GRC Annual Meeting
September 25-28th, 2005; Reno, NV, USA

Power Production from a Moderate -Temperature Geothermal Resource

Joost J. Brasz
Carrier Corporation
Syracuse, NY 13221
joost.j.brasz@carrier.utc.com

Bruce P. Biederman
United Technologies Research Center
East Hartford, CT 06108
biederbp@utrc.utc.com

Gwen Holdmann
Chena Hot Springs Resort
Fairbanks, AK 99711
gwen@yourownpower.com

ABSTRACT

Organic Rankine Cycle power production from low temperature resources has inherently a low thermal efficiency. Low efficiency requires increased power plant equipment size (turbine, condenser, pump and boiler) that can become cost prohibitive. The use of ORC power plant hardware derived from air-conditioning equipment overcomes this cost problem since air-conditioning hardware has a cost structure almost an order of magnitude smaller than that of traditional power generating equipment.

Using the HVAC derivative concept a low-cost 200 kW ORC power plant has been developed as a derivative of a standard 350 ton air-conditioning equipment. The corresponding PureCycle^{TM}200 product was introduced in 2004. It uses waste heat exhaust gases and air-cooled condenser equipment. This paper describes the extension of this ORC development work towards power production from moderate-temperature geothermal resources.

1. BACKGROUND (ORIGIN OF THE IDEA)

Compressors in HVAC and refrigeration installations are known to start running in reverse after system shut-down unless special provisions are in place to prevent that. Reverse operation can be detrimental for certain type of positive displacement compressors, e.g. scroll and screw compressors, requiring a check valve in the compressor discharge line to prevent reverse rotation. Turbo-compressors, such as the centrifugal compressors used on water-cooled chillers, can easily be designed to handle temporary reverse rotation following system shut-down. After system shutdown pressure equalization takes place between the condenser and the evaporator. During that process refrigerant flashes/boils in the condenser and condenses in the evaporator, temporarily reversing the original design roles of these heat exchangers with heat now being rejected to the chilled water loop and extracted from the cooling tower water loop. Figure 1 compares the normal operation of the refrigeration or vapor compression cycle with its operation during pressure equalization.

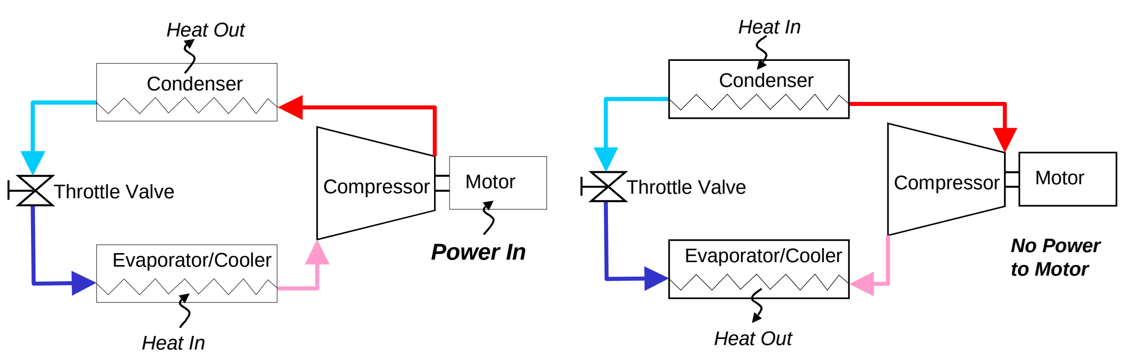

Figure 1. Comparison of a vapor compression system before and after shut down

The rotational speed of the turbo-compressor is immediately reversed after shutdown. The time required for pressure equalization depends on the amount of refrigerant charge in the condenser and evaporator relative to the size of the compressor and whether or not the cooling tower and chilled water pumps are stopped. Pressure equalization takes typically place within half a minute. One of the qualification tests a water-cooled chiller centrifugal compressor has to go through during its development phase is the sudden power failure test. Since the heat transfer between refrigerant and water is reversed during pressure equalization, the entering condenser water is temporarily cooled down and the water entering the cooler/evaporator is temporarily warmed up. Figure 2 compares chiller behavior at normal operating conditions versus that after shut down during pressure equalization.

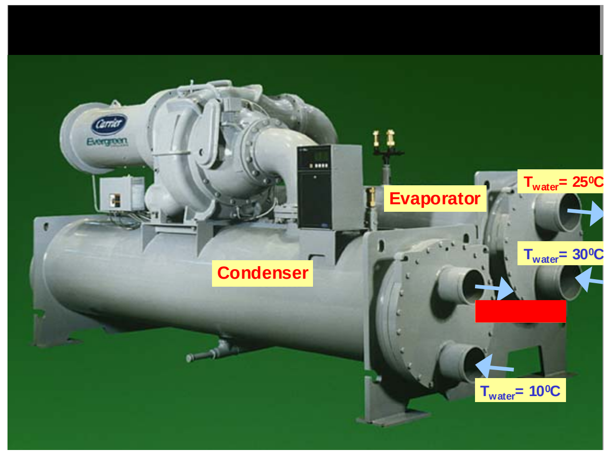

Figure 2. Comparison of normal and shut-down operation of a centrifugal chiller

During the development of the 19XR centrifugal chiller, which uses a compressor with a discrete passage diffuser as opposed to the vaneless diffuser concept used on previous designs, high reverse rotational speed was observed after shutdown. Reverse rotational speeds up to 75% of the original speed were now seen. The explanation for this phenomenon is that the discrete diffuser passages introduced for better pressure recovery in the diffuser and therefore higher compressor efficiency act as perfect turbine nozzles during pressure equalization. This observation triggered the idea to actually use this compressor as a turbine. Figure 3 shows how the impeller/pipe diffuser combination of a centrifugal compressor can act as the perfect nozzle/rotor combination for a radial inflow turbine when flow direction and rotor speed are reversed.

Figure 3. Comparison of centrifugal compressor and radial inflow turbine operation

2. THE COST ADVANTAGE OF USING AIR-CONDITIONING EQUIPMENT FOR POWER GENERATION

Thanks to equipment standardization and high-volume production, air-conditioning and refrigeration equipment is available at a cost of around $ 200 - $ 300 per kW electric motor input. For example, the cost of a 1500 kW centrifugal chiller with a 300 kW electric motor varies, depending on options, from $ 60,000 to $ 90,000. The equipment cost of multi-megawatt conventional power generation equipment is an order of magnitude higher ($ 1,200 – 1,500 per kW generator output). Even higher cost is encountered for smaller (100 to 2000 kW) distributed power generation equipment. Reciprocating engines are the exception at $ 500 per kW, but these engines have emission problems and suffer from high maintenance costs.

Industrial processes generate large amounts of waste heat. Waste-heat-driven steam power plants are often not economical, especially for capacities below 5 MW and for low-temperature waste-heat streams. In those cases waste heat power recovery has been attempted with Organic Rankine Cycle (ORC) machines. Due to high cost of the equipment, the penetration of this technology has been limited to specific niche markets such as geothermal. Moreover, most ORC applications have been heavily subsidized. The reason for high equipment cost is that current ORC systems utilize low-volume power equipment hardware. Waste heat power recovery systems are inherently limited in thermal efficiency due to the relatively low temperature of waste heat. Consequently, a waste heat power generating ORC system requires larger capacity components (boiler, condenser, turbine and pump) for equivalent power output than conventional fuel fired power generation equipment. This causes high overall system cost. Efforts to improve the ORC cost structure by focussing on thermal efficiency enhancements have not been successful in bringing the system cost down to a level that would allow a large market penetration. The absence of fuel cost means that the economically correct metric to be used for waste-heat power recovery systems is its cost per unit of power generating capacity ($\$/kW_{el}$). Better efficiency is only beneficial as far as it results in lower equipment/installation cost since the waste heat is free.

3. R-245fa, THE ENABLING REFRIGERANT FOR AIR-COOLED ORC SYSTEMS

Given the lower cost structure of HVAC equipment versus power generating equipment, and the apparently good turbine action of the centrifugal compressor during power outages, it was decided to design an ORC system using HVAC hardware to the maximum extend possible. Only minor equipment modifications - not fundamentally affecting the equipment - were allowed. For example, modifying equipment to achieve higher cycle efficiency by going to higher boiler temperatures was only allowed if the resulting improvement in efficiency would result in a lower-cost overall product without too much additional development work.

Air-conditioning equipment is only cost effective if it is used to its full design capability. Temperature/working fluid combinations that result in a turbine power output less than the power input of the existing compressor would not fully utilize the potential of this compressor hardware during turbine operation and would therefore result in higher equipment cost per unit power delivered. Conversely, temperature/working fluid combinations that result in a turbine power output higher than the power input of the corresponding compressor hardware would exceed the mechanical limits (e.g. gear and shaft torque limits and bearing loading limits) of the original compressor design. Modifications to overcome those limits were only allowed if the net cost per unit power delivered would reduce, again without too much additional development work.

In order to preserve the cost advantage of the HVAC compressor as an ORC turbine it was found that the maximum temperature and pressure the turbine is seeing should be within the capabilities of the existing compressor housing. Moreover, to take full advantage of the given compressor hardware in turbine operation the power density of the turbine should be equal to that of the compressor. This allows unaltered use of the electrical and mechanical components of the centrifugal compressor. In other words, if a 200 kW compressor is used in an ORC application, the pressure-flow characteristics of the working fluid in the ORC system have to result in a 200 kW turbine output. Using R-134a, the working fluid used for the chiller application, as working fluid for the ORC application, would result in unacceptably high operating pressures requiring major redesign. Therefore, a lower-pressure working fluid is required for the higher- temperature operational regime of the organic Rankine cycle.

The non-flammable and non-toxic refrigerants promoted in the past as attractive fluids for organic Rankine cycle systems [1] have all been outlawed because of their ozone layer depletion potential. The HFC refrigerants introduced during the last decade to replace the CFC and HCFC refrigerants, such as R-134a, R-407C and R410A, have relatively low critical temperatures resulting in low ORC cycle efficiencies [2]. These refrigerants also would result in very high evaporator and turbine inlet pressures. This only leaves flammable and sometimes also highly toxic hydrocarbon fluids, such as pentane and toluene, as the fluids currently used on ORC installations. Apart from the flammability and toxicity issues, these fluids would, If applied to existing HVAC compressor hardware, fail to achieve the required turbine power density given their low vapor density at moderate temperature levels. These fluids would need larger turbine and condenser equipment than is available in the HVAC industry.

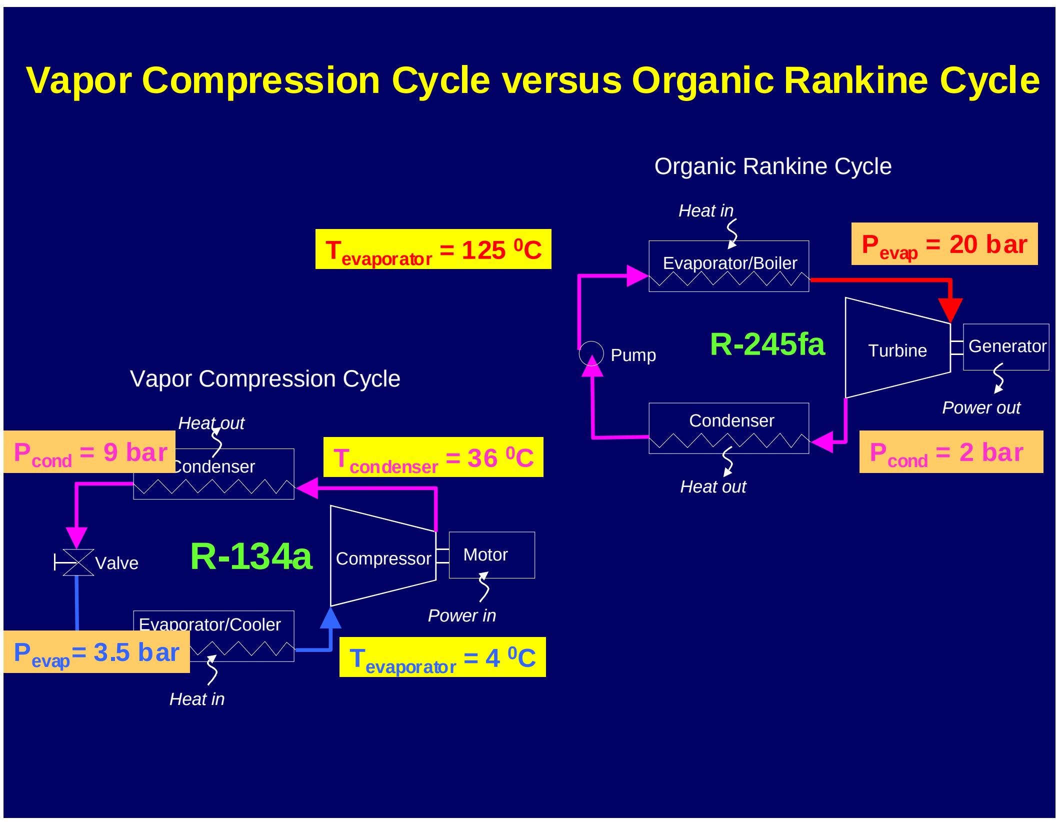

Figure 4. The vapor compression cycle at refrigeration temperature levels has pressures and power densities similar to the organic Rankine cycle at waste-heat- temperature conditions

Recently, R-245fa , a non-flammable, low-pressure (relative to R-134a) HFC refrigerant has been introduced for the foam blowing industry [3,4]. The relatively high critical temperature of this fluid means that it would be an efficient working fluid for a moderate temperature ORC system [2].

R-245fa is the fluid that allows usage of mass-produced existing HVAC compressor and heat exchanger hardware. The current offering of HVAC derived organic Rankine cycle systems uses hot gas as a heat source and ambient air as the heat sink. Minor modifications were required to existing HVAC hardware to obtain the optimized ORC components. The air-cooled condenser for the ORC system is derived from the Carrier commercial unitary product line with only slight circuiting adjustments to account for the lower density of R245fa in the condenser. The R245fa evaporator is a derivative from the generator in the gas-fired absorption product line. The turbine is a slightly modified centrifugal compressor design [5].

4. EXISTING INSTALLATIONS

Three 200 kW power producing installations using different waste heat sources have been in operation since January 2004. The exhaust heat of an Pratt and Whitney FT12 gas turbine is used as heat input source for the organic Rankine cycle in East-Hartford, CT. A second installation uses the heat from a landfill flare in Austin, TX while the exhaust heat from three Jenbacher reciprocating engines powers the third installation in Danville, IL. Figure 5 shows pictures of these installations.

East Hartford, CT | Austin, TX | Danville, IL

Figure 5. Existing HVAC-derived ORC installation

After successful continuous operation since January 2004, the ORC product has been offered for sale in the US in August 2004. UTCPower sells the product under the PureCycle^{TM}200 trademark name, in close cooperation with Carrier Corporation [6].

5. LOWER TEMPERATURE ORC APPLICATIONS

Cost-effective ORC operation requires a minimum temperature difference between evaporator and condenser saturation temperatures of about 50K or 100 ^{0}F. If air-cooled condensers are used, as in the PureCycle^{TM}200 system, nominal condenser saturation temperatures of around 35 ^{0}C or 100 ^{0}F are encountered. As a result evaporator/boiler saturation temperatures from 90 to 120 ^{0}C (200 –250 ^{0}F) are required for cost-effective ORC operation.

Lower saturated condenser temperatures enable power recovery from lower temperature heat sources utilizing ORC technology. These lower temperature applications increase the similarity between existing air-conditioning equipment and ORC hardware. If the operating temperatures of the ORC system approach those of the air-conditioning system the unit was derived from, the original refrigerant used the corresponding centrifugal chiller becomes the preferred working fluid for the organic Rankine cycle at these temperature levels. R-245fa is the working fluid that at the normally higher ORC condenser and evaporator saturation temperatures achieves power-density similarity with a conventional R-134a centrifugal chiller. R-245fa would result in lower power output for the same turbine at reduced evaporator/condenser temperatures due to the lower densities and pressures. To obtain power density similarity at these lower saturation temperatures between compressor and turbine operation the original centrifugal chiller refrigerant R-134a should be used as ORC working fluid. Figure 6 shows side-by-side conventional chiller operation and low temperature ORC operation utilizing the same working fluid.

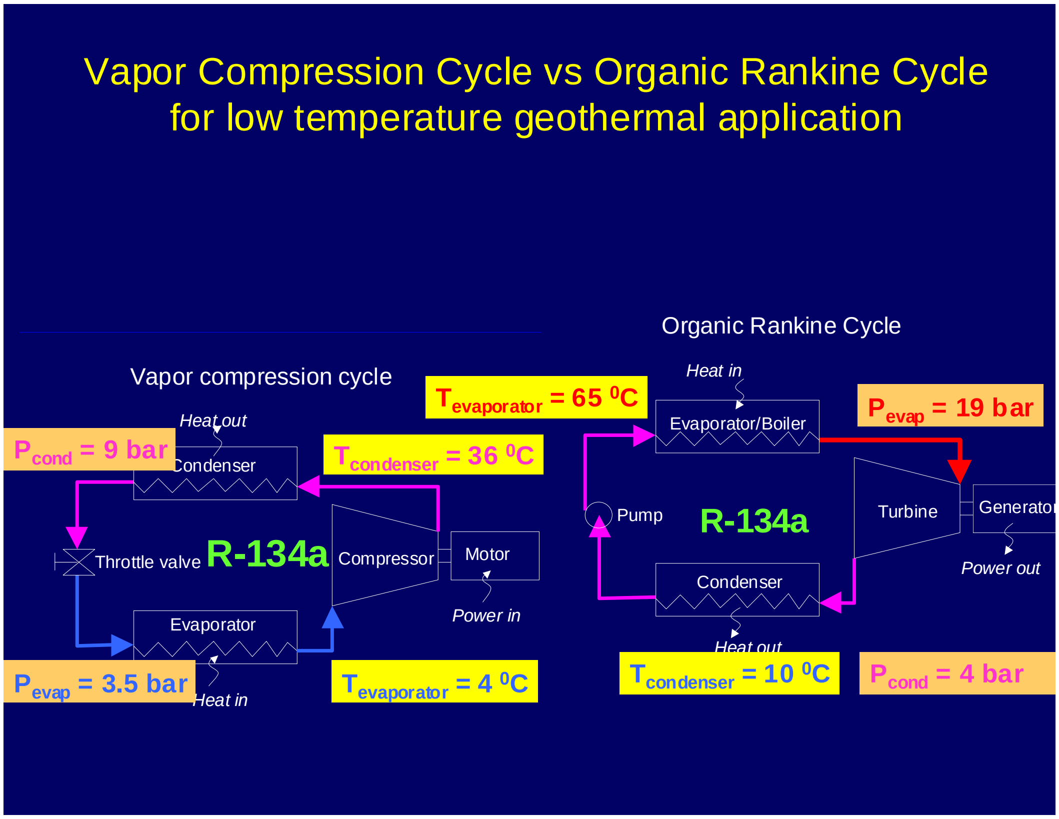

Figure 6. The vapor compression cycle at refrigeration temperature levels has pressures and power densities similar to the organic Rankine cycle at low geothermal heat resource temperature conditions

One such application for utilizing an Organic Rankine cycle to recover power from a low-temperature geothermal heat source is the proposed ORC system for Chena Hot Springs in Alaska. The geothermal wells at Chena Hot Springs produce hot water of around 74 ^{0}C (165 ^{0}F). This temperature is in general too low for ORC duty. However, given the year-round availability of 3 ^{0}C (37 ^{0}F) river water ORC-operation with an 8% thermal efficiency is possible. Figure 7 shows the temperature entropy diagram of the cycle anf Figure 8 shows a picture of the unit with evaporator and condenser entering and leaving water temperatures.

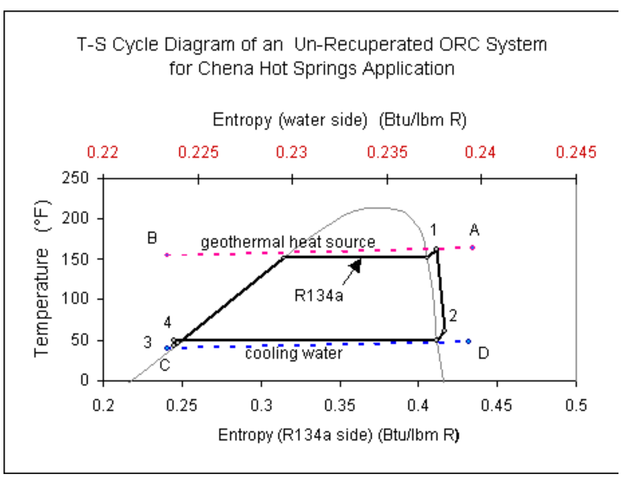

Figure 7. Temperature-entropy diagram of the proposed Chena Hot Springs ORC unit

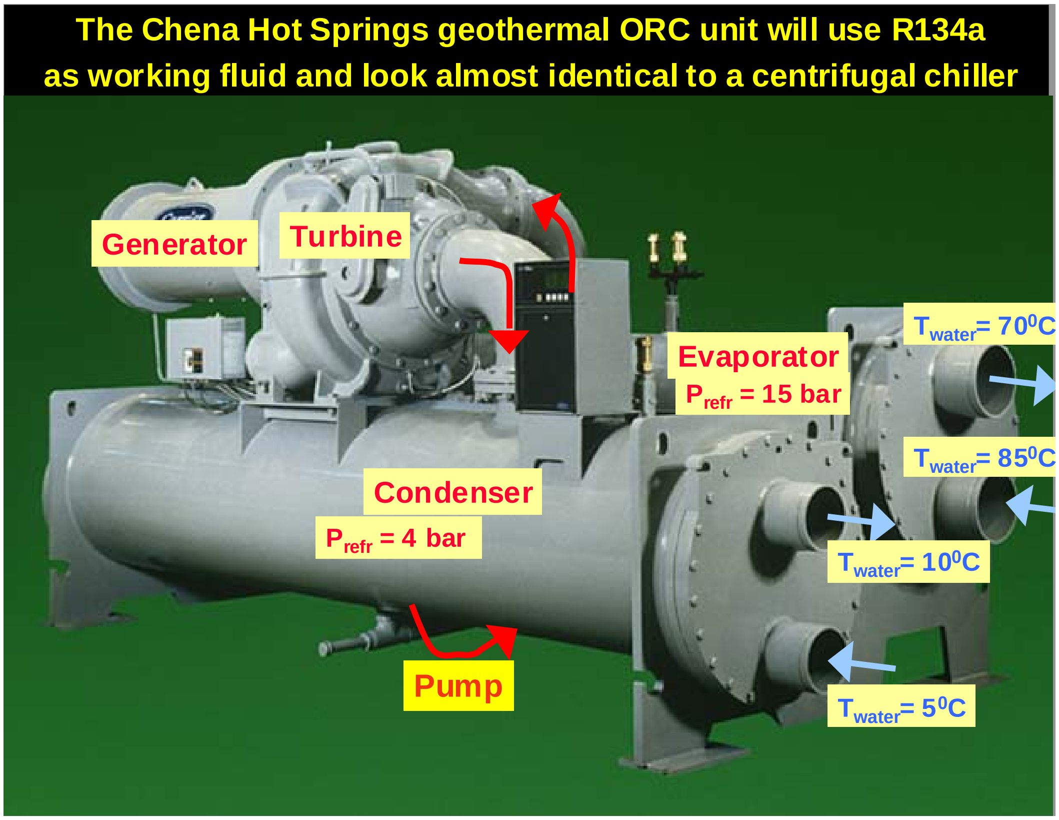

Figure 8. The Chena Hot Springs geothermal chiller derivative ORC unit

6. CONCLUSIONS

1. An Organic Rankine Cycle (ORC) turbine has been developed as a derivative of an existing centrifugal compressor with a discrete passage diffuser used on water-cooled chillers.
2. In order to operate at the higher temperature levels required for ORC duty when using engine exhaust or flare heat as heat input into the ORC unit, a lower pressure refrigerant was required to keep working pressures at acceptable levels.
3. Power density matching between compressor and turbine operation is needed for cost-effective utilization of existing compressor hardware as a radial inflow turbine. This can be achieved by switching from R-134a for HVAC compressor operation to R-245fa for ORC turbine operation when encountering higher temperature waste heat sources.
5. For moderate geothermal resource temperatures the power producing ORC system can use the original refrigerant used in centrifugal chillers at its working fluid, increasing the similarity between HVAC and ORC equipment. Such a unit is currently under construction for testing at Chena Hot Springs in the near future.
4. The cost advantage of HVAC equipment over power generation equipment allows an economically viable product despite the inherently (= second law of thermodynamics) low thermal efficiency of low temperature waste heat power recovery.

7. REFERENCES

1. Smith, I.K., *The Choice of Working Fluids for Power Recovery from Waste Heat Streams*, Transactions by the Institute of Marine Engineers of Conference on Organic Fluids for Waste Heat Recovery in Ships and Industry, pp. 8-18, January 7-8, 1981.
2. Brasz, L.J., Bilbow, W.M., *Ranking of Working Fluids for Organic Rankine Cycle Applications*, paper R068 presented at the 10th International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, Indiana, July 12-15, 2004.

3. Zhong, B., Bowman, J.M., Williams, D., *HFC-245fa: An Ideal Blowing Agent for Integral Skin Foam*, Paper presented at the International Conference and Exposition Polyurethanes Expo 2001, Columbus, Ohio, September 30 - October 3, 2001.
4. Zyhowski, Sr, G.J., Spatz, M.W., Yana Motta, S. *An Overview of the Properties and Applications of HFC-245fa*, Paper presented at the Ninth International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, Indiana, July 16-19, 2002.
5. Brasz, J.J., *Transforming a Centrifugal Compressor into a Radial Inflow Turbine*, paper C060 presented at the 17^{th} International Compressor Engineering Conference at Purdue, West Lafayette, Indiana, July 12-15, 2004.
6. http://www.utcfuelcells.com/utcpower/products/purecycle/purecycle.shtm