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A Transcritical CO2 Turbine-Compressor

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A TRANSCRITICAL CO_2 TURBINE-COMPRESSOR

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ABSTRACT

A transcritical turbine-compressor was analyzed for CO_2 refrigeration and heat pump systems and for CO_2 liquefaction. The turbine is an axial flow design based upon previous two-phase turbines applied for refrigerant energy recovery by the authors. The turbine output shaft provides direct drive of a full flow boost compressor, enabling a low cost, hermetic turbine-compressor assembly. Results for a small (6 ton) CO_2 refrigeration or heat pump cycle showed an increase in the coefficient of performance of 39% above a simple throttle valve cycle.
A description of the technology and its experience and a summary of the performance calculations are provided.

1. INTRODUCTION

The need for refrigeration systems using environmentally benign refrigerants has become greater in the last decade because of the dual drivers of ozone depletion and global warming. "Optimal" refrigerants that were engineered to maximize the cycle efficiency by minimizing the expansion throttling loss are no longer usable. The major air conditioning manufacturers have adopted refrigerants which either have a high vapor pressure and hence throttling loss (e.g. R-134a, R-407C, R-410A) or which have partial ozone depletion potential and high toxicity level (40 ppm AEL) and are due to be phased out (e.g. HFC-123).
The choice of CO_2 as a refrigerant has many benefits. It is a natural, non-toxic substance with no ozone depletion potential. Additionally, there are volumetric and heat transfer advantages and the refrigerant has a low cost. A disadvantage is that the large pressure difference for compression and expansion results in a high throttling loss, leading to a poor cycle efficiency compared to current HCFC refrigerants (Robinson and Groll, 1998). However, a transcritical CO_2 refrigeration cycle with an expander was found to have the same or better performance as an HCFC cycle with expansion valve. The gain in CO_2 refrigerant cycle efficiency resulting from energy recovery with a 60% efficient expander was as much as 33% compared to a CO_2 cycle with a throttling valve and 25% compared to the same throttling valve cycle with maximum internal heat exchange, ibid.
Another application involving CO_2 is liquefaction of gaseous CO_2 for transportation, storage or sequestration.
The need for an expander to recover energy from the expansion is clearly indicated if the CO_2 refrigeration cycle is to be widely deployed. However, the expansion, which starts in the supercritical region, enters the two-phase region, producing over 50% liquid by mass.
Until recently, no two-phase expanders had been developed for commercial applications. Attempts to use radial inflow machines in the wet region have not been successful due to poor performance and erosion from the liquid centrifuging outward. Attempts to use positive displacement machines have not been successful due to high cost and size and reliability issues.
However, recent advances in two-phase nozzle technology and two-phase impulse turbine design have resulted in commercial turbines to recover energy from two-phase refrigerant expansion (Brasz, 1995, Hays and Brasz, 1996, Hays and Brasz, 1998). The resulting units are efficient, compact and inexpensive. They have been included as OEM equipment on over 125 large commercial chillers (Hays 1999).

The other requirement for expanders to improve the efficiency of a refrigeration system is a cost effective method to use the generated shaft power. This has been accomplished by installing the two-phase turbine on the outboard shaft of the compressor motor, unloading the power required for refrigerant compression (Hays and Brasz, 1996).
Another method is a hermetic two-phase turbine-compressor, where the two-phase turbine shaft power directly drives a centrifugal boost compressor, reducing the power required by the main compressor (Hays, 2003). This method has the potential for the lowest manufacturing cost and highest reliability for smaller systems.
A third option is the generation of electric power from the expansion, which can be used to reduce the net power to the compressor. Recent advances in high-speed generators may make this option useful for larger systems.

2. DESCRIPTION OF TECHNOLOGY

The two-phase axial-flow turbine is comprised of two basic elements:

- A two-phase nozzle to convert thermal and pressure energy into directed kinetic energy.
- Axial-flow turbine blades designed to maximize kinetic energy transfer from the high velocity two-phase mixture.

A two-phase nozzle is a nozzle in which a liquid and a gas mixture, a flashing liquid, or a condensing gas, at high pressure, is expanded to low pressure and high velocity. The gas phase may either be the vapor of the liquid being accelerated, in which case the flow is termed one-component, or a different chemical species from the liquid, in which case the flow is termed two-component.
In the two-phase nozzle the gas is accelerated by the pressure difference between the inlet and exit. The gas shears the liquid phase into small droplets producing a large surface area and hence good coupling between the phases. The gas transfers momentum to the liquid droplets resulting in a high velocity homogeneous mixture.
The basic conservation equations and those of droplet formation and breakup and heat transfer and momentum exchange between phases have been programmed and utilized to predict nozzle efficiency and perform nozzle design.
This nozzle code has been verified many times for a variety of working fluids, both in the laboratory and in the field and has been used to successfully design two-phase nozzles for a variety of applications. These include nozzles for geothermal and waste-heat-recovery turbines using water as the working fluid, as well as for axial-flow turbines in chiller applications utilizing refrigerants.
A typical two-phase nozzle profile generated by the code is shown below in Fig. 1. Note that the nozzle is of a converging-diverging geometry with a throat because the flow exiting the two-phase nozzle is usually supersonic. An additional feature shown is a shedder to strip liquid from the wall into the bulk stream where it can be efficiently atomized.

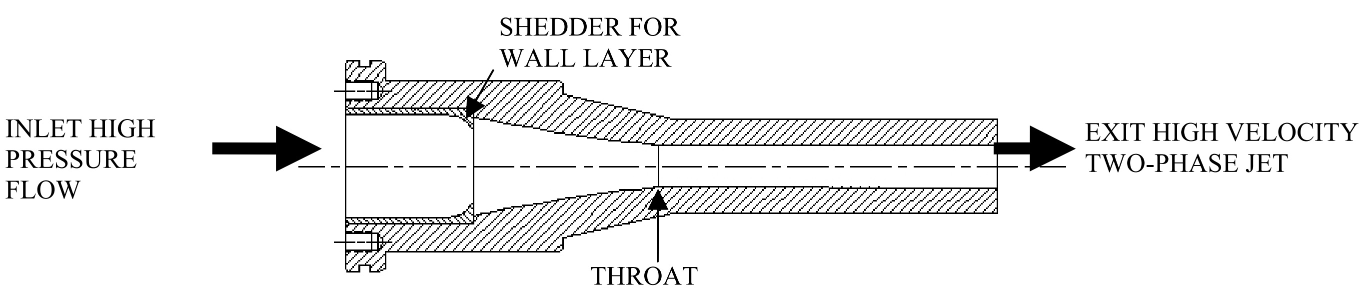

Fig. 1. Typical Two-phase Nozzle Profile Generated from Computer Code

The jet from the two-phase nozzle impinges upon the axial-flow turbine blades. The two-phase stream impinges on the turbine blades as shown schematically in Fig. 2. Here the liquid separates from the gas and forms a layer on the blade. If the blade has a long axial dimension, the liquid flow separated on the blade (which has a tendency to travel on a straight path) will exit the blading at an angle that is different than the blade leaving angle and direction of thrust. Thus it is important in the blade design to keep the blade chord as small as is practicable.

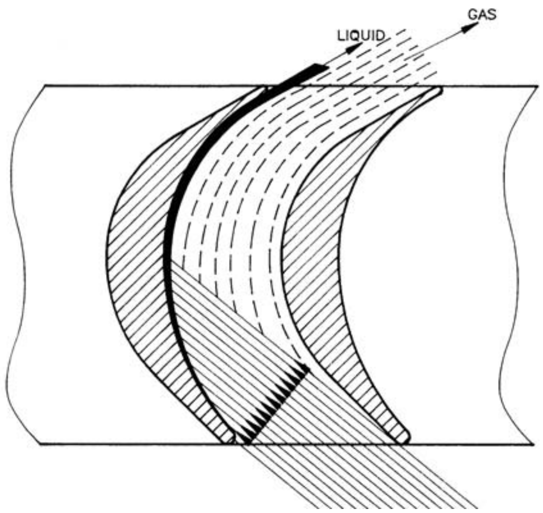

Figure 2. Schematic of Two-Phase Jet Interaction with Axial Blading

A key feature of the impulse turbine design is the provision of an axial path for the two-phase flow. As discussed previously radial inflow machinery will centrifuge liquid in a direction counter to the flow. This liquid can collect between the nozzles and rotor blades producing severe erosion. In the axial design the bulk of the liquid leaves the rotor with a swirl to enable collection on the casing wall. The small fraction that is centrifuged is collected on a shroud, which also directs the flow to the casing wall.

An axial flow two-phase turbine having a single nozzle was also designed and tested for a high lift heat pump application using R-134a. A schematic drawing of this unit is shown in figure 3.

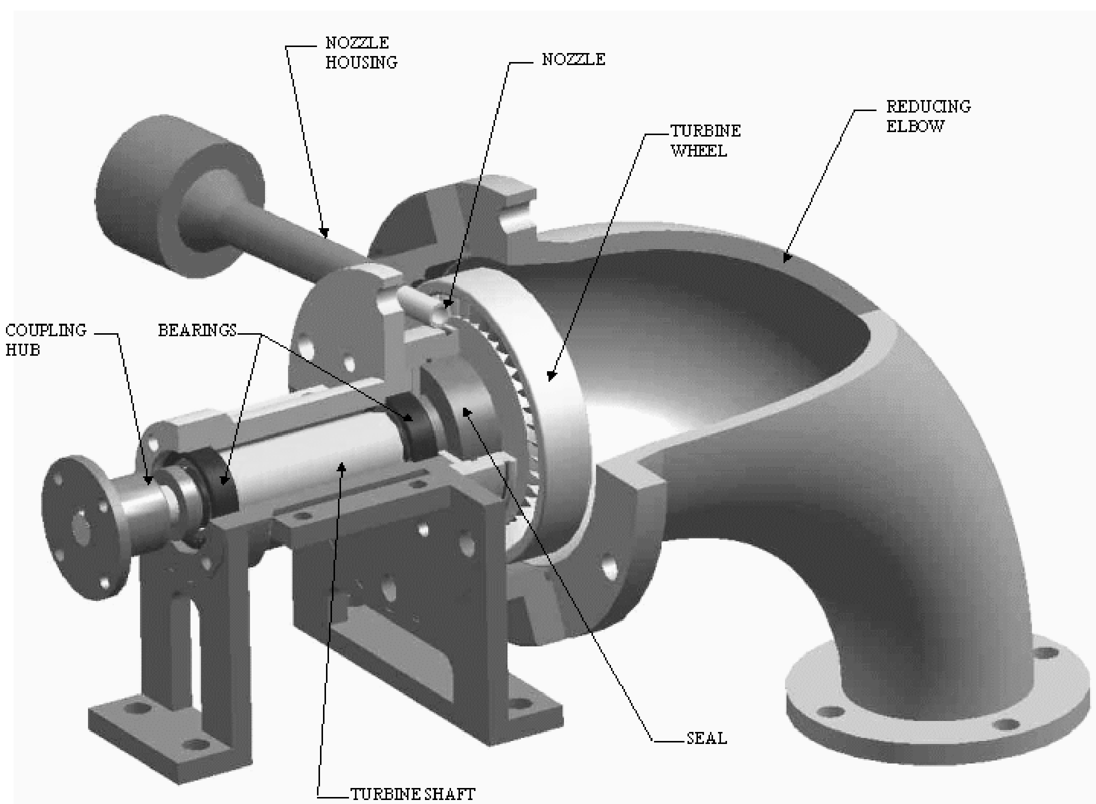

Figure 3. Cut-away Drawing of Two-Phase Axial flow Turbine for Heat Pump

A photograph of the test unit is given in figure 4. The rotor diameter is 2.8 inches. The blade height is 0.3 inches. It was operated at a maximum speed of 12,800 rpm. The measured efficiency of 56% at the power output of 310 watts was close to the predicted efficiency of 61%. A relatively large degree of subcooling at the inlet was probably responsible for the deviation from predicted performance which was made for saturated conditions.

Figure 4. Photograph of Two-Phase Axial Flow Turbine for Heat Pump

3. APPLICATION OF TRANSCRITICAL TURBINE TO CO_2 REFRIGERATION SYSTEMS

A CO_2 refrigeration-heat pump cycle using a transcritical turbine and boost compressor is shown in figure 5. Supercritical CO_2 from the main compressor is cooled in the heat exchanger. The cooled supercritical CO_2 is expanded in the transcritical turbine into the two-phase region. The liquid CO_2 in the two-phase flow is evaporated in the evaporator. The CO_2 gas enters the boost compressor which is driven by the shaft of the transcritical turbine. The pressure is increased in the boost compressor from where the CO_2 flows to the main compressor where the pressure is increased to the maximum cycle pressure.

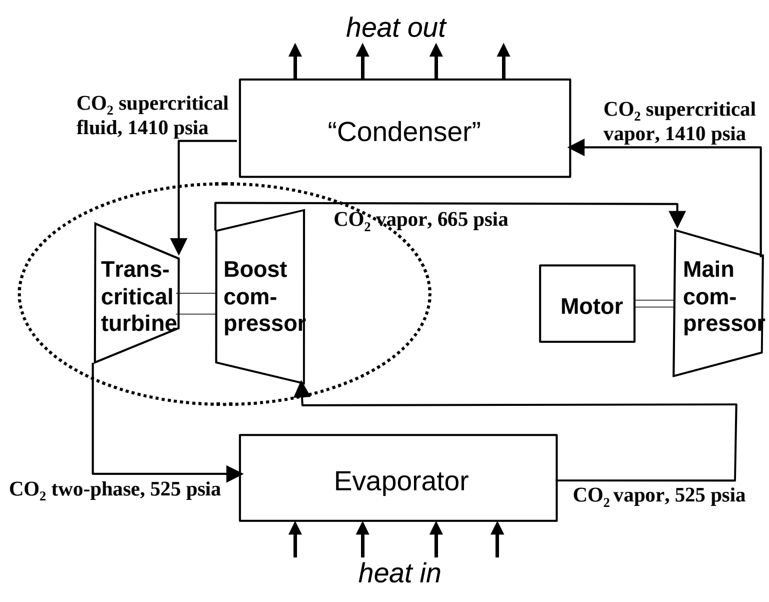

Figure 5. The CO_2 Refrigeration-Heat Pump Cycle with a Transcritical Turbine Powering the Boost Compressor

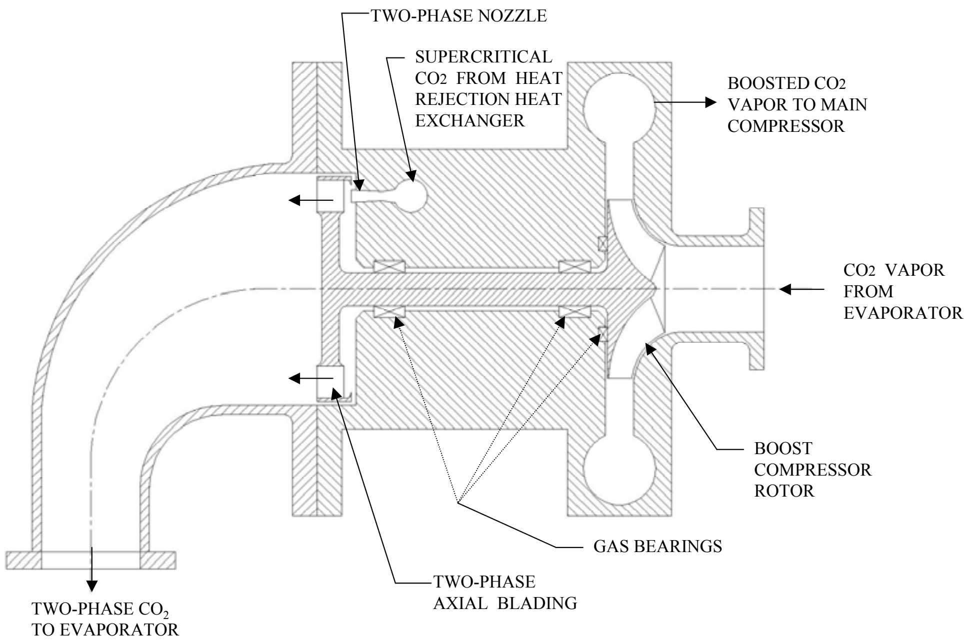

Figure 6. Schematic of Transcritical CO_2 Turbine and Boost Compressor

Evaluation of a transcritical CO_2 turbine (TCT) was made using cycle conditions from Robinson and Groll, 1998. Figure 6 shows a general arrangement of the TCT with a compressor boost compressor. Supercritical CO_2 from the heat rejection heat exchanger enters the two-phase nozzle. The fluid is expanded in the nozzle forming a two-phase mixture at high velocity. The flow is directed on the axial flow blading transferring energy to the rotor. The flow swirls from the blades, the liquid being collected on the outlet pipe walls as the flow is turned. The cooled two-phase mixture leaves through a vertical elbow and flows to the evaporator.

The rotor power drives a boost compressor rotor on the same shaft. The full flow from the evaporator enters the boost compressor. The pressure is boosted to a value above the evaporator pressure, reducing the power requirements of the main compressor. Reduction of the pressure ratio required for the main compressor has the secondary benefit of increasing the main compressor efficiency. The unit would be hermetically sealed and readily manufactured using castings.

For purpose of the analysis the heat exchanger outlet conditions were selected to be:

$T_1 = 104$ F
$p_1 = 1410$ psia
$m = 1440$ lbm/h

The expansion conditions chosen were:

$p_2 = 525$ psia
$T_2 = 34.7$ F

A two-phase nozzle efficiency of 94% gives a spouting velocity of:

$V_b = 562$ ft/s

The liquid fraction at the exit of the nozzle is:

$1 - x_2 = .552$

The nozzle exit diameter is:

$$d_2 = (\ 1.27m / \rho_{2m} V_b\ )^{1/2}$$
$$= .103 \text{ in}$$

The Transcritical CO_2 Turbine was analyzed for a nominal 6 Ton refrigeration unit. Speed was optimized considering the TCT efficiency and the compressor efficiency. At a speed of 110,000 rpm the turbine efficiency was 69%. The specific speed for the compressor, $n_s = .42$, and the specific diameter, $d_s = 7.0$, give a boost compressor efficiency in the 80% range.

The overall cycle advantage compared to a flash valve system is 1.39 for 6 Tons.

The results from the turbine analysis code are provided in Table 1.

Table 1. Axial Turbine Design Calculations.

Rotor Speed	110,000	rpm
Rotor Power	1952.2	Watts
Rotor Efficiency	0.737	
Liquid Mass Flow Rate	0.2222	lbm/sec
Gas Mass Flow Rate	0.1778	lbm/sec
Nozzle Exit Diameter	0.1020	inches
Number of Nozzles	1	
Blade Spacing	0.0299	inches
Blade Height	0.1010	inches
Meanline Radius	0.3000	inches
Outer Radius	0.4500	inches
Blade Chord	0.1079	inches
Number of Blades	63	
U/C	0.5138	
Gas Blade Efficiency	0.8500	
Windage Power	135.97	Watts
Jet Power	2649.6	Watts
Rotor Power	1952.2	Watts
Blade Efficiency	0.788	
Rotor Efficiency	0.737	

Assuming an isentropic compressor efficiency of 80%, and using the NIST refrigerant property tables, the power from the TCT, 4.63 BTU/lb, would result in a pressure boost of the full vapor flow rate to 665 psia. The enthalpy at this point is 189.9 B/lb, the temperature is 66.9 F and the entropy is 0.4414 BTU/lbm^{0}R.
The main compressor power (for 80% isentropic efficiency) required to increase the boost pressure of 665 psia to 1410 psia is 16.60 BTU/lbm. For the flash valve expansion the main compressor power required to increase the pressure from the 525 psia evaporator pressure is 21.0 BTU/lbm. The TCT exhaust stream has an enthalpy of 133.1 BTU/lbm. The refrigeration from the TCT exhaust flow stream is 52.22 BTU/lbm. The exhaust stream from a throttle valve has a higher enthalpy, 137.7 BTU/lbm. The refrigeration from the throttling valve option is 47.6 BTU/lbm. Thus the increase in cooling per unit of power input is:

$$COP_{tct}/COP_{tv} = (52.2/16.6)/(47.6/21.0) = 1.39$$

The above example was done for approximately 6 Ton of cooling. Increasing the size will improve the performance of the TCT because of partial admission effects and the decrease in the ratio of windage loss to shaft power output. The increase in COP will vary depending on the final cycle conditions. However, the above increase is representative of the cycle efficiency advantages that can be realized with a transcritical CO_2 turbine and boost compressor utilized in place of the expansion valve.

5. CONCLUSIONS

A transcritical turbine compressor design based upon successful two-phase turbine experience has been presented. The use of this transcritical turbine compressor can enable a substantial reduction in the power requirements for CO_2 refrigeration and heat pump systems (Amend and Hays, 2002) as well as CO_2 liquefaction systems (Hays, 2004). The resulting environmental and energy conservation advantages should dictate development of this concept and near term application to beta sites.

REFERENCES

1. Robinson, D. and Groll, E., *Efficiencies of Transcritical CO_2 Cycles With and Without an Expansion Turbine*, Int. J. Refrig., Vol. 21, No. 7, pp. 577-589, 1998.

2. Brasz, J.J., *Improving the Refrigeration Cycle with Turbo Expanders*, Proceedings of the 19th International Conference of Refrigeration, The Hague, Netherlands, Volume IIIa, pp 246-253, 1995.
3. Hays, L. and Brasz, J.J., *Two-Phase Turbines for Compressor Energy Recovery*, Proceedings of the 1998 International Compressor Engineering Conference at Purdue, Vol. II, pp. 657-662, July 1996.
4. Hays, L. and Brasz, J.J., *Two-Phase Turbine as Stand-Alone Throttle Replacement Units in Large 2000 – 5000 Ton Centrifugal Chiller Installations*, Proceedings of the 1998 International Compressor Engineering Conference at Purdue, Vol. II, pp. 797-802, July 1998.
5. Brasz, J.J., *Performance Characteristics of Two-Phase Flow Turbo-Expanders used in Water-Cooled Chillers*, Proceedings of the IMechE International Conference on Compressors and their Systems, London, England, pp. 171-180, Sept 13-15, 1999.
6. Hays, L., *History and Overview of Two-Phase Flow Turbines,* Proceedings of the IMechE International Conference on Compressors and their Systems, London, England, pp. 159-170, Sept 13-15, 1999.
7. Hays, L., *Transcritical Turbine and Method of Operation,* U.S. Patent No. 6,644,062 B1, November, 2003.
8. Amend, W., and Hays, L., *Design of a Two-Phase Turbine for a High Lift Heat Pump*, Final Report, Douglas Energy Company, Placentia, CA, May 2002.
9. Hays, L., *Gas Liquefaction System Using a Transcritical Turbine*, patent pending, 2004.