AIR-SOURCE INTEGRATED HEAT PUMPS (AS-IHPS) WITH SMART ZONING FOR RESIDENTIAL SPACE AND WATER HEATING IN COLD CLIMATE

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ABSTRACT

This paper aims at assessing the energy savings potential of cold climate Air-Source Integrated Heat Pumps (AS-IHPs) for Canadian households. The AS-IHP studied in this paper consists of a highly efficient air source heat pump (ASHP), which can supply both space heating and cooling in extreme weather conditions, and of a heat pump water heater (HPWH) providing domestic hot water (DHW) that is integrated with the ASHP. The system features zoned air distribution to improve thermal comfort and energy performance. The simulation study using TRNSYS shows that the system results in annual energy savings of 42 % to 49 % compared to a conventional forced air system commonly installed in Canadian houses. Zoned systems deliver thermal comfort improvements in addition to energy savings.

INTRODUCTION

Canadian households accounted for 16% of the national secondary energy use in 2010, among which 63% were contributed by space heating and 17% by water heating (NRCan-OEE, 2013). Typical heating systems found in Canadian households include furnaces, boilers or electric baseboards, which have efficiency between 0.7 and 1. A national survey conducted by Natural Resources Canada in 2011 reveals that 88% of the households were using the aforementioned heating systems, while only 2.5% were utilizing heat pumps with a higher efficiency (NRCan, 2011). The vast majority of dwellings equipped with a central system employ a single thermostat, often located in a central location, to control the Heating, Ventilation and Air-Conditioning (HVAC) system for the whole house. This control method can lead to stratification between different floors, and significant temperature differences between rooms, causing thermal discomfort for occupants.

Cold climate air source heat pumps (CC-ASHP), which include Variable Refrigerant Flow (VRF) heat pumps, can be used in different zoning configurations and are a promising way to increase the energy efficiency of residential HVAC systems. The CC-ASHP is an advanced heat pump, which can operate in extreme weather conditions with high energy efficiency. A test of a CC-ASHP in Toronto, Ontario, demonstrated that the installed heat pump was capable to maintain thermal comfort at ambient temperatures as low as -24 °C and the simulation study under five Canadian cities showed that the average coefficient of performance (COP) of the CC-ASHP during heating season was 3 (Safa, 2012). A test of a CC-ASHP has been carried out in the Canadian Centre for Housing Technology (CCHT) in 2012 demonstrating that the heat pump could maintain thermal comfort when the temperature was below -21 °C with a COP greater than 1.5, delivering significant cost savings in areas supplied with low-cost electricity (Sager et al., 2013). A simulation-based assessment also showed that the annual energy savings of a CC-ASHP reached 41% compared with baseboard heating in the very cold climate of Yukon (Energy solutions centre, 2013). Another test performed in Connecticut, USA showed that the CC-ASHP had the capability of saving up to 70% of energy consumption compared with electric resistance heating (Bugbee et al., 2013). It should be noted that all these experimental and simulation studies did not investigate zoning configurations specifically.

A zoned space heating or cooling system can not only improve thermal comfort of each zone in a house, but also distribute the heating or cooling capacity of the system to the zones with higher loads. This improved thermal comfort and flexibility can also result in energy savings, but this is not always the case (Wilcox et al., 2011). A simulation study of a zoned cooling system based on the CCHT test house (see below) using ESP-r demonstrated that the zoning system with control strategy was effective to reduce and shift peak power demand (Lomanowski & Haddad, 2010).

OBJECTIVE

The objective of the paper is to assess the energy savings potential of a zoned Cold Climate AS-IHP system in a typical Canadian home through a simulation study using TRNSYS. The studied heat pump system is a VRF system zoned with 3 indoor units, each serving one floor in the house. A standard system composed of a gas furnace and a central air conditioner is used as the baseline case for comparison. A non-zoned VRF system with only 1 indoor unit is also investigated. The paper presents the selected house, the three systems and the custom TRNSYS components used for the study.

Performance comparisons are then presented in terms of annual energy consumption and thermal comfort.

The objective of this simulation study is to compare different systems in a typical North-American house using standardized (and simplified) assumptions on user behaviour. Modifications to the house structure (e.g. to add thermal mass) and more detailed user behavioral models are not addressed.

SELECTED HOUSES

The houses selected for the modeling work are the twin houses of the CCHT in Ottawa, Canada. They are identical, unoccupied houses built in 1998 to provide an experimental platform to assess the energy performance of innovative building technologies and energy systems (Swinton M. et al., 2001). The houses are typical North-American wood-frame buildings with brick facing, constructed according to the Canadian standard R-2000. The houses have two floors above ground, a basement, an (unfinished) attic, and an attached garage. The livable area is approximately 210 m^2 excluding the basement. Home automation systems are installed to simulate occupancy by activating appliances, lights, water valves and incandescent bulbs (for internal gains due to humans) based on repetitive daily schedules. Each house is fully instrumented with two data acquisition systems tracking more than 20 energy meters and 250 sensors. Figure 1 shows a photo of the twin houses.

Figure 1 CCHT twin houses (www.ccht-cctr.gc.ca)

The design heating load of the houses at -25 °C is 12.9 kW (44,000 Btu/h) and the design cooling load at 31 °C is 6.95 kW (23,714 Btu/h) (Gusdorf, 2003). The infiltration rate is 1.5 air change rate per hour at 50 Pa. The HVAC system installed in the reference house of the CCHT is a conventional central forced-air system consisting of a gas furnace with a capacity 19.8 kW (67,500 Btu/h), a 2-ton (7.03 kW) air conditioner and a HRV unit operating with constant speed. Table 1 presents the main characteristics of the houses (Lomanowski et al., 2010).

Table 1 CCHT houses characteristics

FEATURE	DETAILS
Liveable area	210 m^2 (2 storeys)
Insulation	Attic: R=8.6 m².K/W; Walls: R=3.5 m².K/W; Rim joints: R=3.5 m².K/W
Basement	Poured concrete, full basement Floor: concrete slab, no insulation Walls: R=3.5 m².K/W in a framed wall
Windows	Low-e coated, argon filled windows Area: 35 m^2 total, 16.2 m^2 south facing
Exposed floor over garage	R=4.4 m².K/W with heated/cooled plenum air space between insulation and sub-floor
Airtightness	1.5 h^{-1} @ 50 Pa

COMPARED SYSTEMS

The systems to be compared in the study, as mentioned before, are the standard system, the zoned VRF system with 3 indoor units and the non-zoned VRF system with 1 indoor unit.

Standard system: Gas furnace + Air conditioner

The standard HVAC system is a common system for Canadian houses, as it is installed in the CCHT reference house. It is used as a baseline case for comparison. Figure 2 shows the air flow diagram of the standard system. The conditioned zones in our study include the basement, the first floor and the second floor as shown in the figure below. The garage and attic are included in the model but they are not conditioned zones.

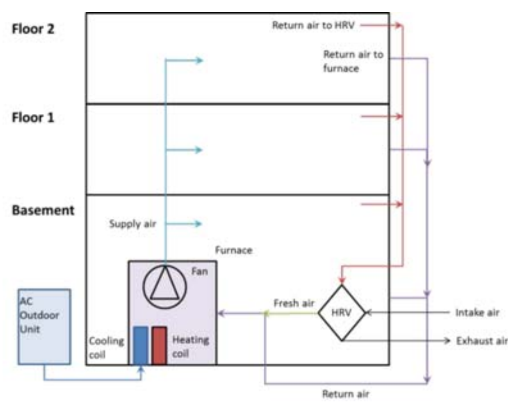

Figure 2 Schematic of standard system

The annual fuel utilization efficiency (AFUE) of the gas furnace is assumed to be 0.8, which is typical for existing non-condensing units (Manning et al., 2007). Its capacity is oversized by 50% compared to the design heating load, as it is commonly done in practice to allow for safety factors and faster recovery times. The seasonal energy efficiency ratio (SEER) of the air conditioner is assumed to be 12 and its total capacity is 7 kW, again corresponding to typical Canadian systems. The impact of indoor and outdoor conditions on the air conditioner performance is modeled using typical performance maps.

The cooling coil of the air conditioner is located within the furnace air-handler, so the cooling and heating systems both use the furnace fan to distribute the air to each zone. Part of the return air goes to an HRV unit while the rest returns directly to the furnace. The conditioned fresh air from the HRV is mixed with the return air into the furnace air handler. The HRV system with a nominal effectiveness of 0.84 runs continuously; the furnace fan runs at low speed to circulate the air all year round and at a higher speed when heating or cooling is activated. A fixed ratio of supply air is distributed to the different floors.

DHW is supplied by a separate gas water heater with an annual overall efficiency of 0.67, typical of existing (non-condensing) systems.

AS-IHP: zoned VRF system (VRF-3Z)

With the zoned VRF system, the house is split into 3 control zones, namely the basement, the first floor and the second floor. Each zone is served by a dedicated indoor unit, which is activated only when its control zone requires space heating or cooling. The outdoor unit is activated when one or more indoor unit(s) is (are) working. The simultaneous heating and cooling operation of the VRF system is not modeled in this study. The fresh air is conditioned through the HRV first and then goes to the indoor units, in the same configuration as for the standard system. The VRF indoor units have efficient ECM motors and are assumed to run continuously with a constant flowrate.

A Heat Pump Water Heater (HPWH) with a rated COP (rated Energy Factor in the US/Canada) of 2.66 is ducted into the return air of the VRF system to provide free cooling in summer and hot water for the whole year. The HPWH has its own fan and diverts part of the return air flowrate, which is then fed back into the return plenum before the indoor units.

Figure 3 presents the diagram of the air flow of the VRF system with three indoor units.

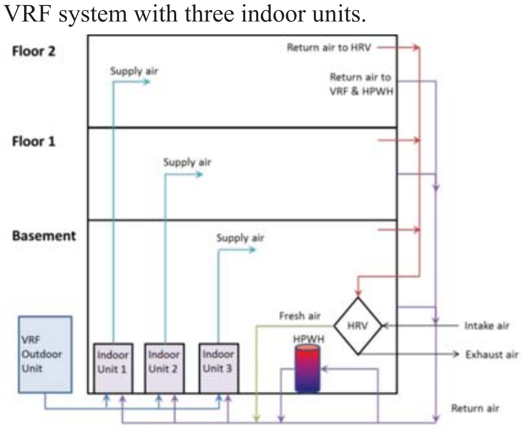

Figure 3 Schematic of VRF with 3 indoor units

Details on the models used for the HPWH and the VRF system are provided in separate sections below.

AS-IHP: non-zoned VRF system (VRF-1Z)

A non-zoned VRF system with 1 indoor unit is included in the comparison to assess the impact of zoning separately from the impact of the VRF technology. In contrast with the zoned VRF system coupled with 3 indoor units, the non-zoned VRF system supplies conditioned air with only 1 indoor unit. The indoor fan is assumed to run continuously to provide some degree of air mixing between the zones. A fixed ratio of the supply air is provided to the different floors, as for the standard system.

The HRV and HPWH system are the same as the zoned VRF system. Figure 4 shows the schematic of the non-zoned VRF system.

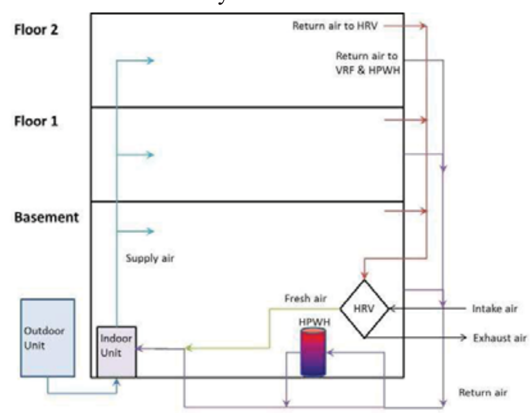

Figure 4 Schematic of non-zoned VRF

Details on the VRF modeling are provided in the next section.

VRF HEAT PUMP MODEL

Philosophy of the new TRNSYS components

Most of the heat pump models currently available for TRNSYS are designed to model devices having a single-speed compressor. In the existing models when the equipment is on, it operates at full load condition and its capacity and COP vary only as a function of current inlet conditions. The variability is described in one or more performance map data files. Such data is fairly available from manufacturers in a relatively standardized format and modeling heat pump performance as a function only of inlet conditions (instead of as a function of the zone load that they must meet) allows the models to be used for much more direct comparison against measured conditions.

The heat pump in the present study is distinct from existing models in that it contains a variable speed compressor and that it couples multiple indoor units with a single outdoor unit. Manufacturers provide some performance data for VRF devices although it is by no means as comprehensive as the data provided for single-speed devices. A new TRNSYS component was written to make use of the data files provided by a particular manufacturer. In this case two data files (one for heating performance and one for cooling performance) are required for each indoor unit. This allows for the indoor units to be of different sizes (capacities) as appropriate to the zones in which they are installed.

The indoor unit cooling performance data file contains total and sensible full-load capacity as a function of the indoor unit entering air dry and wet bulb temperatures and as a function of the outdoor air dry bulb temperature. An example is provided in Figure 5 in a normalized form. If, for example, the rated capacity of the indoor unit is 7.0 kW Total (4.7 kW sensible), the capacity for an outdoor temperature of 40 °C with indoor conditions at 32.2 °C DB and 23.9 °C WB will be 8 kW total, 4.5 kW sensible (respectively 1.14×7 and 0.96×4.7).

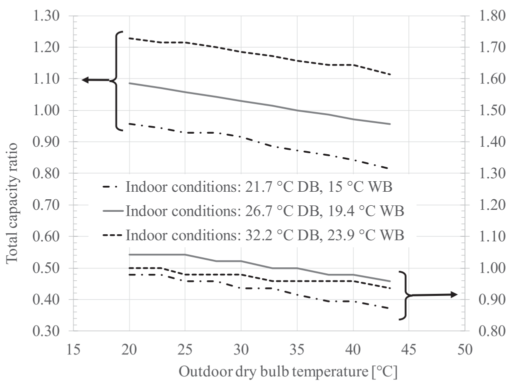

Figure 5 Total and sensible cooling capacity ratio for a VRF indoor unit

The indoor-unit heating performance file contains total capacity as a function of the indoor unit entering air dry bulb temperature and the outdoor air wet bulb temperature. Figure 6 shows an example of performance curves for an indoor unit. Assuming a 7.9 kW rated unit, the heating capacity will range from 4.4 kW to 9.6 kW in the operating conditions covered by the Figure.

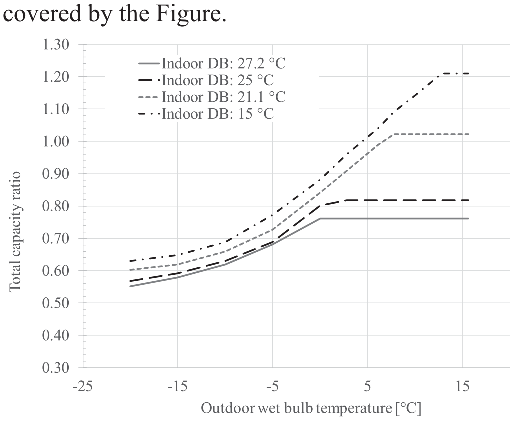

Figure 6 Heating capacity ratio for a VRF indoor unit

The capacity of each indoor unit is also corrected for the refrigerant piping length between that unit and the central (outdoor) unit using catalog data from another file, as shown in Figure 7.

The air flow rate in each of the indoor units is assumed be controlled by a single speed fan. In reality the indoor units include fans that can run at various speeds but it is unclear how the fan speed is set by the unit controls. The ability to run the indoor unit fans at multiple speeds will be added to a later version of the model as experimental data becomes available to make the control algorithms clear.

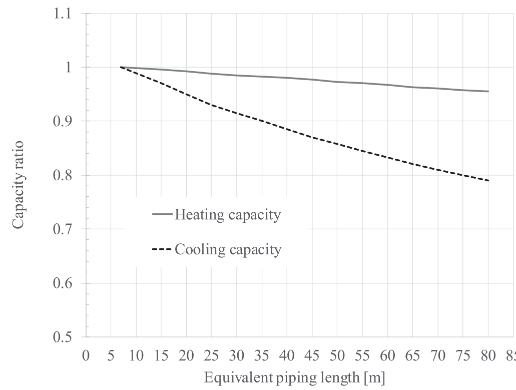

Figure 7 Capacity ration vs. piping length for a VRF indoor unit

The TRNSYS component determines the amount of energy required by each indoor unit in order to bring that indoor unit's entering air conditions to a user-defined supply temperature. The available manufacturer literature is not clear on the internal controls for this supply temperature – it could be set to a constant value, adapted to maintain a constant temperature difference across the unit, or set by a more complicated control strategy. As experimental data becomes available in later stages of the project the component will be modified to simulate the actual controls as best they can be determined.

The total energy requirement of all indoor units is then imposed on the outdoor unit. In the best of all possible worlds, if some of the indoor units were heating while others were cooling, the outdoor unit would see only the net load of the indoor units. In reality the outdoor unit will see something more than the net load but at this point in the project it is unclear how to determine the load seen by the outdoor unit when the indoor units are operating in mixed mode. Consequently it was decided that mixed mode operation would not be allowed in the initial model. The outdoor unit requires two performance maps (one for cooling and one for heating) to adjust its capacity and power to the indoor and outdoor conditions. Figure 8 shows an example of the heating performance map (the cooling performance map is similar to Figure 5 except that it represents the total cooling capacity and the power ratio, not the sensible capacity). In Figure 8, a unit having a 19.3 kW rated heating capacity and a rated heating power of 4.8 kW (i.e. a rated COP of 4), the COP would have a COP between 2.1 and 5.3 in the operating range covered by the Figure, and a heating capacity between 11.0 kW and 22.3 kW.

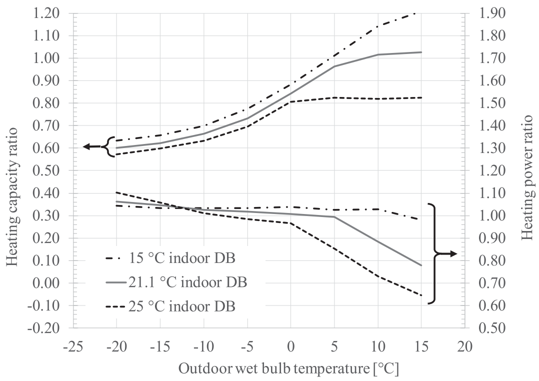

Figure 8 Heating capacity and power ratio for a VRF outdoor unit

Finally, the outdoor unit requires a performance data file that contains values of heating capacity, cooling capacity, compressor power in heating, and compressor power in cooling mode as a function of the fraction of the outdoor unit's rated capacity that is currently required by the sum of the indoor units. An example illustrating the part-load operation of an outdoor unit is provided in Figure 9. The figure shows that for the represented outdoor unit, a drop of more than 20 % in COP should be expected for part-load operation under 50 % of the available capacity.

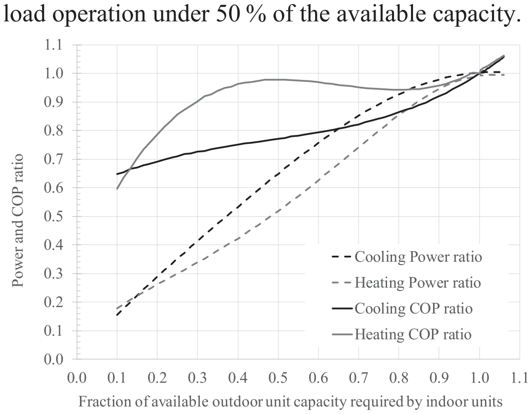

Figure 9 Power and COP ratio, VRF outdoor unit

Operating range vs. performance map range

Some studies have shown that VRF outdoor units, including the ones modeled in our study, are capable of operating down to -25 °C outdoor dry bulb temperature and even below. The experimental part of our work will aim at verifying and complementing the available performance maps to assess the unit performance in dynamic and extreme conditions. In this modeling study, the available performance maps were not extrapolated for heating at low outdoor temperature, so auxiliary heating was implemented in the model. It was assumed to be electrical backup heat with an efficiency of 1. The backup heat is controlled to maintain the desired air supply temperature if the VRF capacity is insufficient or if the machine is outside of the performance map conditions.

HEAT PUMP WATER HEATER MODEL

The Heat Pump Water Heater (HPWH) model uses a performance-map approach for the heat pump and separate TRNSYS components for other parts of the system (storage tank, auxiliary heater and controls). The TRNSYS component used to model the heat pump performance, known as Type 994, is presented in (Maguire, 2012). It relies on a performance map obtained from empirical data (Hudon et al., 2012). The performance parameters (air-side sensible and latent cooling, water-side heating, compressor and fan power) are related to the tank water temperature and to the inlet air wet bulb temperature.

The modeled unit has a rated heat output of 1.28 kW and a rated COP of 2.66, with 50 °C water and 14.1 °C inlet wet bulb temperature (e.g. corresponding to 21 °C dry bulb and 47 % relative humidity). Heat is rejected by a wrap-around condenser located around the bottom part of the tank.

The heat pump component is coupled to a standard TRNSYS Type modeling a storage tank (TESS Type 534). The modeled tank has a volume of 173 L and an overall loss coefficient of 1.14 W/K. The heat rejected by the heat pump is injected into the bottom part of the storage tank, and the average tank temperature is fed back to the heat pump component, as recommended in (Maguire, 2012). The HPWH is assumed to be set to maximize the heat pump operation. An auxiliary electric resistance heater (4.5 kW) located in the top part of the tank is only activated when the heat pump cannot maintain the setpoint. The HPWH fan power is taken into account, and it is assumed that the HPWH imposes the air flowrate. That flowrate depends on operating conditions and has a rated value of 64 L/s.

The simulated DHW draw profile is the one used in the CCHT houses (Swinton et al., 2001). The profile consists of identical days that include draws for shower, taps, bath, dishwasher and cloth washing machine. The total daily volume is 256 L/day.

OTHER SIMULATION PARAMETERS

The house is modeled using TRNSYS multizone building, known as Type 56. The infiltration is modeled in detail using the Alberta Infiltration Model (AIM-2, Walker & Wilson, 1998). Basement losses are modeled using a detailed 3-D ground model known as TESS Type 1244, and the ground is preconditioned by running a 2-year simulation and discarding the first year.

A time step of 3 minutes is used, and the selected weather file is the Canadian Weather for Energy Calculation (CWEC) data file for Ottawa, ON. The cooling season was adjusted by trial and error, it starts on June 1st and ends on September 21st. Heating is activated outside of that period. All simulations use the same thermostat settings: 22 °C for space heating, and 24 °C for space cooling. A 2 °C deadband is used for all configurations. It is centered on the setpoint so that, for example, space heating aims at maintaining the temperature between 21 °C and 23 °C.

The VRF system configurations include a HPWH which is ducted into the return air. In order to obtain a more straightforward comparison with the reference scenario, the VRF indoor fans were left running continuously.

Schedules for lighting, appliances, occupancy and water draws are the ones implemented at the CCHT (Swinton et al., 2001) and are repeated every day.

RESULTS

Overview of the energy use

Table 2 presents an overview of the on-site energy use for the 3 configurations. The last row includes the total end-use energy (sum of electricity and gas without any equivalence factor). This is for example the approach used in defining the Energuide Rating Scale for Canadian homes (NRCan-OEE, 2005). Using the total figure, the energy savings of the VRF-1Z configuration amount to 49 %, while the VRF-3Z delivers 42 % savings. A few points are worth mentioning:

- The large share of fan power in the VRF configurations is a direct result of the decision to operate the indoor units continuously at the same speed. This assumption may not be appropriate but it was selected in order to obtain a fair comparison with the standard system.
- The share of space heating in the total energy use is not significantly different in the 3 configurations, just above or under 50 %. But the integrated HPWH in the two VRF configurations brings the share of DHW from 20 % to 10 %.
- The HRV energy use corresponds to a continuous power draw of 115 W, as an identical device was assumed in all cases.
- The present study focuses on annual energy performance and comfort under simple operating assumptions. The benefit of zoning in terms of implementing setbacks/setups in different zones at different times and to use the VRF technology for peak demand reduction and shifting will be investigated in a next phase of our work. Again, the continuous operation of 3 fans in the VRF-3Z configuration has a significant impact on the energy use and may not be appropriate in realistic implementations.

The next sections will investigate the energy and comfort performance of the different configurations in more detail.

Table 2 Annual energy use overview [GJ]

	STD	**VRF-1Z**	**VRF-3Z**
HRV fan	3.6	3.6	3.6
All other fans	12.0	10.1	13.3
Fans subtotal	**15.6**	**13.7**	**16.9**
Gas furnace	64.9	-	-
VRF (heating mode)	-	20.1	27.9
VRF auxiliary heating	-	7.4	4.8
Space heating subtotal	**64.9**	**27.5**	**32.7**
Air conditioner	3.5	-	-
VRF (cooling mode)	-	1.7	1.5
Space cooling subtotal	**3.5**	**1.7**	**1.5**
Gas water heater	24.6	-	-
HPWH (incl. auxiliary)	-	6.9	6.8
DHW subtotal	**24.6**	**6.9**	**6.8**
Lighting and appliances	**12.0**	**12.0**	**12.0**
Total gas	89.5	-	-
Total electricity	31.1	61.8	69.9
Total (Gas and elec.)	**120.6**	**61.8**	**69.9**

Energy use vs. thermal loads and HPWH impact

Figure 10 shows an energy balance of the house and HVAC systems for the STD and VRF-1Z configuration.

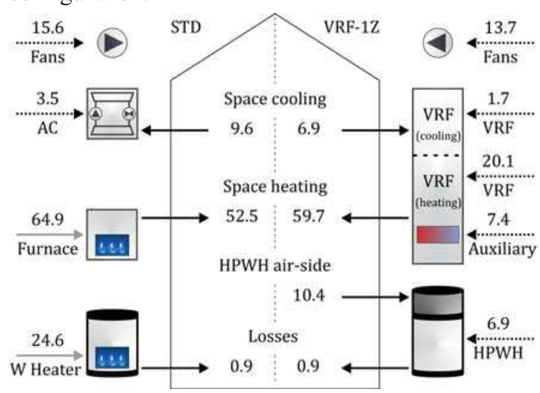

Figure 10 Yearly balance for STD and VRF-1Z [GJ]

Not shown in the figure are the “source-side” heat transfers for the air conditioner and VRF heat pump, and the useful energy transferred to DHW, which is the same in all cases (16.5 GJ/year).

Figure 10 shows that the HPWH uses 10.4 GJ of heat from the house, which corresponds to the air-side heat transfer of the heat pump. About 1 GJ of heat is given back to the house with tank losses. The heat removed from the house is beneficial during the cooling season, and the results show that the space cooling needs are reduced by 2.7 GJ, so a simple estimate of the free cooling effect of the HPWH is in the order of 2.7 GJ or 26 % of the air-side heat transfer. On the other hand, the space heating load increases by 7.2 GJ, so we can estimate that 69 % of the heat removed by the HPWH results in an increase in the space heating demand. These are rough estimates that neglect the impact of other variations such as different heat gains from fans between the different configurations. Other system designs (e.g. HPWH located in an unconditioned room or only partially ducted) would probably deliver different results.

The annual COP of the VRF system is 4.1 in cooling, and it is close to 3.0 in heating when only the operation of the heat pump is taken into account. If the auxiliary heat is included, the annual COP of the space heating system is 2.2.

Figure 11 shows a similar figure but comparing the zoned (VRF-3Z, left) and non-zoned (VRF-1Z, right) configurations. The results show that a large part of the increase in energy use between VRF-1Z and VRF-3Z can be attributed to an increase in the space heating load. As will be shown below, the zoned configuration delivers better thermal comfort in all zones, including the basement which is colder than the upper floors. This results in an increased space heating load. Another difference is the fan power, which has been discussed above.

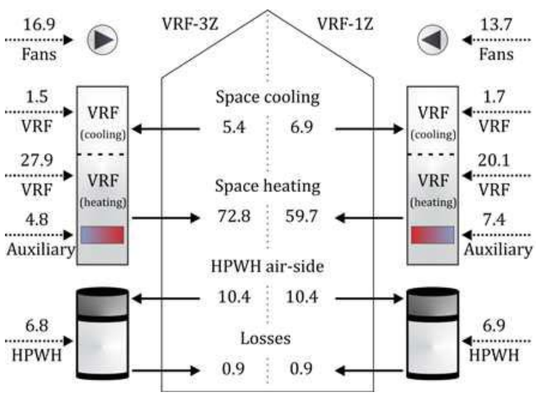

Figure 11 Yearly balance for VRF-3Z and -1Z [GJ]

Temperature control and thermal comfort

Figure 12 shows the air temperature in the 3 zones for the coldest day of the year for the 3 system configurations. In the CWEC data file for Ottawa, Jan 27 is the coldest day with a minimum temperature of -25 °C and a maximum temperature of -16.7 °C. The temperature profiles show that both the STD system and the VRF-1Z system keep the basement uncomfortably colder than the upper floors. The 2nd floor sometimes gets too warm on milder, sunny days. The figure also show different cycling times for the 3 systems, which are related to their capacities and controls. The zoned VRF system (VRF-3Z) maintains the room temperature within the deadbands in each of the 3 zones.

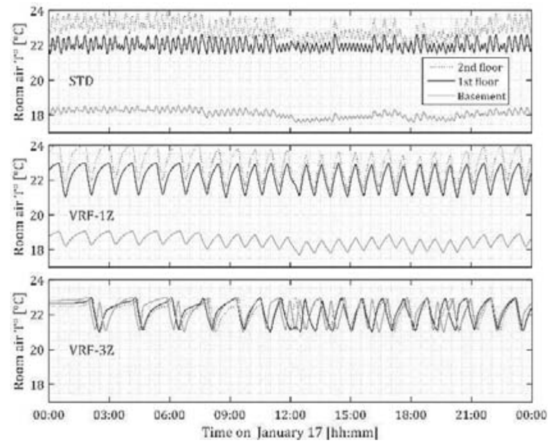

Figure 12 Temperature of the 3 zones for the coldest day of the year, for the 3 systems

Similar trends are observed for the hottest day of the year (20 July), and generally speaking during the heating and cooling seasons.

The impact of poor temperature control in the basement and second floor is evidenced by Figure 13, which shows histograms of Fanger's Predicted Mean Vote. The PMV should normally lie between -0.5 and 0.5, which corresponds to 10 % Predicted Percentage Dissatisfied (PPD).

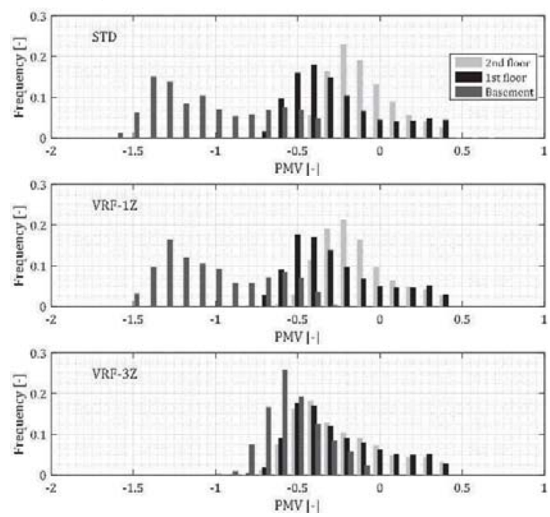

Figure 13 PMV histogram for the 3 zones

The average Predicted Percentage of Dissatisfied people (PPD) for the 1st floor during the heating season is very similar for the 3 systems (between 8.6 % and 8.8 %). However, the average PPD in the basement during the heating season is above 35 % for STD and VRF-1Z, while it is at 12.5 % for VRF-3Z.

During the cooling season, the VRF-3Z system does not deliver a significant improvement on PPD values, which are under 10 % for the upper floors and slightly above 10 % for the basement.

DISCUSSION AND CONCLUSIONS

This paper assessed the annual energy performance of three different HVAC systems, which are a standard forced-air system, a zoned VRF system with 3 indoor units, and a non-zoned VRF system with 1 indoor unit. The study is based on TRNSYS simulations using a detailed building model and new component models for the VRF system.

The selected house is one of the CCHT experimental twin houses, which have three floors including basement, the first floor and the second floor. The weather data used was a typical year for Ottawa, ON.

The standard system is a typical conventional forced air system installed in Canadian houses, consisting of a furnace and an air conditioner. It is a non-zoned system with a single thermostat placed in the centre of the living room on the first floor. It is used as a baseline case for the comparison.

The zoned VRF system has three thermostats in each floor and each indoor unit supplies space heating and cooling to each floor. The non-zoned VRF system only has one thermostat installed in the first floor.

The comparison shows that the non-zoned VRF system delivers annual energy savings of 49 % compared with the standard system; and the zoned VRF system saves 42 %. The latter system provides better thermal comfort than the two other configurations throughout the year for the basement and second floor zones. These results were obtained using a typical North-American house and simplified, “average”, assumptions for user behaviour.

Further work will aim at verifying and improving the performance data on the VRF systems by comparing simulation results with experimental data. Simulations will also be used to investigate the potential of zoning strategies including setback and setups to deliver further energy savings and peak demand reduction.

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