DUAL-ENERGY SOURCE HEAT PUMP

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Abstract: Heating performance of conventional air-source heat pumps dramatically drops at outdoor temperatures below -5°C, and therefore supplementary heating sources are required. In cold climates, if electricity is used as a back-up heating source, at outdoor temperatures below -12°C it may coincide with the grid peak power demand. Additional energy sources, other than electricity, could consequently help in avoiding the use of an expensive, high-quality energy source for residential comfort heating. This paper presents a variable speed heat pump with a non-electrical back-up energy source. It highlights the original design and features, as well as the main energy performance. In the conventional air-to-air heating mode, the heat pump balance point has moved from -5°C to -9°C. Between the new balance point and the shutdown outdoor temperature (-12°C), the compressor operates about 70% of the time at maximum speed, with net heating gains exceeding 2 kW, but with lower coefficients of performance. In the combustion gas-to-air heating mode, activated below outdoor temperatures of -12°C, a propane burner provides heat to a very compact add-on evaporator. In this mode, fuel consumption is 76% lower and the heating power supplied 13.3% higher compared to those of similar dual-energy heat pumps.

Key words: heat pump, dual-energy heating, cooling, energy efficiency

1 INTRODUCTION

Air-source heat pumps are highly efficient devices for space heating and cooling. In cold and very cold climates, the winter heating loads are greater than the summer cooling loads. In addition, the heating performance of conventional air-source heat pumps sharply drops at outdoor temperatures generally below -5°C. As a result, most North American manufacturers use add-on electrical heaters installed in the duct work of home forced hot air systems. They operate whenever additional heat is required and the heat pump is out of operation for significant periods of time. Moreover, when outdoor temperatures fall below -12°C, most electrical grids in cold and very cold climates may achieve peak power demand loads at a relatively high cost for each kW generated. But, with increasing energy and electrical peak demand costs, operating electrical heaters can become too expensive (Guilbeault 1987). Using fossil fuel fired back-up heaters integrated in the heat pump indoor units as an alternative may raise issues such as insufficient space for incorporating the supplementary heater in the indoor unit, soot-free combustion and clean indoor atmosphere, and requirements for discharging the combustion gases outside. Setting it up apart from the heat pump requires separate ducting and controls, and renders the installation as well as the operation and maintenance of the supplementary heater expensive in addition to wasting space. The concept presented in this paper consists in the integration of a fossil energy source within the refrigeration circuit of the heat pump. It allows heat pumps to operate efficiently with a back-up energy source other than electricity at outdoor temperatures below -12°C. This concept may help reduce grid peak power demand while achieving high energy performance in the space heating mode by using readily stored and available fuels such as propane gas or heating oil.

2 BACKGROUND

2.1 Fuel fired vaporizer

Several methods for adding a second non-electrical heat source to heat pump systems were developed in the past. In the system represented in Figure 1.1 (Sisk and Veyo 1978), an outdoor refrigerant heater including a burner and a *vaporizer*, as well as a liquid pump are integrated within the refrigeration circuit of a typical air-to-air heat pump.

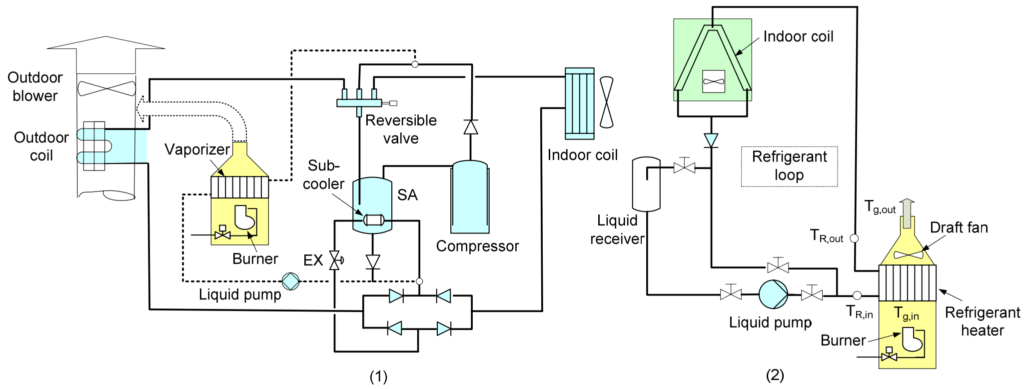


Figure 1: (1) Dual-energy source heat pump with fuel-fired supplementary heater (Sisk and Veyo 1978); (2) Refrigerant gravitational and pumping system (Minea 1988)

The *vaporizer* is positioned downstream of the burner so that the hot gases from the burner heat the coil and vaporize the liquid refrigerant. The outdoor blower can be used for inducing the required draft air for the burner. In the conventional space heating mode, the refrigerant liquid passes, via a check-valve manifold set, through the sub-cooler and expansion valve EX, and vaporizes inside the outdoor coil. For operation in the cooling mode, the 4-way reversing valve reverses the refrigerant flow. The indoor coil is used as an evaporator and the outdoor coil as a condenser. In the back-up heating mode, the liquid refrigerant pump is activated and the burner is fired. The refrigerant pump supplies enough flow to avoid fluid overheating and decomposition. The *vaporizer* therefore receives liquid refrigerant from the condenser. The generated vapor is directed via the reversible valve to the indoor coil to condense and release the heat to the indoor space, while the compressor is by-passed and stopped.

2.2 Gravitational & pumping system

Figure 1.2 shows another refrigerant pumping system that can also operate the closed refrigerant loop by gravity (Minea 1988). This system used R-22 and CFC-11 as refrigerants but new high-temperature refrigerants such as R-236fa and R-245fa can be used. In both the gravitational and pumping modes, strict control of the combustion gas temperatures and refrigerant flow rates was achieved. In the *gravitational mode*, the burner provided 8.3 kW to the refrigerant heater, which operated with a thermal efficiency of 85% and overall heat transfer coefficients of 10.5 W/m^{2}K. The average velocity of the combustion gas through the refrigerant heater varied from 0.37 m/s to 0.56 m/s, while the average air flow rate through the indoor coil was 448 L/s (0.45 m^3/s). The refrigerant flow rate (2.9 kg/min) provided relatively high superheating values (7 to 12°C). To increase the thermal power supplied in the add-on mode, the refrigerant flow rate can be enhanced by increasing the vertical distance between the add-on coil and the indoor air coil levels. In the *heat pump mode* with the combustion gas fan running, the burner provided thermal outputs between 11 and 13.5 kW. The add-on heating coil achieved thermal efficiencies of about 90% and overall heat transfer coefficients of 19.2. For the same heat transfer surface but with combustion gas velocities 1.45 to 13.6 times higher, the overall heat transfer coefficients increased 2.7 to 5.2 times. A liquid pump was selected to supply liquid refrigerant at a rate much greater than the rate at which the refrigerant is evaporated. It was actually 60% higher compared to the gravitational mode, and zero superheating was achieved within the evaporator. Table 1 shows other experimental parameters and performance for both gravitational and pumping operating modes of the system represented in Figure 1.2.

Table 1: Gravitational and pumping add-on systems without compressor (Minea 1988)

Propane flow rate	**Combustion gases temperatures**		**Refrigerant parameters**					**$(LMTD)_m$**
	$T_{G,in}$	$T_{G,out}$	$T_{R,in}$	$T_{R,out}$	**Flow rate**	**Saturated pressure**	**Saturated temperature**	**-**
kg/h	**°C**	**°C**	**°C**	**°C**	**kg/min**	**kPa (r)**	**°C**	**°C**
Gravitational mode								
0.628	314	67	42.7	53.2	2.9	1641	46	86.7
Pumping mode								
0.907	396	60	51.7	54.4	4.8	2082	56	79.6

2.3 Heat-augmented heat exchanger

Another heat pump concept able to operate at low ambient temperatures includes a capillary tube, and operates as a conventional heat pump during both the space heating and cooling modes (Figure 2.1) (Vaart 1982). A third operating mode allows the heat pump to operate at temperatures below those that would render conventional heat pumps totally inefficient. The outdoor unit contains a *heat-augmented* coil and a fuel-fired burner that generates heat to vaporize the liquid refrigerant. The *heat-augmented* coil operates as a condenser in the cooling mode and as an evaporator in air-to-air (conventional) using a bidirectional capillary tube as an expansion device. The defrosting mode is provided without reversing the thermodynamic cycle because the fuel-fired burner supplies heat to defrost the outdoor evaporator. The *heat-augmented mode* (back-up), the third operating mode, renders the heat pump fully operative at temperatures well below those that would render conventional heat pumps virtually inoperative and/or inefficient in the heating mode of operation (Vaart 1982). In this mode, the outdoor air blower shuts down. The heat generating source can be a fossil fuel source placed at the bottom of the “A-coil” located outside of the building. The refrigerant flow is identical to that of the conventional heating mode. When the outdoor blower is de-energized, the gas burner is energized by igniting the gas. The flames are spread out across the bottom of the A-coil and the heat is absorbed almost totally, while the burning process approaches 100% efficiency. The compressor can use the relatively high heated low pressure vapor phase of the refrigerant, which would be impossible to achieve in the absence of the additional heat provided by the heat source. Another concept consists in a conventional air-to-air heat pump equipped with a parallel *glycol closed-loop* and heat exchanger (Figure 2.2) (Minea 1991). This add-on dual-energy heat pump provides additional heat to the house at very low ambient temperatures. The burner is located inside the outdoor unit. The heat pump does not operate in the back-up heating mode, and, as in the previous system, in the defrosting mode, the fuel burner supplies heat to the outdoor evaporator without reversing the thermodynamic cycle. Such a system has been developed and successfully *field* tested in Eastern Canada to reduce the electrical peak power demand at outdoor temperatures below -12°C (Parent et al. 1991).

3 ADD-ON CONCEPT

The refrigeration circuit of a conventional 8.75 kW (nominal cooling capacity) air-to-air heat pump has been modified by including a new add-on heating unit. The dual-energy source heat pump thus obtained includes an indoor unit with variable speed compressor, suction accumulator, 4-way reversing valve and controls (Figure 3). The second cabinet, located outdoors, contains an outdoor finned coil with air fan, as well as expansion valve EX1 with by-pass and check valve CV1.

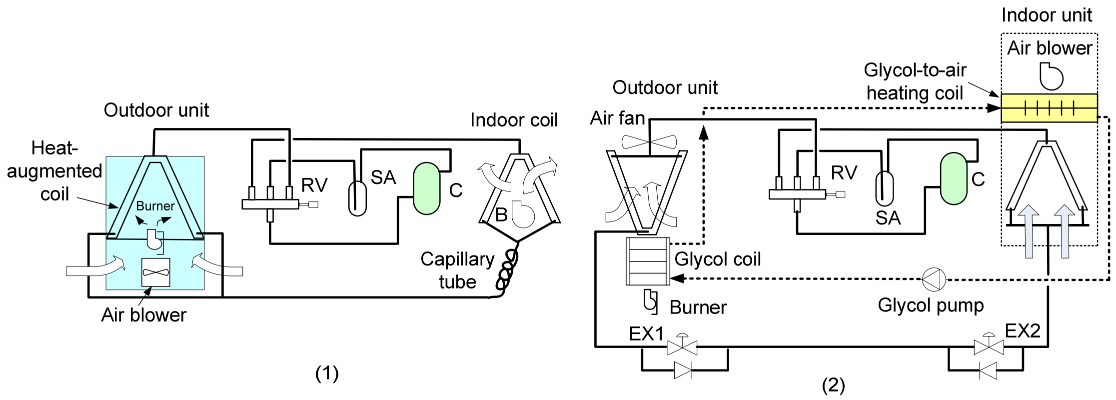


Figure 2: (1) Conventional dual-energy source heat pump with "A" coil and capillary tube (Vaart 1982); (2) Dual-energy source heat pump with parallel glycol loop (Minea 1991). C: compressor; RV: reversible valve; SA: suction accumulator

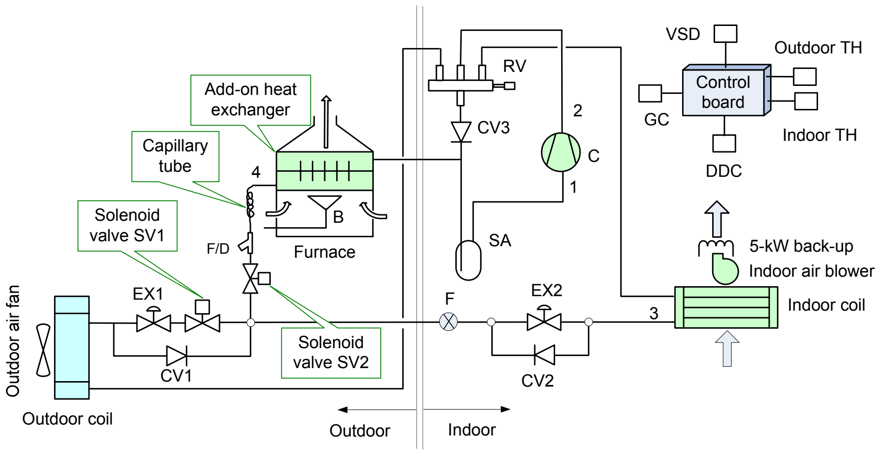


Figure 3: Schematic of the dual-energy source heat pump
B: burner; C: variable speed compressor; CT: capillary tube; EX: expansion valve; F: refrigerant flow meter; F/D: filter-drier; RV: reversible valve; SA: suction accumulator; SV: solenoid valve; GC: gas controller; VSD: variable speed drive; TH: thermostat; DDC: demand defrost controller

The third cabinet is an add-on unit, also located outdoors. It includes a small furnace with a gas-fired burner and a compact add-on combustion gas-to-refrigerant heat exchanger. It preheats, vaporizes and superheats the refrigerant in the back-up heating mode. Being located outdoors, this add-on unit does not use valuable indoor space and the burned gases do not contaminate the indoor atmosphere. Finally, the fourth cabinet located inside the house includes a finned indoor coil with indoor air blower, expansion valve EX2 and by-pass with check valve CV2. Two additional solenoid valves, SV1 and SV2, make it possible to bypass the outdoor coil and to supply low pressure refrigerant liquid to the add-on heat exchanger via the capillary tube installed upstream of solenoid valve SV2. Finally, check valve CV3 allows the refrigerant vapor leaving the add-on coil to bypass the 4-way reversible valve and to flow, via suction accumulator SA, to the compressor suction line in the back-up heating mode.

4 OPERATING MODES AND CONTROLS

The dual-energy source heat pump can operate in both conventional air-to-air space heating/defrosting and cooling modes, as well as in the gas-to-air back-up heating mode. In the conventional *air-to-air heating mode*, solenoid valve SV1 is open and solenoid valve SV2 is closed. The outdoor coil operates as an evaporator and the indoor coil as a condenser. In this mode, the heat source medium is the outdoor air. The frequency of the electrical current supplying the compressor can reach 90 Hz according to the actual space heating demand. At outdoor temperatures between the typical balance point (-5°C) and the shutdown point (-12°C), a 5 kW back-up electrical heating coil supplies additional heat to the indoor air. It compensates the lower heating capacity of the heat pump and increases the temperature of the air supplied. When defrost is required, reversible valve RV reverts to the cooling position, and the outdoor heat exchanger becomes a condenser. In the defrost mode, the compressor runs at constant speed (60 Hz), and the 5 kW electric back-up coil is energized. It avoids supplying cold air to the house and improves indoor comfort. At temperatures equal to lower than -12°C, the system operates in the combustion gas-to-air (add-on or back-up) heating mode. Reversible valve RV is set in the heating position, the solenoid valve SV1 closes and solenoid valve SV2 opens. Compressor C starts and the liquid refrigerant flows from the indoor coil (condenser) to the add-on coil via solenoid valve SV2 and the capillary tube. After a few seconds, burner B is switched on and the combustion gases begin to heat the add-on heat exchanger. In this mode, the compressor runs at constant speed (60 Hz), the defrosting cycle is disabled, and the fan of the outdoor coil shuts down. The refrigerant is preheated, vaporized and superheated within the add-on heat exchanger. Check-valve CV3 allows the superheated vapor to enter the compressor via suction accumulator SA. The indoor coil operates as a condenser, as long as the outdoor temperature is bellow -12°C. In the summer, the heat pump operates in the *air-to-air cooling mode*. The reversible valve RV is in the cooling position and solenoid valves SV1 and SV2 are closed. The compressor can run at variable speed (from 30 to 60 Hz) according to the home actual cooling demand. The speed of the indoor fan is modulated to control the relative humidity of the indoor air supplied. The outdoor coil operates as a condenser discharging in the ambient air the sensible and latent heat recovered inside the house. The *electronic board* (Figure 3) controls the compressor variable speed drive (VSD) based on the temperature difference between the thermostat set point and the actual indoor temperature. The design of the electronic control board is compatible with indoor programmable electronic thermostats. The electronic board also controls burner operation via a gas control module (GC), the defrosting mode via a defrost demand control (DDC) module, the speed of the indoor air blower, and opens and closes solenoid valves SV1 and SV2.

5 ADD-ON HEAT EXCHANGER

5.1 Description

The *add-on heat exchanger* is a horizontal finned copper coil where the refrigerant evaporates through a forced convective process. Back-up energy is supplied by the propane gas burner that takes its combustion air from the ambient atmosphere. Prior to entering the add-on coil the refrigerant flows through a ½” tube integrated within the furnace wall (Figure 4). The combustion gases temperatures and thermal powers required to heat the ambient combustion air depend on the inlet temperature of the ambient combustion air and the excess air factors. In cold and very cold climates, it is assumed that the ambient combustion air enters the furnace at temperatures between -30°C and -12°C. Because the fusion temperature of copper is 1083°C, the excess air factor has to be higher than 3. Excess factors lower than 3 are avoided by appropriate design and operation of the propane furnace. The actual combustion excess air is adjusted to obtain combustion gas temperatures at the add-on heat exchanger inlet of about 310°C, while the thermal power required to preheat the outdoor excess air varies around 1 kW. In the gas-to-air back-up heating mode (see section 6.2), the expansion device is a capillary tube. A capillary tube has been chosen because it is inexpensive and does not have any moving parts, and hence does not require maintenance. The capillary tube doesn't need any adjustment because the flow conditions are very stable and it isn't liable to clogging because a filter-drier is used upstream (see Figure 3) to prevent the entry of moisture or any solid particles. The flow is assumed to be steady and one-dimensional, and divided into a liquid region where the pressure decreases linearly down to the flash point, and a two-phase region characterized by increasing refrigerant velocity and pressure drop. For the designed refrigerant flow rate in the gas-to-air back-up heating mode, the analytical calculation gave a 6-tube capillary tube, 279 mm in total length with each tube having a 1.37 mm inside diameter.

5.2 Design

It was assumed that in cold climates, the space heating demand for a medium size home in the add-on heating mode is about 15.5 kW. If the heat pump compressor power input is 1.9 kW, the required thermal power of the add-on heat exchanger will be $\dot{Q}_{add-on} = 13.6\,\mathrm{kW}$. With a calorific power (Lower) of 46 348 kJ/kg and an add-on heat exchanger thermal efficiency of 98%, propane gas consumption will be about 1 kg/h.

The convective vaporization heat transfer coefficient on the refrigerant side is calculated using the following correlation (Ciconkov, 2001):

$$h_{cv} = C \frac{G^{0.1} \dot{q}_{heat}^{0.7}}{d^{0.5}_{eq,i}} \tag{1}$$

where $G = 1282\,\mathrm{kg/m^2}s$ is the refrigerant mass velocity through the add-on heat exchanger tubes defined as the ratio between the refrigerant flow rate and the tube flow area (Table 1); $\dot{q}_{heat}$ - the heat flux density related to the total fin heat transfer area ($68000\ W/m^2$), and C, a constant based on the refrigerant (R-22) thermodynamic properties at the actual saturation temperature :

$$C = \varphi(k_f, \mathrm{l_{fg}}, \rho_\mathrm{f}, \rho_g, T_{sat} = 23°C, ...) = 0.187 \tag{2}$$

Based on equation 1, the refrigerant convective vaporization heat transfer coefficient will be $h_{cv} = 9631.6\,\mathrm{W/m^2}K$. The gas-side free convection heat transfer coefficient is assumed to be $h_G = 89\,\mathrm{W/m^2}K$. For the finned tube, cross-flow add-on heat exchanger, the following thermal power balance equation can be written for constant properties of both fluids (Incropera and DeWitt, 2002):

$$\dot{Q}_{add-on} = \eta_B\, \dot{m}_G\, c_{p,G}(T_{G,in} - T_{G,out}) = \dot{m}_{ev}\, \Delta h_R = UFA_G(LMTD)_m \tag{3}$$

where h_R is the refrigerant mass enthalpy, $\eta_B = 0.96$ - thermal efficiency of the add-on coil and $F = 0.97$ - the correction factor for the cross-flow add-on heat exchanger with unmixed fluids. This equation ignores heat losses to the surrounding area, as well as kinetic and potential energy changes. The combustion gases mass flow rate is calculated as follows:

$$\dot{m}_G = \frac{\dot{Q}_{add-on}}{\eta_{add-on} c_{p,G}} \tag{4}$$

where $c_{p,G} = 1000\ J/kgK$ is the average specific heat of combustion gases across the add-on heat exchanger.

Based on the refrigerant enthalpy entering and leaving the add-on coil in the permanent add-on heating mode (Figure 4.3), equation 3 allows calculating the refrigerant flow rate as follows:

$$\dot{m}_R = \frac{\dot{Q}_{add-on}}{\Delta h_R} = \frac{\dot{Q}_{add-on}}{h_1 - h_5} \tag{5}$$

Where h_1 and h_5 are the refrigerant mass enthalpies (kJ/kg) at the outlet and the inlet of the add-on coil, respectively. The overall heat transfer coefficient will be:

$$\bar{U} = \frac{1}{\frac{1}{h_{cv}} + \frac{1}{h_G}} \tag{6}$$

Table 1: Designed parameters of the add-on heat exchanger

Parameter (symbol, unit)	Calculated value
Required thermal power ($\dot{Q}_{add-on}$, kW)	13.6
Refrigerant mass velocity (G, kg/m^3)	1282
Heat flux density ($\dot{q}_{heat}$, W/m^2)	68,000
Refrigerant-side heat transfer coefficient (h_{cv}, W/m^{2}K)	9631.6
Gas-side heat transfer coefficient (h_G, W/m^{2}K)	89
Combustion gases mass flow rate ($\dot{m}_G$, m^3/s)	0.05
Steady-state refrigerant mass flow rate ($\dot{m}_R$, kg/s)	0.085
Overall heat transfer coefficient ($\bar{U}$, W/m^{2}K)	88.2
Logarithmic mean temperature difference ($LMTD_m$, °C)	73
Required gas-side total heat transfer area (A_{req}, m^2)	2.16

With combustion gas entering the add-on heat exchanger at 300°C and leaving it at 29°C, the required gas-side total heat transfer area (exterior heat exchange surface of fins + tubes) will be:

$$A_{req} = \frac{\dot{Q}_{add-on}}{UF(LMTD)_m} \tag{7}$$

where the logarithmic mean temperature difference is:

$$(LMTD)_m = \frac{T_{G,out} - T_{G,in}}{\ln \frac{T_{G,out} - T_{sat}}{T_{G,in} - T_{sat}}} \tag{8}$$

The selected add-on exchanger is a 0.1524 m (width) x 0.3048 m (length), 2-row, finned copper coil with 9.1875 mm (ID 3/8”) circular horizontal tubes, and 106 0.2 mm thick fins (Figure 4).

The condenser thermal power in the permanent add-on heating mode is:

$$\dot{Q}_{CD} = \dot{m}_R (h_2 - h_5) = \dot{Q}_{add-on} + W_C \quad (9)$$

Where h_2 is the refrigerant mass enthalpy (kJ/kg) entering the add-on coil.
Equation 9 allows calculating the required compressor input power:

$$W_C = \dot{Q}_{CD} - \dot{Q}_{add-on} \quad (10)$$

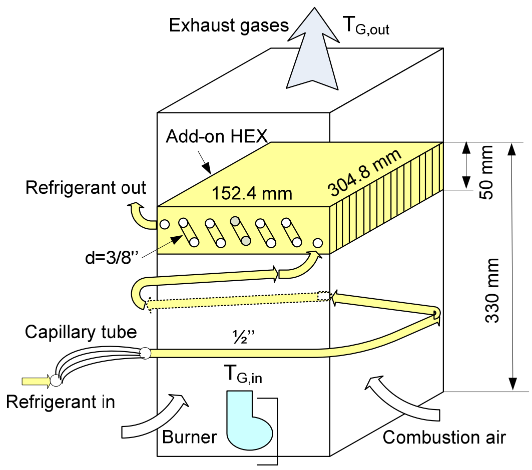


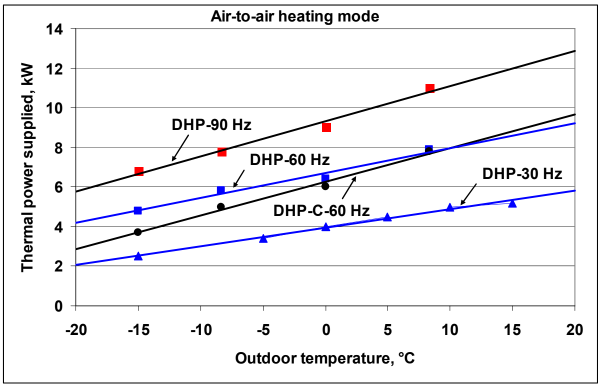
Figure 4: Schematic representation of the add-on heat exchanger and propane furnace. HEX: heat exchanger; T: temperature; G: gas; in: inlet; out: outlet

6 EXPERIMENTAL VALIDATION

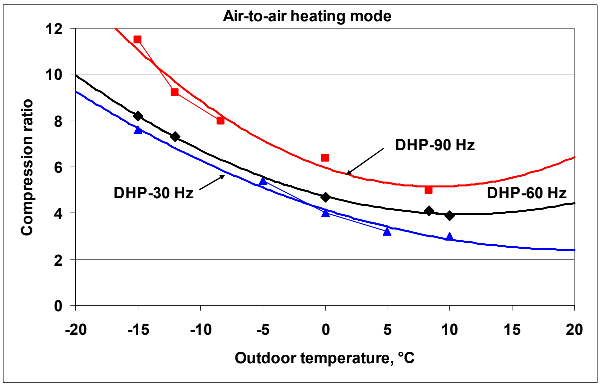
The laboratory prototype of the new dual-energy source heat pump (DHP) was extensively tested in both the conventional air-to-air and add-on gas-to-air heating modes.

6.1 Air-to-air heating mode

The advantages of heat pumps are obvious when they can modulate the compressor frequency to avoid transitory phases. However, in cold and very cold climates, heat pumps are mainly used for heating and, to meet heating demand, they could become constant speed units. In the conventional *air-to-air heating mode*, the results obtained don't provide new qualitative findings, they are generally well known. As expected, in this mode, the thermal power supplied by the indoor coil (condenser) increases with compressor speed at a given outdoor temperature, as shown in Figure 5.1. For example, at outdoor temperatures of -5°C, the thermal power supplied at 60 Hz is 6 kW, and at 90 Hz it is 8.4 kW. The gains achieved in heating thermal power as compared to the supplementary electrical power used by the compressor are between 20% and 40% when the frequency increases from 60 to 90 Hz. However, by increasing the compressor speed from 60 Hz to 90 Hz, the heat pump coefficient of performance (COP), defined as the ratio between the thermal power supplied to the indoor air and the compressor electrical power input, decreases by about 20% (Figure 6). At the same time, the compressor suction pressures drop and the compression ratios increase by about 30%. At very low outdoor temperatures, undesirable compression ratio values occur causing abnormal overheating of the variable speed scroll compressor. The problems with high compressor outlet temperatures at 90 Hz should not be ignored. This phenomenon can be mainly explained by the compressor type (scroll) and the cooling of the engine by discharge gases. At high speed, refrigerant flow is not high enough to dissipate all electrical energy consumed by the compressor, and the load adjustment becomes critical to avoid overheating of the compressor. Compared to the conventional dual-energy source heat pump (DHP-C) shown in Figure 2.1, the COPs of the new dual-energy heat pump (DHP) at 60 Hz are clearly higher over the entire outdoor temperature range (Figure 6). Even at 90 Hz, the COPs of DHP are higher than those of DHP-C.



(1)



(2)

Figure 5: Air-to-air heating mode at different compressor speeds; (1) Thermal power supplied; (2) Compression ratios. DHP: new dual-energy source heat pump; DHP-C: conventional dual-energy source heat pump

When defrosting is required in cold and humid weathers, the fuel-fired burner supplies heat for defrosting the outdoor coil. In this case, the fan of the outdoor coil shuts down and the heat pump continues to operate in the heating mode. The temperature of the outdoor coil rises and the ice that has built up melts. During the defrosting periods the heat pump thermodynamic cycle isn't reversed and therefore the indoor comfort, as well as the overall performance is improved.

6.2 Gas-to-air add-on heating mode

At outdoor temperatures below -12°C, the dual-source heat pump operates only in one stable, permanent add-on heating mode. In this mode, the compressor runs at a constant speed (60 Hz) with very interesting energy performance. Tables 2 and 3 show the main refrigerant parameters and system performance measured in this mode. The average temperature of the hot combustion gases leaving the furnace combustion chamber is 310°C. After the vaporizing and superheating the refrigerant (R-22) inside the add-on heat exchanger, the exhaust gases leave the furnace at 29°C (Figure 7.1). The compressor runs at a constant speed (60 Hz) and provides a constant refrigerant flow rate (5.1 kg/min) with very stable superheating (5°C) and sub-cooling (2.1°C) values (Figure 7.2). The indoor air flow rate is 562 L/s and its temperature inside the indoor coil increases by about 25°C. The compressor electrical power input is 1.9 kW, the thermal power supplied to the indoor air is 15.5 kW, and the average coefficient of performance is higher than 8. Compared to all-electric heating systems, common in Eastern Canada, the grid electrical power demand for each typical residence is reduced by 12.8 kW. If, for example, 100 000 houses were equipped with such a dual-energy heat pump, the grid power demand would be reduced by at least 1.28 GW during peak power demand periods. For Hydro-Québec, one of the largest Canadian hydroelectric utility, this may represent about 3.5% of its total installed electrical power in 2008.

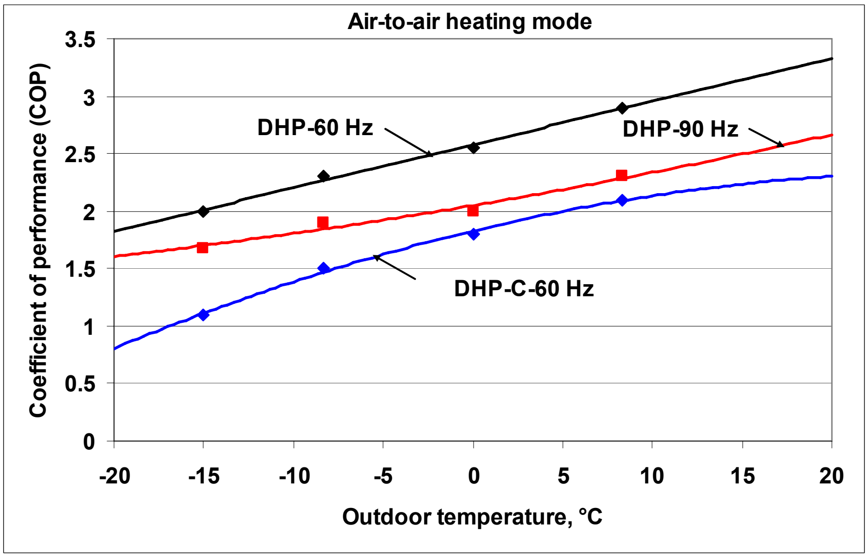


Figure 6: Coefficients of performance in air-to-air heating mode. DHP: new dual-energy source heat pump; DHP-C: conventional dual-energy source heat pump

Table 2: Refrigerant parameters in permanent add-on heating mode

Parameter (unit)	Value
Compressor suction temperature (°C)	28
Compressor discharge temperature (°C)	79.5
Compressor suction pressure (kPa)	894
Compressor discharge pressure (kPa)	1 852
Condenser inlet temperature (°C)	79.2
Condensing temperature (°C)	50.5
Condenser outlet temperature (°C)	48.4
Mass flow rate (kg/min)	5.08
Capillary tube inlet temperature (°C)	46.3
Evaporating temperature (°C)	23
Add-on heat exchanger outlet temperature (°C)	28
Add-on heat exchanger superheating (°C)	5
Condenser sub-cooling (°C)	2

7 OPERATIONAL ISSUES

Laboratory tests, as well as field monitoring of eighteen dual-energy source prototypes during two consecutive heating seasons, showed that the electrical energy savings for home heating varied between 32% and 40%, while reducing by more that 15 kW the peak electrical power demand per house. Average propane consumption was 550 liters per house, per heating season. At outdoor temperatures below -12°C, the average utilization factor of the propane gas burner was approximately 35%. The study proved that installing the compressor indoors facilitates system maintenance in cold and very cold climates, while lowering thermal losses. However, locating the compressor unit indoors requires proper noise and vibration insulation of the cabinet. It has to be preferably installed in the basement or a closed mechanical room. Furthermore, the outdoor installation of add-on unit with its auxiliary heater does not use valuable indoor space and does not contaminate the indoor atmosphere. However, careful adjustments and controls of fuel burning and combustion excess air supply are required to prevent the refrigerant/oil mixture from overheating and deteriorating.

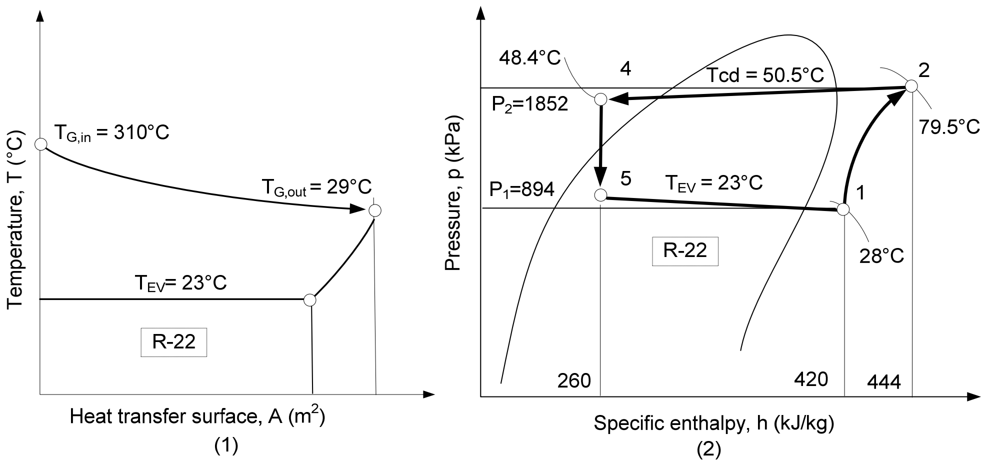


Figure 7: Measured thermodynamic parameters of the add-on coil in the gas-to-air back-up heating mode. (1) T-A diagram; (2) p-h diagram

Table 3: System performance in add-on heating mode

Parameter (unit)	Value
Propane gas flow rate (kg/h)	1.08
Propane lower calorific power (kJ/kg)	46,348
Propane burning thermal power (kW)	13.9
Add-on HEX input thermal power (kW)	13.6
Burner overall thermal efficiency (%)	98
Combustion gas temperature entering add-on coil (°C)	300
Combustion gas temperature leaving add-on coil (°C)	29.5
Condenser air flow rate (L/s)	562
Condenser indoor air temperature difference (°C)	25
Condenser thermal power output (kW)	15.5
Compressor electrical power input (kW)	1.9
Coefficient of performance (COP)	8.15

If frequently exposed to the excessive temperatures of the combustion gases, the add-on coil and soldered joins can be damaged. It was also observed that the fins' copper material deteriorates over time, and therefore new fins are required with higher fusion temperatures and resistance to combustion gas corrosion. On the other hand, if the solenoid valve located on the refrigerant liquid line is broken and doesn't open prior to entering the add-on heating mode there might be destruction hazards of the add-on coil. Also, refrigerant leakage at the add-on heat exchanger must be avoided. It can produce noxious compounds inside and/or around the gas furnace. For additional protection, in this study, the gas ignition system allowed three successive ignition attempts, but additional alarms in case of failure are required. Finally, it could be noted that the use of variable speed compressors and motor drives increases the total cost of dual-energy heat pumps. That may limit the market potential for such systems. Local liquid propane gas suppliers and storage devices are also required.

8 CONCLUSIONS

This paper presents a variable speed dual-energy source heat pump with a compact gas-to-refrigerant *add-on* heat exchanger. It is dedicated to cold and very cold climates. Some experimental results in the conventional air-to-air and permanent back-up heating modes are also presented. The new dual-energy source heat pump lowered the typical outdoor air temperature balance point from -5°C to about -9°C. Laboratory tests showed that thermal power in the *air-to-air heating mode* is higher than that of equivalent conventional heat pumps. The design improvements reduced and/or eliminated the use of an electrical back-up between the new balance point and the shutdown point. Between the new balance point and the shutdown point (-12°C), the compressor operated between 60% and 80% of the time at maximum speed, with net savings in heating thermal power of approximately 2.1 kW. Compressor speed modulation avoided transitory regimes and improved indoor comfort. In the *gas-to-air add-on heating mode*, propane gas consumption was approximately 76% lower than that of conventional dual-energy heat pumps, and the heating thermal power supplied increased by 13.3%. The thermodynamic behavior in this mode was remarkably stable. Laboratory and field experiments showed that installing the compressor indoors facilitates maintenance and lowers thermal losses. However, careful adjustments and controls of fuel burning and combustion excess air supply are required to prevent the refrigerant from overheating and deteriorating, and/or the destruction of the coil fins if exposed to excessive combustion gas temperatures. It is also necessary to improve the reliability of all heat pump components, use new GWP-free refrigerants, and reduce and/or eliminate eventual refrigerant leakage within the 4-way reversible valve in the gas-to-air add-on heating mode.

9 NOMENCLATURE

h_R - refrigerant mass enthalpy, kJ / kg

$c_{p,G}$ - combustion gases specific heat, kJ / kgK

$T_{G,in}$ - combustion gases inlet temperature, °C

$T_{G,out}$ - combustion gases outlet temperature, °C

k_f - liquid thermal conductivity, W / mK

l_{fg} - refrigerant vaporization mass enthalpy, kJ / kg

ρ_f - refrigerant saturated liquid density, kg / m^3

ρ_g - refrigerant saturated vapour density, kg / m^3

T_{sat} - saturated temperature, °C

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