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L. G. Hays  
*Biphase Energy Company*

J. J. Brasz  
*Carrier Corporation*

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Two-Phase Turbines for Compressor Energy Recovery

Lance G. Hays
Biphase Energy Company
Placentia, CA 92670
and
Joost J. Brasz
Carrier Corporation
Syracuse, NY 13221

Abstract
Application of a two-phase turbine for the recovery of compressor energy in refrigeration cycles is described. Reduction in compressor power and size is estimated for refrigeration applications with HFC134a, HCFC22, HCFC123 and propane. A description of the two-phase turbine is provided and a comparison of predicted and measured efficiency is given for air and water tests and for a commercial HFC134a unit. Losses identified in the air and water tests and potential efficiency improvements are presented.

Introduction
Many two-phase-flow adiabatic expansion processes occur where significant quantities of energy are irreversibly degraded during the throttling of gas and liquid mixtures. Fluids with available energy that could have been used to produce shaft power are simply transformed to a lower temperature and pressure without doing useful work. Examples of such energy-degrading processes are the flashing and separation of oil and natural gas; the flashing of geothermal brine and steam to produce steam for power production; and the flashing of refrigerants or hydrocarbon fluids through an expansion valve to lower the temperature of the mixture.

In the latter case, this energy degradation is known as the throttling loss of the vapor compression refrigeration cycle. Reduction of this loss by substituting the expansion valve with a two-phase expander has two beneficial effects for the closed-loop refrigeration cycle:
1. Useful shaft power is created by the power recovery turbine that can be used to offset the power required by the compressor.
2. Frictional heating is reduced by the amount of energy recovered in the two-phase turbine resulting in the generation of more liquid and less vapor after expansion and therefore an increased refrigeration effect of the cycle. As a result more cooling is obtained for the same compressor mass flow rate.

In the past, the design of two-phase-flow expanders operating in the predominantly liquid region of the vapor dome was thought to be technically too difficult and economically unattractive. However, recent developments in two-phase nozzle technology [1] and subsequent two-phase reaction and impulse rotor technology developments [2,3], overcame these technical barriers. Two-phase turbines are now commercially applied as stand-alone units for increased geothermal power production [4], and for energy recovery as well as separation of oil and gas mixtures in oil production platforms [5]. Last year, two-phase power recovery turbines were introduced by a major HVAC manufacturer as a standard product option on its line of centrifugal chillers [6], removing the largest irreversibility of the refrigeration cycle [7].

This paper will discuss the benefit of two-phase turbines for the refrigeration application.

The Compression Refrigeration Cycle with a Two-Phase Turbine
The basic compression refrigeration cycle is illustrated by the solid lines in Figure 1. Cold refrigerant is evaporated in a heat exchanger by the heat load, for example, water being chilled. The resulting cold gas, 1, enters a compressor and is compressed to the pressure at which the gas will be condensed. The high pressure gas, 2, is condensed in a heat exchanger by the coolant, for example, cooling tower water.

The resulting high pressure liquid, 3, is flashed to a two-phase mixture across an expansion valve or other friction producing device, lowering the temperature. The low temperature mixture, 4, flows to the evaporator where the liquid is evaporated, closing the cycle.

The process is illustrated on a temperature-entropy diagram in Figure 2. Expansion from 3 to 4 occurs along an isenthalpe, $h_3 = h_4$, producing no power. The amount of cold liquid produced by the expansion is proportional to the entropy difference $s_1 - s_4$. 657
Substitution of a two-phase turbine for the expansion valve is illustrated by the dashed lines in Figure 1. The balance of the cycle is identical, the only exception being that the high pressure liquid, 3, is flashed in the two-phase turbine to point 4a, which is at the same pressure as point 4 in the original cycle.

As shown in Figure 2, the new cold liquid state point 4a has a lower enthalpy, $h_{4a}$ and entropy $s_{4a}$, than the original state point 4. The amount of power generated is proportional to the enthalpy drop $h_3 - h_{4a}$ and the refrigeration effect of the cycle increases by the same amount.

**Predicted Improvement for Refrigeration Cycles with Two-Phase Turbines**

The decrease in required cycle power and the additional cooling resulting from application of the two-phase turbine was predicted for several refrigerants and various operating conditions. Those analyzed include HFC134a, HCFC22, HCFC123, and propane. An isentropic compressor efficiency of 85% was assumed for all cases. Two turbine efficiency levels were used in the calculations: 55% - a value predicted and measured for the HFC134a two-phase turbine currently in production, and 70% - a value believed to be representative of next generation turbine designs currently under development.

Figure 3 shows the reduction in compressor power with the application of directly coupled two-phase turbines for the four refrigerants. The three fluorocarbon refrigerants were analyzed for an evaporator temperature of 45 °F and a condenser refrigerant saturation temperature varying from 90 to 120 °F. The results for a 55% efficient turbine are a decrease in compressor power from 5.9% to 10.0% for HFC134a, from 5.5% to 8.7% for HCFC22 and from 3.8% to 6.5% for HCFC123. An increase in two-phase turbine efficiency from 55% to 70% would increase the power savings at 90 °F condenser saturation temperature from 5.9% to 7.6% for HFC134a, from 5.5% to 7.0% for HCFC22 and from 3.8% to 5.0% for HCFC123.

The benefit of the power recovery turbine increases with the difference saturation temperature between condenser and evaporator. Propane is used in industrial chillers where lower temperatures are required. As shown in Figure 3, the energy savings for this application are much larger than for the comfort air-conditioning application of water-cooled chillers. Application of a two-phase turbine having 55% efficiency to a propane system with the evaporator at -3 °F and the condenser saturation temperature at 120 °F reduces the compressor power by 18.2%.

In a specific oil field application, to condense and remove natural gas liquids, a 6190 HP compressor is used in a propane chiller at the conditions of the preceding paragraph. Addition of a two-phase turbine would reduce the required compressor power to 5062 HP. A portion of the reduction in power arises from the shaft power generated by the two-phase turbine. The turbine power can be applied directly to the compressor shaft or it can be used to generate electricity that can be fed to the compressor motor. In the example above, the power generated is 12.7% of the compressor power. In addition, the liquid fraction produced is increased, increasing the cooling effectiveness of the system. For the oil field example, the liquid propane output is increased by 6.3%. Thus, in addition to the overall energy savings of 18.2% the size of the compressor and its motor can be reduced by 6.3% while achieving the same cooling duty. If the two-phase turbine is mechanically coupled to the compressor, its motor can be reduced by 18.2% instead of 6.3%.

**Two-Phase Turbine for an HFC134a Chiller**

A two-phase turbine applied to a 500 ton HFC134a centrifugal chiller is shown during a set-up for air and water testing in Figure 4. Liquid HFC134a enters a two-phase nozzle, 1. The refrigerant is flashed during expansion in the nozzle, 2. The resulting two-phase jet, 3, impinges upon impulse blades, 4, and a separating shroud, 5. Liquid refrigerant is separated on the blades and shroud and the flow direction is reversed. The separated gas also traverses the blades generating power. The HFC134a turbine used on the 19XT centrifugal chiller [6] has six two-phase nozzles fed by a liquid refrigerant manifold. The rotor is attached to the shaft of the compressor motor. The power generated directly unloads the mechanical requirements of the motor. Figure 5 shows a cut-away of 19XT turbine/compressor unit.

**Two-Phase Turbine Performance Analysis**

A generic one-dimensional code has been developed for design and performance analysis of the two-phase turbine. The program can analyze turbine performance of two-phase single-component fluids, such as flashing refrigerants, or two-phase liquid-gas mixtures, such as air-and-water. The program analyzes nozzle and rotor performance sequentially. The nozzle portion of the code calculates heat and momentum transfer, droplet breakup, vapor-liquid slip, and wall friction in the nozzle. The rotor portion calculates the relative flow angles; momentum loss for impingement on the blades; frictional loss in the separated liquid films forming on the pressure side of the blade; losses due to divergence of the rotor exit flow resulting from impingement momentum losses; and losses for that portion of the liquid flow that is centrifuged to the shroud and stagnates at shroud velocity (the so-called stagnation fraction). The stagnation fraction has to be specified to calculate the stagnation loss and can be used as a fudge factor to match prediction with test data.

The code can be run in either a design or an analysis mode.
Comparison of Predicted and Measured Performance

Initial experimental work on the 19XT turbine consisted of testing various nozzle designs with HFC134a. The purpose of these tests was to confirm nozzle performance with HFC134a as predicted by the code, since original code validation was done with nitrogen-and-water, steam-and-water, HCFC22, CFC113 and air-and-water mixtures. These HFC134a nozzle tests were carried out in 1993 at Biphase Energy Company in Placentia, CA. Excellent agreement was obtained between predicted and measured nozzle performance. Nozzle efficiency at design operating conditions was confirmed to be around 84%.

Complete turbine testing with HFC134a, both as a stand-alone unit where the power generated in the turbine was directly measured from the output of the two-pole induction generator and as an integral unit with HFC134a, was carried out at the Carrier Commercial Applied Development Laboratories in Syracuse, NY from late 1993 through early 1995. The overall efficiency of the HFC134a production turbine (which is the product of the nozzle and the turbine efficiency) was found to be between 50 and 55%. Consequently, indirectly derived HFC134a rotor efficiencies range from 60 to 65%, substantially below the predicted rotor efficiencies of around 75% predicted without stagnation losses.

Use of a stagnation fraction \( f_0 \) of 0.30, meaning that 30% of the liquid leaves the rotor at rotor speed, reproduces the experimentally derived rotor peak efficiency of 65% (see Figure 6). The following physical phenomena could cause stagnation losses:

1. Liquid droplet flow reflection from the leading edges of the blades. The model assumes that all liquid and vapor traverses through the rotor, but what fraction of the flow is reflected and never makes it through the rotor?
2. Stagnation of liquid flow at the shroud. What fraction of the flow is stagnated at the shroud?
3. The rotor model is a one-dimensional mean streamline model. However, hub-to-shroud radius ratio of the rotor is 0.67. Since the rotor blades were straight (identical blade angle from hub to shroud) a substantial fraction of the flow impinges at non-optimum angles.

In order to quantify the potential contribution of each of these loss mechanisms it was decided to construct a two-phase rotor test facility at Biphase Energy Company. The excellent agreement for both HFC134a nozzle test results and previous air-and-water mixture nozzle test results with the prediction of the code enabled the design of an air-and-water nozzle and define a set of air-and-water inlet pressure conditions which would result in two-phase nozzle exit flow conditions (jet droplet velocities) equal to the conditions of the HFC134a two-phase jet. Having achieved representative rotor inlet conditions, rotor performance can be tested in this open-loop air-and-water test environment allowing easy flow visualization for low test loop capital investment and operating cost.

The HFC134a representative air-and-water turbine rotor test facility was constructed at Biphase Energy Company in 1995. The loop has been operational since mid 1995. Initial flow visualization studies indicated qualitatively three loss mechanisms not accounted for in the original rotor model: inlet flow reflection, liquid stagnation at the shroud and radial inlet and outlet exit flow divergence. Subsequent quantitative flow measurements revealed that at peak efficiency point, 10% of the total flow was reflected from the inlet and 4% stagnated on the shroud.

In order to reduce these losses a new shroud has been designed to collect and recover energy from reflected and stagnated flow. The one-dimensional code has been modified to enable design of twisted leading edges allowing optimum incidence from hub to shroud. Shortened axial blade length rotor designs will reduce the residence time of the liquid flow on the blades, which currently corresponds to about 35 degrees of rotation, and are expected to reduce the stagnation losses. With these changes the efficiency for the next generation 19XT HFC134a turbine is anticipated to increase to 65-70%. Since these improvements are achieved by a rotor design only, existing turbine units can easily be retrofitted for improved performance.

Conclusions

Two-phase turbines have been applied commercially to reduce the compressor power required for HFC134a centrifugal chillers. The demonstrated efficiency and cost appear capable of providing substantial savings for other commercial and industrial refrigeration applications. Testing to date has revealed two-phase turbine rotor loss mechanisms. Reduction of these losses would result in higher performance turbines leading to larger system efficiency gains.

References


![Figure 1. Application of two-phase turbine (dashed) to vapor compression refrigeration system (solid).](image1)

![Figure 2. Comparison of conventional vapor compression system (solid) to addition of a two-phase turbine (dashed).](image2)
Figure 3. Reduction in compressor power with the application of two-phase turbines for four refrigerants.

Figure 4. Two-phase turbine during air-and-water testing.
Figure 5. Cut-away of the 19XT turbine/compressor unit.

Figure 6. Comparison of predicted versus measured turbine rotor efficiencies.