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ABSTRACT

A new concept of heat pump has been developed for Domestic Hot Water (DHW) generation. The concept is based on recovery of heat on lukewarm wastewater recovered from sinks, showers, and bath tubes. The recovered lukewarm water is stored at atmospheric pressure in a tank. The typical average temperature of wastewater is of 31°C. The DHW is produced instantaneously by the heat pump and stored in a tank, and hot water is delivered at 61°C. Refrigerant blends showing different temperature glides have been developed in order to improve energy efficiency of the heat pump system. Simulations using dedicated software allowed calculating the annual energy consumption. Moreover, specific control has been developed taking into account the daily needs of DHW in hotels. Annual COPs of 7 have been reached.

1. INTRODUCTION

Currently, the improvement in the quality of life causes increasing energy demand. In France, nearly 8% of residential energy use is for domestic water heating. Therefore, it becomes essential to reduce the DHW generation consumption.

Many systems exist to produce DHW using different heat sources. Combined systems constitute large applications. Xu *et al.* (2006) studied a heat pump water heater with a solar-air source. Kuang and Wang (2005) worked on a direct-expansion solar assisted heat pump system. Nekså *et al.* (1998), Stene (2005), and Yokoyama *et al.* (2007) treated an air source CO_2 heat pump. Hepbasli (2007) worked on the assessment of solar assisted DHW integrated ground-source heat pump system. Baek *et al.* (2005) studied and analysed a heat pump heating system using wastewater as a heat source for saunas.

The purpose of this study is to design a heat pump using wastewater as heat source to generate DHW at very high efficiency.

2. SYSTEM DESIGN

A wastewater heat pump system (WWHP) is a DHW generation system for residential and collective habitation presented in Clodic (2005). This system is developed in the study for a hotel of 150 rooms with 60% of occupation ratio in Lyon, France. The hot water generation is semi-instantaneous during daily peak demand. Else it is a semi-accumulated system. Most often, the DHW is generated during night time hours, which allows using low-cost off-peak electricity.

2.1 System description

Fig. 1 shows the system diagram. The WWHP is composed of a single-stage direct expansion heat pump, additional preheating plate heat exchanger (PHX), a city-water storage tank (CWST), a hot-water storage tank (HWST), an automatic self-cleaning water filter, a wastewater atmospheric pressure storage tank (WST), and three circulating pumps.

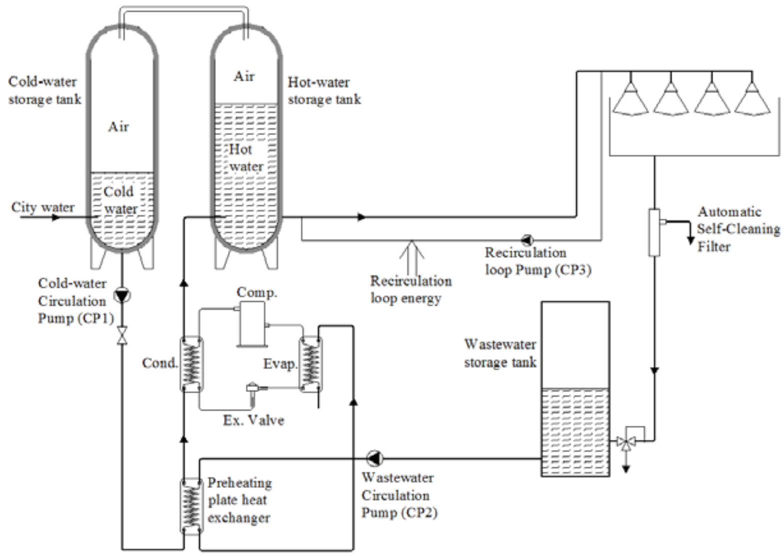


Figure 1: Schematic diagram of the wastewater heat pump system.

Two separated water circuits exist: Tap Water Circuit and Wastewater Circuit.

Tap Water Circuit: city water is stored in the CWST connected to the HWST. Both tank pressures are regulated by air at the city water pressure (0.5 MPa). During the heat storage period, cold water is pumped to the PHX then flows through the condenser of the heat pump and finally stored in the HWST. The water temperature at the PHX outlet varies between 26°C and 28°C, and 61°C at the condenser outlet. During the period of use, hot water is distributed according to needs. A recirculation loop system is installed to conserve a high temperature in the distribution pipes.

Wastewater Circuit: wastewater coming from sinks, showers, and bath tubes passes through the Automatic Self-Cleaning Filter directly to the WST. During the heat storage period, the wastewater is circulated through the PHX, and then flows into the evaporator of the heat pump. The wastewater enters the PHX at 31°C. Its temperature varies between 21°C and 26°C at the evaporator inlet.

2.2 Design conditions

Four main parameters have to be known to design an efficient hot water heat pump system: the tap hot-water temperature, the mass flow rate, the volume, and the waiting time. These four parameters depend on people activity and satisfaction on the first hand, and on the hotel occupation on the other hand. The city water temperature, hot-water temperature, and the hot water volume have been measured each 10 minutes all over a year.

Water supply is very high in the morning (125 kg/min) and low in the afternoon (<10 kg/min). Fig. 2 shows the variation of the tap water demand in a typical winter day.

According to French standards, due to Legionella development, the hot water temperature in the HWST should reach 60°C at least for 2 hours, and the lowest temperature in the distribution tubes should be about 50°C. Table 1 shows the operation temperatures for the system design.

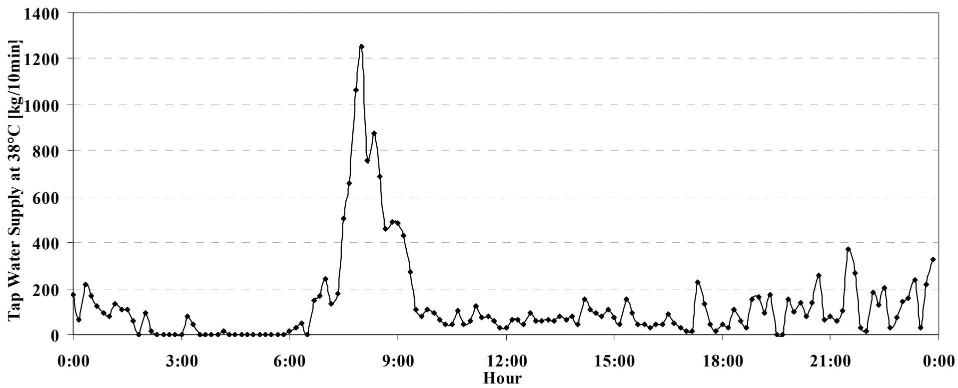


Figure 2: Specific daily amount of tap water.

Table 1: Design conditions.

Item	Design condition	Remarks
Heating temperature	61°C	
Hot water storage temperature	~60°C	
Tap water temperature	38°C	Shower average temperature
Wastewater temperature	~31°C	Considering 7 K for heat losses
Evaporating water outlet temperature	5°C	Water temperature is regulated at 5°C for freeze protection.

2.3 System controls

Tap Water Circuit Control: The hot water supply reduces the stored hot water in the HWST; therefore the water pressure in the tanks is lowered. The city water pressure becomes higher than the storage pressure. Consequently, cold water enters the CWST until the pressure equilibrium is reached. Thus, the cold-water inventory is exactly equal to the quantity of delivered hot water; the water volume and the air pressure are kept constant in the two tanks.

Wastewater Circuit Control: A thermostatic water valve is installed at the inlet of the WST. If the wastewater temperature is colder than the recovered water, it will be sent directly to the drain.

3. HEAT PUMP SYSTEM

The heat pump is used to transfer heat from the wastewater to the stored hot water. The higher the wastewater temperature, the higher the heat pump efficiency because the temperature difference between condenser and evaporator working fluids is reduced.

3.1 Heat pump control

In order to ensure the availability of hot water, the heat pump operates every day at 5:00 AM just before the peak period. At this hour, the electricity cost is low. Also it operates when the hot water level in the HWST becomes lower than the minimum level. Once the heat pump is ON, it stays so until the HWST is full.

The heat pump operation depends also on the wastewater level in the WST. The heat pump is OFF when the wastewater level becomes lower than the minimum.

3.2 Heat pump heating capacity

The heat pump heating capacity depends on the hot water needs (see Fig. 3). During the peak load, WWHP works in a semi-instantaneous mode. This means that the installed heating capacity of the heat pump depends on the storage volume and on the operating time.

For a 150-room hotel, a heat pump of 40 kW is capable to deliver the hot water needs in 4 hours. The running time depends on the amount of energy transferred by the PHX.

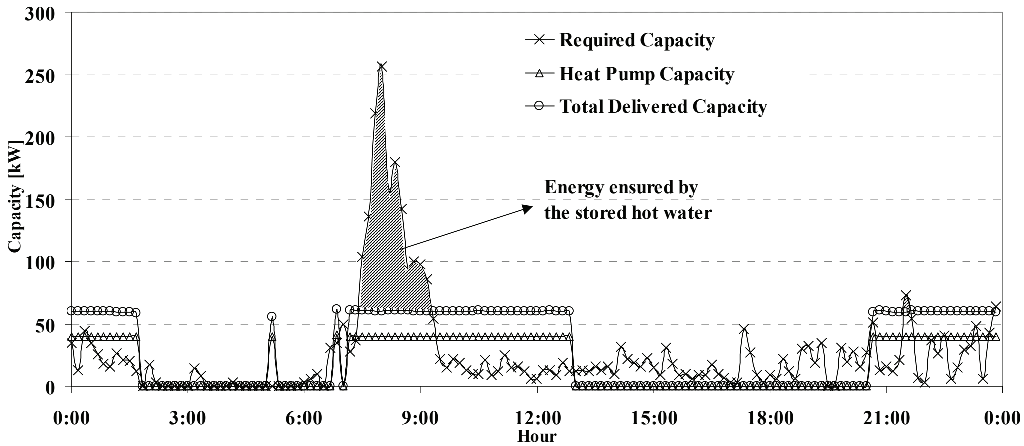


Figure 3: Heat pump heating capacity.

Fig.3 shows the required capacity, the stored capacity, and the HP heating capacity for 2 days. During the peak load, the heat pump and the PHX ensure 60 kW of the needs and the complement is delivered by the stored hot water. The heat pump operates about 2 hours after the peak period to recharge the HWST. Out of the peak period, the required capacity is lower than the heat pump capacity, so it is ensured only by the stored hot water.

3.3 Heat pump performance

The single-stage heat pump is composed of a scroll compressor, a plate condenser, a plate evaporator, and an electronic expansion valve.

The compressor effective efficiency is evaluated based on the manufacturer performance data. For simulation purposes, it is given as a function of the pressure ratio:

$$\eta_{eff} = a_0 + a_1 \cdot PR + a_2.PR^2 + a_3.PR^3 + a_4.PR^4 + a_5.PR^5 \tag{1}$$

Where a_0 = 39.58, a_1 = 53.594, a_2 = -21.204, a_3 = 3.5972, a_4 = -0.2971, a_5 = 0.0097.
The compressor work is calculated by:

$$W_{comp} = \dot{m} \cdot \frac{\Delta h_{is}}{\eta_{eff}} \tag{2}$$

Thermodynamic properties were computed using the database REFPROP7. The thermodynamic cycle is calculated according to the “pinch” method.

The heat pump performance is characterized by the coefficient of performance (COP).

$$COP_{HeatPump} = \frac{Q_k}{W_{comp}} = \frac{\Delta h_k}{\Delta h_{comp}} \tag{3}$$

Equation (3) shows that the heat pump performance depends on the refrigerant thermodynamic properties. For the condensing and evaporating operating temperatures, the suitable refrigerant that leads to the highest heat pump COP should be used.

4. WORKING FLUID SELECTION

4.1 Selection criteria

The refrigerant blend selection is based on the following criteria.

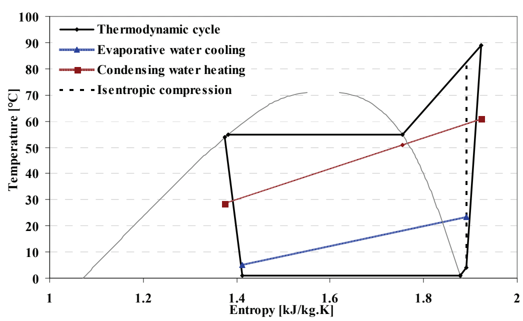
- *Environmental aspect:* Global warming potential (GWP) as low as possible, and zero ozone depletion potential (ODP).
- *Safety aspects:* Non-toxic, non-flammable or moderately flammable.
- *Thermodynamic aspect:* High volumetric heat capacity, limited pressure at the condenser, compatible with oil, and energy efficient for the levels of source and sink temperatures.

4.2 Temperature glides

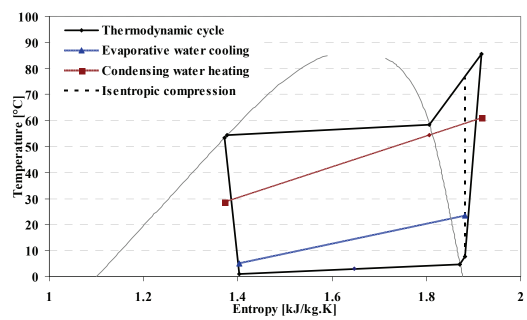
To determine the maximum COP, several blends with different temperature glides have been simulated for the same source and sink temperatures. Table 2 presents the results and Fig. 4 shows the differences between the thermodynamic cycles.

Table 2: Mixture properties.

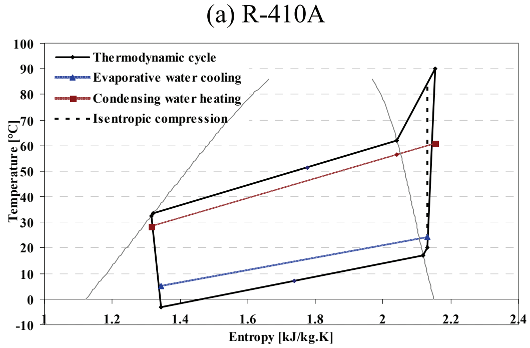
Mixture	Condenser T-s slope [kg.K²/kJ]	Evaporator T-s slope [kg.K²/kJ]	Heat pump COP [W/W]	Comments
R-410A	0.27	0.14	3.55	Refrigerant blend with no temperature glide.
R-407C	9.86	8.14	3.68	Refrigerant blend with low temperature glide.
Blend 1	39.35	26.32	5.44	Parallel condenser temperature glide.
Blend 2	35.14	21.65	5.60	Parallel evaporator temperature glide.



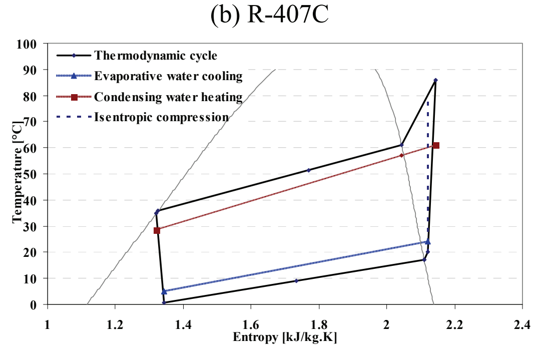
(a) R-410A



(b) R-407C



(c) Blend 1 :Parallel condenser temperature glide



(d) Blend 2 : Parallel evaporator temperature glide

Figure 4: Thermodynamic cycles for different refrigerant blends.

As shown in Table 2, the maximum heat pump performance is reached for blend 2 because the difference of the average condensing temperature and the average evaporative temperature is minimum. In this case, the water temperature glide is parallel to the evaporative refrigerant temperature glide. Consequently, the compressor work is reduced and the overall system performance is improved.

5. ENERGY ANALYSIS

5.1 Total heating energy

One part of the heating energy is delivered by the heat pump and the other part by the PHX. The condenser heating energy is obtained using:

$$E_k = m_w \cdot c_{p_w} \cdot \left(T_{w,out,k} - T_{w,in,k}\right) \tag{4}$$

And the PHX heating energy is calculated by:

$$E_{PHX} = m_w \cdot c_{p_w} \cdot \left(T_{w,in,k} - T_{CW}\right) \tag{5}$$

The total heating energy is calculated by:

$$E_{Heating} = E_k + E_{PHX} \tag{6}$$

Fig. 5 shows the distribution of required heating energy for each month of the year. The amount of energy delivered by the PHX is proportional to the city water temperature. For low temperature city water, the PHX delivers more energy. In winter, more than 30% of the total heat supply is provided by the PHX. Otherwise, it is lower than 20% in summer. However, the heat pump supplied energy is more sensitive to the demand of hot water. In July for example, the energy provided by the heat pump is low because the hot water demand is also low. The percentage of energy provided by the PHX is low because the city water temperature is high.

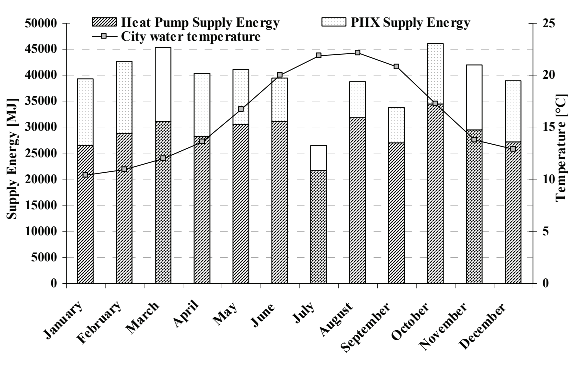


Figure 5: Energy demand for each month.

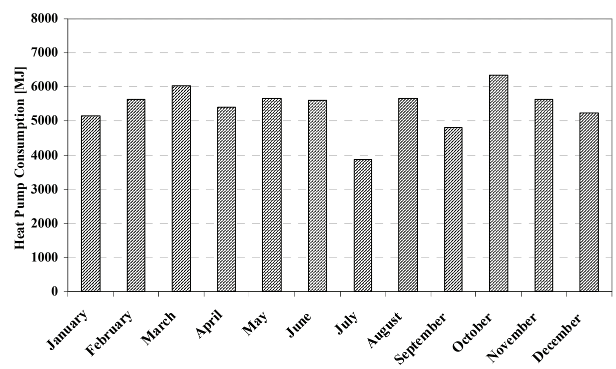


Figure 6: Energy consumption for each month.

5.2 Components energy consumption

The total energy consumption is the sum of all component consumptions.

$$E_{consumption,total} = E_{CP1} + E_{CP2} + E_{CP3} + E_{comp} \tag{7}$$

Where:

$$E_{comp} = \frac{E_k}{COP_{HeatPump}} \tag{8}$$

The energy consumption of circulating pumps is obtained from the manufacturer energy consumption data as a function of the mass flow rate for different pressure drops.

Fig. 6 shows the energy consumption of the heat pump compressor. It depends on the energy demand. The monthly variation of the heat pump consumption is similar to the variation of the heat pump supply energy because the heat pump COP remains almost constant all over the year. However, the total energy consumption of the circulating pumps constitutes only about 4.5% of the compressor consumption.

6. OVERALL SYSTEM PERFORMANCE

The overall system performance depends on three main conditions: the mass supply, the wastewater temperature and the city water temperature. Therefore, a dynamic analysis is carried out all around the year. Given the hot water needed volume, and the city water temperature, we assume that the water temperature is used at about 38°C.

The overall system performance is determined by the system COP.

$$COP_{system} = \frac{E_{Heating}}{E_{consumption,total}} \tag{9}$$

Table 3 shows the results of energy analysis by month.

Table 3: Monthly energy analysis results.

Month	Heating demand [MJ]	Supply energy [MJ]		Energy consumption [MJ]				COP System [W/W]
		Heat pump	PHX	Compressor	CP 1	CP 2	CP 3	
January	38774	26426	12899	5163	87	49	96	7.29
February	42017	28875	13888	5628	95	54	96	7.28
March	44923	31155	14293	6025	100	58	106	7.23
April	39833	28346	12022	5418	89	54	102	7.13
May	40179	30553	10529	5660	90	59	106	6.95
June	38381	31153	8340	5595	86	62	102	6.76
July	25275	21721	4765	3869	57	43	106	6.50
August	37582	31849	6887	5670	83	64	106	6.54
September	32786	27073	6785	4819	74	54	102	6.71
October	45325	34442	11560	6356	101	67	106	6.94
November	40980	29618	12397	5650	93	56	102	7.12
December	38282	27291	11741	5230	86	51	106	7.13
Total	464336	348502	126107	65083	1040	670	1236	**6.98**

The system COP varies slightly with the season. The overall COP of the system decreases in summer due to the low heating load and the high city water temperature. Fig 7 shows the overall COP variation all over the year.

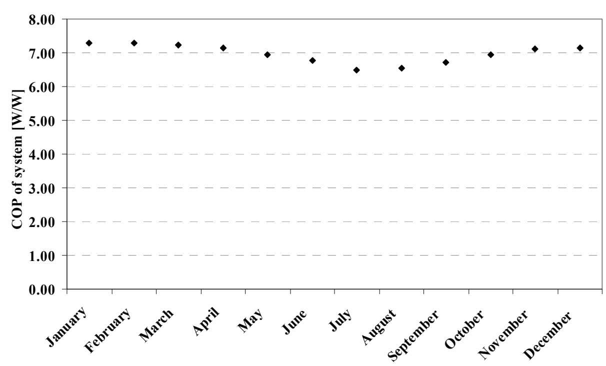


Figure 7: System COP for each month.

The annual system COP is sensitive to the heat pump efficiency. For the analyzed operating conditions, the cycle efficiency referred to Carnot is about 0.71 and the annual system COP is 7.

7. CONCLUSION

A heat pump system, using wastewater recovered from showers and sinks, was designed for DHW generation. French standards concerning hot water storage and distribution are respected.
The hot water is stored at 60°C and the tap water temperature is delivered at 38°C. The recovered water temperature is about 31°C.

Performance analysis was performed by a computer simulation:
- The PHX, installed upstream the heat pump, preheats the city water and transfers about 18% to 33% of the total heating capacity for free.
- Due to high demand of hot water and low city water temperature in winter, the heating energy transferred in the PHX is higher than the PHX heating energy in summer. Therefore, the monthly system COP is higher in winter.
- The heat pump COP is higher when using a refrigerant blend with adapted temperature glide. The highest COP is reached when the water temperature glide is parallel to the evaporative refrigerant temperature glide.
- The overall annual COP of the system is 7 for the given manufacturer compressor effective efficiency.

In future work, the system COP has to be determined for different cycle efficiencies and for different wastewater temperatures.

NOMENCLATURE

c_p	Heat capacity	(kJ/kg.K)
E	Energy	(kJ)
h	enthalpy	(kJ/kg)
m	Mass	(kg)
$\dot{m}$	Mass flow rate	(kg/s)
PR	Pressure ratio	(-)
Q	Heating capacity	(kW)
T	Temperature	(K)
W	Power	(kW)
Greek symbols		
Δ	Difference	(-)
η	Efficiency	(%)

Subscripts

comp	compresser
CP	circulating pump
CW	city water
eff	effective
in	inlet
is	isentropic
k	condensation
out	outlet
w	water

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