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Determination of a domestic hot water annual performance factor of an air to water heat pump

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ABSTRACT

HSPF (heating seasonal performance factor) is an important coefficient that characterizes the seasonal performance of heat pumps for heating; and it does not take into account the production of domestic hot water. As such, it is also possible to calculate a performance factor that characterizes the annual performance of the heat pump for domestic hot water production.

The aim of this study is to calculate this factor for an air to water heat pump in different climate zones in France. For this, the system consisting of the heat pump and the storage water tank has been modeled and the calculation has been carried out on a simulation tool. The simulation has been realized over a year with hourly outdoor air temperature data. Furthermore, the hot water requirements vary; so different domestic hot water tapping cycles have been studied.

Key words: air source heat pump, domestic hot water, annual performance.

1. INTRODUCTION

Most common air to water heat pumps cannot supply water at a temperature greater than 50°C. However, recent heat pumps can produce hot water up to 65°C, even at low outdoor air temperature, with the use of a two-stage compression cycle. Thus, it is possible to produce domestic hot water (DHW) without adding an immersion heater in the storage water tank.

The goal of this study is to calculate the DHW annual performance factor of such heat pumps. The DHW annual performance factor corresponds to the ratio of the heating energy delivered and the electrical energy consumed by the heat pump over a year. The gap between the DHW consumption and the heating energy delivered by the heat pump is mainly due to the heat losses of the water tank.

The DHW consumption is usually determined by a binning analysis method. Haglund Stignor *et al.* (2004) presented an application of this method. The DHW consumption can also be calculated by simulating the hot water production system (Morrison, 2004). In this study, the calculation of DHW consumption is also based on simulation. The hot water production system (Figure 1) consists of a high temperature heat pump and a water storage tank with an internal heat exchanger. Thereafter, the models used for each component are presented. The results of simulations and the influence of parameters such as the tapping cycles and weather conditions on the DHW annual performance factor are analyzed.



Figure 1: Domestic hot water production network.

2. A model for each component of the system

2.1. The air to water heat pump

For this study, an air to water high temperature heat pump is considered because common air to water heat pump cannot provide water at a temperature higher than 50°C. This heat pump is equipped with a vapor injection compressor and uses R-407C as refrigerant. Consequently, this heat pump can reach a water temperature of 65° C at the outlet of the condenser. The heating capacity and the power input of the heat pump are correlated to the water and the air temperature at the inlet of the heat pump. The correlation is represented by a bilinear interpolation of measured data (Figure 2).



Figure 2: Bilinear interpolation of the heating capacity and the power input of the heat pump.

With heat pumps, the domestic hot water is not produced instantaneously but with a water storage tank. The water tank is equipped with an internal heat exchanger in which flows hot water heated by the heat pump, so the heat pump does not directly heat up the water in the tank. Thereafter, the models used for the water tank and the internal heat exchanger will be explained.

2.2. The water tank

The water tank is considered as a cylinder with external insulation. It exchanges heat by conduction and convection with the ambient air temperature fixed to 20°C. As such, the heat losses of the tank are taken into account. A uniform water temperature in the tank is assumed, i.e. as if the water in the tank were always well mixed, so the water tank temperature is equal the water temperature at the outlet of the water tank.

Subsequently, different equations for calculation are presented. First, the overall heat transfer coefficient on the lateral surface of the water tank is calculated,

$$U_{lat} = \frac{1}{A_{lat}} \left[\frac{1}{2\pi\pi_{tank} \lambda_{insulator}} ln \left(\frac{D_{tank} + 2\delta_{insulator}}{D_{tank}} \right) + \frac{1}{\pi (D_{tank} + 2\delta_{insulator}) H_{tank} h_{air_free_convection}} \right]^{-1}$$
(1),

and than the overall heat transfer coefficient at the top and the bottom of the water tank,

$$U_{bottom} = U_{top} = \left[\frac{1}{h_{air_free_convection}} + \frac{\delta_{insulator}}{\lambda_{insulator}}\right]^{-1}$$
(2).

In the equations (1) and (2) the heat transfer coefficient $h_{air_free_convection}$ is fixed at 10 W.m⁻².K⁻¹. Thus, with the equations (1) and (2), it is possible to calculate an overall heat transfer coefficient of the water tank,

$$U_{tank} = \frac{\left(U_{lat}A_{lat} + 2U_{top}A_{lop}\right)}{A_{tank}}$$
(3).

It is also necessary to calculate the thermal capacitance of the water tank,

$$m_{1}c_{1} = \frac{\pi D_{tank}^{2}}{4} H_{tank} \rho_{water} c_{p,water} + H_{tank} \pi D_{tank} \delta_{steel} \rho_{steel} c_{steel}$$
(4).

So as to determine the temperature at the outlet of the water tank, an energy balance is applied on the water tank,

$$m_{1}c_{1}\frac{dT_{tank,water}}{dt} = \dot{m}_{draw-off}c_{p,water}\left(T_{tank,water,in} - T_{tank,water}\right) + U_{tank}A_{tank}\left(T_{ambient} - T_{tank,water}\right) + \dot{Q}_{IHEX}$$
(5),

with \dot{Q}_{IHEX} the heat transfer rate of the internal heat exchanger.

Consequently, it is possible to take into account the inertia of the water tank and the heat losses of the water tank, but also to calculate the evolution of the water temperature at the outlet of the water tank in transient conditions.

2.3. The internal heat exchanger

The water in the tank is heated up with an internal heat exchanger through which hot water circulates. This water comes from the heat pump. The internal heat exchanger is usually a helically coiled tube, but for the determination of the correlation, it will be considered as a horizontal tube. For the calculation of the overall heat transfer coefficient, a forced convection inside the tube and a free convection outside is assumed. First, we will calculate the thermal capacitance of the steel of the heat exchanger,

$$m_{steel}c_{steel} = \left[\left(\frac{D_{ext}}{2}\right)^2 - \left(\frac{D_{int}}{2}\right)^2 \right] \pi L_{IHEX} \rho_{steel} c_{steel}$$
(6),

and the thermal capacitance of the water inside the heat exchanger,

$$m_{water}c_{p,water} = \left(\frac{D_{int}}{2}\right)^2 \pi L_{IHEX} \rho_{water} c_{p,water}$$
(7).

So, it is possible to determine the thermal capacitance of the internal heat exchanger by the equation (8) while taking into account the thermal inertia of the internal heat exchanger.

$$m_2 c_2 = m_{steel} c_{steel} + m_{water} c_{p,water}$$
(8)

The water temperature at the outlet of the internal heat exchanger is calculated by performing a heat balance on the internal heat exchanger,

$$m_2 c_2 \frac{dT_{IHEX,water}}{dt} = U_{ext} A_{ext} \left(T_{tank} - T_{IHEX,water} \right) + \dot{m}_{int} c_{p,water} \left(T_{IHEX,water,in} - T_{IHEX,water,out} \right)$$
(9).

However, to carry out the calculation, the overall heat transfer coefficient U_{ext} of the internal heat exchanger has to be determined. This coefficient is a function of the internal and the external heat transfer coefficient. The internal heat exchanger coefficient is determined by using internal flow heat transfer correlation. The correlation used depends on the flow conditions i.e. on the Reynolds number,

$$Re_{D_{int}} = \frac{4\dot{m}_{int}}{\pi D_{int} \mu_{water}}$$
(10).

If $Re_{D_{int}} > 2500$, the correlation of Petukhov Popov (1963) is used,

$$Nu = \frac{(f/2)Re_{Dint}Pr}{1.07 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)}$$
(11),

and the friction factor is calculate by a correlation proposed by Shah (1987),

$$f = 0.00128 + 0.1143Re_{D_{int}}^{-0.311}$$
(12).

And if $Re_{D_{int}}$ is lower than 2500, the flow is laminar and the equation (13) proposed by Kays (1980) is used.

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$$Nu = 3.66$$
 (13)

So, it is possible to calculate the internal heat transfer coefficient with the equation (14).

$$h_{int} = \frac{\lambda_{water} N u}{D_{int}} \tag{14}$$

On the external side of the heat exchanger, free convection is considered and the internal heat exchanger is assumed to be a horizontal tube, so it is possible to use the correlation of Churchill and Chu (1975) that is verified if $Ra_{D_{ext}} \leq 10^{12}$.

$$Nu = \left[0.60 + \frac{0.387 Ra_{D_{ext}}^{1/6}}{\left[1 + (0.559/Pr)^{9/16}\right]^{8/27}}\right]^2$$
(15)

Thus, it is possible to calculate the external heat transfer coefficient with equation (16).

$$h_{ext} = \frac{\lambda_{water} N u}{D_{ext}}$$
(16).

Finally, knowing the different heat transfer coefficient, it is possible to calculate the overall heat transfer coefficient U_{ext} with the equation (17) and uses it in the equation (9).

$$U_{ext} = \frac{l}{A_{ext}} \left[\frac{l}{(\pi D_{int} L_{IHEX} h_{int})} + \frac{l}{(\pi D_{ext} L_{IHEX} h_{ext})} + \frac{l}{2\pi L_{IHEX} \lambda_{steel}} ln \left(\frac{D_{ext}}{D_{int}} \right) \right]^{-l}$$
(17)

2.4. Control for the production of domestic hot water

The standard EN 13203-2, (2006) defines different tapping cycles depending on the use. These tapping cycles are expressed in terms of energy consumption of hot water over time. In the standard EN 13203-2, five different tapping cycles are defined, but in this study, only the tapping cycles $n^{\circ}2$ and $n^{\circ}3$ are considered (Figure 3).



Figure 3: Tapping cycle n°2 et n°3 according to the standard, NF EN 13203-2 (2006).

With the defined tapping cycle and knowing the water temperature at the inlet and at the outlet of the water tank, it is possible to determine the mass flow rate of water through the water tank with the equation (18).

$$\dot{m}_{draw-off} = \frac{Q_{DHW}}{\Delta t \times c_{p,water} (T_{tank,out} - T_{tank,in})}$$
(18)

The temperature in the water tank controls the on/off of the heat pump. The set point corresponds to the temperature at which the heat pump starts. If $T_{tank} \ge T_{setpoint} + 5^{\circ}C$, the heat pump stops and if $T_{tank} \le T_{setpoint}$, the

heat pump starts. Furthermore, a minimum working time of 300 s and a minimum stop time of 600 s are fixed for the heat pump.

3. Results

3.1. Results of the simulation

The simulations have been performed under Matlab/Simulink with the toolbox Simbad 4 (CSTB, 2004). The results have been evaluated using a 1 min time step model. For the first result, the different parameters are fixed as indicated in the Table 1. At first, the simulation was performed with the French standardized weather condition in the city of Trappes (Figure 4). The simulation was performed over a year in order to evaluate the annual DHW consumption. The gap between the energy delivered by the heat pump and the DHW consumption corresponds to the heat losses of the water tank. In this case, the DHW APF was found to be 2.48. Regarding to the DHW consumption, a consumption of 2.13 MWh was found. The heating energy at the outlet of the heat pump is equal to 2.32 MWh, thus the water tank presents annual heat loss energy of 8%, hence the importance of the insulation of the water tank.

hair_free_convection	$10 \text{ W.m}^{-2}.\text{K}^{-1}$	H _{tank}	1.4 m
λ_{steel}	$15 \text{ W.m}^{-1}.\text{K}^{-1}$	D _{tank}	0.47 m
$\lambda_{insulator}$	$0.03 \text{ W.m}^{-1}.\text{K}^{-1}$	V _{tank}	250 L
$c_{p,water}$	4180 J.K ⁻¹ .kg ⁻¹	L _{IHEX}	1.7 m
c _{steel}	500 J.K ⁻¹ .kg ⁻¹	D _{IHEX}	0.0254 m
$\delta_{insulator}$	0.05 m	T _{setpoint}	50°C
$\delta_{tank,steel}$	0.005 m	T _{cold water}	15°C
$ ho_{water}$	1000 kg.m ⁻³	Tapping cycle	n°2
ρ_{steel}	8000 kg.m ⁻³	Weather area	Trappes

Table	1.	Input	data	for	the	simul	lation
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Figure 4: Outdoor air temperature distribution of three climate area in France (RT2005, 2006).

The Figure 5 presents results of simulation on one day, such as the evolution of the water temperature in the water tank. The water temperature at the outlet of the heat pump can reach 65°C. On Figure 5, it can be seen that there is a gap between the temperature at the outlet of the water tank and the temperature at the outlet of the internal heat exchanger. By increasing the surface of the internal heat exchanger, this gap may be reduced. Thus, the water temperature at the outlet of the heat pump will decrease and the COP will be improved.



Figure 5: Evolution of the water temperature at the outlet of the heat pump and inside the water tank.

3.2. Effects of different parameters on the results

As the COP of the heat pump is function of the outdoor air temperature, simulations are performed for different weather zones in France. These weather zones are defined by the RT2005 (2006), which is a French thermal regulation. Calculations are performed for each weather zones. With the input data of the Table 1, it can be seen on Figure 6 that the DHW AFP varies between 2.46 and 2.60 in France.



Figure 6: DHW APF for different weather area in France.

In order to see the effect of 4 factors (Table 2), a complete experimental design has been performed. Each factor has been varied between a maximum and a minimum level; the purpose being to evaluate the effect of these factors on responses such as the DHW consumption, the DHW APF and the heat losses of the water tank.

	$T_{tank,in}$ (°C)	T _{tank,setpoint} (°C)	V_{tank} (L)	Tapping cycle
Minimum	14	49	200	n°1
Maximum	16	51	300	n°2

Table 2: Maximum and minimum level of each factor.

The results are summed up in the Table 3, it can be seen that an increase of the set point temperature reduces the DHW AFP by 3%. The DHW AFP is slightly sensitive to the temperature of the cold water and the volume of the water tank. On the other hand, the volume of the tank can significantly increase the heat losses of the water tank.

Table 3: Effects of factors on the DHW consumption, the DHW APF and the heat loss energy of the water tank.

	DHW consumption	DHW APF	Heat losses of the water tank
$T_{tank,in}$ +2°C	~0%	~0%	~0%
$T_{setpoint}$ +2°C	~0%	-3%	7%
V_{tank} +100L	3%	~0%	89%
Tapping cycle n°2 to n°3	167%	2%	-38%

4. CONCLUSIONS

This study has allowed determining the annual performance factor of a high temperature heat pump for the production of DHW. For different French climates, a mean value of 2.5 has been found. This value shows the significant benefit of high temperature heat pump to produce DHW. Nevertheless, the real benefit of high temperature heat pump is to produce not only domestic hot water but also heating with the same device.

Subsequently, the model used for the water tank should be improved by considering the stratification. Furthermore, it would be possible to reach higher temperature in the water tank by increasing the size of the internal heat exchanger.

NOMENCLATURE

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Α	Area	(m^2)
с	Specific heat	$(J.K^{-1}.kg^{-1})$
C_p	Specific heat at constant pressure	$(J.K^{-1}.kg^{-1})$
Ď	Diameter	(m)
h	Convection heat transfer coefficient	$(W.m^{-2}.K^{-1})$
Н	Height	(m)
L	Length	(m)
m	Mass	(kg)
m	Mass flow rate	(kg/s)
Q	Energy	(J) or (kWh)
Q	Heat transfer rate	(W)
t	Time	(s)
Т	Temperature	(°C)
U	Overall heat transfer coefficient	$(W.m^{-2}.K^{-1})$
V	Volume	(m^3)
δ	Thickness	(m)
λ	Thermal conductivity	$(W.m^{-1}.K^{-1})$
μ	Viscosity	(kg/m.s)
ρ	Mass density	(kg/m^3)
Nu	Nusselt number	
Pr	Prandtl number	
Ra	Rayleigh number	
Re	Reynolds number	
APF	Annual performance factor	
COP	Coefficient of performance	
DHW	Domestic hot water	
IHEX	Internal heat exchanger	
int	Internal	
ext	External	

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