen liquefaction equipment has attracted more and more attention. Hydrogen turbo-expander is the most critical and core component of the large-scale low-temperature hydrogen liquefier recognized by the international cryogenic refrigeration industry. Its efficiency and stability determine the first performance of the entire liquid hydrogen system. The density of hydrogen is small, so it will bring new problems different from others, such as high speed, leakage, low efficiency, etc., resulting in more severe working conditions and increased difficulty in development.

AMBs has the advantages of fast response, sensitive control, low power loss, no mechanical wear and no loss of process gas, and with the continuous breakthrough of technology, the cost is getting lower [3]. Compared with oil bearings, AMBs are free of pollution. Compared with gas bearings, AMBs have a stronger bearing capacity [4]. The working conditions experienced by the hydrogen turbo-expander is very complicated, and the control of the turbine speed needs to be accurate and timely. The brake function is mainly to maintain the turbine speed and accept the power of the expansion and output of the corresponding mechanical work. Turbine brakes are primarily available in oil brakes, fan brakes and electromagnetic brakes [5]. As an electronically controlled non-contact braking technology, eddy
current braking has the characteristics of smooth braking, simple control, fast response, and easy to be accurately controlled. The first step of the current work is to verify the application effect of the AMBs in the hydrogen turbo-expander. Therefore, the AMBs hydrogen turbo-expander using a fan to brake is designed first, and the whole machine is processed and tested. The research on the braking effect of eddy current brake is done separately in the paper.

The main contents of this paper are as follows: section two describes the overall structure of the turbo-expander. In section three, the working impeller is designed and simulated by Ansys CFX. In section four, the main parameters of the AMBs are determined. Section five shows the results of Modal analysis and unbalanced responses. Stress analysis and deformation analysis is done in section six. In section seven, the eddy current brake is designed and simulated.

2. Hydrogen turbo-expander overall structure
The main structure of the hydrogen turbo-expander currently designed is shown in figure 1. The turbine is vertically arranged. The impeller is mounted on the lower end of the shaft, and the fan is mounted on the upper end. Both of them are fixed by nuts. The turbo-expander is used in hermetic system. In order to reduce the loss of refrigeration capacity caused by gas leakage, a labyrinth seal is provided on the impeller side and the fan side, respectively. The axial AMB is placed in the middle of the two radial AMBs. The thrust disk and the spindle are integrated. The turbine is at cryogenic temperatures and the bearings are at room temperature. In order to reduce the heat leakage, the heat-insulator made of rigid polyurethane foam is used at both the ends, which has better heat insulation effect in low temperatures [6]. The Spiral grooves are arranged on the rotating shaft. To avoid contamination, use hydrogen as the seal containment gas. At the cryogenic temperature end, the gas seal has also been used. The pressure of the leaked leaking hydrogen decreases under the action of throttling in the spiral grooves. The room temperature seal gas enters from the inlet to the mid piece of the grooves and provide a pressure slightly higher than that of the low temperature hydrogen (about 0.1 MPa higher), to prevent the leakage of low-temperature hydrogen. The leaked hydrogen, mixing with the seal gas, fills the cavity of bearings housing.

![Figure 1. Structure components of hydrogen turbo-expander.](image)

3. Aerodynamic design and simulation
The aerodynamic design is the first and one of the most crucial parts of the turbo-expander design. The turbo-expander designed in this paper adopts a semi-opening radial reaction turbine, which has the advantages of higher enthalpy-drop ratio, high rotational speed, simple structure, and high thermal efficiency.

3.1. 1D design and 3D modeling
The initial design parameters, such as the flow rate, the inlet temperature, the inlet pressure, and the expansion ratio are then determined according to the process. Then select the absolute and the relative
flow angles of the impeller inlet, the impeller speed coefficient, the nozzle speed coefficient, and other parameters, and then optimize the impeller diameter ratio and the reaction degree, and calculate the heat according to the parameters after the optimization to obtain the basic parameters of the turbine.

After determining the original size, three-dimensional modeling is performed in the Ansys Bladegen. The nozzle adopts pneumatic cascades of type TC-P2, which has small flow loss, significant speed coefficient, and excellent aerodynamic performance. However, its processing is complicated and requires high processing precision. It is convenient to create the meridian plane and ensure the installation angle of the impeller by Ansys Bladegen. Then the software will create a 3D model and provide access to Ansys CFX. The basic parameters of the turbine are shown in Table 1.

**Table 1. Main parameters of the hydrogen turbo-expander.**

| parameter                      | value  |
|-------------------------------|--------|
| Flow rate (g/s)               | 50     |
| Inlet pressure (MPa)          | 1      |
| Inlet temperature (K)         | 80     |
| Outlet pressure (MPa)         | 0.7    |
| Outlet temperature (K)        | 73.7   |
| Rotational speed (rpm)        | 100000 |
| Impeller diameter (mm)        | 42     |
| Isentropic efficiency of turbine | 74.2% |

**3.2. Simulation and analysis**

It can be seen from figure 2 that there is apparent leakage vortex in the impeller. It can be seen from figure 3 that there is a tremendous pressure difference from the pressure surface to the suction surface in the tip clearance. The pressure difference generates a lateral leakage, and a leakage vortex is generated after the leakage flow is mixed with the mainstream. The main factors affecting the leakage are the impeller speed and the nozzle installation angle because they affect the mainstream airflow angle, thereby changing the velocity triangle at the turbine inlet. When the velocity triangle does not constitute the right triangle, a lateral leak occurs, which will form a leakage vortex. Since the operating conditions and materials limit the rotational speed, the leakage vortex can be eliminated by adjusting the nozzle installation angle [7].

It can be seen from the velocity stream that there is a large velocity at the outlet of the nozzles, and the fastest point is up to 494 m/s. But the mass of this part of the airflow is minimal and disappears quickly after entering the impeller passages. The leading edge of the TC-2P nozzles is thinner, and this airfoil design improves its aerodynamic performance at high subsonic speeds. At shallow installation angles, even supersonic flow can be achieved.

![Figure 2. 3D velocity streamline map.](image1)

![Figure 3. 3D pressure contour map.](image2)
4. The structure design of axial and radial AMBs
Because of the pressure difference among the different parts of the impeller, an axial force is applied to
the rotor. The empirical formulas calculate it to get 430N by integration and direction is directed to the
impeller end [8]. Similarly, the fan also produces an axial force, which is calculated in the same way.
The vector sum of the rotor gravity, the axial force of the work impeller and the fan is the axial force of
the rotor. The axial force of the rotor was calculated to be 85N, and the design value of the axial bearing
maximum capacity is taken as 120N. According to the unbalance and the rotation speed of the rotor, the
maximum radial capacity is taken as 20N. Based on the capacities of AMBs, the structure parameters of
AMBs can be figured out by electromagnetism formulas [3]. The stator of the axial AMB is made into
a folding type to reduce the outer diameter of the thrust disk and thus increase the maximum rotation
speed. The parameters are shown in table 2.

| parameter                                              | Value       |
|--------------------------------------------------------|-------------|
| Radial AMB                                             |             |
| Number of magnetic numbers                             | 8           |
| Load capacity (N)                                      | 20          |
| The interior/outer diameter of a stator (mm)            | 36/77       |
| Axial length (mm)                                      | 6.5         |
| Maximum current (A)                                    | 6.5         |
| The radial air gap between bearing and rotor (mm)       | 0.5         |
| The radial air gap between auxiliary bearing and rotor (mm) | 0.1       |
| Axial AMB                                              |             |
| Load capacity (N)                                      | 120         |
| The interior/outer diameter of the thrust disk (mm)     | 35/50       |
| Maximum current (A)                                    | 6.5         |
| the axial air gap between bearing and rotor (mm)        | 0.4         |
| The axial air gap between auxiliary bearing and rotor (mm) | 0.2       |

5. Rotor dynamic design and simulation

5.1. Modal analysis
According to the natural stiffness calculation formula, the stiffness of the AMBs is set to 150N/mm [9].
Rotor dynamics analysis was performed by Dyrobes and Ansys Modal, respectively. Because in
engineering practice, the key point is the modes during the rotation process, so the axial stiffness is
ignored. Besides, because the bearing damping is small at high frequencies and affects the critical speed
little, it is ignored as well.

Figure 4 shows the results which are obtained by Dyrobes and Ansys Modal severally, ensuring the
reliability of the simulation. The mode shape appears to be tapered at the first-order critical speed, and
the translational motion occurs in the second order. Bending occurs in the third order. There is 140%
safe margin between rated speed (1666.7Hz) and vortex frequency of the first-order bending mode so
the rotor can be seen as a rigid rotor. The safe margin meets the API Standard 617 [10]. In the operating
speed range, the modal excitation responses of the low-frequency band is much larger than the excitation
response of the high-frequency band.
Figure 4. (a) 1st critical speed mode shape with modal frequency 65.06Hz by Ansys Modal; (b) 1st critical speed mode shape with modal frequency 61.59Hz by Dyrobes; (c) 2nd critical speed mode shape with modal frequency 89.09Hz by Ansys Modal; (d) 2nd critical speed mode shape with modal frequency 87.35Hz by Dyrobes; (e) 3rd critical speed mode shape with modal frequency 4007.90Hz Ansys Modal; (f) 3rd critical speed mode shape with modal frequency 3999.15Hz by Dyrobes.

5.2. Unbalance responses

The amount of unbalance is applied to the rotor according to G2.5 standard, and the Dyrobes software is used to analyze the vibration responses and bearing force in the low-frequency range. The assumed axial positions of the residual imbalances are at the two ends of the shaft. Because the rotor unbalanced responses are most sensitive to the unbalance at two ends of the shaft, it allows for the consideration in most extreme case.

The amplitudes depending on different speed, is shown in figure 5. The peaks of the amplitudes appear at 3700 pm and 5200 rpm respectively. It is worth noting that the amplitudes of lower and upper auxiliary bearings at 3700 rpm are 0.18 mm and 0.17 mm respectively, exceeding the air gap of them. When the turbine starts and stops the stiffness of AMBs should be increased temporarily. In the start-stop stage, raising the current of the radial bearings instantaneously can increase the stiffness to 300N/mm and the drastic unbalanced response will be delayed at 3700 and 5200rpm. The displacement of the impeller will meet the peak of 0.015mm at 4100rpm, which is much smaller than the distance
between blade tip and volute support of 0.1mm. The stiffness return to 150N/mm after passing 5500rpm, and then the fierce vibration stage in starting or stopping is crossed. The two peaks of the lower radial AMB are 12N and 9N, respectively. The two peaks of the upper bearing are 12N and 9.5N, respectively. Both the maximum bearing forces of two radial AMBs is within the capacity of itself.

![Amplitudes of lower sensor in different rotation speed](image1)

![Amplitudes of upper sensor in different rotation speed](image2)

![Amplitudes of lower radial auxiliary bearing in different rotation speed](image3)

![Amplitudes of upper radial auxiliary bearing in different rotation speed](image4)

**Figure 5.** (a) Amplitudes of lower sensor in different rotation speed; (b) amplitudes of upper sensor in different rotation speed; (c) amplitudes of lower radial auxiliary bearing in different rotation speed; (d) amplitudes of upper radial auxiliary bearing in different rotation speed.

![Support force of lower radial AMB in difference rotational speed](image5)

![Support force of lower radial AMB in difference rotational speed](image6)

**Figure 6.** Support force of lower radial AMB in difference rotational speed.

**Figure 7.** Support force of lower radial AMB in difference rotational speed.

**6. Stress analysis**

By the analysis of the deformation whose results is displayed in figure 8, it is found that the maximum deformation appears at the top of the blade, which is 0.026 mm. In this paper, the clearance between volute and blade tip is 0.1mm. Therefore, the shape deformation is not enough to cause a scrape. The fan and the work impeller have little deformation due to low material density and small mass inertia.
Through stress analysis, it can be concluded from figure 9 that the maximum stress occurs at the rotor thrust disk, and the maximum stress is 148.28MPa. The stress of each component on the rotor is less than the yield strength of the material, meeting the safety requirements.

7. Eddy current brake
An eddy current brake for auxiliary speed regulation is designed in accordance with the output shaft work of the turbo-expander. The structure of the eddy current brake is shown in figure 10. Wind the energized solenoid in the direction of the central axis. A magnetic field is excited by copper coils, and the boss on the rotor introduces the magnetic field into the external static metal conductor to generate an eddy current on the metal conductor. The induced magnetic field generated by the eddy current products a braking torque to the rotating shaft.

The magnetic stimulation is done by Ansys Maxwell. The braking power near the operating speed is shown in figure 11. It can be seen that the braking power reaches more than 10% of the impeller output power, which can quickly adjust the impeller speed and assist the braking.

8. Conclusion
A high-speed hydrogen turbine expander with AMBs is designed with a rated speed of 100,000 rpm and an efficiency of 74.2%. Under the rated working condition, the design and three-dimensional numerical simulation of the impeller have been carried out. The defects in the impeller flow field have been found, the reasons have been analyzed, and improved methods have been proposed. Modal analysis and unbalanced responses analysis has been performed on the rotor. A countermeasure is proposed for the modal excitation response of the low-frequency band, and at the same time, there is enough margin for the working speed and the critical bending speed. Deformation analysis and stress analysis of the rotor ensure that the deformation and material stress is within safe limits. An electromagnetic eddy current
brake that can be used for turbo-assisted auxiliary braking and rapid speed regulation was designed. These designs and results provide a basis for subsequent improvements in design and experimentation. Based on the existing work, the eddy current brake which can match the turbine power absolutely will be invented in the future and replace the braking fan. The novel design will combine AMBs and eddy current brake and remove fan finally, which will give it some unique advantages. Some existing turbo-expanders employees gas bearings and gas brake, including fan and compressor [11] [12]. The novel design will eliminate bearing gas, seal gas, and brake gas, reducing the total amount of gas to improve system efficiency. And complex structures and related pipelines will be left out such as labyrinth seal and heat exchanger of brake, which will simplify the system. A hydrogen turbo-expander adopts oil bearings and oil brake [13]. In contrast to it, the novel design can eliminate the oil contaminate and the oil circuit is also removed, which will reduce the volume of the expander and simplify the system. Compared with all of existing ones, the novel turbo-expander can change stiffness more quickly and accurately and the disturbance resistance is stronger. And it can regulate the speed more timely when disturbance occurs and stabilize the rotor with eddy current brake.

Currently, the hydrogen turbine expander with AMBs has been designed and will be manufactured. The experiment will then be carried out by the end of 2019. The hydrogen turbo-expander, which uses both AMBs and eddy current brake, will be manufactured and tested in the future.

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