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ELECTRICAL POWER GENERATION WITH LOW-TEMPERATURE ORGANIC RANKINE CYCLE MACHINES

BY

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Abstract. By 2050, global energy consumption is projected to grow by 71%. At the same time, energy-related carbon dioxide emissions are expected to rise by more than 40%. In this context, waste and renewable energy sources may represent alternatives to help reduce fossil primary energy consumption. This paper focuses on the theoretical design and performances of a 50 kW laboratory beta-prototype Organic Rankine Cycle (ORC) machine using low-enthalpy heat as industrial waste or deep geothermal renewable energy at temperatures varying between 85 and 125°C. The experimental study shows that the electrical power generated and the overall net conversion efficiency of the ORC machine depend on parameters as the inlet temperatures of the waste (or renewable) heat and cooling fluids, as well as on the control strategy under variable operating conditions.

Key words: electrical power generation; industrial waste heat recovery; energy efficiency.

1. Introduction

The use of fossil fuels (coal, oil, natural gas) in industry, as iron and steel, pulp and paper, and chemical, contributes to the emission of pollutants that are damaging to the environment. On the other hand, up to 50% of the

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consumed energy is lost into the environment at relatively low temperatures (< 370°C) in the form of stack gases, vapor or liquid effluents. At such temperatures, the wasted heat cannot efficiently be used to produce electricity using steam-based power generation such as conventional Diesel, Stirling or Rankine cycles (Sonntag *et al.*, 2003). Consequently, different energy conversion techniques are required to efficiently use low-grade "free" waste heat resources for power generation. Among these alternatives, Organic Rankine Cycle-based (ORC) machines, similar to basic Rankine power plants, do not use water, but rather vaporize high-molecular-mass organic fluids with boiling points below that of water (Leibowitz *et al.*, 2006). ORC machines can use various types of low-grade industrial waste heat or renewable (solar, geothermal, biomass,) energy sources available at temperatures between 85°C and 350°C.

2. Experimental Set-Up

A pre-commercial small footprint, skid-mounted prototype based on the ORC cycle was designed (Leibowitz *et al.*, 2006), built (ElectraTherm, 2012) and installed on a test bench (Fig. 1) (Minea, 2014). It can generate up to 50 kW of electrical power by recovering heat from waste heat in liquid form at temperatures ranging from 85°C to 125°C.

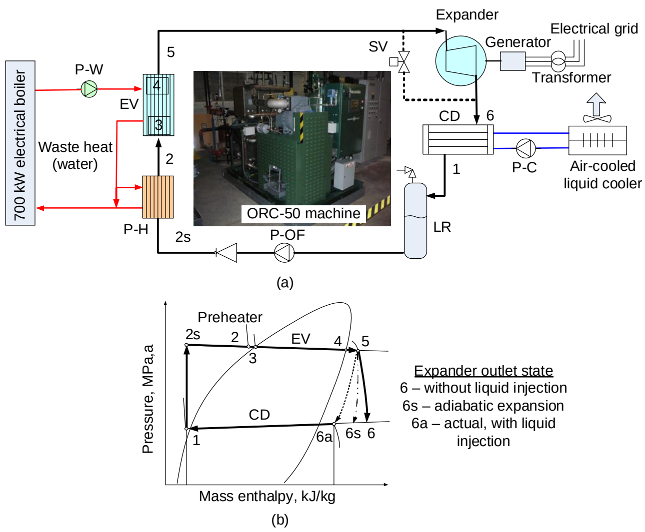

Fig. 1 – *a* – Schematic of the experimental set-up; *b* – typical thermodynamic cycle in the p-h diagram; SV – solenoid valve.

As shown in Fig. 1, the prototype includes a pre-heater (P-H) and an evaporator (EV) intended for preheating and vaporizing the organic fluid, respectively, a single-stage twin screw expander directly coupled (*i.e.* without gearbox) to an asynchronous alternator (generator), and a stainless steel plate condenser (CD). The induction alternator generates electric power at 460 Volts, which is transferred to the local electrical grid via a 25 kV/600V electrical transformer. The liquid mixture (glycol/water 50%) pump (P-C) circulates the cooling fluid within a closed-loop to be cooled by an ambient air-cooled liquid cooler. The organic fluid pump P-OF returns the organic fluid liquid accumulated in the liquid receiver LR into the preheater. A 700 kW electric boiler simulates the thermal waste heat source, and the pump (P-W) circulates the waste heat thermal carrier (water) in a closed-loop.

The working fluid is HFC-245fa, a low-pressure, high-temperature refrigerant, with relatively high enthalpy of vaporization and density, high chemical stability, safe in regards to explosion, flammability, and health, and available at relatively low prices. Moreover, HFC-245fa becomes superheated during the expansion, thus avoiding condensation of the organic fluid as it passes through the expander (Maizza & Maizza, 2001). Its Ozone Depletion Potential is zero, but its Global Warming Potential is relatively high (1900).

The experimental bench was comprehensively instrumented with thermocouples, power and pressure transducers, flow meters for the working fluid, and both heat source and sink thermal carriers. A data transmission system and associated analysis software were set up to monitor/control system operation. All parameters were scanned at 15 second intervals, then averaged and saved every minute, to help determine the cycle’s instantaneous and overall thermodynamic performance.

3. Design of Thermodynamic Cycle

The low-pressure organic fluid leaving the condenser as a saturated or sub-cooled liquid (state 1) accumulates inside the receiver in equilibrium with its vapour phase (Fig.1). Then it enters the working fluid feed pump (P-OF) where its pressure is adiabatically raised to the saturation (evaporating) pressure (state 2s), prior to entering the preheater/evaporator. The multistage feed pump is driven by a variable frequency drive in order to supply the flow rate required (11,...,12 kg/s) to achieve a superheating amount below 5°C at the evaporator outlet. Also, up to between 5% of the organic fluid flow rate is injected into the expander for cooling and lubrication purposes.

The waste heat fluid carrier (water) enters the preheater/evaporator where it transfers heat to the organic fluid during the preheating, evaporation and superheating processes. At the evaporator outlet, 20% of the water total flow rate (7 to 12 kg/s) enters the preheater to preheat the organic fluid from 2s up to (near) the saturated state 2a *prior* entering the evaporator where it will vaporise and superheat at a constant pressure (process 2a-3-4-5). The preheater/evaporator thermal power recovered from the waste heat source is given by:

$$\dot{Q}_{\mathrm{preheat+evap}} = \dot{m}_{\mathrm{org}} \left(h_5 - h_{2s} \right), \tag{1}$$

where: $\dot{m}_{\mathrm{org}}$ is the organic fluid mass flow rate, [kg/s]; h_5 and h_{2s} – organic fluid specific enthalpy leaving and entering the preheater/evaporator, respectively, [kJ/kg].

The high-pressure saturated (state 4) or superheated vapour (state 5) leaving the evaporator enters the twin screw expander. It accelerates the 50 kW (460 V) asynchronous induction generator (alternator) connected to the local electrical grid. Inside the screw expander, the working fluid expands creating pressure and temperature drops at the exit port, thus converting the thermal energy of the high pressure vapor into mechanical work. Without any liquid injection, the organic fluid would leave the expander at state 6 but, because of this process, it actually leaves the expander at state 6a, closer to the vapor saturated wet curve.

The power generated by the twin screw expander is calculated as follows:

$$\dot{W}_{\mathrm{exp}} = \dot{m}_{\mathrm{org}}\left(h_5 - h_{6s}\right)\eta_s \eta_m, \tag{2}$$

where: $\eta_S = (h_5 - h_6)/(\ h_5 - h_{6s})$ is the expander isentropic efficiency (≈ 70%), η_m – mechanical efficiency, h_5 and h_{6s} – working fluid specific enthalpies at the expander inlet and outlet in the ideal expansion process, respectively, [kJ/kg].

The working fluid enters the condenser at state 6a, close to its saturated state, where it is condensed at constant pressure and temperature (process 6a-1) to become a saturated or sub-cooled liquid (state 1). The condensing (latent) heat (enthalpy) is transferred from the vapor to the cooling fluid (10 to 15 kg/s) circulating within a water/glycol (50% by weight) closed loop linked to an air-cooled liquid cooler. The condenser thermal power is expressed by the following equation:

$$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{org}}\left(h_{6a} - h_1\right), \tag{3}$$

where: h_{6a} and h_1 are the organic fluid specific enthalpies entering and leaving the condenser, [kJ/kg]. The condensed organic fluid at state 1 is stored inside the liquid receiver, pumped back to the preheater/evaporator assembly and then a new cycle begins.

Since the exergy destruction rate in the feed pump is relatively small, the pumping process 1,…,2s is considered as isentropic (adiabatic), and the pump power input can be expressed as:

$$\dot{W}_{\mathrm{pump}} = \frac{\dot{m}_{\mathrm{org}}\left(h_{2s} - h_1\right)}{\eta_{\mathrm{pump}}}, \tag{4}$$

where: η_{pump} – the feed pump isentropic efficiency (≈ 90%), h_1 and h_{2s} – organic fluid specific enthalpies at the fed pump inlet and outlet in the ideal case, respectively, [kJ/kg].

4. Results

The organic fluid feed pump ran at full range variable speed with the current frequency varying between 0 and 60 Hz in order to get small amounts of superheat at the evaporator outlet and adjust the power generated to the system actual temperature lifts.

A. Temperature lifts

Experimental tests were conducted with the waste heat thermal carrier (water) and cooling fluid entering the ORC-50 machine at temperatures varying between 85°C and 115°C, and 15°C to 30°C, respectively. Under these thermal boundary conditions, the temperature lifts (*i.e.* the difference between the waste heat and cooling fluid inlet temperatures) varied between 55°C and 100°C (Fig.2).

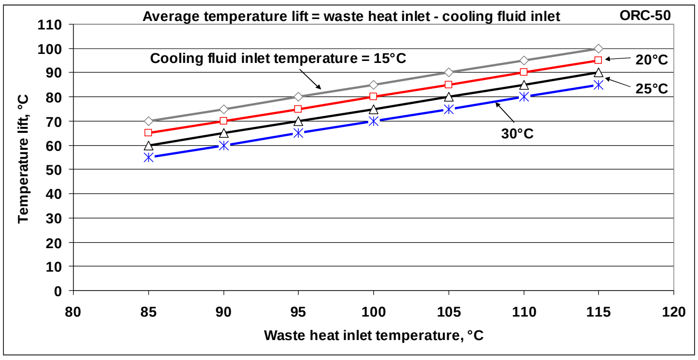

Fig. 2 – Experimental temperature lifts.

B. Net power output

Fig. 3 shows that the net power output increased linearly with the waste heat inlet temperatures when the cooling fluid inlet temperatures were kept constant. It can be seen that a maximum net power of 47 kW was generated with waste heat entering the ORC machine at 115°C and cooling fluid at 15°C, *i.e.* 94% of the maximum design output power at a temperature lift of 100°C.

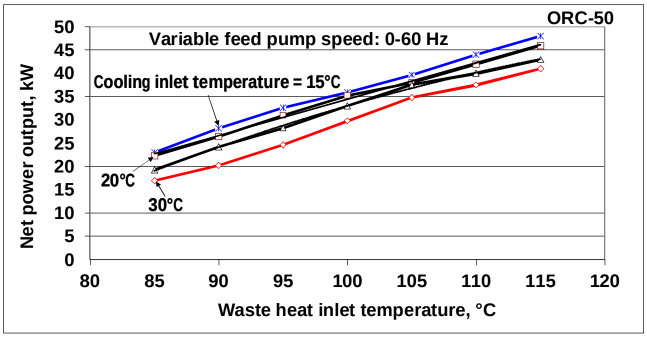

Fig. 3 – Net electric power output as a function of waste heat and cooling fluid inlet temperatures.

Fig. 4 *a* shows the thermodynamic cycle of a representative run (test AD-14) with waste heat and cooling fluids entering the evaporator and condenser at constant temperatures of 105°C and 20°C, respectively. Under these operating conditions, the expander net power output was 39.9 kW, *i.e.* about 79% of the maximum design power output of the ORC-50 machine.

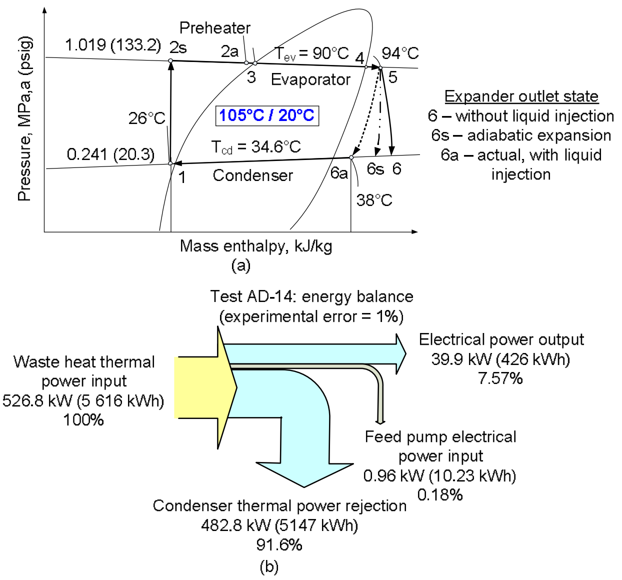

Fig. 4 – *a* – Thermodynamic cycle of test AD-14 using waste heat and cooling fluid inlet temperatures of 105°C and 20°C, respectively; *b* – experimental energetic balance; T_{cd} – condensing temperature; T_{ev} – evaporating temperature

The energetic balance of test AD-14 was achieved with a net heat-to-electricity conversion energetic efficiency of about 7.6% (Fig. 4 *b*).

B. Energetic conversion efficiency

The net heat-to-electricity energetic conversion efficiency rate (η_{net}) of the ORC machine is a dimensionless number defined as the ratio of the net electrical power output ($\dot{W}_{\text{gross, exp}} - \dot{W}_{\text{pump}}$) to the sum of the preheater plus evaporator ($\dot{Q}_{\text{preheat+evap}}$) thermal power inputs. As the electrical power required by the feed pump is about 1% of the expander power output, it has been neglected, and, thus:

$$\eta_{\text{energ}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{preheat+evap}}} \approx \frac{\dot{W}_{\text{gross,exp}}}{\dot{Q}_{\text{preheat+evap}}}, \tag{5}$$

where: $\dot{W}_{\text{net}} \approx \dot{W}_{\text{gross,exp}}$ is the expander net electrical power output, *i.e.* the gross power generated by the expander less the (neglected) feed pump electrical power input. The preheater plus evaporator thermal power inputs are expressed as follows (see Fig. 1):

$$\dot{Q}_{\text{preheat+evap}} = \dot{m}_{\text{waste}} \overline{c}_{p,\text{waste}} \left(T_{\text{waste}}^{\text{IN}} - T_{\text{waste}}^{\text{OUT}} \right). \tag{6}$$

It can be seen from eq. 5 that any parasitic electrical energy consumption (or power input), such as that of the waste heat (water) and cooling fluid (water/glycol) circulating pumps and of the air-cooled cooler fans, was not taken in consideration. However, in actual industrial applications, such parasitic power has to be analysed carefully and, if possible, eliminated or substantially reduced. This approach is sometimes possible in practice because many industrial sites are already equipped with circulating pumps for waste heat (hot water) and cooling fluids, as well as with cooling towers or other similar cooling devices.

Fig. 5 shows that, if the cooling fluid inlet temperature remains constant, the heat-to-electricity net conversion efficiency, as defined by eq. 5, increases along with the waste heat inlet temperature. It can be seen, for example, that with cooling fluid inlet temperature of 15°C, the net conversion efficiency was 6.5% with waste heat inlet temperature of 85°C, and 9% when it attained 115°C. By linearly extrapolating the measured values, the net conversion efficiency may increase to about 10% by using waste heat entering the ORC machine at 125°C.

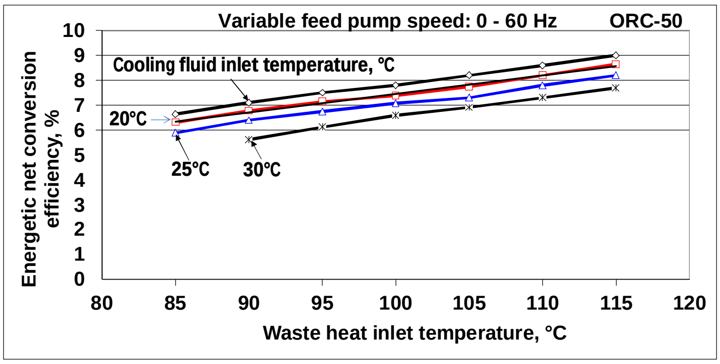

Fig. 5 – Energetic net conversion efficiency as a function of waste heat inlet temperatures and cooling fluid inlet temperatures constant at between 15°C and 30°C.

C. Exergetic conversion efficiency

Similar to heat-to-electricity net conversion efficiency, the exergetic net conversion efficiency can be defined as:

$$\eta_{\text{energ}} = \frac{\dot{W}_{\text{net}}}{\dot{E}_{\text{available}}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{preheat+evap}}\left(1 - T_a / T_{\text{waste}}^{\text{IN}}\right)}, \tag{7}$$

where: $\dot{E}_{\text{available}}$ is the exergy flux available in the waste heat source at the preheater/evaporator inlet, [kW]; $\left(1 - T_a / T_{\text{waste}}^{\text{IN}}\right)$ – Carnot factor, *i.e.* the maximum amount of waste heat input which can be transformed into mechanical work; T_a – ambient absolute temperature, [K]; $T_{\text{waste}}^{\text{IN}}$ – waste heat inlet temperature, [K].

By ignoring the potential and kinetic energies, the maximum reversible work per unit mass flow, equal to the decrease in flow availability plus the reversible work that can be extracted from an ORC cycle operating between the waste heat inlet absolute temperature ($T_{\text{waste}}^{\text{IN}}$) and the ambient absolute temperature (T_a), is defined as:

$$e = (h - h_a) - T_a(s - s_a), \tag{8}$$

where: *a* subscript refers to the dead-state, usually the environment temperature, but here, it is the state of the cooling fluid entering the condenser.

Table 1 summarizes the energetic and exergetic net conversion efficiencies of five representative tests as functions of the waste heat inlet and outlet temperatures (at a constant flow rate of 11.6 kg/s) for a reference ambient temperature of 20°C representing the cooling fluid temperature entering the condenser.

Table 1
Energetic and Exergetic Net Conversion Efficiencies

Test	$T_{\text{waste}}^{\text{IN}}$, [°C]	T_{lift}, [°C]	$\dot{W}_{\text{net}}$, [kW]	η_{energ}, [%]	η_{exerg}, [%]
AD-1	85	65	22.3	6.62	3.95
AD-3	90	70	26.4	6.98	4.11
AD-8	95	75	31	7.2	4.29
AD-9	100	80	35	7.38	4.38
AD-14	105	85	39.9	7.57	4.43

Note: cooling fluid inlet temperature constant at 20°C.

D. Impact of superheating

The impact of evaporator superheating on the ORC-50 machine's operating parameters and energy performance was experimentally investigated by setting the feed pump speed to vary between 0 and maximum 37 Hz, while varying the cooling fluid inlet temperatures between 15°C and 30°C. By fixing the feed pump speed at maximum 37 Hz, it was found that, with waste heat inlet temperatures higher than 105°C, this maximum speed became insufficient to keep the evaporator superheating at optimum values (*i.e.* 4 to 5°C). In other words, with waste heat entering the ORC-50 machine at temperatures lower than 105°C, the speed of the feed pump varied between 0 and 37 Hz and was able to provide optimum superheating amounts for all cooling fluid inlet temperatures provided. However, with waste heat inlet temperatures higher than 105°C, excessive superheating amounts have been provide because the maximum speed of the feed pump was fixed at 37 Hz. It can be noted that this maximum current frequency wasn't arbitrary chosen, but it was determined after many experimental trials and associated data analysis.

First, it can be seen that with waste heat inlet temperatures above 105°C and feed pump speed fixed at maximum 37 Hz, the pressure of the superheated vapour at the expander inlet port dropped steeply (Fig. 6 *a*), while the expander inlet temperatures began to increase (Fig. 6 *b*). In order to better illustrate these phenomena, in all the running tests represented in Figs. 6 *a* and 6 *b*, the cooling fluid inlet temperature was kept constant at 20°C while the waste heat inlet temperatures varied from 85°C to 115°C by 5°C increments.

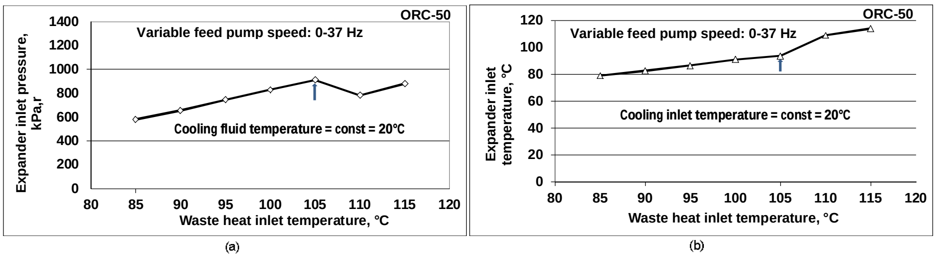

Fig. 6 – Expander inlet pressure (*a*) and temperature (*b*) as functions of waste heat inlet temperatures, at variable feed pump speed (0 to 37 Hz) and fixed cooling fluid inlet temperature (20°C).

Second, by setting the feed pump speed at maximum 37 Hz, the organic fluid flow rate sharply dropped at waste heat inlet temperatures above 105°C for all cooling fluid inlet temperatures (Fig. 7 *a*). On the other hand, the evaporator superheating reached values as high as 14°C to 19°C and 23°C to 25°C with waste heat inlet temperatures of 110°C and 115°C, respectively (Fig. 7 *b*).

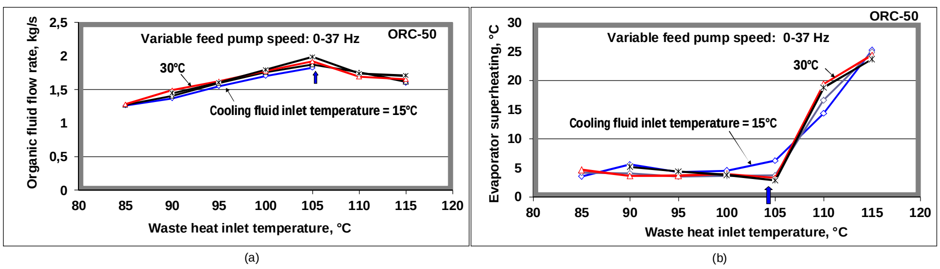

Fig. 7 – Organic fluid flow rate (*a*) and evaporator superheat (*b*) as a function of waste heat inlet temperatures, at variable feed pump speed (0 to 37 Hz) and fixed cooling fluid inlet temperature (15°C).

As a direct consequence of reducing the organic fluid flow rate and of excessively increasing the evaporator vapour superheat, the net power output (Fig. 8 *a*) as well as the heat-to-electricity energetic net conversion efficiency rate (Fig. 8 *b*) both stopped increasing at waste heat inlet temperatures above 105°C, for all cooling fluid inlet temperatures.

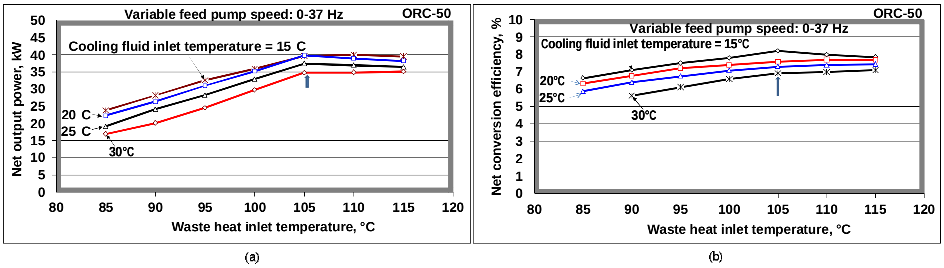

Fig. 8 – Net power output (*a*) and net conversion efficiency rate (*b*) as a function of waste heat inlet temperatures, at a variable feed pump speed (0 to 37 Hz) and cooling fluid inlet temperatures at between 15 and 30°C.

E. Cycle improvements

More or less practical and/or theoretical improvements could be made to ORC machines to increase energy performance and reliability. First of all, by appropriately selecting the working (organic) fluids, based on the machine's actual operating conditions, as well as by using an advanced design and selecting the size of the expander, the net conversion efficiency rate may be increased. The working fluids, such as HFC-134a, HFC-245fa, n-pentane and silicon oils, have relatively high critical points and achieve optimal performance in term of cycle efficiency.

Among several alternatives, screw expanders, developed for relatively low-scale ORC machines (< 250 kWe), have fixed built-in volume ratios and can use wet fluids with limited superheat at the inlet supply. There are also small screw expanders which can manually or automatically vary the volume ratio within a nearby range to improve efficiency. According to Leibowitz *et al*. (2006), by taking full advantage of the potential of screw expanders, it is possible to produce ORC units for heat recovery from low-temperature heat sources with outputs as low as 50 kW at economically viable costs. However, any risks of oil and/or working fluid leakage through the shaft seal must be limited or avoided.

To improve the net energetic or exergetic conversion efficiency rate of ORC machines, various methods have been proposed, such as using cascade evaporators with high and low-pressure expanders, or two-cycle concepts with fluids having different properties. However, such improvements may significantly increase the initial costs of the systems.

In the case of dry organic fluids entering the expander close to their saturated state (*i.*e. with small superheat amounts), the cycle can eventually be improved by integrating a regenerator, a counter-current heat exchanger installed between the expander and the condenser. In this cycle, since the fluid does not reach a two-phase state at the end of the expansion process, its temperature is higher than the condensing temperature and it can thus be used to preheat the liquid before it enters the evaporator. Such a process is intended to reduce the thermal power required from the waste heat source to generate the same electrical power, while lowering the overall irreversibility and increasing the conversion efficiency. By using low-temperature heat sources giving at low expander inlet pressures, the amount of waste heat required could be 7.5% lower and the second-law efficiencies of regenerative ORC machines may be approximately 12% higher than those of basic ORC cycles. However, a significant factor in the decision to use or not a regenerator should be the impact of the additional irreversibility in form of pressure drop introduced before the pre-heater and after the expander.

To improve the efficiency and reliability of ORC machines, a number of practical, simple measures have to be carefully applied. First, if high-temperature refrigerants (*i.e.* refrigerants having relatively high critical temperatures compared to those of conventional ones) are employed as working fluid (*e.g.* HFC-245fa, of which the critical temperature is 154.05°C), inlet temperatures of the cooling fluid below certain values (in this example, bellow about 10°C) must be banned in order to avoid condensing pressures below the atmospheric pressure (vacuum). This situation may eventually allow incondensable gases from the ambient air to enter the system and increase the pressure of the working fluid above its saturation values corresponding to the actual process temperature, whether the machine is running or not. The incondensable gases may also increase the liquid sub-cooling at the condenser outlet as well as the expander outlet pressure and thus reduce the net power output. At lowest waste heat source inlet temperatures (*e.*g. at 85°C), the impact may be even higher. By bleeding the liquid receiver prior to each operating period, the overpressures caused by incondensable gases can be drastically reduced and/or eliminated.

Second, before starting the ORC machine, it would be useful to control both the waste (hot) and cooling (cold) fluid flows through the pre-heater/evaporator and condenser, respectively. This precaution may help manage the organic fluid migration within the heat exchangers and other components (expander, liquid receiver, etc.) in order to provide adequate starting sequences.

Third, the feed pump variable speed as well as the expander vapour by-pass and liquid injection have to be carefully controlled during the starting sequences in order to ensure optimum superheating at the evaporator exit and avoid undesirable on/off cycles. For example, during the starting periods it may be helpful to allow the feed pump to run at a given, fixed speed during a certain period of time prior beginning operating at variable speed.

Fourth, in actual industrial applications, the parasitic power consumptions have to be analysed carefully and, if possible, eliminated or substantially reduced. This approach is sometimes possible because many industrial sites are already equipped with waste heat and cooling fluid circulating pumps, as well as with cooling towers or other similar devices.

Finally, for air-cooled ORC systems, the cooling fluid inlet temperature depends on the ambient temperature and cannot be adjusted easily. The challenge consists in finding combined cooling methods for lowering the condensing temperatures during the hottest periods of the year locally.

5. Potential Applications

The increase in fossil fuel prices, the growing availability of industrial waste heat and of renewable energy (solar, geothermal, biomass, etc.) open a huge market for heat recovery machines based on the ORC cycle, if they could be manufactured and installed at competitive costs (Leibowitz *et al*., 2006). The competitiveness of ORC machines also depends on the price of conventional fuels (*e.g*. natural gas, oil, electricity), as well as on their utilization factor that must be at least of 5,000 h per year.

Applications of ORC machines include heat recovery from industrial processes, internal combustion and reciprocating engines (from both jacket cooling water and combustion gas exhaust), medium size gas turbines, deep geothermal, oil and gas coproduced fluid wells, solar energy sources, biomass boilers, combined heat and power plants, and gas compression stations.

Some large-scale, high-temperature plants as those for aluminum processing lose up to 50% of the energy used for the electrolytic process, about 2/3 of which is lost through the walls of hundreds of furnaces in each plant and 1/3 from the combustion gases. The energy lost through the walls is difficult to recover, but the heat from electrolysis gases at about 100°C,…,130°C is easier to recover. After filtering and purifying the exhaust gases, the waste heat could be recovered and used for power generation, thus enhancing process efficiency and reducing greenhouse emissions. The power generated may avoid the need to purchase extra electricity to increase aluminum production, while the energy intensity of the process can be reduced, and the overall competitiveness, improved.

In most of internal combustion engines, only 30,…,40% of the total fossil thermal heat input is turned into mechanical work, while the remaining 60,…,70% leaves the engine as waste heat, mainly through the jacket water cooling system and the exhaust pipe. By recovering the waste heat from engine liquid coolants and/or from the exhaust gases of the internal combustion engines in order to convert it into electrical power, the overall efficiency of these devices could exceed the present limits.When fuel costs are high and internal combustion engines are used for electrical production, ORC machines will save fuel costs by allowing the engine to operate at a lower fuel input rate for the same electrical output. Biomass, mainly used locally because of its low energy density, and bio-gases, are another renewable energy sources available in many off-grid combined heat and power plants.

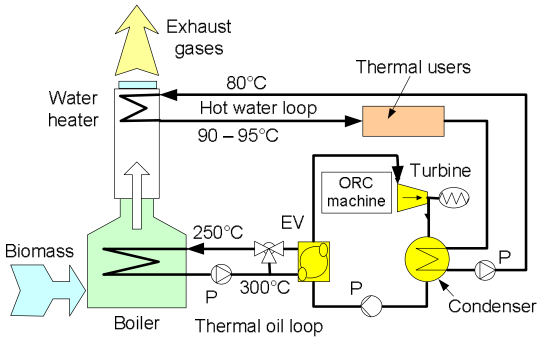

Fig. 10 – Principle of a power and heat generation plant using biomass renewable energy; ORC – Organic Rankine Cycle machine; P – circulating pump.

In the combined power and heat (CPH) generation plant shown in Fig. 10 the biomass boiler is fed by virgin wood chips. Thermal oil (mineral or synthetic), heated at 300°C inside the CHP plant's boiler, flows through a closed loop and is cooled-down to 250°C inside the evaporator EV of the ORC machine. The heat recovered allows generating electrical power through the turbine of the ORC machine.

At full load, the ORC machine's net heat-to-electricity efficiency could attain 18%, and at 50% load, it may drop at 16.5%.

To increase the overall energy efficiency of such a ORC machine, the heat rejected by the condenser at around 80°C could be further heated up to 90,…,95°C by the aid of boiler's exhaust gases and, then, used for industrial (*e.g*. wood drying, greenhouses and fish farms heating, etc.) and/or district heating.

Gas compression stations also generate a lot of waste heat mainly from compressors' inter-cooling heat exchangers. Recovering such a waste heat presents a double benefit in both producing electrical power from the heat generated and reducing the cooling power loads on the existing systems.

Finally, the ORC machines are perfectly suited to exploit low-enthalpy deep-geothermal heat sources at temperatures ranging between 85,…,350°C, and also for converting solar energy into electricity.

6. Conclusions

This paper succinctly presents a number of experimental results obtained with a small-scale beta-prototype 50 kWe ORC machine using HFC-245fa as a working fluid. It converts low-grade waste heat or renewable energy at inlet temperatures ranging between 85°C and 125°C into electricity, using a cooling fluid at inlet temperatures varying from 15°C to 30°C. To achieve superheating amounts lower than 4,...,5°C at the evaporator outlet, the speed of the organic fluid feed pump has been set to theoretically vary between 0 and 60 Hz. Under these thermal operating conditions, the net power output and net heat-to-electricity energetic conversion efficiency rate varied between 22.3 and 39.9 (electric) kW, and from 6.62% to 7.57%, respectively. With waste heat inlet temperatures higher than 115 °C, the net conversion factor may reach 10% or more at 125°C with cooling fluid inlet temperature of 15°C.

The results obtained show that both net power output and energetic conversion efficiency depend on the cooling fluid inlet temperatures, *i.e.* on ambient thermal conditions. For example, at the same waste heat source inlet temperature (ex: 90°C), the expander electrical power output increased by about 28.3% when the cooling fluid inlet temperature dropped from 30°C to 15°C.

In the future, R&D programs are required to identify technological options to recover energy from industrial plants to evaluate their efficiency and costs, and find the most efficient application fields. ORC machines could become energy efficient and environmentally friendly alternatives allowing the industrial plants to reduce their energy consumption, and, thus, become less dependent on the fluctuating prices of primary energy sources.

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REFERENCES

Leibowitz H., Smith I.K., Stosic N., *Cost Effective Small Scale ORC Systems for Power Recovery from Low Grade Heat Sources*. Proc. of IMECE2006, 2006 ASME Internat. Mechan. Engng. Cong. a. Exposition, Chicago, Illinois, USA, November 5-10, 2006.

Maizza V., Maizza A., *Unconventional Working Fluids in Organic Rankine Cycles for Waste Energy Recovery Systems*. Appl. Thermal Engng., *21*, 381-390 (2001).

Minea V., *Power Generation with ORC Machines Using Low-Grade Waste Heat or Geothermal Renewable Energy*. Appl. Thermal Engng., *69*, 143-154 (2014).

Sonntag R.E., Borgnakke C., Van Wylen G.J., *Fundamentals of Thermodynamics*. John Wiley & Sons, 2003.

* * * *ElectraTherm ORC Primer Series 4000 Green Machine*. ElectraTherm, 2012, www.electratherm.com.

* * * *Stricker Associates Inc. Market Study on Waste Heat and Requirements for Cooling and Refrigeration in Canadian Industry* – main report, 2006.

GENERAREA ENERGIEI ELECTRICE CU AJUTORUL MAŞINILOR BAZATE PE CICLUL ORGANIC RANKINE FUNCȚIONÂND LA TEMPERATURI SCĂZUTE

(Rezumat)

Începând de astăzi şi pâna în jurul anului 2050, creşterea consumului global de energie este estimată la 71%. În acelaşi timp, emisiunile de bioxid de carbon asociate consumului de energie ar putea să crească la rândul lor cu mai mult de 40%. În acest context, sursele de energii industriale « pierdute » sau regenerabile ar putea reprezenta alternative putând contribui la reducerea consumului global de energii fosile. Acest articol prezintă principalele caracteristici dimensionale cât şi performanțele unui prototip de laborator bazat pe Ciclul Organic Rankine funcționând cu surse de căldură la temperaturi relativ scăzute. Această maşină termică produce putere electrică de până la 50 kW utilizând surse de caldură de entalpie joasă ca, de exemplu, căldura industrială « pierdută » sub formă de lichid sau gaz, sau energie geotermală la temperaturi variind între 85 şi 125°C. Studiul experimental al prototipului pre-comercial arată că puterea electrică generată, ca şi eficacitatea netă de conversie directă a căldurii recuperate în energie electrică, depind de mai mulți parametri, între care temperatura de intrare atât a căldurii industriale recuperate (sau a fluidului geotermic), cât şi a fluidului de răcire disponibil, ca şi de strategia de control a maşinii termice în condiții variabile de funcționare.