Renewable Power Production of Moderate Temperature Geothermal Heat Using Air Conditioning Hardware

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Abstract: - A system has been developed to enable cost-effective power production from moderate temperature liquid heat sources, defined as temperatures less than 125 ^{0}C. Examples of such heat sources are the warm water from geothermal wells, the warm water/oil mixture from active or abandoned oil and gas wells and the warm jacket water from reciprocating engines that is normally cooled in radiators. The moderate temperature power plant is a vapor power cycle with an organic fluid or refrigerant instead of water/steam as the working fluid. Functionally it resembles the steam cycle power plant: A pump downstream of the condenser increases the pressure of the condensed working fluid. This high-pressure liquid is then vaporized in an evaporator/boiler by extracting heat from some moderate temperature heat source. The high-pressure vapor expands in a turbine, producing power. The low-pressure vapor leaving the turbine is condensed before being sent back to the pump to restart the cycle.
The paper will describe the successful demonstration of this technology in generating 400 kW electrical power from a 73 ^{0}C geothermal; heat source in Fairbanks, AK and will illustrate the potential of this technology for moderate temperature geothermal resources in Greece.

Key-Words: - Organic Rankine Cycle, Geothermal Power, Moderate Temperature Heat, Air Conditioning Hardware, Renewable Power, Waste Heat Power Recovery

1. Introduction

The commercial penetration of waste heat power recovery systems into the market was limited in the past due to the inherently low thermal efficiency as dictated by the second law of thermodynamics for power generation of moderate temperature (< 125 ^{0}C) sensible heat sources (< 125 ^{0}C). The maximum thermal efficiency (assuming all heat transfer, pressure rise and expansion processes are reversible) of a power generation process utilizing a constant temperature 100 ^{0}C (373.15 K) heat source and a constant temperature 15 ^{0}C (288.15 K) heat sink is given by the Carnot efficiency:

$$\eta_{Carnot} = \frac{T_H - T_L}{T_H} = \frac{373.15 - 288.15}{373.15} = 22.8\% \qquad (1)$$

Actual process irreversibilities such as the necessary temperature difference between the working fluid of the cycle and the heat source/heat sink fluids, the inefficiencies of pumps and turbines and the friction losses in the connecting piping result in final thermal efficiencies substantially (30-50% lower) than the Carnot efficiency.

For a sensible heat source, such as the hot water from a geothermal well, the temperature drops when heat is extracted. As a consequence, the average temperature difference between heat source and heat sink diminishes, causing a further reduction in thermal efficiency. It can be shown that the maximum possible efficiency of a sensible heat source where during heat extraction by the power plant the temperature reduces from T_{1H} to T_{2H} and an infinite heat sink at temperature T_L

$$\eta_{\max} = 1 - \frac{T_L}{T_{1H} - T_{2H}} \ln\left(\frac{T_{1H}}{T_{2H}}\right) \qquad (2)$$

Equation (2) can be used to illustrate the lower thermal efficiency resulting from a finite latent heat source compared to the Carnot efficiency given by Equation (1).

If for example, contrary to the constant temperature heat source of 100 ^{0}C assumed by the Carnot cycle, the sensible heat source is cooled down from 100 ^{0}C to 25 ^{0}C and if the ambient temperature at which the condenser rejects its heat remains 15 ^{0}C, the maximum efficiency becomes 13.8%.
The inherently low thermal efficiencies of power equipment for moderate temperature geothermal sensible heat sources require relatively large heat exchanger and machinery equipment per unit of power produced. This affects the power plant cost negatively. However, the temperature levels encountered in these applications approach more the air-conditioning temperature range than the steam power generation temperature range.
This allows the use of lower cost air-conditioning components instead of the more expensive power generation industry hardware. Use of low-cost air-conditioning hardware for moderate temperature power generation can offset the equipment cost penalty that results from the requirement of larger heat exchangers as a result of the inherently lower thermal efficiency.

2. HVAC compressors running in reverse as turbines/expanders

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. 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

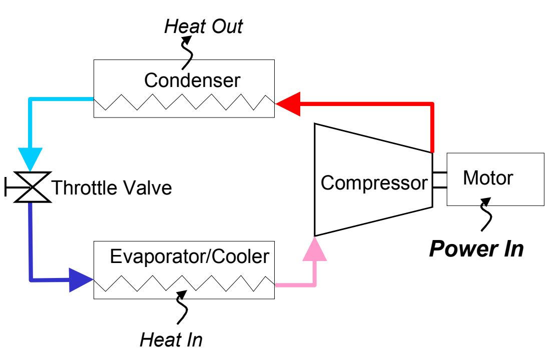


Figure 1a. Vapor compression system before shut down

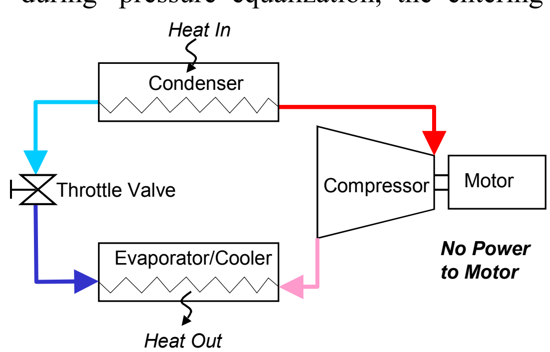


Figure 1b. Vapor compression system during shut down

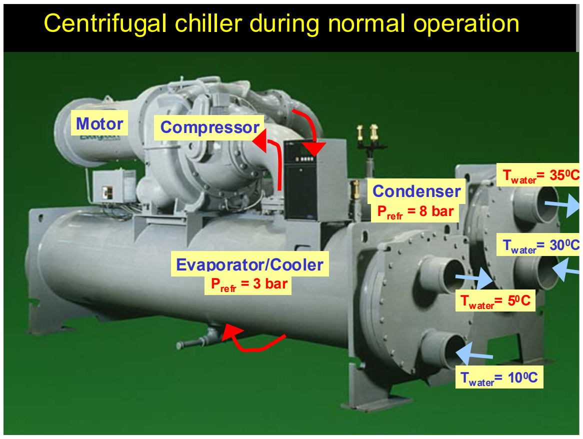


Figure 2a. Centrifugal chiller during normal operation

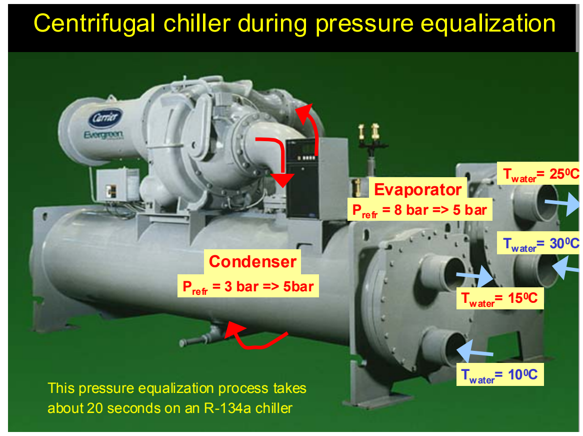
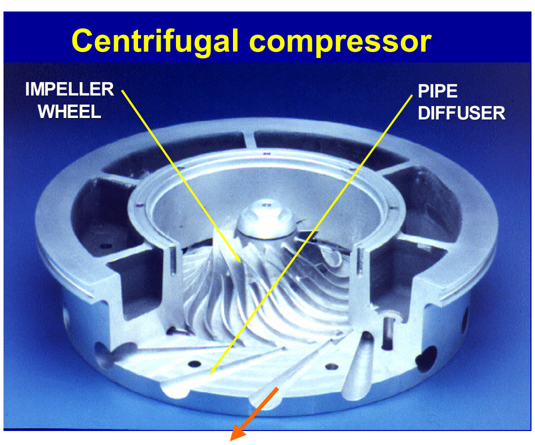


Figure 2b. Centrifugal chiller during pressure equalization

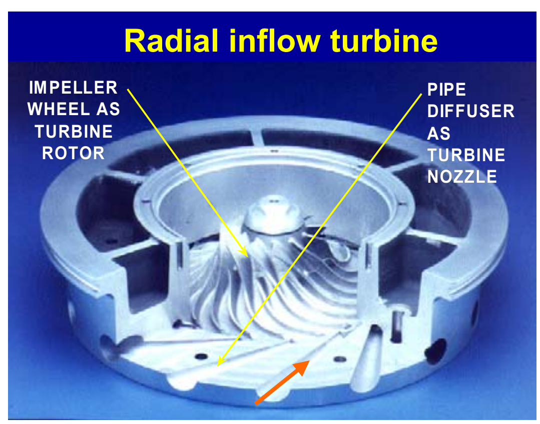
condenser water is temporarily cooled down and the water entering the cooler/evaporator is temporarily warmed up. Figures 2a and 2b compare chiller behavior at normal operating conditions versus that after shut down during pressure equalization.

During the development of a 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. Figures 3a and 3b show 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.



<u>Compressor Operation:</u>
Cut-away Of Impeller
(Spinning Clockwise)
and Pipe Diffuser
(Radial Outward Flow)

Figure 3a. Centrifugal compressor



<u>Turbine Operation:</u>
Cut-away Of Impeller
(Spinning Counter-clockwise)
and Pipe Nozzle
(Radial Inward Flow)

Figure 3b. Radial inflow turbine

3. 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. 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 focusing 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.

4. R-245fa, the enabling refrigerant

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 considered 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 equipment than is available from existing HVAC compressors.

Recently, 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], as shown in Figure 4.

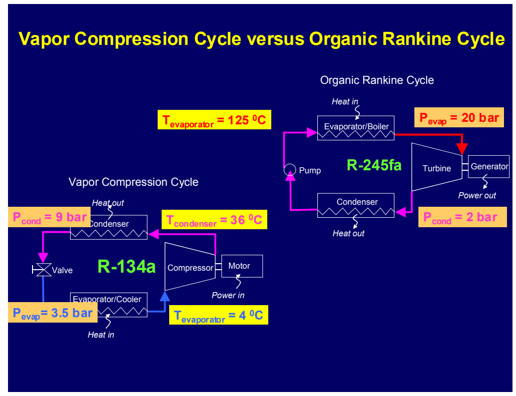


Figure 4. The vapor compression cycle at refrigeration temperature levels has pressures and power densities similar to the organic Rankine cycle at low waste-heat-temperature conditions

5. Adjustments required for a cost-effective transformation

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]. The two main differences are illustrated in Figure 5:

1. The inlet blade height of the turbine rotor is smaller than the compressor impeller exit blade height in order to accommodate the larger change in vapor density experienced by the turbine
2. The turbine rotor has radial inlet blading as opposed to the backswept blading for the impeller of the compressor. The larger turbine pressure ratio results in a

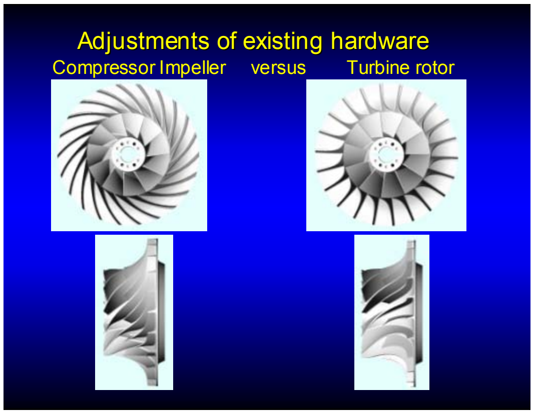


Figure 5. Differences between compressor impeller and turbine rotor design

larger nozzle exit velocity than the corresponding diffuser inlet velocity from the impeller. Radial turbine blading is required to match nozzle exit velocity with rotor tip speed.

5. PROTOTYPE INSTALLATIONS

Initial prototype installations used gaseous waste heat as heat source and ambient air as heat sink. Three 200 kW power plants using different gaseous waste heat sources have been in operation since January 2004. The exhaust heat of a Pratt and Whitney FT12 gas turbine is used as heat input source for the organic Rankine cycle in East-Hartford, CT, shown in Figure 6. Figure 7 shows a field installations using waste heat from a landfill flare and Figure 8 shows an ORC power plant using the exhaust of three reciprocating engines.

Figure 6. ORC power plant using gas turbine exhaust in East Hartford CT

Figure 7. ORC power plant using landfill flare exhaust in Austin, TX

Figure 8. ORC power plant using reciprocating engine exhaust in Danville, IL

Based on the experience with these first air-cooled prototype units and especially the installation issues encountered with gaseous heat sources it was decided to develop a water-cooled, hot-liquid driven power plant, looking a lot start a centrifugal chiller. Prototypes of this configuration have been running since 2005 at United Technologies Research Center using hot liquid and/or steam as heat source. Figure 7 shows a picture of this power plant.

Figure 9. Water-cooled, hot-liquid driven *ORC power plant in East Hartford, CT*

The first field installation of this system has been running since July 2006 at the Chena Hot Springs, near Fairbanks, Alaska [6] making 400 kW of electricity from warm water at 73 ^{0}C. Figure 8 shows a picture of the two power plants running at Chena Hot Springs.

Figure 10. Two ORC power plants (400 kW) at Chena Hot Springs near Fairbanks, AK

6. Conclusions

1. An Organic Rankine Cycle (ORC) turbine was 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 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.
4. The cost advantage of HVAC equipment over power generation equipment allows an economically viable product despite the inherently low thermal efficiency of waste heat power recovery (9-15%).

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