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Technological and Economical Survey of Organic Rankine Cycle Systems

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Abstract

This paper presents an overview of current R&D in the field of small and middle scale Organic Rankine Cycles (ORC's). Major ORC's applications are described and their technical and economical maturity is analyzed. The paper also emphasizes the selection criteria for the expander and for the working fluid.

Keywords: Organic Rankine cycles, heat recovery, biomass, combined heat power, Solar ORC, fluid selection, ORC expanders

1. INTRODUCTION

The interest for low grade heat recovery has been growing for the last ten years, due to the increasing concern over energy shortage and global warming.

An important number of new solutions have been proposed to generate electricity from low temperature heat sources. Among the proposed solutions, the Organic Rankine Cycle (ORC) system is the most widely used. This system involves the same components as in a conventional steam power plant (a boiler, a work-producing expansion device, a condenser and a pump). However, for such a system, the working fluid is an organic component characterized by a lower ebullition temperature than water and allowing reduced evaporating temperatures.

The success of the ORC technology can be partly explained by its modular feature: a similar ORC system can be used, with little modifications, in conjunction with various heat sources. This success was reinforced by the high technological maturity of most of its components, due to their extensive use in refrigeration applications. Moreover, unlike with conventional power cycles, local and small scale power generation is made possible by this technology.

Today, Organic Rankine Cycles are commercially available in the MW power range. However very few solutions are actually suitable for the kW scale. This paper presents an overview of current R&D in the field of small-scale ORC's. Major small-scale ORC's applications are analyzed and their technical and economical maturity is discussed.

2. SMALL AND MIDDLE SCALE ORC TECHNOLOGY APPLICATIONS

2.1. Biomass combined heat and power

Biomass is widely available in a number of agricultural of industrial processes such as agricultural waste or wood industry. It is best used locally for two main reasons : the energy density of biomass is low, which increases transportation costs; and heat and electricity demand are usually available on-site, which makes a biomass plant particularly suitable in the case of off-grid or unreliable grid connection. Local generation leads to smaller scale power plants (<1 MWe) which excludes traditional steam cycles that are not cost-effective in this power range.

The ORC presents several advantages over the traditional steam cycle:

- The boiler operates at a lower temperature and at a lower pressure since it only heats up thermal oil at a temperature of about 300°C and at low pressure. Steam boilers on the contrary need to superheat the steam up to a temperature higher than 450°C in order to avoid droplets formation during the expansion. The pressure of about 60 to 70 bars and the thermal stresses increase dramatically the complexity and the cost of the steam boiler in comparison with a thermal oil boiler.
- The ORC has a lower operating pressure than the steam cycle. This decreases the installation cost and the management of the installation with respect to security standards.

The efficiency of power generation with ORC's is lower than that of traditional steam cycles, and generally decreases with the plant size. Heat demand is therefore a prerequisite to increase the overall plant energy conversion efficiency. This heat demand can be fulfilled by industrial processes (such as wood drying) or space heating. Plant load can be controlled either by the on-site heat demand, or by maximizing power generation. The second solution involves wasting the additional heat but has the advantage of increasing the annual full load operating hours.

A second drawback of the ORC over the steam cycle is the higher flue gases temperature at the exhaust of the boiler: the thermal oil supply temperature is indeed about 300°C, while the temperature of the feed water of a steam cycle is about 100°C. With flue gases leaving at a higher temperature, the boiler efficiency is decreased. This can be addressed by adding additional heat exchangers to preheat the ORC working fluid and the combustion air, as shown in Figure 1.

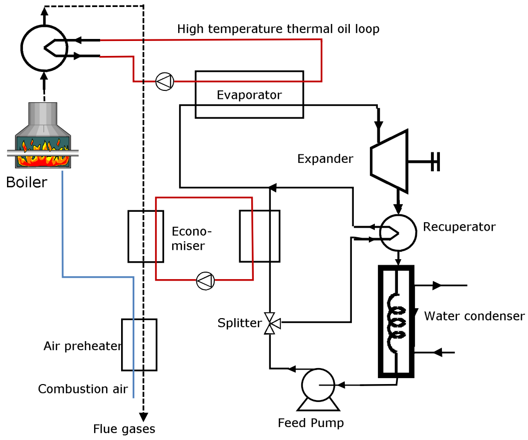

Figure 1: Schematic representation of an ORC CHP system

Concurrent technologies for electricity generation from solid biofuels also include gasification: biomass is transformed into a synthetic gas composed mainly of H_2, CO, CO_2, CH_4. This synthetic gas also contains solid particles and needs to be treated and filtered before being burned in an internal combustion engine or in a gas turbine.
Rentzelas et Al (2008) compared the technology and the costs of Biomass CHP using an ORC or using gasification. They showed that gasification yields higher investment costs (about 75%) and higher operation and maintenance costs (about 200%). On the other hand, gasification shows a higher power-to-thermal ratio, which makes its exploitation more profitable. It should also be noted that ORC is a well-proven technology, while gasification plants actually in operation are mostly prototypes of for demonstration purpose.

2.2. Solar power plant

Concentrating solar power is a well-proven technology: the sun is tracked and reflected on a linear or on a punctual collector, transferring heat to a fluid at high temperature. The heat is then transferred to a power cycle generating electricity. The three main concentrating technologies are the parabolic dish, the solar tower, and the parabolic though. Parabolic dishes and solar towers are punctual concentration technologies, leading to a higher concentration factor and to higher temperature. The best suited power cycles for these technologies are the Stirling engine (small-scale plants), the steam cycle, or even the combined cycle, for solar towers.

Parabolic troughs work at a lower temperature (300°C to 400°C). Up to now, they were mainly coupled to the traditional steam Rankine cycle for power generation (see for example Steinhagen, 2004). The same limitation as in geothermal or biomass power plants remains: steam cycles need high temperature, high pressures, and therefore high installed power in order to be profitable.

Organic Rankine cycles seem to be a promising technology in a view to decrease plant size and investment costs. They can work at lower temperatures, and the total installed power can be reduced down to the kW scale. Technologies such as Fresnel linear concentrators (Ford, 2008) are particularly suitable for solar ORC's since they require lower investment cost, but work at a lower temperature.

Up to now, very few CSP plants using ORC are available on the market:

- A 1MWe CSP plant working with ORC was completed in 2006 in Arizona. The ORC module was provided by ORMAT, uses n-pentane as the working fluid and shows an efficiency of 20 %. The global solar to electricity efficiency is 12.1% on the design point (Canada, 2004).
- A 250 KWe prototype plant was built in Germany in 2005 by GMK. It should be noted that, although this plant aims at simulating a solar system, the heat source was provided by a natural gas boiler. The electrical efficiency of the system was about 15%.
- Some very small-scale systems are actually studied for remote off-grid applications. Figure 2 shows a 1 KWe system installed in Lesotho by the "Solar Turbine Group" for rural electrification. The goal of this project is to develop and implement a small scale solar thermal technology utilizing medium temperature collectors and an ORC to achieve economics analogous to large-scale solar thermal installations. This configuration aims at replacing or supplementing Diesel generators in off-grid areas of developing countries, by generating clean power at a lower levelized cost (~$0.12/kWh compared to ~$0.30/kWh for Diesel) (Quoilin *et Al.*, 2008).

Figure 2 Solar ORC in field trials in Lesotho, southern Africa 2007 (Solar Turbine Group)

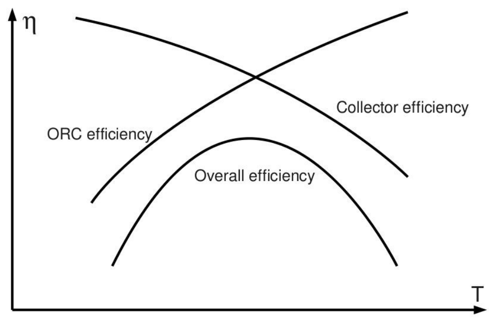

Figure 3 Tradeoff between collector and ORC efficiency

It is worthwhile to note that the choice of the temperature in the collector results of a tradeoff between collector efficiency and ORC efficiency (Figure 3): Increasing the temperature will lead to higher collector ambient heat losses, but also to a higher conversion efficiency.

2.3. Heat recovery on mechanical equipments and industry processes

Many applications in industry reject heat at relatively low temperature. This heat can be converted into heat sources for other on-site applications, or used for space heating (e.g. district heating). For instance, Engin et al. (2004) demonstrated through a case study that 40% of the heat used in cement industry was lost in flue gases, whose temperature varies between 215 and 315 °C. For economical reasons (Hung, 2001), traditional steam cycles wouldn't allow recovering heat in this range of temperatures. A huge potential market is therefore available for the ORC technology in this application field.

2.4. Geothermal energy

The range of temperatures of geothermal heat sources is large. Lowest possible temperature for ORC heat recovery is about 100 °C, while other ORC geothermal plants work at a temperature higher than 200°C.

Higher temperature (>150°C) geothermal heat sources enable combined heat and power generation: the condensing temperature is set to a higher temperature (e.g. 60°C), allowing the cooling water to be used for space heating. The global energy recovery efficiency is therefore increased, at the expense of the electrical efficiency.

2.5. Heat recovery on internal combustion engines

An Internal Combustion Engine only converts roughly one third of the fuel energy into mechanical power. For instance, for a typical 1.4 liter Spark Ignition ICE, with a thermal efficiency ranging from 15 to 32%, 1.7 to 45 kW are released through the radiator (at a temperature close to 80 - 100°C) and 4.6 to 120 kW through the exhaust gas (400 - 900°C) (El Chammas and Clodic, 2005).

The heat recovery Rankine cycle system is an efficient means for recovering heat (in comparison with other technologies such as thermo-electricity and absorption cycle air-conditioning). The idea of associating a Rankine cycle to an ICE is not new and the first technical developments followed the 70's energy crisis. For instance, Mack Trucks (Doyle and Patel, 1976) designed and built a prototype of such a system operating on the exhaust gas of a 288 HP truck engine. A 450 km on-road test demonstrated the technical feasibility of the system and its economical interest: an improvement of 12.5% of the fuel consumption was achieved. Systems developed today differ from those of the 70's because of the advances in the development of expansion devices and the broader choice of working fluids. The literature survey indicated that, at the present time, Rankine cycle systems are under development, but no commercial solution seems to be available yet.

Most of the systems recover heat from the exhaust gas (Endo et al., 2007; Nelson 2008) and, in addition from the cooling circuit (Freymann et al., 2008). By contrast, the system developed by Oomori and Ogino (1993) only recovers heat from the cooling circuit.

The control of the system is particularly complex due to the (often) transient regime of the heat source. However, optimizing the control is crucial to improve the performance of the system. For instance, Honda (Endo et al., 2007) proposed to control the temperature by varying the water flow rate through the evaporator (by varying the pump speed) and to control the expander supply pressure by varying its rotational speed

Performance of the recently developed prototypes of Rankine cycles is promising. For instance, the system designed by Honda (Endo et al., 2007) showed a maximum cycle thermal efficiency of 13%. At 100 km/h, this yields a cycle output of 2.5 kW (for an engine output of 19.2 kW). This represents an increase of the thermal efficiency of the engine from 28.9% to 32.7%.

3. ORC MANUFACTURERS AND MARKET EVOLUTION

ORC manufacturers have been present on the market since the beginning of the 80's. They provide ORC solutions in a broad range of power and temperature levels, as shown in Table 1.

Table 1 Non-exhaustive list of the main ORC manufacturers

Manufacturer	Applications	Power range	Heat source temperature	Technology
ORMAT, US	Geothermal, WHR, solar	200 KWe – 72 MWe	150° - 300°C	Fluid : n-pentane
Turboden, Italy	CHP, geothermal	200 KWe – 2 MWe	100 - 300°C	Fluids : OMTS, Solkatherm Axial turbines
Adoratec, Germany	CHP	315 – 1600 KWe	300°C	Fluid: OMTS
GMK, Germany	WHR, Geothermal, CHP	50 KWe – 2 MWe	120° - 350°C	3000 rpm Multi-stage axial turbines (KKK) Fluid: GL160 (GMK patented)
Koehler-Ziegler, Germany	CHP	70 – 200 KWe	150 – 270°C	Fluid: Hydrocarbons Screw expander
UTC, US	WHR, geothermal	280 KWe	>93°C	
Cryostar	WHR, Geothermal	n/a	100 – 400 °C	Radial inflow turbine Fluids: R245fa, R134a
Freepower, UK	WHR	6 KWe - 120 KW	180 - 225 °C	
Tri-o-gen, Netherlands	WHR	160 kWe	>350°C	Turbo-expander
Electratherm, US	WHR	50 KWe	>93°C	Twin screw expander
Infinity Turbine	WHR	250 KWe	>80°C	Fluid: R134a Radial Turboexpander

Sources: Manufacturers websites; Citrin, 2005; Gaia, 2006; Lorenz, 2006; Holdmann, 2007; Schuster, 2009

The market for ORC's is growing at a rapid pace. Since the first installed commercial ORC plants in the 80's, an exponential growth has been stated. Figure 4 shows for instance the evolution of the installed power and of the number of plants in operation, based on a compilation of manufacturer data over the internet.

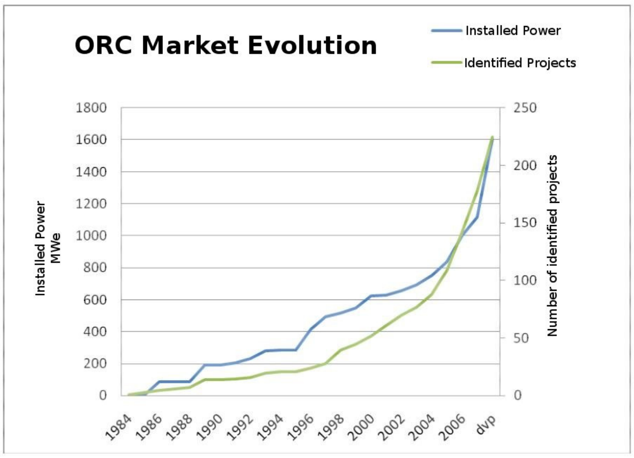

Figure 4 ORC market evolution (data source : Enertime, 2009)

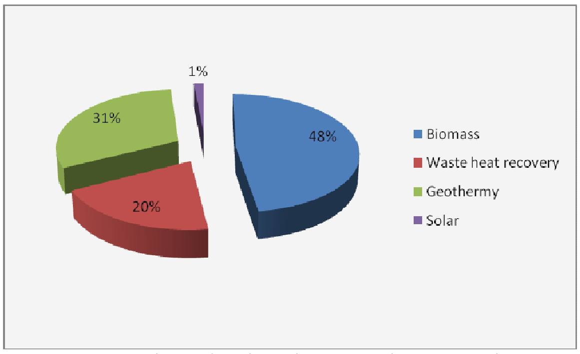

Figure 5 Share of each application in the ORC market

Figure 5 also shows that the ORC is a mature technology for waste heat recovery, biomass CHP and geothermy but is still very uncommon for solar applications. Moreover, as stated in Table 1, those technologies are mainly developed in the MW power range and very few ORC plants are available in the KW power range.

4. SELECTION OF THE WORKING FLUID

The choice of the working fluid for a given application is a key-issue and has been treated in numerous studies. Some general relevant characteristics can be extracted from those studies:

1. Thermodynamic performance: the efficiency and/or output power should be as high as possible for the given heat source and heat sink temperatures. This generally involves low pump consumption and high critical point.
2. Positive or isentropic saturation vapor curve. A negative saturation vapor curve (“Wet” fluid) leads to droplets at the end of the expansion (Quoilin, 2007). The vapor must therefore be superheated at the turbine inlet in order to avoid turbine damages, which decreases cycle performance (Yamamoto et al., 2001). In the case of positive saturation vapor curve (“Dry” fluid), a recuperator can be used in order to increase cycle efficiency.
3. High vapor density: this parameter is of key importance, especially for fluids showing a very low condensing pressure (e.g. silicon oils). A low density leads to very large equipments at the expander and condenser level.
4. Acceptable pressures: as already stated with water, high pressures usually lead to higher investment costs and increasing complexity.
5. High stability temperature: unlike with water, organic fluids usually suffer from chemical deteriorations and decomposition at high temperatures. The maximum heat source temperature is therefore limited by the chemical stability of the working fluid.
6. Low environmental impact and high safety level: the main parameters to take into account are the Ozone Depleting Potential (ODP), the Greenhouse Warming Potential (GWP), the toxicity and the flammability.
7. Good availability and low cost

Some of the main representative papers dealing with the selection of the working fluid for ORC applications are given in Table 2. Fluids are typically compared by fixing the evaporating and condensing temperatures (with respects to the nature of the heat source and sink).

Table 2 Summary of different comparisons of working fluids

Author(s)	Application	Cond. Temp.	Evap. Temp.	Considered fluids	Recommended fluids (in terms of efficiency and/or power)
Saleh et al.	Geothermal	30°C	100°C	alkanes, fluorinated alkanes, ethers and fluorinated ethers	RE134, RE245, R600, R245fa, R245ca, R601
Maizza and Maizza (2001)	n/a	35 – 60°C	80-110	Unconventional working fluids	R123, R124
Liu et al. (2004)	Waste heat recovery	30°C	150 – 200°C	R123, iso-pentane, HFE7100, Benzene Toluene, p-xylene	Benzene, Toluene, R123
El Chammas and Clodic (2005)	ICE	55°C (100°C for water)	60 - 150°C (150 – 260°C for water)	Water, R123, isopentane, R245ca, R245fa, butane, isobutene and R-152a	Water, R245-ca and isopentane
Drescher and Bruggemann (2007)	Biomass CHP	90°C *	250 - 350°C*	ButylBenzene, Propylbenzene, Ethylbenzene, Toluene, OMTS	AlkylBenzenes
Hettiarachchia et al. (2007)	Geothermal	30°C*	70 – 90°C	Ammonia, n-Pentane, R123, PF5050	Ammonia
Lemort et al. (2007)	Waste heat recovery	35°C	60 – 100°C	R245fa, R123, R134a, n-pentane	R123, n-pentane
Hettiarachchia et al. (2007)	Geothermal	30°C*	70 – 90°C	Ammonia, n-Pentane, R123, PF5050	Ammonia
Lemort et al. (2007)	Waste heat recovery	35°C	60 – 100°C	R245fa, R123, R134a, n-pentane	R123, n-pentane
Borsukiewicz-Gozdur and Nowak (2007)	Geothermal	25°C	80 – 115°C	propylene, R227ea, RC318, R236fa, ibutane, R245fa	Propylene, R227ea, R245fa
Fankam et al. (2009)	Solar	35°C	60 – 100°C	Refrigerants	R152a, R600, R290

** Max/min temperature of the heat source/sink instead of evaporating or condensing temperature*

For the particular case of ICE heat recovery applications, the selection of the working fluid is strongly correlated to the choice of the heat source(s). Honda (Endo et al., 2007) selected water as the working fluid of their prototype of Rankine cycle system. The prototype developed by BMW (Freymann et al., 2008) is based on two cycles: the first recovers heat from the exhaust gas and uses water and the second is associated to the cooling circuit and uses ethanol. The system proposed by Cummins (Nelson, 2008) recovers heat from both the exhaust gas and the Exhaust Gas Recirculation (EGR) and used R245fa.

4.1. Temperature profile

The temperature profiles of the heat source and of the heat sink are an essential parameter to take into account when optimizing the performance of an ORC. This is illustrated hereunder on the basis of a simplified simulation model of an ORC, built on the following assumptions:

- Turbine efficiency $\varepsilon_{s,exp} = (h_{su,exp} - h_{ex,exp}) / (h_{su,exp} - h_{ex,exp,s})$ is set to 0.75
- Pump efficiency $\varepsilon_{s,pp} = v \cdot (P_{ex,pp} - P_{su,pp}) / (h_{ex,pp} - h_{su,pp})$ is set to 0.80
- Recuperator effectiveness is set to 0.8
- Temperature pinch points are set to 10 K at both the condenser and the evaporator.
- The heat source and heat sink temperature profiles are evaluated by the temperature differences between supply and exhaust ($\Delta T_{ev} = T_{hf,su,ev} - T_{hf,ex,ev}$ and $\Delta T_{cd} = T_{cf,ex,cd} - T_{cf,su,cd}$).

For the purpose of the modeling, a heat source consisting of hot air at a temperature of 130°C and characterized by a flow rate of 15 kg/s is defined. The heat sink is also assumed to be air, whose supply temperature is 10°C, and flow rate is adapted to maintain the imposed temperature pinch point. The considered working fluid is R245fa. The superheating at the evaporator exhaust is set to 10K, and the subcooling at the condenser exhaust is set to 5K.

Figure 6 shows the T-s diagram of the cycle in three different cases: Case *A* corresponds to a small temperature glide in the evaporator, with a recuperator in the cycle, which is typical of a CHP, or solar plant. Case *B* corresponds to a high temperature glide in the evaporator, without recuperator, which is typical of waster heat recovery application. Case *C* corresponds to a high temperature glide, but using a recuperator.

In case *A*, the heat capacity flow rate in the heat exchangers is high. This allows high and low evaporating and condensing pressures respectively. Increasing the pressure ratio leads to a higher efficiency. Case *B* is typical of a waste heat recovery ORC: the temperature glide of the heat source is very important, since the goal is to recover as much heat as possible from the heat stream. The pinch point limitation leads to a lower evaporating pressure and thus to a lower cycle efficiency (7.8% instead of 12.5%), but the amount of heat recovered is higher and the output power is increased (71 KW instead of 28 KW).

This limitation highlights the necessity of a pinch point analysis for a given application. In general, in heat recovery applications, the objective will be to maximize the output power rather than the efficiency. In contrary, in solar or biomass applications, the heat source can be almost constant and maximizing the output power is therefore similar to maximizing the efficiency,

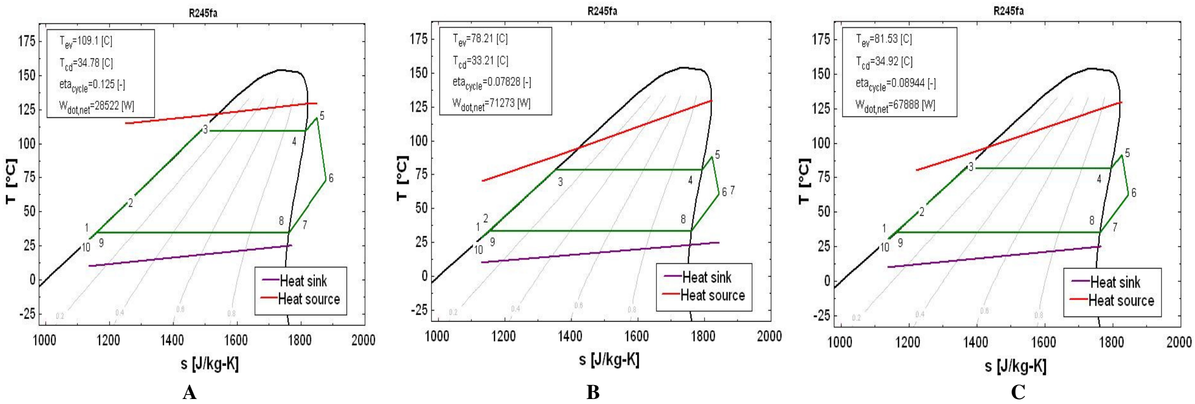

Figure 6 T-s diagram of the cycle with superposition of the secondary fluids temperature profiles

For the same reason, the benefits of a recuperator in the cycle will depend on the considered application.

Comparison of cases *B* and *C* indicates that the recuperator increases the cycle efficiency (from 7.8% to 8.9%) but decreases the output power (from 71 KW down to 68 KW). This is explained by a higher heat source temperature at the exhaust of the evaporator, which reduces the amount of heat recovered, and the output power.

In summary, in heat recovery applications, the output power, and not the efficiency should be maximized, and a recuperator will generally decrease the performance.

4.2. Comparison between working fluids

This section aims at comparing the most commonly used working fluids for three typical applications:

- The first application corresponds to an evaporating temperature of 85°C and to a condensing temperature of 20°C. These temperature levels are typical of a geothermal application.
- The second application corresponds to an evaporating temperature of 150°C and to a condensing temperature of 30°C, which could correspond to a low-temperature solar collector.
- The third application corresponds to an evaporating temperature of 280°C and to a condensing temperature of 100°C, which is typical of biomass CHP plant.

Four different fluids are considered because they seem to be the most common in ORC applications (see tables 1 and 2): R134a, R245fa, n-pentane, and silicon oil. The selected silicon oil is octamethylcyclotetrasiloxane ('D4') and its thermodynamic properties are calculated according to Colonna's Multiparameter Equation of State (Colonna, 2004).

The simplified model introduced in section 4.1 with its parameters is used (still assuming 80% effectiveness for the recuperator). A new performance indicator, the back work ratio (BWR) is defined as the ratio between the works consumed by the pump and produced by the expander. The density at the exhaust of the expander is also considered in order to evaluate the required size of the equipment.

Table 3 gives the cycle performance as a function of the fluid and of the considered application. For high temperatures, the only computed fluid is D4, the other ones being in supercritical state.

Table 3 Cycle performance for 3 different applications

	Fluid	P_{ev} [bar]	P_{cd} [bar]	η_{cycle}	BWR	$\rho_{ex,exp}$ [kg/m³]
T_{cd} = 20°C T_{ev} = 85°C	R134a	29.28	5.73969	10.6%	10.8%	26.2
	R245fa	8.92	1.28839	11.7%	2.9%	6.775
	n-pentane	4.16	0.62557	11.5%	1.6%	1.803
	D4	0.04541	0.0009533	10.3%	0.0%	0.007966
T_{cd} = 30°C T_{ev} = 150°C	R245fa	33.79	1.80767	16.4%	8.0%	8.598
	n-pentane	15.91	0.84297	18.1%	3.9%	2.055
	D4	0.50238	0.001985	15.6%	0.1%	0.01437
T_{cd} = 100°C T_{ev} = 280 °C	D4	8.04243	0.08718	18.6%	2.2%	0.483

Table 3 indicates that R134a and R245fa have a good comparative performance at low temperature levels. They also show the highest back work ratio. N-pentane shows good performance for the second case but with a lower vapor density than R134a and R245fa. The low density becomes prejudicial for the Silicon Oil at low temperature: it is for example 61 times lower than the density of R245fa at a condensing temperature of 30°C, which would lead to oversized expander and condenser.

5. EXPANDER

Performance of the ORC system strongly correlates with those of the expander. The choice of the machine strongly depends on the operating conditions and on the size of the system. Two main types of machines can be distinguished: the turbo and positive displacement types. Similarly to refrigeration applications (*Figure 7*), displacement type machines are more appropriate to the small-scale ORC units, because they are characterized by lower flow rates, higher pressure ratios and much lower rotational speeds than turbo-machines (Persson, 1990).

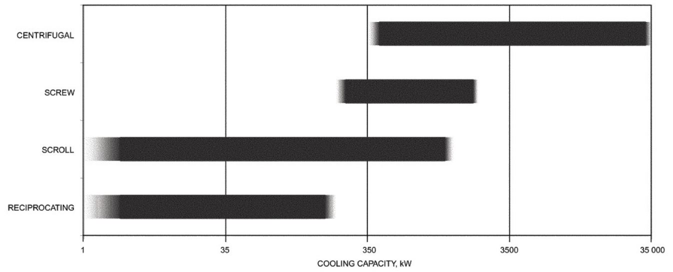

Figure 7: Approximate range of chiller cooling capacity range by compressor type (ASHRAE, 2008)

While technically mature turbomachines are available on the market for large ORC units, it seems that almost all positive displacement expanders that have been used up to now are prototypes, often derived from existing compressors (Zanelli and Favrat, 1994; Yanagisawa et al., 2001; Aoun and Clodic, 2008; Lemort et al., 2008).

Most of the Rankine cycle systems used in automotive applications employ positive displacement machines. One exception is the Rankine cycle system proposed by Cummins (Nelson, 2008), which is associated to a truck engine and uses a turbo machine. Toyota (Oomori and Ogino, 1993) used a scroll expander. BMW (Freymann et al., 2008) initially used 2 vane expanders (one for both cycle) but decided to replace them by axial piston machines showing efficiencies of 55% (which could probably be improved, given that the prototype is non-optimized). Honda (Endo et al., 2007) designed a compact swash plate axial piston type expander, comprising an oil gear pump and a generator motor mounted coaxially with the expander. Steam Rankine cycle recovering heat from high temperature gas, should operate at high evaporating pressure to improve the cycle performance. Most of the existing expander technologies can withstand pressures up to 35 bar (El Chammas and Clodic, 2005). One exception is the swash-plate piston expander developed by Endo et al. (2007) that operates with pressures up to 100 bars.

Another difficulty associated with the use of a positive displacement machine is its lubrication. An oil separator could be installed at the expander exhaust. Unlike with compressors, an oil pump is necessary to drive the separated oil back to the expander suction. Kane (2002) proposed to let the oil traveling with the refrigerant through the system and to install an oil separator at the evaporator exhaust. Separated oil is injected into the scroll expander bearings, while the lubrication of the two spirals relies on the slight inefficiency of the separator. Alternatively, oil-free machines could be used, but generally show lower volumetric performance due to larger tolerances between moving parts (Yanagisawa et al., 2001; Peterson et al., 2008).

In some operating conditions (wet fluids with limited superheat at the expander supply), liquid may appear at the end of the expansion. This could be a threat of damage for piston (reciprocating) expanders but not for scroll and screw expanders, since the latter do not use valves (the timing of suction and discharge is determined by the machine geometry).

Some types of expanders (scroll, screw, vanes) are characterized by a fixed built-in volume ratio. To optimize the performance of the expander, this built-in volume ratio should match the operating conditions (in order to limit under-expansion and over-expansion losses). Volume expansion ratios achieved in Rankine cycle systems are typically larger than those achieved in vapor compression refrigeration systems, which justifies developing adapted design of such expanders rather than retrofitting existing compressors. Generally speaking, piston expanders are more appropriate for the applications with large expansion volume ratio.

Finally, previous studies carried out by the authors also indicated that the tightness of the machine (in case of open-drive expanders) is an important issue (Quoilin, 2007).

6. CONCLUSIONS

A review of ORC applications has been carried out, with a special focus on the temperature levels and on the specificities of each application. The main manufacturers are listed, describing their activity field, the main technological characteristics of their ORC solutions, and their power range. Concurrent technologies of the ORC include gasification and the water steam power cycle. Advantages and drawbacks of each technology were described.

The ORC market is growing exponentially since the beginning of the 80's, mainly in the fields of biomass CHP, geothermal energy and waste heat recovery. The compilation of the available market data shows that actual plants size is mainly limited to the MW scale.

The review of the working fluids pointed out the most widely used working fluids, i.e. R134a, R245fa, n-pentane and silicon oils. The thermodynamic study showed that each fluid is characterized by an optimal temperature range in terms of cycle efficiency and density. In general, the higher the critical point, the higher the optimal temperature range.

Expanders are a key issue in ORC's. Positive displacement machines are preferably used for small-scale applications. At the present time, most of the employed positive displacement expanders are obtained by modifying existing compressors. Turbomachines are mainly designed for larger-scale applications and show a higher degree of technical maturity.

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