13th IEA Heat Pump Conference
April 26-29, 2021 Jeju, Korea

Performance analysis of a vapor-extraction heat-pump system for district heating/cooling applications

Malick Kane[a*], Aziz Mbaye[b]

[a]Laboratory for Thermal Energy (LTE), University of Applied Sciences and Arts of Western Switzerland Boulevard de Pérolles 80, CH-1700, (Fribourg) Switzerland
[b]He-Tech Suisse, Route du Jura 37B – CP 304, CH-1701, (Fribourg) Switzerland,

Abstract

This paper describes a new concept of a high temperature (HT)/low-temperature (LT) network integration by using an advanced heat pump technology able to integrate and manage various renewable and residual energy sources with different temperature levels. It is based on a vapor extraction (VE) compression cycle, which integrates two well-known cycles: a vapor compression heat pump cycle and an organic Rankine cycle (ORC). To perform a comparison with existing heat pumps, an energy/exergy analysis has been carried out by using R134a as a working fluid. Thus, for a given extraction ratio of 25%, results show that an increased heating/cooling capacity of about 30% and a high exergy efficiency of 59.7% can be achieved, compared to the 55.5% obtained for conventional compression heat pumps. A global coefficient of performance (COP) between 2.5 and 3.5 (corresponding to a vapor extraction ratio between 10% and 30%) is also obtained and compared to the 1.3-1.7 values given in the literature for existing thermally driven heat pumps (e.g. absorption heat pumps or double orc-orc heat-pump cycles) working with high temperature levels of hot sources.

Keywords: **:** *District Heating/cooling Network;, Heat pumps; Organic Rankine Cycle; Vapor compression Cycle; Thermally driven heat pump cycle; Vapor extraction compression heat pump*

1. Introduction

Heating and cooling represent around 40-50% of the total end-use energy consumption in Switzerland [1]. In their vast majorities, useful heating and hot water preparation demand in Buildings are supplied by using fossil-fuel centralized energy conversion systems like boiler or combined heat and power units (CHP) driven district heating (DH) networks [2]. These existing types of networks operate in general with relatively low thermal efficiencies related to high operating temperature levels (e.g. around 80-200°C). As an alternative or complement to pollutant control technologies to mitigate the emissions of fossil-fuel-based district heating networks, consumers, public interest groups and some local and state governments advocates a shift away from polluting energy systems to low-grade district heating/cooling systems using residual and/or renewable sources like geothermal, solar thermal, heat-pumps or waste heat [3-5].

* Corresponding author. Tel.: +41 (0)26 429 68 42.
E-mail address: malick.kane@hefr.ch.

Depending on the type of sources, such low-grade district heating/cooling networks operate at temperature levels between 10 and 35°C and use generally decentralized heat pump (HP) units to raise the temperature in buildings for heating and domestic hot water production [6-7]. These advanced networks have the advantage of being able to serve both for heating (up to 35°C in winter) and cooling (15°C in summer) if their temperature levels could be alternately adapted between seasons [9]. However, the time dependence of temperature levels of these renewable energy sources and the increasing numbers of small installations providing both heating or cooling to connected buildings will considerably affect the temperature levels of the network and therefore influence both the performance of thermal networks and decentralized heat pumps units. The high differential temperature between the DH network and the supply temperature level for heating and hot water process in the building corresponds to an important consumption of electricity in the decentralized HP units. The control of these temperature levels can be easily done in the networks by using existing HT district heating networks. Such integration of HT/LT networks can be realized by using an intermediate station unit (or Cluster). The Cluster station system could be represented by a simple heat exchanger to separate HT and LT circuits or an energy conversion system based organic Rankine cycle (ORC) cogeneration units or thermally driven heat pump (THP) cycles [10]. Figure 1 shows an application of thermally driven heat pump cycles using a HT network system (temperature between 80-280°C) to supply heat to a low-grade district heating/cooling network (20-35°C). The cold source of the THP could be renewable like solar thermal or geothermal or residual heat. A new concept of THP system able to integrate HT/LT networks with better efficiency (or COP-coefficient of performance) and can be operated with hot sources at relatively low temperature levels (even below 50°C) is the object of this paper (Figure 3).

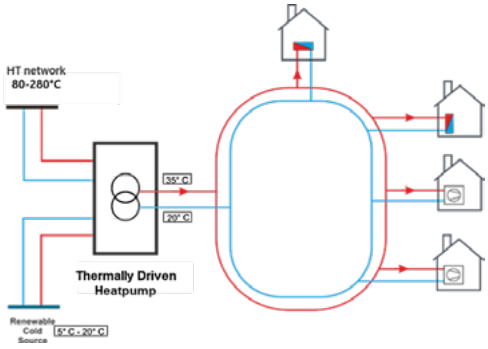


Fig. 1. An Integrated DH network application with thermally driven heat pumps

2. Existing thermally driven heat pump systems

Existing THP technologies such as absorption heat-pumps (single-effect and double-effect) [11,12], ORC driven heat pumps (ORC-ORC) [13] and ejector refrigeration technologies [14] operate in general with high temperature level are in general limited in term of capacity of heating and cooling, and efficiency (coefficient of performance). Figure 2 shows the coefficient of performance (COP) of a broad range of thermally driven heat pumps operating together in the range of hot sources with high temperature levels (between 95 and 275°C) which are more difficult to achieve using residual or renewable energy sources.

A global coefficient of performance between 1.3 and 1.7 have been demontrated for some existing commercial or experimental prototype units working with hot sources at relatively high temperature levels. We can indeed mention a) the gas fired absorption chiller units, commercialized by the company ROBUR S.A and operating at 150°C on the generator with a COP of 1.7 [15] and b) the various solar thermal prototype units based ORC driven heat pumps, demonstrated by several authors. So based on experimental studies on the ORC driven heat pumps, Prigmore and Baraber [16] showed that a COP of 1.5 can be reached with a hot source temperature of 95°C. Strong [17] worked on an ORC driven heat pump that can operates at, with a relatively high temperature level (277°C) with a COP of 1.43. He demonstrated a COP of 1.17 at 229°C. Another experimental study carried out by Biancardi et al. [18] specified a COP of 1.6 for a generator operating at 145°C. Wang et al. [13] have tested an ORC driven machine that produce a COP of 1.5 for a high temperature of 190°C.

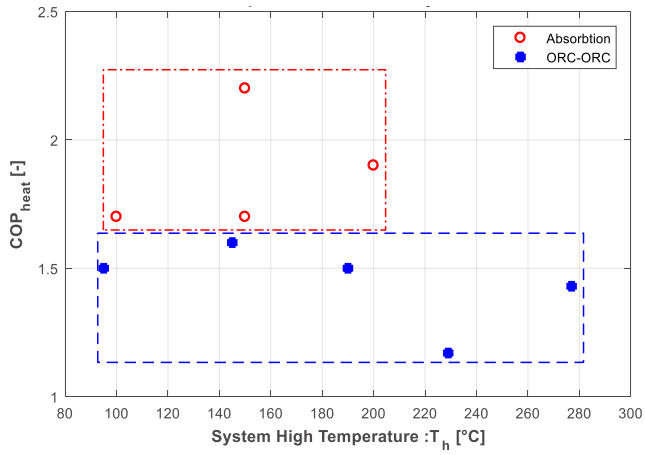


Fig. 2. Coefficient of performance of some existing and theoretical THP based absorption and orc-orc systems given in the literature

Besides those experimental investigations, some other theroretical studies have been performed mainly for absorption cycles. Grossmann [19] studied the performances of both single-effect and double-effect absorption machines for heating applications. His results showed that the single-effect absorption heat pump operating at a temperature level of 100°C can achieve a COP of 1.7 while the double-effect machine delivers a COP of 2.2 at 150°C. Critoph [20] investigated theoretically a forced convection adsorption cycle, which produced a COP of 1.9 at 200°C.

In addition to their low efficiencies and capacity for heating and cooling, such thermally driven heat pumps are limited in term of power density because of the size and the increasing number of components used in the cycle. The new ORC-ORC solution of thermally driven heat-pumps using high speed centrifugal compressor directly coupled with a radial turbine [21] are promising but needs important further developments to demonstrated efficiency, reliability and the ability to work with low temperature heat sources. Theoretical and experimental investigations have been done recently for space heating and hot water preparation in small residential building by using a gas fired boiler. A coefficient of performance of 1.3 has been demonstrated for a differential temperature of 20°C between the evaporator and the condenser and with supercritical thermodynamic cycle using R134a as a working fluid [22].

As the cost-competitiveness of existing thermally driven heat pumps is not yet feasible for applications of low temperature hot sources (below 50°C) because of the costs, efficiency and size of the units, renewable applications will shift towards increasing numbers of vapor compression heat pump technologies mainly adapted for low temperature heating and cooling [6–7]. The solution described in this paper is an original concept of "hybrid drive heat pump systems", developed by the company He-Tech. The hybrid heat pump uses both a primary heat and electricity to run a vapor compression cycle with intermediate pressure extraction for both providing heat from the condenser and recovering heat from the generator. It is based on two well-known cycles: a vapor compression heat pump cycle and an organic Rankine cycle. The system offers more flexibility to integrate and manage different renewable and residual energy sources (such as solar thermal, biomass, geothermal and waste heat) and ability to work with low-temperature heat sources (even below 50°C). One object of this present paper is to evaluate the performances (capacity for heating and cooling, coefficient of performance and exergy efficiency) of such a vapor extraction compression cycle concept and compare results to those of existing heat pump cycles.

3. Vapor-extraction heat pump cycle technology

The proposed new solution of heating/cooling is based on an innovative concept of hybrid drive heat pump technology based on a vapor compression cycle with partial flow extraction (VE heat pump cycle) that would enable to provide cost effective heating. Moreover, the system can provide significant environmental benefits by using renewable energy sources (e.g. solar thermal, geothermal and or biomass) and energy benefits by allowing the valorization of low-temperature residual heat from buildings. It is based on two well-known

thermodynamic cycles: The Organic Rankine Cycle (ORC) and the vapor compression cycle. It is able to perform energy tri-generation, which means, depending on the application, it can provide heat, cold and/or electricity. A flowsheet diagram showing the main refrigerant circuit of the proposed system is given in Figure 3.

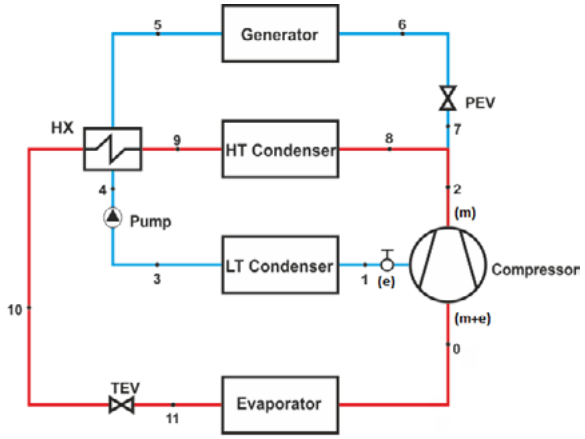


Fig. 3. Flowsheet diagram showing the main refrigerant circuit of a VE heat pump cycle

The new working fluid like R1234yf can be a good solution for such a type of advanced heat pumps. However, for the basis of this comparison with existing vapor compression heat pumps, the refrigerant R134a is considered as a working fluid for calculations. Figure 4 shows the T-s diagram of the VE heat pump cycle.

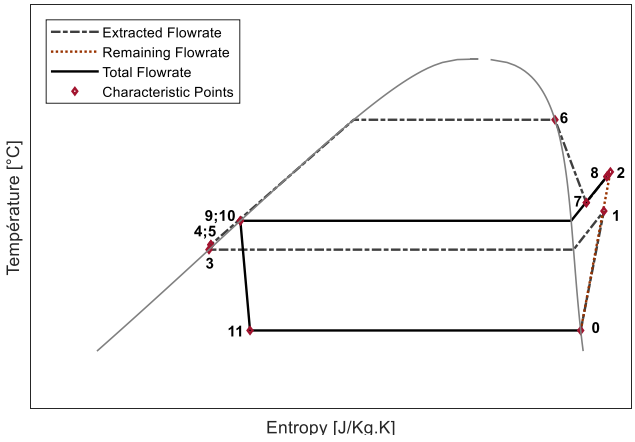


Fig. 4. T-s diagram of a VE heat pump cycle

A portion of the compressed vapor (e) is extracted from the compressor at an intermediate pressure level (fig. 3). After condensation (in the LT condenser), the liquid is pumped to feed the generator via an internal heat exchanger (HX) which acts as a preheater. The fluid vapor produced within the generator is expanded through a pressure expansion valve (PEV) controlling the outlet pressure. After mixing with the high-pressure vapor coming from the compressor (m), cooling and condensation within a heat exchanger (HT condenser), the total liquid (m+e) is cooled into the internal HX, which acts as a subcooler and then expanded through an expansion valve (TEV) into the evaporator. In order to minimize the electricity consumption, the high-pressure expansion valve (PEV) could be replaced by an expander operating at higher temperature levels compared to the PEV and able to generate a part of electricity needed for the heat pump.

To point out the contribution of the heat recovered at the generator in the overall amount of heat produced by the VE heat pump we defined a parameter named extraction ratio (ER) which represents the extracted flowrate (ė) over the compressor inlet flowrate ($\dot{m} + \dot{e}$). For a standard heat pump system, the ER is equal to zero.

$$ER\ [\%] = \frac{\dot{e}}{\dot{m} + \dot{e}} \tag{1}$$

4. Performance analysis and results

As mentioned above, a low-grad district heating (DH) network application is considered where the centralized VE heat pump is used to supply the required heat and to control the temperature of the network. Thermodynamic calculations have been made to compare the system with existing heat pumps (vapor compression standard heat pump and thermally driven heat pumps) used for the same application.

4.1. Model of the vapor extraction heat pump cycle

A simplified R134a thermodynamic model have been considered for the VE-HP. To establish such model, we made the following assumptions:

- The heat exchangers operate at the saturation level (e.g. without superheating on the evaporator and no subcooling on the LT condenser) and the system is considered in steady state;
- The expansion process through the valves TEV and PEV is considered isenthalpic;
- A global isentropic efficiency of 70% is assumed for both the pump and the compressor;
- The motor efficiency and the mechanical efficiency the pump and the compressor stages are set to 1;
- The pressure losses as well as the heat losses are ignored

The models of the VE-HP are then obtained by performing energy and mass balances on each component (Eq. 2 - 5):

$$\dot{Q}_{gen} = \dot{m} \cdot (h_{gen,out} - h_{gen,in}) \tag{2}$$

$$\dot{Q}_{evap} = (\dot{m} + \dot{e}) \cdot (h_{evap,out} - h_{evap,in}) \tag{3}$$

$$\dot{Q}_{LT_cond} = \dot{m} \cdot (h_{LT_cond,in} - h_{LT_cond,out}) \tag{4}$$

$$\dot{Q}_{HT_cond} = (\dot{m} + \dot{e}) \cdot (h_{HT_cond,in} - h_{HT_cond,out}) \tag{5}$$

where $\dot{m}$ is the mass flowrate through the generator; $h_{gen,out}$, $h_{gen,in}$, $h_{evap,out}$, $h_{evap,in}$, $h_{LT_cond,out}$, $h_{LT_cond,in}$, $h_{HT_cond,out}$ and $h_{HT_cond,in}$and h_5 are respectively the refrigerant enthalpy at the outlet and the inlet of the generator, evaporator, low temperature condenser and high temperature condenser and $\dot{e}$ is the extracted flowrate. The extracted flowrate is calculated by using the extraction ratio ER:

$$\dot{e} = \dot{m} \cdot \frac{1 - ER}{ER} \tag{6}$$

To determine the compressor total electric power, we take account of the work delivered to the fluid by the two compression stages:

$$\dot{E}_K = \dot{m} \cdot (h_{K,out,ext} - h_{K,in}) + \dot{e} \cdot (h_{K,out} - h_{K,in}) \tag{7}$$

where $h_{K,in}$, $h_{K,out,ext}$, $h_{K,out}$ are the refrigerant enthalpy at respectively the compressor inlet, extraction outlet and second stage outlet.

In the same way, the power of the pump is calculated by Eq (8):

$$\dot{E}_p = \dot{m} \cdot (h_{p,out} - h_{p,in}) \quad (8)$$

where $h_{p,out}$ and $h_{p,in}$ are the enthalpy at the pump discharge and succion.

In order to assess the contribution of the heat recovered at the generator, two definitions of coefficient of performance (COP) are considered:

- the net electric coefficient of performance (COP_{el}) given by the ratio between the output of condensers and the total electric power consumed by the system (compressor and pump)
- the hybrid (or global) coefficient of performance (COP_{gl}) taking into account both the electric power consumption and the recovered heat rate at the generator

$$COP_{el} = \frac{\dot{Q}_{LT_cond} + \dot{Q}_{HT_cond}}{\dot{E}_k + \dot{E}_p} \quad (9)$$

$$COP_{gl} = \frac{\dot{Q}_{LT_cond} + \dot{Q}_{HT_cond}}{\dot{E}_k + \dot{E}_p + \dot{Q}_{gen}} \quad (10)$$

For the classical heat pump, since the extraction ratio is zero the COP_{el} and the COP_{gl} are the same. In order to be able to compare the two technologies (standard vapor compression cycle and hybrid vapor compression cycle) with the same criterion, an exergy efficiency was defined as in equation 11. $\dot{e}_{LT_cond}$, $\dot{e}_{HT_cond}$, $\dot{e}_k$, $\dot{e}_p$, $\dot{e}_{gen}$ are respectively the heating exergy provided by the LT condenser and the HT condense, the compressor exergy consumption, the pump exergy consumption and the exergy received at the generator; T_{LT_cond}, T_{HT_cond}, T_{gen}, T_o are respectively the temperature of the LT condenser, the temperature of the HT condenser, the temperature of the hot source at the generator and the outside ambient temperature.

$$\eta_{ex} = \frac{\dot{e}_{LT_cond} + \dot{e}_{HT_cond}}{\dot{e}_k + \dot{e}_p + \dot{e}_{gen}} \quad (11)$$

$$\eta_{ex} = \frac{\dot{Q}_{LT_{cond}} \cdot \left(1 - \frac{T_o}{T_{LT_{cond}}}\right) + \dot{Q}_{HT_{cond}} \cdot \left(1 - \frac{T_o}{T_{HT_{cond}}}\right)}{\dot{E}_k + \dot{E}_p + \dot{Q}_{gen} \cdot \left(1 - \frac{T_o}{T_{gen}}\right)} \quad (12)$$

The set of equations is implemented and solved with Matlab®. At each point of the cycle, the refrigerant properties are evaluated using the thermodynamic state equations.

4.2. Performance simulation

A case study of a DH network operating at a temperature level of 35°C and with a heating nominal demand of 100% is considered for the simulation.

Table 1. An example of a table

	Pressure [bar]	Temperature [°C]
Evaporator	3.74	7
LT-condenser	8.87	38
HT-condenser	11.6	48
Generator	26.3	77

The generator of the VE heat pump is connected to the hot source with a maximum temperature of 80°C and which can come from process waste heat, solar thermal, or biomass. The VE heat pump uses the

atmosphere or a geothermal cold source for which the temperature is supposed to be constant at 10°C. Table 1 shows pressure and temperature levels of evaporation and condensation of the working fluid (R134a). The same thermodynamic cycle is used to evaluate the performance of the standard vapor compression heat pump cycle by setting the extraction ratio at 0%.

4.3. Capacity and efficiency of the VE heat pump

Figure 5 shows the variations of the VE heat pump coefficient of performance (blue curves) and exergy efficiency (red curve) in function of the extraction ratio. A net exergy efficiency of about 63% (e.g. 50% extraction ratio) can be achieved for this application. The minimum value of 55.5% corresponding to a zero extraction ratio represents the exergy efficiency of a standard vapor compression heat pump.

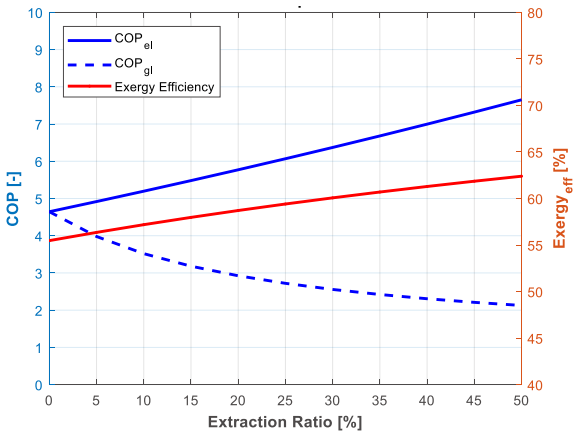


Fig. 5. Coefficient of performances and exergy efficiency of the VE heat pump cycle for various extraction ratio

Note that the global coefficient of performance decrease at high extraction ratio can be explained by the primary heat consumption, which is then too high at the generator of the cycle. As shown in figure 5 the net electric coefficient of performance increase also with the vapor extraction ratio. It can be pointed out that the compressor is used at very partial flow (operation between the LT and HT condensing pressures), requiring then less electric power consumption for the same demand or corresponding to an important gain of heating/cooling capacity when compared to the standard vapor compression cycle with same electricity consumption. Figure 6 represents the variation of heat supply vs electricity consumption with the two configurations. Results show that an increased heating/cooling capacity of about 30% is achieved for a given extraction ratio of 25% (which corresponds also to an exergy efficiency of 59.7% according to the figure 5).

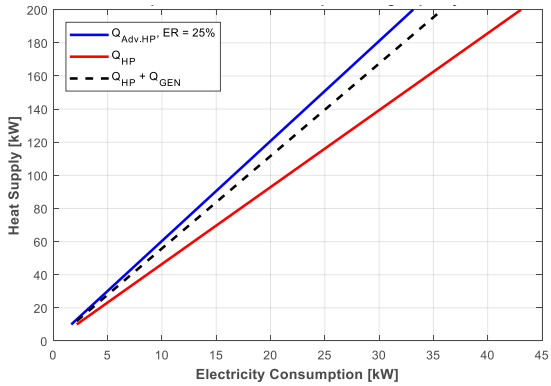


Fig. 6. Heating capacity of the VE heat pump cycle for an extraction ratio of 25% and the standard vapor compression cycle

Other sources of improvements that can influence the capacity of the VE heat pump are related to the internal heat exchanger (HX). The additional subcooling increases the evaporator cooling capacity by reducing the evaporator inlet enthalpy of the fluid. The additional condenser mass flow (**e+e**, for both the LT and HT condenser) increases also considerably the heating capacity of the installation. These added capacities are achieved with less electrical power consumption in the compressor because the amount of vapor extracted (**e**) is compressed only from the suction pressure to the intermediate pressure. Due to this increase in capacity, it could be required to specify a smaller compressor (low displacement) for a given heating and/or cooling demand. For a given extraction ratio of 25%, an increased capacity of 30% is achieved compared to a conventional vapor compression heat pump.

Fluctuations of both the amount of heat recovered at the generator, the hot source temperature and/or outside ambient temperature can be coped with the adaptation of the extraction mass flow rates of the HP working fluid. This flexibility of operation whoud allow to better integrated various sources of renewable energy such as solar thermal application.

4.4. Comparison with commons thermally driven heat pumps

A good way to analyze the performances of such technology is to compare them with those of other tri-therm machines that could be potentially used in this particular application. Figure 7 shows the coefficient of performance of the VE heat pump in function of the temperature levels at the generator compared to existing thermally driven heat pumps.

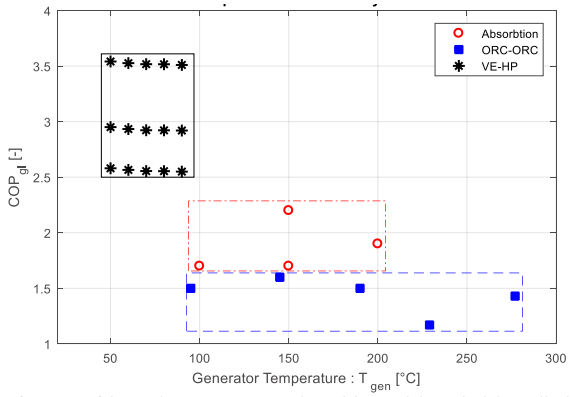


Fig. 7. Coefficient of performance of the VE-heat pump compared to exisitng and theoretical thermally driven heat pumps based absorption and orc-orc systems given in the litterature

Efficiency with vapor extraction heat pump is always higher than conventional thermally driven heat pumps (like absorption heat pump and orc-orc heat pump) when providing the same capacity for heating and cooling. Because, only the small portion (**e**) of refrigerant, extracted from the compressor (less than 30% of the main flow rate) is used to feed the generator, requiring then less heat consumption compared to other systems. A global coefficient of performance (COP) between 2.5 and 3.5 (corresponding to a vapor extraction ratio between 10 and 30%) is obtained and compared to the 1.3-1.7 of existing thermally driven heat pumps working with high temperature levels.

The results show that there is no advantage to work at relatively high temperature levels on the generator of the cycle. Such a situation corresponds in fact to a high difference of temperature between the HT condensing temperature and the generator evaporation temperature reducing then the global coeffient of performance because of losses on the pressure expansion valve (PEV). A high difference of temperature corresponds also to a high difference of pressure on the pump that is accompanied by a significant consumption of the electric power. This makes the cycles work at higher temperature on the generator, which is then detrimental for the coefficient of performance.

5. Conclusions

A new concept of hybrid drive heat pump technology based on a vapor compression heat pump cycle with partial flow extraction has been proposed and applied to control the temperature level of a low-grade district heating/cooling network **(**VE heat pump cycle**)**. The system offers more flexibility to integrate and manage different renewable and residual energy sources (such as solar thermal, biomass, geothermal and waste heat) and ability to work with low-temperature heat sources (even below 50°C). For the comparison with standard vapor compression heat pump the refrigerant R134a have been considered and a simplified thermodynamic model was performed to simulate and evaluate the performances (capacity for heating and cooling, coefficient of performance and exergy efficiency) of such a vapor extraction compression cycle concept and compare results to those of existing heat pump cycles. Results show that an increased heating/cooling capacity of about 30% and a high exergy efficiency of 59.7% (instead of 55.5%) can be achieved for a given extraction ratio of 25%, compared to the 55.5% obtained for conventional compression heat pumps. A global coefficient of performance between 2.5 and 3.5 (corresponding to a vapor extraction ratio between 10 and 30%) is also obtained and compared to the 1.3-1.7 given values in the literature for existing thermally driven heat pumps (e.g. single or double effect absorption heat pumps or double orc-orc heat-pump cycles). Future work could be devoted to experimental investigations on the VE heat pump and a comparative thermo-economical study with other solutions available for district heating/cooling application.

Acknowledgements

The authors gratefully acknowledge the support of InnoSuisse [Grant number 33741.1 INNO-EE].

References

[1] Kirchner A., Kemmler A. Hofer P. Keller M. Jakob M. Catenazzi G.**:** Analyse des schweizerischen Energieverbrauchs 2000-2009 nach Verwendungszwecken. Swiss Federal Office of Energy, (2010).
[2] Rezaie B., Rosen MA.**:** District heating and cooling: review of technology and potential enhancements. Applied Energy, 93 : 2-10, (2012).
[3] Ostergaard PA., Lund H.**:** A renewable energy system in Frederikshavn using low-temperature geothermal energy for district heating. Applied Energy, 88: 479-87, (2011).
[4] Hallera M-Y., Bertramb E., Dottc R., Afjeic T., Ochsd F., Hadorn J-C.: Review of component models for the simulation of combined solar and heat pump heating systems. SHC, Energy Procedia 30 (2012) 611 – 622
[5] Noussan M., Jarre M., Degiorgis L., Poggio A.: Data Analysis of the Energy Performance of Large Scale Solar Collectors for District Heating. Energy Procedia, Volume 134, October 2017, Pages 61-68
[6] Molyneaux A., Leyland G., Favrat D.**:** Environomic multi-objective optimization of a district heating network considering centralized and decentralized heat pumps. Energy, 35(2): 751-758, (2010).
[7] Østergaard P-A and Andersen A-N.: Booster heat pumps and central heat pumps in district heating. Applied Energy 184 (2016) 1374–1388
[8] Henchoz S., Chatelan P., Marechal F., Favrat D.**:** Key energy and technological aspects of three innovative concepts of district energy networks, Energy, 117 (2016) 465-477, (2016).
[9] Kane M., Grosjean S., Kaempf J.: Energy integration and performance evaluation of a new concept of advanced solar district heating/cooling networks for energy supply in buildings. 5th International Solar District Heating/Cooling (SDH) conferences, April 2018 (Graz, Austria)
[10] Lake, Andrew & Rezaie, Behanz & Beyerlein, Steven: "Review of district heating and cooling systems for a sustainable future," Renewable and Sustainable Energy Reviews, Elsevier, vol. 67(C), pages 417-425 (2017).
[11] Wang R.Z. and Oliveira R.G.**:** Absorption refrigeration-An efficient way to make good use of waste heat and solar energy. Progress in Energy and Combustion Science, 32(4): 424-458, (2006).
[12] Schossig P.**:** Thermally driven heat pumps (Foreword). IEA Heat Pump Center Newsletter, 29(1): 18-22, (2011).
[13] Wang H., Perterson R., Harada K., Miller E., Ingram-Goble R., Fisher L., Yih J., and War C.**:** Performance of a combined organic Rankine Cycle and vapor compression cycle for heat activated cooling. Energy, 36(1): 447-458, (2011).
[14] Abdulateef J.M., Sopian K., Alghoul M. A. and Suaiman**:** Review on solar-driver ejector refrigeration technologies. Renewable and Sustainable Energy Reviews, 13(6-7): 1338-1349, (2009).
[15] ROBUR SA, Association Swiss Gas Industry [On ligne]. - 2018. - http://www.gaz-naturel.ch/.

[16] Prigmore D., Barber R.: Cooling with the sun's heat: Design considerations and test data for a Rankine Cycle prototype. Solar Energy.
[17] Strong D.T.G.: Ph.D. Thesis, University of Oxford, 1980.
[18]0Biancardi F.R., Sitler J.W., Melikian G. : Development and test of solar Rankine cycle heating and cooling systems. International Journal of Refrigeration.
[19] Grossmann G.: Solar powered systems for cooling, dehumidication and air-conditioning. Solar Energy.
[20] Critoph R.E.: Forced convection adsorption cycle with packed bed heat regeneration. International Journal of Refrigeration.
[21] Demierre J.: Theoretical and Experimental Study of a Thermally Driver Heat Pump Based on a Double Organic Rankine Cycle. PhD. Thesis No. 5201, Suisse Federal Institute of Technology, EPFL-Lausanne, Switzerland, (2012).
[22] Morvaj B., Evins R., Carmeliet J.: Optimizing urban energy systems: Simultaneous system sizing, operation and district heating network layout. Energy 116 (2016) 619e636