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HSPF CALCULATION OF A SYSTEM ALLOWING SWITCHING BETWEEN A SINGLE-STAGE TO A TWO-STAGE COMPRESSION SYSTEM

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ABSTRACT

In France, 62\% of heating installations in existing single-family dwellings are hydronic systems. A high supply temperature, higher than 65 °C, during the coldest days is needed in order to ensure the comfort. An interesting way to reach high temperature by low air temperature with a heat pump is to use a two-stage compression cycle. But, when the weather becomes warmer, the efficiency of a single-stage compression system becomes better than a two-stage compression system. In order to optimise the heating seasonal performance factor, HSPF, a system allowing switching from a single-stage compression system to a two-stage compression system has been studied.

The aim of this study is to evaluate the gain of switching from a single-stage to a two-stage compression system and to calculate a HSPF in accordance with a house type and heat network. For that the input power and the heating capacity of the heat pump have been expressed in relation to outdoor air temperature and water supply temperature at the outlet of the condenser in a single-stage mode and in a two-stage mode. The system that is studied includes the house, the radiators, and the hydronic circuit. A model describing each component of the heating system has been implemented in a numerical tool, so that, it is possible to simulate, in a quasi-static state, the behaviour of the system, and then to evaluate the COP along a heating season.

\textit{Key words: heat pump, HSPF, control.}

INTRODUCTION

In France, dwellings are classified by their surfaces and their year of construction. An important point is that before 1975, there was no thermal legislation in France and then these houses present a very poor performance thermal insulation. The studied house is assumed to be built after 1980 and presents thermal heat losses of 12.5 kW, when the outdoor air temperature is of -15 °C. The most common emitter in single-family dwellings is the static radiator. Due to the emitter design, a high temperature supply, higher than 65 °C, during the coldest days is required in order to ensure comfort. With such a high temperature, it is not possible to use a classical heat pump. In order to reach this high temperature, a solution consists in using heat pumps, equipped with a compressor with a vapour injection port, and then to take advantage of the two-stage compression cycle. But when the outdoor air temperature is warmer, the heating capacity is significantly higher than the heating demand, and then the heat pump has to run with on/off cycles. An issue is to use a second compressor in parallel (Flach-Malaspina, N, 2004) with a lowest capacity and to switch on this compressor. Furthermore, at a certain outdoor air temperature the performance with a single compressor is better than with a two-stage compression cycle (Clodic, D, Rahhal, C, 2005). This study demonstrates the advantage of such a solution and at which temperature, the switch has to be done in order to reach the highest HSPF, which is the ratio of the total heat delivered over the heating season to the total energy input over the heating season.

1. DEFINITION OF A REFERENCE MACHINE

1.1 Association in parallel of a compressor with an injection port and a single compressor

Nowadays, new generations of heat pumps (Hafner, B, Heikrodt, K, 2005), (Zehnder, M, 2004), which allow reaching a temperature of 65 °C, have been commercialised. For this study, two compressors in parallel have been used rather than one in order to switch from a single-stage compression system to a two-stage compression system, a scheme of the system is presented on Figure 1. Two commercialised heat pumps have been chosen. The first heat pump is equipped with a compressor including an injection port and the second one is equipped with a single compressor. The concept is to associate these two systems and to switch to a single-stage system when the outdoor air temperature is superior to a certain threshold.
Switching to a single-stage system allows reducing heating capacity and then to decrease the number of on/off cycles, Figure 2 shows the evolution of the heating capacity of these two heat pumps.

The heating capacity and the input power have been expressed with a polynomial regression in relation to outdoor air temperature and water temperature at the inlet of the condenser. A complementary assumption has been made in order to use these data for different volume flow rates: the heating capacity and the power input do not change for the same mean water temperature and the same outdoor air temperature.

## 2 MODELS AND CALCULATION OF THE HSPF

### 2.1 Models under Matlab-Simulink integrating the house, the heating system and the weather

#### 2.1.1 Model of the building: Single house: 3R2C

The model uses a single indoor air temperature. A simple model has been chosen in order to address physical phenomena. The building is modelled as an electrical circuit (see Figure 3). The three resistances represent the heat transfer phenomena and the two capacities represent the thermal mass of the house leading to the 3R2C model.
The equation of the model 3R2C:

\[
\begin{bmatrix}
\dot{T}_a \\
\dot{T}_m
\end{bmatrix} = 
\begin{bmatrix}
\frac{1}{K1 + 1/K3} & \frac{1}{K3C1} \\
\frac{1}{K2 + 1/K3} & \frac{1}{C1}
\end{bmatrix}
\begin{bmatrix}
T_a \\
T_m
\end{bmatrix} + 
\begin{bmatrix}
\frac{1}{K1C2} & \frac{1}{C2} \\
\frac{1}{K2C1} & 0
\end{bmatrix}
\begin{bmatrix}
Te \\
P_{input}
\end{bmatrix}
\]

The differential equation (1) is then solved with Matlab. GV is a coefficient, which links the thermal losses, the outdoor and the indoor air temperature respectively Te and Ta:

\[
\text{Thermal heat losses} = GV(Te-Ta)
\]

The advantage of this model is to give an analytical expression of the coefficient GV:

\[
GV = \frac{1}{K1} + \frac{1}{K2 + K3} + m_{air}^c \text{pair}
\]

In this study: K1=1.873 \times 10^{-2} K.W^{-1}, K2=1.485 \times 10^{-3} K.W^{-1}, K3=2.487 \times 10^{-3} K.W^{-1}, C1=5.590 \times 10^{6} K.W^{-1}, C2=7.163 \times 10^{7} K.W^{-1}, V=396.15 m^3, air mass flow rate \( m_{air} \) =0.0468 kg.s^{-1}.

With those assumptions, GV=351 W.K^{-1}, leading to thermal losses of 12.5 kW for the coldest day. The evolution of the heat losses as a function of the outdoor air temperature is presented on Figure 4.

Note: For the results shown on Figure 4, the indoor air temperature is fixed at 20 °C and the climatic conditions are those of Trappes, a city in France, from the 1st October to the 20th May.

![Figure 4: Thermal heat losses of a dwelling with GV=351 W/K.](image)

![Figure 5: Evolution of the indoor air temperature.](image)

Figure 5 shows the inertia of the house: 8 days are required to reach an indoor air temperature of 10 °C when the initial indoor air temperature is equal to 20 °C and the outdoor air temperature is equal to 10 °C without heating. It is possible to calculate the total heating needs on a heating season by integrating the thermal heat losses on the heating season.

\[
\int_{1^\text{st} \text{October}}^{20^\text{th} \text{May}} (Ta - Te) dt
\]

A heating demand of 26 MWh is required on a heating season to ensure an indoor air temperature of 20 °C.

### 2.1.2 Emitters: Radiator

According to the standard NF-EN-442-2 (NF-EN-442-2, 1997), the heating capacity provided by a radiator is calculated by the following formula:

\[
P_{radiator} = KSA\Delta Tm^n
\]

This formula expresses that the heating capacity emitted by the radiator depends on the mean temperature difference between water and indoor air temperature. The influence of water flow rate on the heating capacity emitted by the radiator is negligible. For cast iron radiators, the emission coefficient, n, is equal to 1.31.

For a nominal condition, T water at the inlet = 75 °C, T water at the outlet = 65 °C, Ta=20 °C, the radiator provides a heating capacity of 17650 kW. So the coefficient KS = 105 W/K is considered constant. Thus it is possible to calculate the temperature at the outlet of the radiator for other conditions.
2.1.3 Weather

To calculate the HSPF, Simbad (CSTB, 2004), a program developed by the CSTB under Matlab, has been used. This program includes a database of French cities hour per hour. In this study, the calculation has been done on a heating season, from the 1st October to the 20th May. On Figure 4, the distribution of the outdoor air temperature for the city of Trappes (near Paris) is represented. It can be noticed on Figure 4, that the temperature is below -5 °C for only a few days.

2.1.4 Heat pump: evolution of heating capacity and input power in relation to the water temperature at the condenser inlet and the outdoor air temperature

The input power and the heating capacity of each heat pump have been expressed by a polynomial formula using the water temperature, the heat pump inlet, and the outdoor air temperature.

\[ \text{Heating capacity} = c_0 + c_1 T_{w\_in} + c_2 T_e + c_3 T_e T_{w\_in} + c_4 T_e^2 + c_5 T_{w\_in}^2 + c_6 T_e^3 + c_7 T_{w\_in}^3 \]  \hspace{1cm} (6)

\[ \text{Input power} = c_0 + c_1 T_{w\_in} + c_2 T_e + c_3 T_e^2 + c_4 T_{w\_in}^2 + c_5 T_e^3 + c_6 T_{w\_in}^3 \]  \hspace{1cm} (7)

3 Calculation of HSPF

The program Simbad of the CSTB allows using a certain number of "boxes" under Matlab such as the weather, the building, the hydronic system, the heat pump, and the emitter, Figure 6.

![Diagram of the system simulated under Matlab.](image)

The set point, for the heat pump control, corresponds to the water temperature at the condenser inlet. It is fixed thanks to the heat curve (Figure 7), which links the outdoor air temperature and the temperature at the condenser inlet. The heat curve characterises the temperature required for the hydronic circuit. The temperature at the condenser inlet is fixed at the set point ±1.5 K. According to this controlled temperature the heat pump is operating or not.
The heat curve can easily be found with a balance equation between radiators and the dwelling, the heat curve is represented on Figure 7 and is calculated using formula (8).

\[ T = f(\text{Te}) = T_a + \left( \frac{G V}{K S} \right) \left( \frac{1}{n} + \frac{G V}{2 \text{m}_w c_p w} (T_a - \text{Te}) \right) \]  

(8)

In this case, the water volume flow rate is fixed to 850 L.h⁻¹, which corresponds to a difference of temperature of 12.5 K when the heating capacity is equal to 12 kW. Assuming a constant mass flow rate of the heating water, the water temperature at the condenser outlet is higher than the required supply temperature, however with a second compressor (HP1 Figure 7) with a lower capacity, the temperature at the condenser outlet comes closer to the required temperature. Thus the number of on/off cycle is reduced. When the COP is calculated according to the heat curve, it is possible to see (Figure 8) that for an outdoor air temperature higher than 0°C the HP1COP is higher compared to the compressor COP with the injection port (HP2).

The next step is to calculate the HSPF by using the model developed under Matlab. Two cases are compared:
- 1ˢᵗ case: the heat pump with the injection port compressor (HP1)
- 2ⁿᵈ case: the heat pump with the injection port compressor and the classical compressor in parallel (HP1 + HP2), and there is a switch to HP2 when the outdoor air temperature is above 0 °C. The results of the simulations are summarized in Table 1.

On Figure 9, the influence of the inertia of the hydronic network is seen: when the heat pump switches off, the water temperature at the condenser inlet is still increasing and when the heat pump switches on, the water...
temperature at the condenser inlet is still decreasing. Note: the volume of water in the hydronic network is of 70 L.

![Figure 9: Sample on a range time of 1 hour of the evolution of the water at the inlet and at the outlet of the heat pump, the indoor air temperature, the temperature of outdoor air and the on/off of the compressor.](image)

**CONCLUSIONS**

This study shows (Table 1) that by adding a single compressor in parallel to an injection port compressor, it is possible to gain 8.3% on the HSPF.

<table>
<thead>
<tr>
<th></th>
<th>HSPF</th>
<th>Total heating capacity on the heating season</th>
<th>Mean indoor air temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>HP1</td>
<td>2.63 ±0.5%</td>
<td>27.7 MWh</td>
<td>20.3 ±0.4</td>
</tr>
<tr>
<td>HP1 + HP2</td>
<td>2.84 ±0.5%</td>
<td>26.3 MWh</td>
<td>19.7 ±0.8</td>
</tr>
<tr>
<td><strong>Gain</strong></td>
<td>+8.3%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In order to estimate if this gain is cost effective, a calculation has to be done in order to verify if the over cost of the second compressor is rapidly paid back by the gain on the HSPF. A point has not been addressed in this paper: the heat exchangers, which have designed for two compressors are oversized when the single compressor is operating, leading to a possible energy gain, which has not been evaluated here. In conclusion, the energy gains as calculated are possibly underestimated.

**NOMENCLATURE**

- $c_p$: thermal capacity of water (J.kg$^{-1}$.K$^{-1}$),
- $C_1$: heat capacity representative of width walls (J.K$^{-1}$),
- $C_2$: heat capacity representative of the inside air, the light walls (J.K$^{-1}$),
- $G_V$: heat loss for a difference of 1 K between the outdoor and the indoor air temperature (W.K$^{-1}$),
- $K$: global heat exchange coefficient of emitter (W/K.m$^2$),
- $K_1$: thermal resistance representative of the direct transfer between the inside and the outside through the light walls (doors and windows) (K.W$^{-1}$),
- $K_2$: thermal resistance representative of transfer between the width walls and the outside air (K.W$^{-1}$),
- $K_3$: thermal resistance representative of transfer between the inside air and the width walls (K.W$^{-1}$),
- $m$: mass flow rate (kg.s$^{-1}$),
n: emission coefficient of the emitter,
P: thermal heat loss (W),
P_{input}: heating gains including: the heating system, the solar gains, and ventilation (W),
P_{radiator}: capacity exchange between the water of the emitter and the room (W),
S: surface of emitter (m²),
Ta: indoor air temperature (°C),
Te: outdoor air temperature (°C),
Tm: average wall temperature (°C),
Tw_in: water temperature at the condenser inlet (°C),
T_w: temperature of water at the emitter inlet (°C),
T: temperature of water at the emitter outlet (°C),
V: volume of the house (m³),
∆Tm: mean temperature difference between water and air interior temperature (K)

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