Performance evaluation of an air-to-air heat pump coupled with temperate air-sources integrated into a dwelling

Bruno Filliard, Alain Guiavarch, Bruno Peuportier

PERFORMANCE EVALUATION OF AN AIR-TO-AIR HEAT PUMP COUPLED WITH TEMPERATE AIR-SOURCES INTEGRATED INTO A DWELLING

Bruno Filliard, Alain Guiavarch, and Bruno Peuportier[1]
MINES ParisTech, CEP – Center for Energy and Processes,
60, boulevard Saint-Michel 75272 Paris Cedex 06, France
[1]e-mail: bruno.peuportier@mines-paristech.fr, Tel.: +33 1 40 51 91 51

ABSTRACT

An inverter-driven air-to-air heat pump model has been developped and implemented in the thermal simulation tool COMFIE, in order to compare the seasonal performance of a variable capacity air-to-air heat pump coupled with temperate air sources (crawlspace, attic, sunspace, heat recovery ventilation, earth-to-air heat exchanger) with the performance of a conventionally installed heat pump. The empirical model of the heat pump is presented in this paper, including full-load and part-load model at rating and non rating conditions, and also frosting conditions.

Several coupling configurations are studied and applied on a case study: a French typical residential house. The influence of the climatic region is evaluated, giving indications on energy saving using such systems.

INTRODUCTION

The building sector generates impacts on the environment, being for instance the first energy consumer sector in France responsible for up to 23% of the global CO_2 emissions of the country (ADEME, 2006). Reducing these impacts is possible by combining technical solutions to improve the energy efficiency of buildings (e.g. high insulation level, efficient ventilation with heat recovery system) and to use renewable energies.

Besides, the air-to-air heat pump market is in expansion in Europe and France (Forsén et al., 2008). Air-to-air heat pumps save energy compared to e.g. direct electric heating (Karlson et al., 2006), but the global energy including electric plant efficiency is questioned.

An air-to-air heat pump performs poorly at low ambient temperatures, and its performance decreases in the coldest part of winter when the heating load is greater. Both the COP and the heating capacity of the system decrease as the outdoor temperature decreases. Auxiliary resistance heaters must then be used to provide the reduced capacity, reducing the overall performance of the system. Moreover, defrost cycles are necessary in a specific range of outdoor conditions, lowering the heat pump performance. This is of particular interest for electricity utilities, because peak loads generally coincide with the coldest ambient temperatures. Attempts to improve the heat pump efficiency and to reduce the peak-loading problem have led to the coupling of heat pumps with milder temperature air-sources (Ternes, 1980), (Smith et al., 1981) (Ternes, 1982), (Wasserman et al., 1983), (Wasserman et al. 1984) (Lubliner et al., 2007) (Lazzarin et al., 1986) (SØrensen, 2002) (Hastings et al., 2000) (Zondag, 2003). The literature review shows that the main investigations have examined two types of configuration, one using crawlspace, the other using air-collectors.

But different components of a building envelope can contribute to increase the heat pump's heat-source temperature, as for instance sunspaces or attics, and to some extent crawlspaces and earth-to-air heat exchangers. Other sources such as the exhaust air of a heat recovery ventilation system can be used to preheat the heat pump air-source. This allows the heat pump to operate in more favourable conditions and with a higher efficiency.

A global approach is needed to explore the energy potential of such combination and evaluate the relevance of coupling an air-source heat pump with milder air sources from the building environment. Ambient conditions and solar energy both influence the heating load and the air-source temperature that interact directly with the heat pump running conditions and performance. Besides, the heat pump running conditions, and particularly the air flow rate conditions needed at the outdoor unit, interacts directly with the air flow rate of the milder source, influencing the building heating load.

The global model used to investigate the coupling question is presented. The building model is briefly presented, whereas the heat pump model is described in detail. Then, different coupling configurations (Figure 6.a-6.c) are studied in the case of a French typical dwelling, and typical climates. At last, the results are exposed and then discussed.

BUILDING AND HEAT PUMP MODELING

Building model

The thermal dynamic simulation software COMFIE is a multi-zone simulation tool based on the finite volume method, reduced after modal analysis (Peuportier et al., 1990). It can be applied on many types of buildings (single-family houses, appartment buildings, tertiary buildings).

The tool enables to model multiple components of a building envelope such as crawlspaces, attics, sunspaces or solar air-collectors. It also integrates efficient ventilation systems with heat recovery . An earth-to-air heat exchanger module has been developped to offer the possibility to study load reduction and comfort effect of such systems during winter or summer (Thiers et al., 2008).

Heat pump model

The heat pump model presented in the following sections is based on a steady-state empirical model. The steady-state behaviour of the heat pump model can be justified through the building dynamic behaviour. The heat pump time constant is around a few or ten minutes. This time constant is small compared with the building time constant, which is around several hours. Thus, the behaviour of the heat pump on a hourly basis, can be considered as a sequence of steady-state behaviours. The heat pump model considers full load and part load conditions (Roujol et al. 2003).

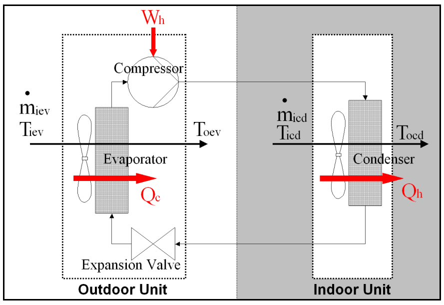

Figure 1 – Heat pump energy balance

A first set of equations is used to calculate the full load performance and a second set is used to correct the result of the first set and deduce the part load performance. A third model accounts for performance degradation due to frost formation on the outdoor exchanger. The whole model is then complemented with a control model, to take into account changing the outdoor fan speed at part load conditions in the case of an inverter-driven heat pump. This is of a particular interest when milder air sources are used.

1. Full-load model

The full load heating capacity $Q_{h,fl}$ given by equation (1) is function of the rating value of the capacity $Q_{h,rat}$ and of the air inlet temperature at both indoor and outdoor heat exchangers (at rating and non rating conditions). The rating conditions are those imposed by standard NF EN 14511 (Standard, 2004) for heat pump performance measurements. It indicates the net heating capacity at full load conditions and the absorbed energy rate $W_{h,fl}$ which includes compressor and both indoor and outdoor fans consumption. Rating conditions are as follows: 20°C air inlet temperature at the indoor unit, while air inlet temperature at the oudoor unit is 7°C.

$$\frac{Q_{h,fl}}{Q_{h,rat}} = 1 + A_1(T_{icd} - T_{icd,rat}) + A_2(T_{iev} - T_{iev,rat}) \qquad (1)$$

Parameters A_1 and A_2 of the heating capacity model are fitted by the least square method using manufacturers' data from a variable capacity air-to-air heat pump.

The second equation determines the Coefficient Of Performance ($COP_{h,fl}$), from the rating value COP_{rat} . It's an second-order polynomial function depending on air inlet temperature in both indoor and outdoor heat exchangers, according to equations (2) and (3).

$$\frac{1}{COP_{h,fl}} = \frac{W_{h,fl}}{Q_{h,fl}} = \frac{W_{h,rat}}{Q_{h,rat}}.\left(1 + B_1.\Delta T + B_2.\Delta T^2\right) \qquad (2)$$

where

$$\Delta T = \frac{(T_{iev} + 273.15)}{(T_{icd} + 273.15)} - \frac{(T_{iev,rat} + 273.15)}{(T_{icd,rat} + 273.15)} \qquad (3)$$

The parameters B_1 and B_2 are also determined by the least square method using manufacturer's data. Two sets of parameters are calculated according to the shape of the COP curve in terms of the cold source temperature. One set of parameters for outdoor temperatures lower than 6°C and another set of parameters for outdoor temperatures upper than 6°C, to take into account the slope change due to frosting formation on the evaporator.

Coefficients of determination (R^2) from the heating capacity and the COP model regression versus manufacturers' data have been calculated and are respectively 0,93 and 0,95. Moreover, the coefficients of variation (CV) have also been calculated and are respectively 2,5% and 15%. Both coefficients indicate that the full load model fits well with the manufacturers' data at full load conditions.

2. Part-load model

There are two main ways to control the heating capacity of a heat pump and reach the building heating load when the full-load heating capacity is greater than the building load : on/off cycles and inverter driven compressors. The model presented in this paper corresponds to an inverter-driven compressor.

A few empirical models exist in the literature to take into account the part-load running effect on the heat pump performance in on/off cycle e.g. (Parken et al., 1977), (Bettanini et al., 2003). But very few papers consider the part-load effect on inverter-driven heat pumps (Marchio et al., 2003), (Bettanini et al., 2003).

The performance of heat pumps at part-load conditions are not given in manufacturers' catalogues. A test procedure has been launched by the Swedish Energy Agency (Energimyndigheten) in partnership with the Technical Researh Institute of Sweden (SP). Since 2004, the SP Institute has tested and compared more than twenty air-to-air heat pumps at full load, but also at part load conditions. In addition to the classical rated performance (7°C ambient air, 20°C indoor air at full-load conditions) announced by the manufacturers, the SP Institute gives information on heat pumps performances at 50%, 75% and 100% of the full heating capacity. They also measure the full load performance at non-rating conditions (2°C ambient air, 20°C indoor air), but also at full and 50% part-load running conditions. These figures are available online on the website of the Swedish Energy Agency (Johansson, 2006) and are partly presented in figure 2. A linear function of the part load factor (PLF) has been deduced from these tests to develop the part-load model of the inverter-driven heat pump.

The part-load model for inverter-driven heat pumps is composed of three part as seen in figure 2 and is compared to a conventional on-off controled heat pump:

- On/off control when the part load ratio (PLR) is lower than 30%;
- Inverter control for PLR from 30% to 100%;
- Inverter control over 100% of the heating capacity .

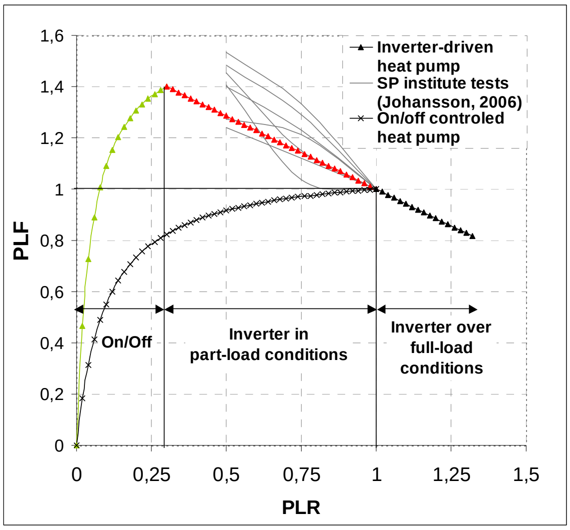

Figure 2 – PLF versus PLR

The PLR is defined as follow:

$$PLR=\frac{BHL}{Q_{h,fl}} \qquad (4)$$

where BHL is the building heating load, and $Q_{h,fl}$ is the heating capacity at full load.

The part-load factor (PLF) in the case of on/off control is given by equation (5):

$$PLF=\frac{COP_r}{COP_{rat}}=\frac{PLR}{PLR+\alpha} \qquad (5)$$

where α is the standby power fraction defined as follows:

$$\alpha=\frac{W_{sb}}{W_{h,rat}} \qquad (6)$$

where W_{sb} is the standby power input, and $W_{h,rat}$ is the total power input of the heat pump, including compressor and fans consumption.

In the case of an inverter control there is a linear relationship between PLF and PLR:

$$PLF=a.PLR+b \qquad (7)$$

This linear relationship takes place for PLR from 30% to 100%, and is assumed to be valid for PLR up to 130%. The trend from 30% to 100% has been deduced from (Johansson, 2006). The part over 100% until 130% is an hypothetic trend, assuming a linear relationship for this interval.

3. Defrost cycle

Frost accumulation on the evaporator exchanger occurs from around 7°C to 3°C, depending on the air humidity. In such conditions, defrosting cycles take place, leading to a significant reduction of the COP and the heating capacity (Rivière et al., 2007).

A simple model is applied to the heating capacity through a degradation coefficient (Cd) to take into account the defrost cycle impacts on the heat pump performance. It is based on (Schibuola , 2000) experiments (see model-1 in figure 3) and is assumed to equal a constant value of 0,9.

$$T_{iev} < 6°C: \qquad Q_{h,r}(T_{iev})=Cd.Q_{h,fl}(T_{iev}) \qquad (8)$$

$$T_{iev} > 6°C: \qquad Q_{h,r}=Q_{h,fl} \qquad (9)$$

where $Q_{h,r}$ is the real heating capacity of the heat pump accounting for defrost degradation, and Cd is the coefficient of degradation due to defrost cycles. An illustration of the impact of defrosting cycles on the calculated heating capacity is given in figure 3.

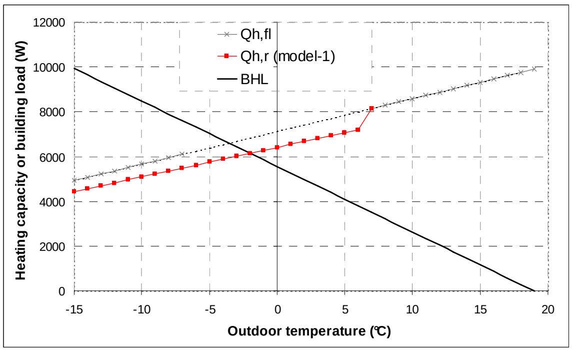

Figure 3 – Defrost effect on full load heating capacity

4. Back up resistance

The studied heat pump is equipped with two electric back up resistances sized according to the heat pump (e.g. two units of 1,25 kW for a 4 kW heating capacity heat pump).

These resistances are used when the heat pump capacity is too low to reach the building heating load. This can happen at very low outdoor temperatures, even if the inverter is set to 130% of the full load capacity, the heating capacity required is not sufficient. In this case, a first stage is turned on, and if it is not sufficient, then both units are used.

5. Air flow control

This model has been developed in the particular case of inverter-driven heat pumps. The outdoor fan speed varies according to the frequency of the compressor. When the compressor frequency increases, the outdoor fan speed is also changed to high speed, and inversely.

The model is based on the first principle application on the heat pump system and the assumption of a constant air temperature difference through the evaporator at any air flow rate, that leads to equation (13).

$$\dot{m}_{iev} = \dot{m}_{iev,rat} \cdot \frac{Q_{c,r}}{Q_{c,rat}} \qquad (13)$$

The heat exchange through the evaporator is given by equation (14).

$$Q_{c,rat} = Q_{h,rat} - W_{h,rat} \qquad (14)$$

6. Auxiliary fans

When the outdoor unit of the heat pump can't be directly placed into the space used as a milder source, the use of additional fans is required to move the air until the outdoor unit.

The consumption of additional fans is taken in account in the overall coefficient of performance of the global system, assuming that the fan consumption is proportional to the air flow rate, considering two cases:

- Standard fan: 0,25 $W.m^{-3}.h^{-1}$
- Low consumption fan: 0,1 $W.m^{-3}.h^{-1}$

CASE STUDY

Description of the dwelling

The studied house is a single family house with a useful area of 135 m², and occupied by 4 inhabitants. The living area is composed of a hall, a living-room, a kitchen, a bed room, and a bathroom downstairs, and three bedrooms, and a second bathrom upstairs (figure 4 and 5). This house was built in the seventies, and fully retrofitted. The thermal envelope characteristics are given below. The ground-floor external walls are made of (from the external to the internal face) a 4.5 cm concrete layer, 10 cm of expanded polystyrene, and 5 cm of plaster panel. The first-floor structure is made of a 5 cm rigid sandwich panel of polyurethane foam, 12 cm of expanded polystyrene, and a 1 cm plaster panel.

The pitched-roof wall is insulated with 18 cm of mineral and 1 cm of sheetrock panel.

The 20 cm concrete slab on the crawlspace is well insulated by 20 cm of expanded polystyrene to avoid heat loss from the house to the crawlspace.

All windows are double glazed with low emissivity glass. The U-values of the main building envelope elements previously described are given in table 1.

Table 1 – U-values of the dwelling envelope elements

WALL TYPE	**U-VALUE** ($W.m^{-2}.K^{-1}$)
External ground-floor wall	0,34
External first-floor wall	0,15
External pitched-roof wall	0,22
Slab	0,19
Upper ceiling	0,20
Windows	1,2 (g-value=0,72)

A sunspace facing south has been added with low emissiviy double-glazing. An earth-to-air heat exchanger (EAHE) has also been modeled. It is composed of one 50 meter long polyvinyl chloride pipe (diameter: 200 mm) buried at 2 meters depth in the soil providing a 500 $m^3.h^{-1}$ air flow rate.

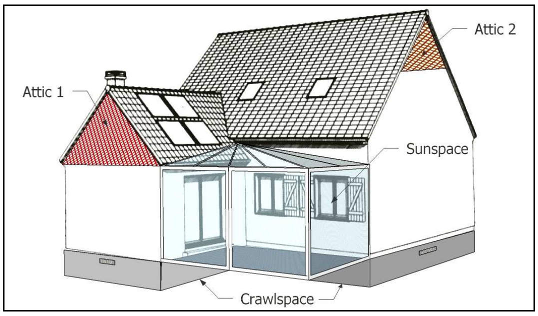

Figure 4 – The studied house

The total air change rate of the heated area is 0.6 $vol.h^{-1}$, provided up to 0.5 $vol.h^{-1}$by a mechanical heat recovery ventilation (HRV) system with a rated efficiency of 75%. The remaining 0.1 $vol.h^{-1}$ is provided by infiltration with an efficiency of 0%. This leads to a global efficiency of the HRV of 62%, including air infiltration.

The volume and the infiltration air change rates of the milder air-sources are given in Table 2.

If the outdoor unit is placed in one of these spaces, the air change rate is adapted to satisfy the air change rate needed to proper running conditions of the heat pump. The conservation of mass is strictly observed, that is to say that all incoming air flow including infiltration, equals the outcoming one.

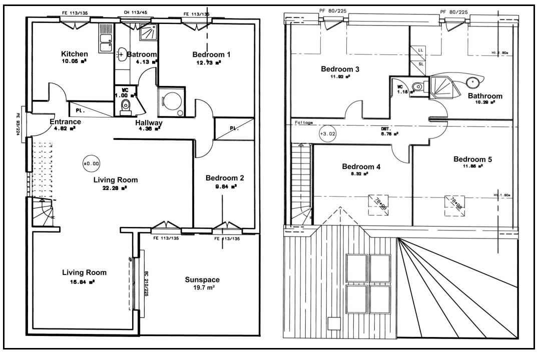

Figure 5 – First floor (left) and second floor (right) plans

Table 2 – Volumes and infiltration air-change rates of the milder air-sources

UNHEATED SPACE	**VOLUME** (m^3)	**INFILTRATION AIR CHANGE RATE** ($vol.h^{-1}$)
Crawlspace	44	1
Attic1	19	1
Attic2	46	1
Sunspace	49	0.2

Coupling configurations

With the global model previously described, the seasonal performance of an air-to-air heat pump coupled with temperate air sources is compared to the performance of a conventionally installed heat pump (Figure 6.a).

Two coupling configurations are studied as illustrated on figures 6.b and 6.c:

- First configuration: the heat pump outdoor unit is placed outside near the perimeter of the crawlspace (1) and uses the crawlspace air as a heat-source, sending outside the air from the crawlspace after having exchanged heat with the evaporator. The air in the crawlspace can be pre-heated by other sources: the exhaust air from the ventilation system (4), the air from the sun space (3) when its temperature is greater than the crawlspace one, or the preheated air from a earth-to-air heat exchanger (5).
- Second configuration: the outdoor unit is placed in the attic (2), which can use the exhaust air coming from the ventilation system (4), or the sun-space air (3).

Two types of heat pumps are used and examined in these coupling configurations: inverter-driven and on-off controlled air-to-air heat pumps. Their rating characteristics are as follows:

- Inverter: 4 kW heating capacity, 2 stages of 1,25 kW back up electric resistance, the air flow rate at the outdoor unit varies from 500 $m^3.h^{-1}$ to 1250 $m^3.h^{-1}$ according to the PLR.
- On-off: 4 kW of heating capacity, 2 stages of 1,25 kW back up electric resistance, 1250 $m^3.h^{-1}$ constant air flow rate at the outdoor unit.

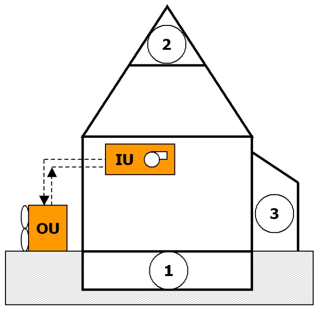

Figure 6.a – Conventionally installed heat pump

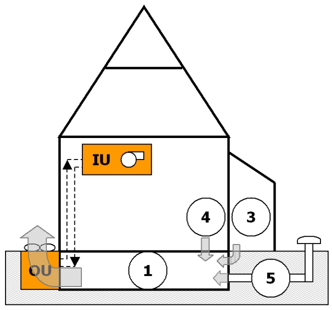

Figure 6.b - First configuration (1)=(3)+(4)+(5)

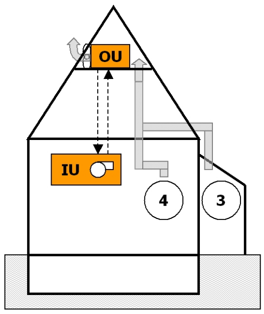

Figure 6.c – Second configuration (2)=(3)+(4)

RESULTS AND DISCUSSION

The thermal simulation software enables to study the interest of using each envelope component as a heat-source for the heat pump. An example of the simulated air temperatures for the coldest week in Trappes (near Paris) is given in figure 7.

The graph below shows the different sources' potential. The temperature difference between the air-source and the ambient air can vary from around 2°C for the attic to more than 10°C for the sunspace.

However, the potentiality varies according to the envelope characteristics (especially with the thermal mass) and the air-change rates of the source.

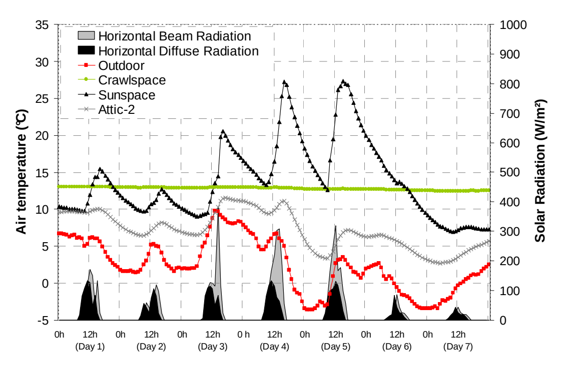

Figure 7 – Milder air-sources temperature for the coldest week of a typical year in Trappes climate (near Paris)

When the heat pump is placed near the heated space, the air flow rate required at the outdoor unit leads to an increase of the air-change rate in the milder source, lowering the air-source temperature and increasing the building heating load (BHL) as it can be seen in figure 8. However, the rise of the BHL can be limited by a good insulation of the connecting wall.

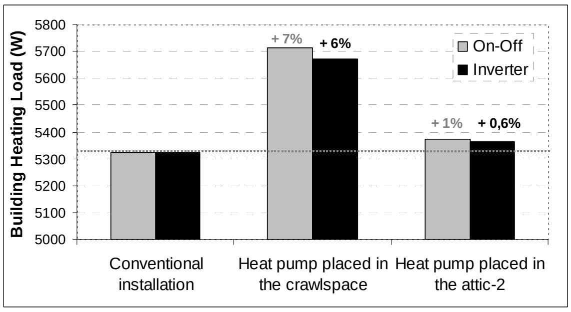

Figure 8 – Building heating load according to the configuration

For both inverter and on-off controlled heat pumps the increase of the BHL is up to 7% when the heat pump is placed in the crawlspace, and hardly 1% in the attic.

But the heat pumps are running using a warmer heat source. The heat capacity delivered by the heat pump is increased, while the consumption of the system is lowered leading to a COP increase. According to figure 9, when the inverter-driven heat pump is placed in the crawlspace (having a higher thermal mass than the attic) the total annual consumption is reduced by up to 12%, and only by 3,5% if the heat pump is placed in the attic. In both cases, the use of milder air sources lowers the use of back up resistances.

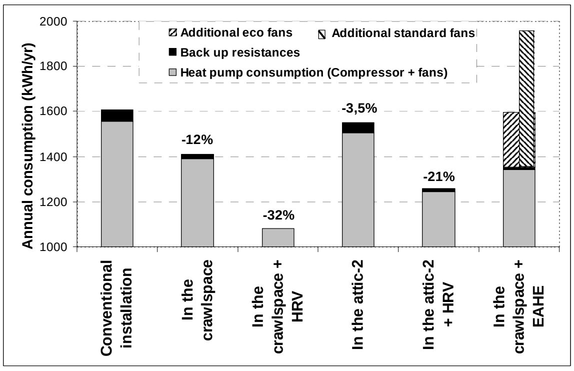

Figure 9 – Inverter-driven heat pump annual consumption in different configurations in Trappes.

The use of back up resistances can even be avoided if the exhaust air of the heat recovery system (HRV) is blown directly into the crawlspace or in the atttic. But the risk of condensation would have to be studied. These coupling configurations lead to significant energy saving. The annual global consumption of the system is respectively lowered up to 30% and 21%.

The use of an earth-to-air heat exchanger (EAHE) can also lower the consumption of the heat pump and the back up resistance. However, in the later case, the use of additional fans is required, leading to increased global consumption according to the fans technical characteristics. When a low consumption fan is used in the EAHE, the global energy balance of the coupled system is equal to the conventional installed system.

The respective impacts on the seasonal COP can be seen in figure 10. When the heat pump is placed directly in the crawlspace or in the attic, the consequences on the COP are respectively an increase of 21% and 4%.

In the case of the crawlspace pre-heated by the HRV, as previously explained, the annual global consumption is lowered up tho 30% (figure 9), while the COP is increased up to 50% compared to the conventional installation (figure 10).

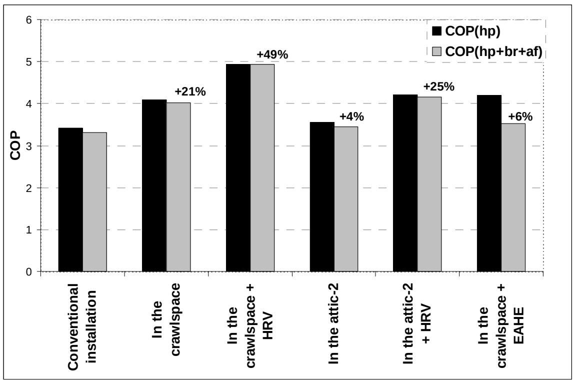

Figure 10 – Inverter-driven annual COP (heat pump only; heat pump + back up resistances + additional fans) in Trappes.

In the case of the crawlspace pre-heated by the EAHE, the consumption of the heat pump is lowered, but the global balance depends on the fans characteristics, with at least a 6% global COP increase in the case of an low consumption fan. With the use of standard fans, the coupling configuration is not even interesting. Nevertheless these results greatly depend on the EAHE sizing and characteristics.

The comparison on different climates requires adequate sizing of the heat pump, according to the building load:

- Nancy (East of France, 3024 Celsius degree days): 6 kW rating heating capacity, $1800m^3.h^{-1}$ rating air flow at the outdoor unit ;
- Trappes (near Paris, 2704 Celsius degree days): 4 kW rating heating capacity, $1240m^3.h^{-1}$ rating air flow at the outdoor unit ;
- Nice (mediterraean climate, 1359 Celsius degree days): 3 kW rating heating capacity, 900 $m^3.h^{-1}$ rating air flow at the outdoor unit .

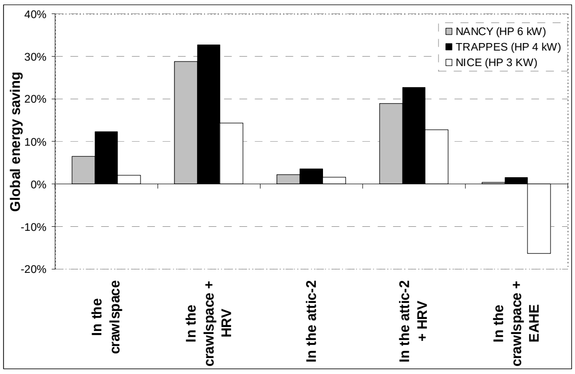

Figure 11 - Annual energy saving for different climates (Nancy, Trappes, Nice) compared to a conventional installation.

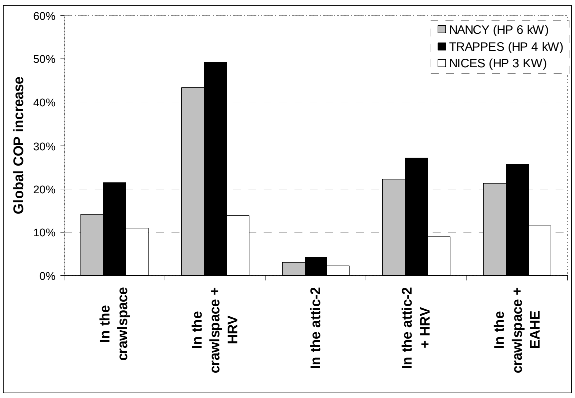

Figure 12 – COP increase for different climates (Nancy, Trappes, Nice) compared to a conventional installation.

The simulation results, presented in figure 11 and 12, show that significant energy saving and seasonal performance improvement can be achieved for cold climates (e.g. a 50% increase of the COP and 30% of energy saving in the case of a crawlspace preheated with a HRV). Thus, all coupling results presented here are more relevant in cold than in hot climates.

CONCLUSION

A heat pump model has been developed and implemented in a building simulation tool to assess the relevance of coupling an inverter-driven air-to-air heat pump with milder air-sources from the building envelope.

The interest of using milder sources as a heat source for a heat pump depends on their volume and thermal mass (eg. the crawlspace, having a higher thermal mass than the attic, seems to be a better source). The larger is the space with higher thermal mass, the better is the performance of the heat pump.

According to these simulation results coupling is more interesting for inverter-driven than for on-off controlled heat pumps. This is mainly due to the fact that the air flow at the outdoor unit is controlled according to the heating capacity delivered by the heat pump.

Moreover, when a heat pump is placed near the heated space, the building heating load is increased. However, the heat pump benefits from a milder heat source, and can deliver more heating capacity for less consumed energy. This can lead to a significant increase of the annual coefficient of performance.

Characteristics of the climates have a great influence on the results. It appears that coupling is more relevant in cold than in temperate climate.

PERSPECTIVES

The heat pump model has not yet been validated on experimental data, but measurements are expected in the coming months to validate particularly the part-load and the defrost cycle models.

Extended configurations using solar air collectors or PV panel to preheat the air of the attic have to be examined. Such configurations could improve both heat pumps and PV panels performance.

NOMENCLATURE

SYMBOL	MEANING	UNIT
α	Standby power fraction	%
A_1, A_2, B_1, B_2, a, b	Regression coefficients	-
BHL	Building heating load	W
Cd	Defrosting degradation coefficicient	-
Cp_{air}	Air specific heat	$J.kg^{-1}.K^{-1}$
CV	Coefficient of variation	-
$_{fl}$	Full-load running conditions	-
g-value	Average window solar factor including the frame	
$_{h}$	Heating mode	-
$_{icd}$	Air inlet at the condenser	-

$_{iev}$	Air inlet at the evaporator	-
$\dot{m}$	Air flow rate	$kg.s^{-1}$
PLF	Part load factor	-
PLR	Part load ratio	$W.W^{-1}$
Q_c	Cooling capacity at the evaporator	W
Q_h	Heating capacity at the condenser	W
$_r$	Real capacity or COP	-
R^2	Coefficient of determination	-
$_{rat}$	Rating conditions	-
T	Temperature	°C
W_h	Input power in heating mode	W

REFERENCES

ADEME, 2006. Les chiffres clés du bâtiment. From the French Environment and Energy Management Agency. (in french).

Bettanini, E., et al., 2003. Simplified models to simulate part load performances of air conditioning equipments. 8th International IBPSA conference, Eindhoven, Netherland, August 11-14, 2003.

CSTB, 2006. Méthode de calcul Th-CE, Version 7.3. The French Scientific Center for Building Physics. (in French)

Forsén, M., Nowak, T., Ransquin, J., 2008. European Heat Pump Statistics: Outlook 2008. Version 1.5. EHPA (European Heat Pump Association) Publication.

Hasting, S.R., Mørck, O., 2000. Solar air systems, A desgin Handbook. Solar Heating and Coolling Executive Committee of the International Energy Agency. Published by James & James (Science Publishers) Ltd, London, UK.

Johansson, H.A., 2006. Energimyndigheten testar: Luft-luftvärmepumpar. Swedish Energy Agency. <http://www.swedishenergyagency.se/web/biblshop.nsf/FilAtkomst/ET2006_56.pdf/$FILE/ET2006_56.pdf?OpenElement >

JRAIA, 2006. JRA 4048: Annual Performance Factor of the package air conditioners. JRAIA, The Japan Refrigeration and air conditioning industry association. Established on february 23, 2001, revised on march 27, 2006.

Karlson, S., Lidbom, P., Axell, M., Lindberg, U., 2006. Air-air heat pumps evaluated for Nordic circumstances. IEA Heat Pump Centre newsletter 24, no.24-04, Retrofit heat pumps for buildings, 2006, pp. 18-25.

Lazzarin, R.M., Schibuola, L., 1986. Energy Roof as Heat Pump Source/Sink. International Journal of Refrigeration, vol. 6., pp.108-112.

Lubliner, M., Hadley, A., Parker, D., 2007. HVAC Improvements in Manufactured Housing Crawlspace-Assisted Heat Pumps. Ashrae Transactions, vol. 113, part 2.

Marchio, D., Filfli, S, 2004. SPLIT - Modèle simplifié d'un système Split. Version 1.2, Centre énergétique et procédés, Ecole des Mines de Paris, novembre 2004. <http://www-cep.ensmp.fr/francais/themes/syst/pdf/modeles_de_systemes/SPLIT_V1-2.pdf> (in French)

Peuportier, B., Blanc-Sommereux, I., 1990. Simulation Tool with Its Expert Interface for the Thermal Design of Multizone Buildings. International Journal of Solar Energy, Vol. 8, pp. 109-120.

Riviere, P., et al., 2007. Preparatory study on the environnemental performance of residential room air conditioning appliance (airco and ventilation) for the EcoDesign European Project. Technical analysis of existing products. Draft report of Task 4 (V4). October 2007

Roujol, S., Marchio, D., 2003. Few parameter chiller model for an hourly based evaluation of annual building energy consumption. EUROVENT-CECOMAF Review, February 2003, no. 50.

Schibuola, L. 2000. Heat pump seasonal performance evaluation: a proposal for european standard. Applied Thermal Engineering 20 (2000): 387-398

Smith, L.O., Reid, R.L., 1981. Evaluation of a Ground Coupled Air-to-Air (Crawl space) Heat Pump. Ashrae Transactions, vol. 87, part 2., pp.405-417.

Sørensen, H., 2002. Photovoltaics/thermal Solar Energy Systems. Status of the Technology and Roadmap for Future Development. Task 7. Report IEA PVPS T7-10 2002.

Standard, 2004. Air conditioners, liquid chilling packages and heat pump with electrically driven compressors for space heating and cooling. NF EN 14511. AFNOR 2004.

Ternes, MP., 1980. Crawl Space-Assisted Heat Pump. Progress Report, Oak Ridge National Laboratory, Oak Ridge, TN, USA.

Thiers, S., Peuportier, B., 2008. Thermal and Environmental Assessment of a Passive Buiding Equipped with an Earth-to-Air Heat Exchanger in France. Solar Energy, vol. 82, pp.820-831.

Wasserman, D., Reid, R.L., McGraw B.A., 1983. Performance Evaluation of a Crawlspace Heat Pump Installation. Ashrae Transactions, vol. 89, part 1a., pp.144-155.

Zondag, H.A., Combined PV-air collector as heat pump air preheater. Netherlands Energy Research Foundation ECN. <http://www.ecn.nl/docs/library/report/2002/rx02065.pdf>