Design and Setup of the Micro-Turboexpander Transcritical CO2 System

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ABSTRACT

The relative low COP of transcritical CO_2 systems caused by the throttle valve limited the further development. The replacement of a throttle valve with a turboexpander, an ejector or a vortex tube is leading to the way for improving the COP of a transcritical CO_2 system. In this paper, a transcritical carbon dioxide refrigeration cycle with different expansion devices was analyzed theoretically using the first and second laws of thermodynamics. A transcritical CO_2 system with micro-turboexpander for refrigerating capacity of 15 kW was designed and set up. Two turboexpanders with 10.6 mm radial reaction turbine wheel and 10.1 mm rotor diameter were designed for the 15 kW transcritical CO_2 refrigeration system, and the rotating speed is about 100 krpm. The proposed turboexpanders outlet states are CO_2 two-phase flow and subcooled CO_2 liquid flow, respectively. This study will lay a foundation for the application of turboexpanders in transcritical CO_2 systems.

1. INTRODUCTION

Due to harmful effects of the synthetic refrigerants on the environment, CO_2 regains much attention as a potential refrigerant. Carbon dioxide has various advantages over conventional refrigerants, such as non-flammability and non-toxicity, easy availability, high volumetric refrigerant capacity, and excellent heat transfer properties. Since the critical temperature of carbon dioxide is usually lower than heat rejection temperature of heat pump and air-conditioning, the transcritical vapor compression cycle is employed in these systems. However, the main disadvantage of CO_2 cycle is the low COP due to the huge energy loss in the throttling valve. In order to reduce throttling loss, a large number of researches have been carried out on several cycle modifications such as using internal heat exchanger, expansion turbine, multi-staging, ejector and vortex tube（Agrawal , 2007; Aprea and Maiorino, 2008; Baek et al., 2005;Banasiak et al. ,2012; Cho et al.,2009; Deng et al., 2007; Groll and Kim, 2007; Haiquing et al., 2006;Li et al., 2000; Robinson and Groll, 1998; Sarkar , 2008; Sarkar, 2010）.

Turboexpanders have been widely used in gas liquefaction devices (below 120K) and reversed Brayton cycle. A turboexpander has many advantages, such as high efficiency, high compactness, easy maintenance, no- leakage and high reliability. If the high efficiency and high reliability of the turboexpander can be guaranteed, turboexpander is the best choice for replacing the throttling valve in refrigeration system. The current research on turbo expander is mainly focused on low temperature cycle. Little attention has been directed toward employing turboexpander in CO_2 refrigeration system. The feasibility of using centrifugal compressor and expander in a car conditioner was analyzed by Bjφrn (1999). Lance and Brasz (2004) has developed a two-phase axial-flow turbine consisting of a two-phase nozzle and axial-flow turbine blades. TØNDELL (2006) tested an impulse expander for CO_2 refrigeration system. The tested turbine was a radial outflow impulse turbine. In China, the research on CO_2 turboexpander is mainly focused on review and theoretical analysis, little work has been conducted on the key technologies of two phase turboexpander (transcritical).

2. COMPARISON OF CYCLES WITH DIFFERENT EXPANSION DEVICES

The transcritical CO_2 cycles are simulated numerically within a wide range of operation conditions. Table 1, summarizes the basic assumptions and input parameters of the system simulation and analysis. Table 1 shows the basic parameters of the cycle in simulation.

Table 1: Parameters used in the simulation

Parameters	Value	Parameters	Value
η_{com}(%)	75	T_3（K）	313.15
η_{noz}(%)	80	P_{gas}（MPa）	10.0
η_{exp}(%)	65	T_e（K）	278.15
ε(%)	85	T_r（K）	283.15

Based on these assumptions, we simulated the transcritical CO_2 cycles with different expansion devices by FORTRAN code. The calculation parameters are as follows: y=0.5, T_3=313.15 K, P_{gas}= 10.0 MPa, T_{water}=300.15K, P_{int}=5.7 MPa and Q_0=15kW. The results show that the maximum COP improvement can be achieved by using turboexpander. Employing the turboexpander in the basic transcritical CO_2 cycle may leads to an increase of 32.2% in COP. The COP of the ejector cycle is 5.8% higher than that of the basic cycle. And the COP of Maurer cycle and Keller cycle are 3.5% and 1.0% higher than the basic cycle, respectively. The exergy loss in the basic cycle is the largest, followed by the vortex tube cycle and the ejector cycle. The expander cycle has the smallest total exergy loss which is 29.3% smaller than that of the basic cycle.

The exergy loss of expander is 87.8% smaller than that of the throttling valve. And exergy loss of expander is 62.9% and 63.8% less than the exergy loss of the vortex tube of Maurer cycle and Keller cycle, respectively. The exergy loss of expander is 63.2 % less than that of the ejector.

3. DESIGN OF THE TEST RIG

And as shown in Section 2, the design refrigerating capacity of transcritical CO_2 refrigeration system is about15KW. Figure 1 shows the schematic of the transcritical CO2 refrigeration system with a turboexpander.The turboexpander was installed parallel to the electronic expansion valve. The needle valves are used to adjust the inlet pressure、the outlet pressure and the mass flowrate of the turboexpander.The inlet temperature of the turboexpander can be controlled by adjusting the temperature and mass flow rate of cooling water.The temperature at the expander outlet can be adjusted by the temperature and the mass flow of the chilling water.Due to the high shaft speed of the turboexpander, the gas bearing was adopted for both the radial and axial bearing.

The test rig of the transcritical CO2 refrigeration system with an turboexpander in this paper is composed of three parts, the refrigeration cycle, the water supply part and the data acquisition system. The presented system is composed of a semi-hermetic reciprocating compressor, an evaporator, an electronic expansion valve, a receiver, a gas cooler and an oil separator as shown in Fig.2. Details about the components are listed in table2.

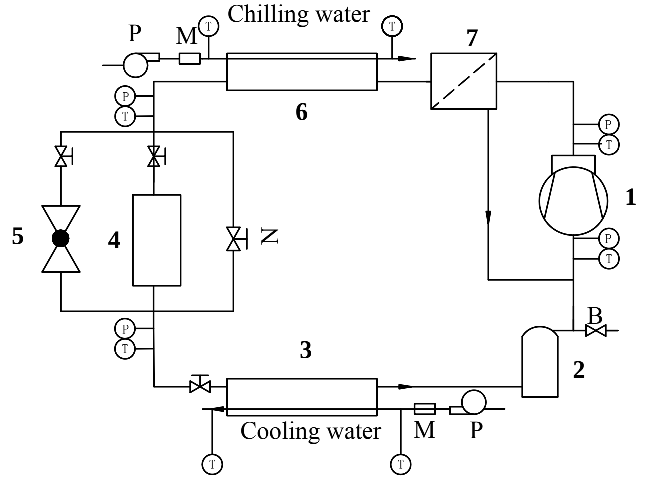

Figure 1: Schematic of the experimental system for CO_2 cycle with an turboexpander
1 compressor; 2 receiver; 3 evaporator; 4 turboexpander; 5 electronic expansion valve;
6 gas cooler; 7 oil separator; Ⓣ temperature sensor; Ⓟ pressure sensor; P pump;
M water flow meter; N needle valve; B ball valve

Figure 2: Picture of the transcritical CO_2 refrigeration system

Table 2: Main Components

Name of components	Main characteristic
Compressor	Semi-hermetic reciprocating compressor; Swept volume: 8.3 m^3/h.
Gas-cooler	Plate heat exchanger; Heat transfer area: 2.09 m^3
Evaporator	Plate heat exchanger; Heat transfer area: 1.81 m^3
Expansion valve	Electrically expansion valve and regulating valve.
Receiver	Inner volume: 4.9×10^{-3} m^3.
Oil separator	Inner volume: 11.7×10^{-3} m^3.

The heat generated from the evaporator and the gas cooler were controlled and regulated by two circulating water systems. And the water inlet temperature was regulated in the range of 5℃ to 90℃ with a precision of ±0.5℃. The T-type thermocouples which have the accuracy of 0.5 ℃ (-10℃ to 150 ℃) were used to measure the temperature. And with an accuracy of ±2.5% (0MPa to 15MPa), the pressure sensors were employed to acquire system pressure data. With an accuracy of 0.15%(AC 0.0 -500.0V,0.030-40.00A) a digital power meter were used to monitor the electrical power supplied for compressor. The accuracy of the turbine flowmeter is ±0.5% (0.8 m^3 to 8.0 m^3). A data acquisition device was used to record the data.

Table 3: The operating parameter of the test rig

Parameters	Values
inlet water temperature (gascooler)(℃)	30.2
water flow rate (gascooler)(m^3/h)	1.99
inlet water temperature (evaporator)(℃)	18.7
water flow rate (evaporator)(m^3/h)	1.85
cooling capacity (kW)	16.6
power of the compressor (kW)	11.5
refrigerant mass flow rate (kg/s)	0.23

Table 3 shows the operating parameter of the test rig ,when the gas cooler water inlet temperature was 30.2 ℃,and the evaporator water inlet temperature was 18.7 ℃.It can be seen that the test refrigerating capacity is about 16.6 KW, the deviation between the design and the test data is 10.7 %. The design refrigerating capacity agrees well with the test data.

4. TURBO EXPANDER DESIGN

4.1 Feasibility analysis on the turboexpander development

The main obstacle of using a turboexpander in the CO_2 refrigeration cycle lies on: 1) the high shaft speed. For example, the designed shaft speed is 35,000 rpm for the CO_2 system with a nominal cooling capacity of 3000 W. 2) absence of the internal mechanism and the design theory of the two phase turboexpander with high liquid ratio. In recent years, there has been great development in material, manufacturing, designing, assembling, especially the novel bearings such as gas bearing, magnetic bearing. All these make it possible to use turboexpander in a CO_2 refrigeration cycle.

4.2 Design condition

There are three general types of turbines: impulse turbine, axial-flow reaction turbine and radial reaction turbine. The radial reaction turbine is the most efficient one. In this paper, two turboexpanders (the state of CO_2 at the outlet of the expanders are two-phase fluid and the subcooled liquid respectively) for the CO_2 system with a nominal cooling capacity of 15 kW were developed. Table 4 shows the design conditions of turboexpander.

Table 4: Design conditions

Parameters	Values
Cooling capacity(KW)	15
Inlet pressure of expander(MPa)	10.0
Inlet temperature of expander(K)	313.15
Discharge pressure of expander(MPa)	4.0/7.3
Degree of Reaction	0.45
Refrigerant mass flow rate (kg/s)	0.12

4.3 Main dimensions and parameters of the turbo expanders

Two turboexpanders were designed. One of the expander was called full expansion turboexpander, and the state of CO_2 at the outlet of this turboexpander is two-phase fluid. The other one is called semi-expansion turboexpander, in which the state of CO_2 at the outlet of this expander is subcooled liquid. Table 5 shows the main dimensions and parameters of the semi-expansion turboexpander. Table 6 shows the main dimensions and parameters of the full expansion turboexpander.

Table 5: Main dimensions and parameters of the full-expansion turbo expander

Description	Dimension
Impeller inlet diameter	10.6mm
Outlet outer diameter for impeller	6.6mm
Outlet inner diameter for impeller	3.6mm
Number of blade	6
Blade inlet height of impeller	0.6mm
Number of nozzle	13
Height blade of nozzle	0.4mm
Nozzle outlet angles	15°
Shaft speed	183596rpm

Table 6: Main dimensions and parameters of the semi- expansion turbo expander

Description	Dimension
Impeller inlet diameter	10.1mm
Outlet outer diameter for impeller	5.9mm
Outlet inner diameter for impeller	4.0mm
Number of blade	6
Blade inlet height of impeller	0.6mm
Number of nozzle	13
Height blade of nozzle	0.4mm
Nozzle outlet angles	20°
Shaft speed	111742rpm

5. PERFORMANCE PREDICTION OF CYCLE WITH EXPANDER

Table 7 shows the COP of the transcritical CO_2 cycle with different expanders. The calculation parameters are the same as the design conditions. The work recovery isentropic efficiency is 50% under design conditions. From table 7, it can be seen that replacing the throttling valve by the expander can improve the COP of the basic cycle. And The COP of cycle with full expansion turboexpander is about 23.1% to 28.3% higher than that of the basic cycle. The COP of cycle with semi-expansion turboexpander is about 7.3% to 9.3% higher than that of the basic cycle.

Table 7: The COP of the transcritical CO_2 cycle with turbo expander

Basic cyclc COP	Isentropic efficiency of the expander	Full expansion		Semi-expansion	
		COP	Improvement ratio	COP	Improvement ratio
2.310	60%	3.01	24.9%	2.48	7.3%
	65%	3.03	25.7%	2.49	8.0%
	70%	3.04	26.3%	2.52	9.3%

6. CONCLUSIONS

The transcritical cycle with expander and throttling valve was analyzed based on the first and second laws of thermodynamics. The results show that replacing the throttling valve by the expander can decrease the exergy loss and improve the COP greatly. A transcritical CO2 system with micro-turboexpander for refrigerating capacity of 15 kW was designed and set up. The test refrigerating capacity is about 20 kW, the deviation between the design and the test data is 10.7 %. Taking the high shaft speed and the difficulties in designing high liquid ratio expander into consideration, two solutions called full expansion and semi-expansion were proposed. The designed CO2 impeller diameter of the turboexpander is about 10mm, and the shaft speed is nearly10,000 rpm. This study lays a foundation for replacing the throttling valve with the turboexpander and the employment of two phase turboexpander with high liquid ratio.

NOMENCLATURE

Nomenclature		Subscripts	
p	Pressure (MPa)	com	Compressor
T	Temperature (K)	gas	Gas cooler
y	Cold mass fraction	noz	Nozzle
Q0	Cooling capacity	int	Intermediate cooler
ε	Heat exchanger effectiveness	exp	Expander
η	Isentropic efficiency (%)	r	Refrigerated object
water	Water inlet to desuperheater	3	Inlet of the gas cooler

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