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A reverse turbo-Brayton cycle cryocooler for ZBO storage of liquid hydrogen in space

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Abstract—Liquid hydrogen is a high-specific-impulse propellant which is widely used in aerospace. Hydrogen storage is a key problem in the long-distance space exploration. In recent years, zero boil-off (ZBO) storage of cryogenic propellants (LH2, LO2, LCH4) is a significant storage technology for space exploration where the cryocooler is used to achieve ZBO. However, it is difficult to achieve the large cooling capacity at 20 K in space. The reverse turbo-Brayton cycle cryocooler has the obvious advantages of large cooling capacity, low vibration, high reliability and long life which is used as the cryocooler in ZBO system. In this study, a reverse turbo-Brayton cycle cryocooler combining nitrogen and helium cycles is designed to provide the cooling capacity at 20 K and 90 K for liquid hydrogen and liquid oxygen. The nitrogen cycle in the system can provide the cooling capacity of 20 W at 90 K for liquid oxygen storage while the helium cycle in the system can provide the cooling capacity of 5 W at 20 K for liquid hydrogen storage. There are three centrifugal compressors and one expander in the nitrogen cycle. The expander is coaxial with the one of compressors which can recover the expansion work. There are four centrifugal compressors and one expander in the helium cycle. A heat exchanger is used to connect the nitrogen cycle and helium cycle. The nitrogen cycle can provide the cooling capacity for the pre-cooling of the helium cycle. In addition, the cryocooler can also provide the cooling capacity of 200 W at 90 K for liquid oxygen and 200 W at 120 K for liquid methane by the nitrogen cycle. The reverse turbo-Brayton cycle cryocooler in this study can achieve the ZBO storage of liquid hydrogen, oxygen and methane in space. This work can provide the theoretical guidance for the design of ZBO storage in the long-distance space exploration.

Keywords—zero boil-off; hydrogen storage; reverse turbo-Brayton cycle cryocooler

I. BACKGROUND

Propellant storage in orbit is a technical basis for many space technologies such as long-term spaceflight. Cryogenic propellants used in aviation have low boiling point and are easy to evaporate. In order to realize long-term storage in orbit, thermal insulation is needed to be realized. Foam insulation and variable density multi-layer insulation as shown in the above technologies can achieve a high degree of thermal insulation in the low temperature region, but they still cannot completely isolate heat leakage, which leads to slow evaporation of propellants and rising storage pressure[1]. Periodic exhaust to reduce the tank pressure has great technical obstacles. Firstly, its carrying capacity is greater than the rated liquid demand, which significantly increases the spacecraft load and launch cost. Secondly, the gas-liquid two phases cannot be separated automatically in space microgravity environment, so gas-liquid separation is required during exhaust, which increases the technical complexity. Finally, the propellant gas discharged in the microgravity environment diffuses around the spacecraft, causing security risks. Therefore, long-term storage of cryogen in space requires a system to avoid loss of the propellant [1].

A propellant zero boil-off storage (ZBO) system proposed by NASA avoids this problem by efficiently controlling the temperature of cryogenic propellants such as liquid hydrogen, liquid oxygen, and liquid methane in orbit and eliminating propellant gasification. On the basis of the insulation device arranged on the periphery of the storage tank, a cryogenic refrigerator is equipped to proactively produce the cold volume of the liquid storage temperature region, and maintain its low temperature by absorbing the liquid storage heat through the heat exchanger. The heat produced by the refrigerator is discharged to space from the radiator in the form of thermal radiation. This combination of passive adiabatic and active refrigeration can effectively control the temperature of propellant and completely prevent liquid storage gasification to achieve ZBO storage of propellant [2].

After a series of development, ZBO system technology has experienced the cryogenic refrigerator coupled with copper blade cooling propellant, cryogenic refrigerator coupled with heat pipe cooling propellant, cryogenic refrigerator coupled with storage tank forced convection cooling propellant, and developed to the cryogenic refrigerator coupled with cryogenic cooling screen to achieve ZBO of propellant. In this technology, the outer side of propellant tank is equipped with multi-layer insulation device (MLI) to reduce heat leakage and achieve passive insulation. The cryogenic refrigerator adopts the reverse Brayton refrigeration form, and the low temperature working medium absorbs heat through the large area cooling screen (BAC) wrapped outside the storage tank to realize the long-term ZBO storage of liquid hydrogen and liquid oxygen and other propellants[1].

At high cooling loads, the turbo-Brayton cycle has a clear advantage over other cycles in system performance, size, and mass. In addition, the continuous–flow nature of the Brayton cycle is ideal for cryogen storage missions where the cycle gas can be directly interfaced with a broad area cooling (BAC)system attached to the storage tank [3].

The reverse Brayton refrigeration cycle uses single-phase helium gas as the working medium. Three centrifugal compressors in series compress and circulate the circulating gas. A rear cooler built into the compressor housing takes away the heat generated by the compressor, which is repelled into space using a passive radiator. The high-pressure air flow leaves the compressor and the post-cooler and enters the regenerator, and the low-pressure air flow returning from the cold end of the cryocooler is precooled. After leaving the recuperator, the high-pressure circulating gas expands through the turbine to produce refrigeration. The expansion process of the turbine cools the circulating gas, generating shaft power that can be dissipated or restored at the warm end of the cooler. Turbine power recovery is selected to improve system efficiency. This recovery is accomplished by converting shaft power into electrical energy in an alternator. The turbine outlet flow is then directed to a large cooling load interface where the circulating gas absorbs heat from the hydrogen tank and enters the low-pressure side of the cooler. Use an accumulator at the warm end of the cryocooler to limit the maximum pressure of the cryocooler at room temperature [4].

Compressor is the key component of a reverse Brayton cryocooler, which determines the power consumption of the system [5]. High speed centrifugal compressors have advantages of operation stability, very low vibration, and long service life, and is a promising candidate. A 20K-20W ZBO three-stage high speed centrifugal compressor designed by NASA is a major breakthrough of high speed mini centrifugal helium compressor in recent years. The compressor design is a modified version of Creare’s 400-500 W class permanent magnet compressor[5]. Each compressor is identical except for the aerodynamic design of the impeller and diffuser. Based on a new permanent magnet motor design, motor capacity extends from 400 W to 500 W. The journal bearing diameter is 6.4 mm and the impeller diameter is 19 mm. With the help of self-lubricating air bearing, its speed can exceed 6,300 rev/s with a total pressure ratio of 1.45. The three stage efficiency of the compressor exceeds 63% [6].

II. THE REVERSE TURBO-BRAYTON CYCLE SYSTEM

A. *System parameters*

A schematic of the reverse turbo-Brayton cycle system is shown in Fig. 1 The turbo-Brayton cycle cryocooler we designed combining nitrogen and helium cycles.

In helium cycle , The mass flow rate is 4 $g \cdot s^{-1}$, and the pressure ratio is about 1.58 through four-stage compression. The cooling capacity of 5 W in the temperature zone of 20 K can be obtained by using the turbine expander to achieve expansion temperature drop at the low temperature side. After each stage of compression (C-101, C-102, C-103, C-104), the working medium enters the stage after cooler, and the heat is discharged to reduce the temperature to 300 K. After the last stage is compressed, it flows into the regenerator R, and the heat exchange temperature with the low-temperature measuring medium decreases from 300 K to 21.55K, due to the flow resistance pressure drops from 482 kPa to 472 kPa. Then enter the turbine expander T to expand and cool to 19.15K, 316 kPa and enter the large area cooling plate (BAC) to absorb heat. The high and low temperature zones of the system are 300 K and 20 K, respectively. The total power consumption of the compressor is not higher than 2000 W, the designed Carnot efficiency is 3.5%, and the isentropic efficiency of each stage of compression is not less than 67%.

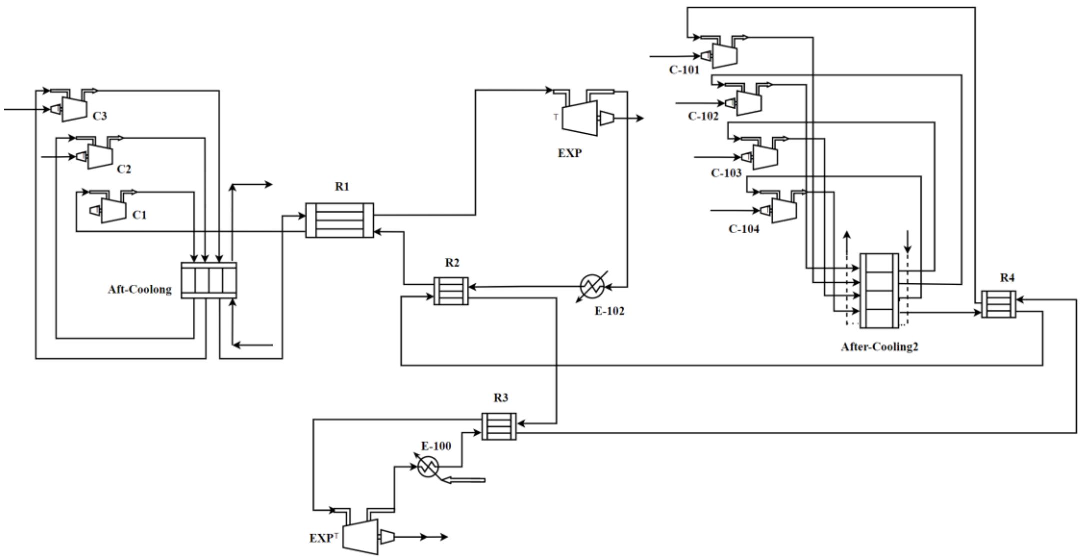

Fig. 1. The reverse turbo-Brayton cycle with nitrogen and helium cascade refrigeration

TABLE I. HELIUM COMPRESSOR DESIGN PARAMETERS

parameter	C-101		C-102		C-103		C-104	
	inlet	***outlet***	***inlet***	***outlet***	***inlet***	***outlet***	***inlet***	***outlet***
Pressure/kPa	303	344	342	388	386	435	433	484
Tempreture/K	299	321	300	322	300	321	300	321
Pressure ratio	1.137		1.133		1.126		1.119	
Power consumption /W	<500		<500		<500		<500	
Efficiency	>67%		>67%		>67%		>67%	

The nitrogen cycle is similar to the helium cycle, The mass flow rate is 32 g·s^{-1}, and the pressure ratio is about 3.77 through three-stage compression. The nitrogen cycle system can provide 20 W cooling. And also nitrogen can achieve expansion temperature drop at the low temperature side by using the turbine expander. The working medium enters the stage after cooler after each stage of compression, and the heat is discharged to reduce the temperature to 300 K. After the last stage is compressed, it flows into the regenerator aft-cooling. Then enter the turbine expander T to expand and cool to 90.15K, 138 kPa. The total power consumption of the compressor is not higher than 4.61 kW, and the isentropic efficiency of each stage of compression is not less than 76%.

In this system, the pressure losses of R1, R2, R3 and R4 are 2 kPa, 1 kPa, 1 kPa and 2 kPa, respectively. The total power consumption of hole reverse turbo-Brayton cycle system is about 6.43 kW. Based on the above system pressure ratio and efficiency requirements, the helium compressor design thermal parameters are shown in the following table:

A. *System optimization*

The two-stage compressors are arranged symmetrically on both sides of a single compressor, as shown in Fig. 2. The cooling between the two-stage compressors is carried out to reduce the working medium temperature. Interstage cooling airflow is used to cool the bearing and the motor stator. The working medium with a mass flow of no more than 5% is led from the second stage outlet to the inner cavity of the compressor shell, and then discharged to the first stage outlet pipeline of the compressor from the outlet at the other end of the shell. The weight of a single helium centrifugal compressor is no more than 4.5kg, and the size is simple. There are two compressors with four-stage compression.

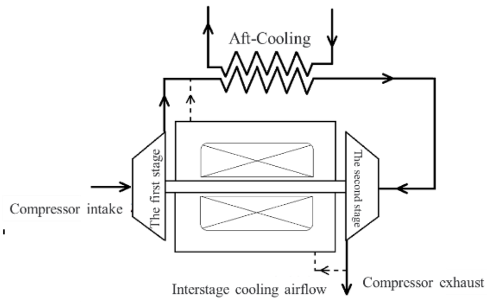

Fig. 2. Schematic of a two-stage centrifugal compressor

In addition, in order to improve the efficiency of the whole system and reduce energy consumption, a high-speed compression-electric-drive turbine expansion integrated machine is used in the system. The output work of the turbine expander compensates part of the compressor power during the working process.

III. DESIGN OF HIGH-SPEED CENTRIFUGAL COMPRESSOR

The impeller, diffuser and volute constitute the flow channel of the compressor, which determines the compression performance of the helium compressor and is the key point of design. Basic dimensions of impeller are shown in TABLE II.

TABLE II. HELIUM COMPRESSOR DESIGN PARAMETERS

parameter	C-101	C-102	C-103	C-104
Inlet diameter (wheel cover) /mm	14.67	14.11	13.83	13.55
Inlet blade height/mm	2.14	1.99	1.94	1.89
Outlet blade height /mm	1.16	1.1	1.1	1.1
Wheel diameter /mm	25.3	23.9	22.9	21.9

It has higher efficiency and adopts closed rear impeller. Each stage has 8 main blades and 8 small blades without pre-rotation. The blade thickness, blade Angle and inlet and outlet flow velocity of leading and trailing edges are shown in TABLE III. , where the blade Angle is the value at 1/2 meridian plane. The three-dimensional model of the first-stage impeller is shown in Fig. 3.

TABLE III. IMPELLER BLADE PARAMETERS OF CENTRIFUGAL COMPRESSOR

parameter	C-101	C-102	C-103	C-104
leading edge leaf angle/°	34.3	34.8	34.1	33.5
Caudal margin leaf angle /°	37.3	68.6	73.2	77.9
Leading edge leaf thickness /mm	0.8	0.8	0.8	0.8
Caudal margin leaf thickness/mm	0.8	0.8	0.8	0.8
Inlet airflow velocity /m·s^{-1}	77.8	77.8	74.4	71.3
Outlet airflow velocity /m·s^{-1}	224.2	228.9	225.4	221.1

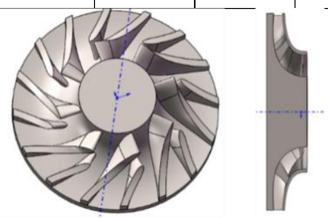

Fig. 3. Three-dimensional model of impeller of first stage centrifugal compressor

Compared with vaneless diffuser, vaned diffuser can achieve higher diffuser effect in shorter radial size, so vaned diffuser is adopted. Blade leading edge angle and inlet flow angle should be as close as possible to reduce impact loss.

Four-stage diffusers all have 15 blades. For example, the blade shape of the first-stage diffuser is shown in Fig. 4. See TABLE IV. for blade thickness, blade Angle and other parameters.

Fig. 4. The first stage has a vane diffuser

TABLE IV. DIFFUSER PARAMETERS OF CENTRIFUGAL COMPRESSOR

Parameters	**C-101**	**C-102**	**C-103**	**C-104**
Inlet diameter /mm	25.7	23.9	23.3	22.3
Outlet diameter /mm	45.0	42.6	42.0	41.0
Leaf leading edge angle /°	18.1	18.2	17.7	17.2
Angle of the caudal margin of the leaf blade /°	38.7	38.8	38.2	37.7
Leaf leading edge thickness /mm	0.4	0.4	0.4	0.4
Leaf caudal margin thickness /mm	1.7	1.7	1.7	1.7
The leaves are leafy and tall / mm	1.4	1.3	1.3	1.3

After exiting the diffuser, the working medium enters the volute of asymmetric circular section for further diffusing. Take the first stage as an example, see Fig. 5 for the view of the whole flow channel and its meridional plane. The larger outlet cross-sectional area can reduce the flow rate of working medium to less than 40 m·s-1, and the volute is slightly bent inward to reduce the radial size of the whole machine.

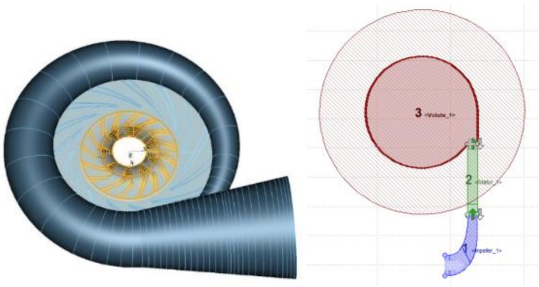

Fig. 5. First stage full flow channel design results

IV. INTERNAL LEAKAGE CHARACTERISTICS AND CORRESPONDING MATHEMATICAL MODEL MODIFICATION OF MICRO COMPRESSOR

A. Leakage in centrifugal compressor

Internal leakage refers to the leakage between the internal components of the compressor body, which is called internal leakage because it does not leak out of the system. Internal leakage clearance is shown in Fig. 6, including the clearance of the wheel back and the clearance of the wheel cover. The design uses a closed impeller. The existence of the impeller wheel cover prevents the tip clearance leakage, but in the outer gap of the wheel cover, driven by the pressure difference between the inlet and outlet of the impeller, the working medium leaks from the outlet of the impeller to the inlet and outlet, which is called the wheel cover leakage. The clearance between the back of the impeller and the diffuser leads to the compressor chamber. The pressure of the compressor chamber in this design is affected by the outlet pressure of the two-stage impeller and adjusted by the inter-stage cooling air flow. As the pressure of the low-pressure chamber is greater than the outlet pressure of the side impeller, the working fluid in the chamber leaks to the outlet of the impeller along the clearance of the back of the wheel. If the pressure of the high-pressure cavity is less than the outlet pressure of the impeller, the working medium at the outlet of the impeller leaks to the cavity through the clearance of the back of the wheel.

Internal leakage changes the flow of working fluid in impeller and stationary parts, resulting in extra power consumption and reduced compression efficiency. This is because (1) Part of the impeller compression medium leaks but does not enter the diffuser, and the impeller needs to compress the working medium higher than the design flow, that is, internal leakage loss. (2) The leaking working medium flows to the inlet of impeller and diffuser, changing the local average working medium state parameters and velocity, making the above parameters deviate from the design value and thus reducing the efficiency.

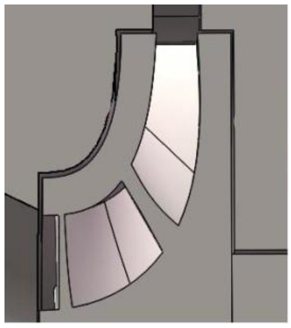

Fig. 6. Wheel back wheel cover clearance screenshot

Internal leakage clearance as shown in Fig. 7, the institute for miniature centrifugal helium compressor is one of the major characteristics of the conventional compressor round the back wheel cover gap and the relative magnitude of the impeller blade height, and relative magnitude of the internal leakage flow rate is higher, the influence of the leakage of working medium inside the impeller flow of import and export cannot be ignored. Wheel cover leakage is subscript "lk, s", wheel back leakage is subscript "lk, h"; The inlet gas of the impeller mixed with the leakage gas is subscript "in '" and the inlet gas of the diffuser is subscript" 2 ". The mass flow of the gas in the impeller conforms to formula (1) (2), where $\dot{m}$ quarter is the design flow:

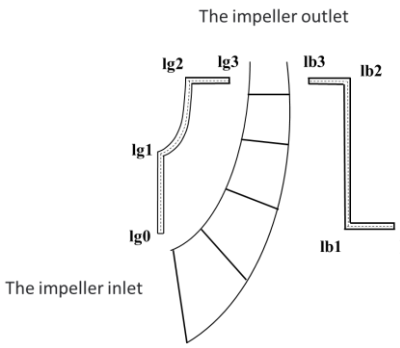

Fig. 7. Diagram of internal leakage gap

$$\dot{m}_{imp} = \dot{m}_{in} + \dot{m}_{lk,s} \quad (1)$$

$$\dot{m}_{imp} = \dot{m}(1 + \beta_{lk,s} + \beta_{lk,h}) \tag{2}$$

B. *Effect of internal leakage on compression*

In order to explore the influence of internal leakage on the compression process of centrifugal compressor, the differences of thermal properties, flow parameters and overall compression results of centrifugal compressor with and without internal leakage were compared and analyzed. CFD method can provide inlet and outlet sections of impeller, diffuser and volute, as well as various working fluid parameters of impeller blade and diffuser blade leading edge section. Therefore, CFD was used to simulate the leakage clearance between the wheel cover with wheel back and the compression basin without internal leakage to compare the influence of internal leakage on the compression process. The comparison results are shown in Fig. 8.

It can be seen that the existence of internal leakage leads to the change of inlet flow Angle of the impeller of the compressor and the pre-rotation, and the loss in the impeller passage increases, so the total pressure decreases and the integral stage pressure ratio decreases. Fig. 9 the first stage and the second stage were compared with and without internal leakage in the CFD simulation at the total pressure change, visible level for low voltage due to leakage flow on the back wheel exhaust gas and impeller flow in a different direction, leakage of working medium mixed with the original working medium impact each other increases the flow losses, and high pressure stage wheel back leakage in the opposite direction so no this phenomenon. In addition, internal leakage causes the mass flow m˙_imp to deviate from the design operating condition.

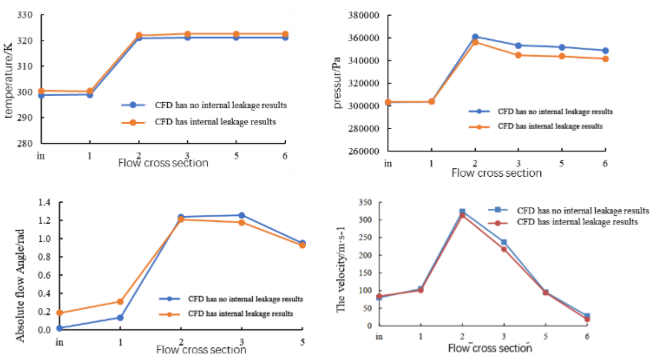

Fig. 8. Influence of internal leakage on compression process

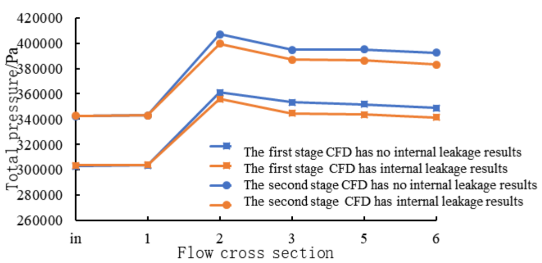

Fig. 9. Comparison of total pressure changes in the first and second levels with or without internal leakage

The above influences can be divided into two categories: (1) the influence of the wheel cover leakage on the flow prerotation and state parameters at the impeller inlet; (2) the influence of the wheel back leakage on the flow loss and state parameters at the impeller outlet.

C. *Simulation results and analysis*

- Solution of model without internal leakage

The original mathematical model and CFD software were used to predict and simulate the compressor. The original mathematical model did not consider the influence of internal leakage, and the CFD flow channel model also did not include internal leakage clearance, so as to verify the accuracy of the original mathematical model. The working medium parameters of each section are shown in Fig. 10 when internal leakage is ignored.

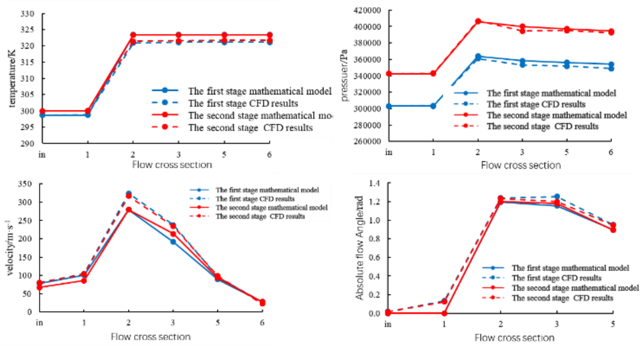

Fig. 10. Solution results of original model without internal leakage

The original mathematical model is basically consistent with the CFD results, and the main differences are reflected in the flow Angle of the leading edge section of the blade, the flow loss in the diffuser section, and the flow rate at the impeller outlet.

- Solution of model with internal leakage

One of the characteristics of high-speed micro-centrifugal compressors is that the relative size of internal leakage gap is larger and the internal leakage has a more significant impact on the compression process. After verifying the accuracy of the original mathematical model, some modifications are made to the model to reflect the influence of internal leakage and adapt to the characteristics of large internal leakage of high-speed micro-compressors. Taking the first stage as an example, the original mathematical model and the modified mathematical model are shown in Fig. 11 for the comparison of the predicted values of section parameters of each flow channel and the corresponding CFD simulation values.

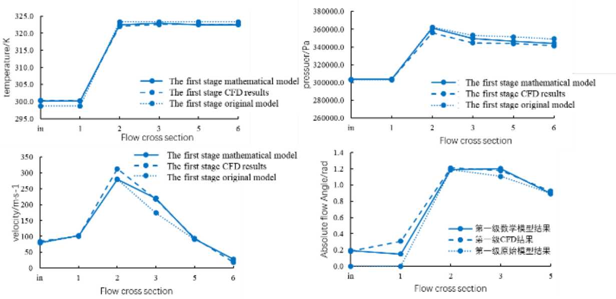

Fig. 11. Solution results of mathematical model with internal leakage

The total temperature of inlet and outlet, flow Angle of each section and total pressure of diffuser section in the modified mathematical model are closer to the CFD value.

The total pressure ratio and total temperature ratio of the whole stage are more accurate based on CFD results.

- Performance under variable working conditions

An important use of the mathematical model is to quickly predict the compression effect when the compressor deviates from the design condition. For working points of different mass flows at different speeds, mathematical models were used to solve and fit the characteristic curves of variable working conditions respectively. The pressure ratio results are shown in Fig. 12, and the adiabatic efficiency results are shown in Fig. 13 and Fig. 14. The characteristic curves of the mathematical model and CFD simulation are obviously different when they deviate far from the design condition.

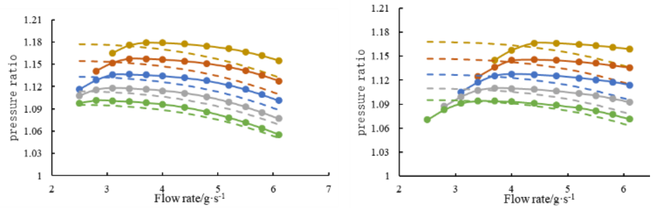

Fig. 12. Pressure ratio under variable conditions

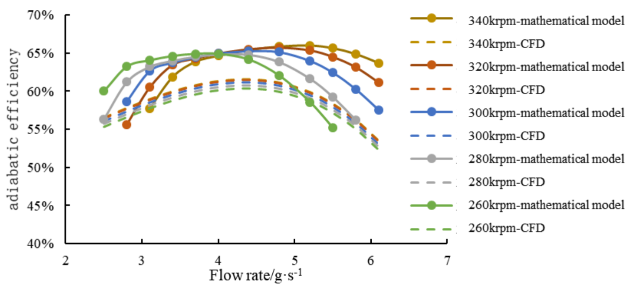

Fig. 13. Predictions of adiabatic efficiency under variable conditions

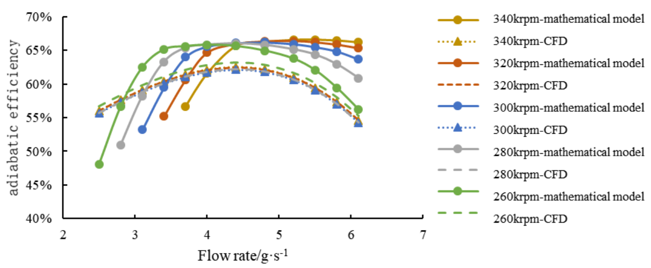

Fig. 14. Predictions of adiabatic efficiency under variable conditions

V. Conclusion

Propellant storage in orbit is a technical basis of many space technologies such as manned spaceflight. An reverse Brayton system is designed in this paper. A high-speed miniature centrifugal compressor is designed for ZBO system, and the reliability of the design is verified by CFD simulation. Based on the existing performance prediction methods of centrifugal compressors, a one-dimensional mathematical model is established, and the mathematical model is modified considering the characteristic of large internal leakage of high-speed micro-helium centrifugal compressors. The prediction results of the model at design points and off-design points are compared with the CFD simulation results.

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