IEA HPP Annex 28

North American Test Procedures and Calculations for Residential Heat Pumps with Combined Space and Domestic Hot Water Heating

Canada Interim Report

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Summary

This intermediate report gives an overview of the North American test procedures and calculations for residential appliances with separate and combined space and domestic hot water heating, and a short market survey of combined operating heat pump systems (Task 1). The hot water heat pumps (HWHP) markets in Canada and United States are relatively weak, and declined during the last two decades, notably because the total installed cost is more than double than of conventional equipment. The hot water heat pump have not marketed aggressively in Canada, and such a result awareness of product was very low or non-existent before 1995. However, the residential exhaust-air heat pump market has increased since 1995 when regulations for mechanical ventilation became mandatory. The combined appliances, while still low in number of sales, are developing a niche in new construction where the winter climate is not too severe. The alternate operation mode is not yet very common, but simultaneous space cooling or heating, and hot water heating units containing refrigerant-to-water heat exchangers, are finding greater use and are now offered by a few manufacturers. In North America, most ground-source systems are installed with desuperheaters while few air-source units are so equipped. Depending of electricity costs relative to competing fuels, the HWHP may consume 40 % to 70 % less energy than conventional electric and gas or oil water heaters under similar heating conditions. However, studies shown that payback periods are typically 5 to 15 years, especially in Canada where energy prices, both electricity and natural gas are still relatively low. Institutional factors affecting market penetration are building codes, zoning ordinances, distribution/marketing channels, and decision makers. Existing codes make no specific references to hydronic (air-to-water or water-to-water) heat pumps, but hydronic home heating systems are covered by mechanical codes. In the North American standardisation, comprehensive test procedures and calculation methods exist for simultaneous operation of air-to-air heat pumps with desuperheaters that could be used for extended to other systems, including space-cooling and space-heating only, for single-speed, dual- and variable-speed compressors, and combined space and hot water heating heat pumps. The available equations for seasonal energy efficiency ratios and heating seasonal performance factors are reliable and based on bin method. They could be used as basis for developing similar calculation methods for Europe and Japan standards. The actual heating load hours, regional heating loads, and distribution of actual cooling load hours throughout the United States are given for six climatic regions, region IV being the climatic region which is the basis for the published HSPF ratings in the United Sates, and region V for standard HSPF ratings in Canada. Some standards give meteorological data required directly in the calculation procedures. This report finally gives some indications about heating and domestic water requirements, water quality, scale and corrosion, safety devices and legionnaire's disease.

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1. Introduction

The Annex 28 aims to investigate the testing of the most common combined heat pump systems with heating of domestic hot water by de-superheating, sub-cooling of the condensate or cascade hot water heat pump. Decisive for these systems is the overall efficiency for both tasks, but – notably in Europe - the existing test procedures are almost restricted to the separate space heating or cooling, and separate heating of domestic hot water.

The first objective of the Annex 28 is to work out a test procedure to calculate the overall seasonal performance factor, and the second, to develop a method to calculate the seasonal performance factor for the heat pumps with hydronic heat distribution and alternative or simultaneous domestic hot water production.

The specific objectives of the task 1 are to investigate the most common systems for separate space and domestic hot water heating, find out the needed parameters to be measured, gather the temperatures required in different countries for domestic hot water, the daily profile of the domestic hot water consumption, and assess the existing standards on testing and design heat pumps for separate space and domestic hot water heating.

This first country report involves North-American (United States and Canada) survey of domestic hot water systems and market, and an evaluation of the existing standards and regulations.

2. System Survey

2.1 Market Shares

Heat pump market in Canada, having a mix of continental (cold winters, hot summers) and maritime (mild winters, cool summers), annual heat pump shipments are typically in the range of 30,000 units. Of these, about two-thirds are air-source heat pumps. The remaining one-third is water-to-air heat pumps used in geothermal closed–loop systems. On the other hand, the hot water heat pumps (HWHP) markets in Canada and United States are relatively weak, and declined during the last two decades. For example, in the United States, in 1984, there were 17 companies for hot water heat pumps, while in 1993, there were only 6 (six). It was suggested that hot water heat pumps had not captured a larger percentage of the annual water heater market because the total installed cost is more than double than of conventional equipment. Another reason was that air conditioning contractors are uncomfortable with plumbing systems and plumbers are uncomfortable with the refrigeration cycle. However, the combined appliances, while still low in number of sales, are developing a niche in new construction where the winter climate is not too severe and where running time of air conditioners are long (i.e., South of United States and Hawaii).

2.2 Systems on the Market

Domestic water heating represents one of the major energy costs in residential sector. There are four types of equipment that dominate the *space and water heating equipment* market in North America (**Table 2.1**). The selection of these systems is based largely on the fact that most of North America is heating dominated, and that heating with natural gas is usually less expensive than heating with electricity. In fact, sales figures indicate that, if natural gas is available at reasonable rates, gas is preferred for space heating in North America.

Table 2.1 – Space and Water Heating Dominated Equipment in North America

Function	Dominant System
Space Heating	Fossil Furnace/Boiler
Water Heating	Fossil Water Heater
Space Cooling	Electric Air Conditioner
Space Heating and Cooling	Fossil Furnace plus Air Conditioner

A domestic hot water heating system generally has a heat energy source (fuel, electrical, recovered heat or solar), a heat transfer equipment of direct or indirect type, a distribution system and terminal usage devices. The main factors to be determined for hydraulic and thermal design are the temperature, flow rate, pressure and water quality.

In Canada, the most residential water heating equipment is of *direct type*, i.e. electric, gas or oil-fired. The *electric water heaters* are generally of the automatic storage type, consisting of a tank with one or more heating elements. Thermostats controlling heating elements may be of the immersion or surface-mounted type.

Because of increasing costs and general public awareness of energy management, there has been a reappearance of the heat pump water heaters (HWHP) in North America, first introduced in the 1950's. Several types of HWHs where originally developed to help reduce the costs of heating water. Heat pump water heaters (HPWH) use a vapour-compression refrigeration cycle to extract energy from an air or water source to heat water up to a maximum output of 60°C. Most HPWH are air-to-water, and thus it provides a potential useful cooling effect and dehumidifies the air. These systems are more efficient where the inlet water temperature is low and the entering air is warm and humid. In Canada, these advantageous conditions are normally meet during the summer, but the cold and dry weather periods are much more longer.

It should be reasonable to divide North American heat pump water heating equipments into three categories :

(i) desuperheating heat pumps for domestic hot water ;
(ii) dedicated condensing heat pumps for domestic hot water ;
(iii) dedicated condensing heat pumps for swimming pool heating.

The *electric heat pumps* comes into play primarily under five circumstances:

(i) natural gas is not available and/or (ii) natural gas is not economically competitive;
(ii) vending access for a gas appliance is not available;
(iii) operating economics are dominated by cooling, and/or
(iv) there are strong need for cooling and heating in a single appliance.

2.2.1 Alternate operation mode

These systems are not yet very common in Canada and United States.

2.2.2 Simultaneous operation mode

These systems provide simultaneous space heating or cooling, and domestic hot water requirements at the same time, currently by using desuperheating heat exchangers to extract the heat at different temperature levels from the heat pump.

2.2.2.1 Combined space cooling and hot water heating

In this category, there are residential ambient air-source and exhaust-air hot water heat pumps.

Ambient air-source hot water heat pumps

The ambient air-source heat pump water heater (**Figure 2.1**) was developed in the 1970s and resembles the conventional electric resistance storage water heater [**2**]. However, it is equipped with a small heat pump unit, normally mounted on top of the storage tank. The heat pump condenser is immersion-type, while the evaporator, located in an enclosure on top of the tank, cools warm re-circulated room or ambient air.

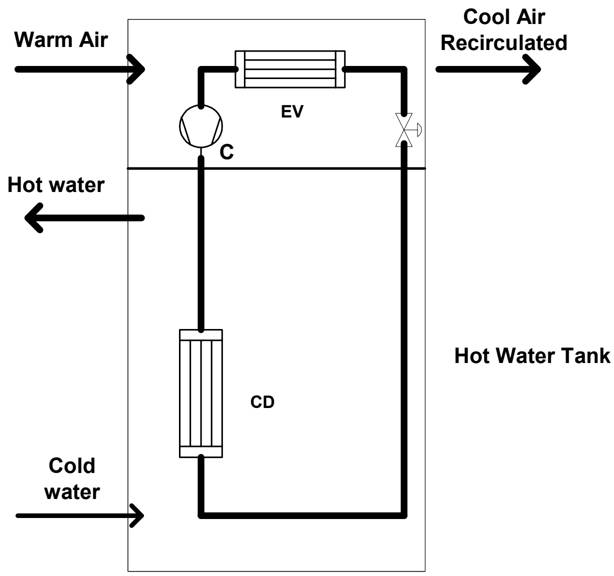

Figure 2.1 – Ambient air-source heat pump water heater
EV – evaporator; CD - condenser

Residential exhaust-air heat pumps

In the residential exhaust-air hot water heat pump (**Figure 2.2**), house warm air is cooled by the unit's evaporator and exhausted outdoors, thus providing ventilation with heat recovery. Make-up air is brought in separately. The condenser output produces domestic hot water.

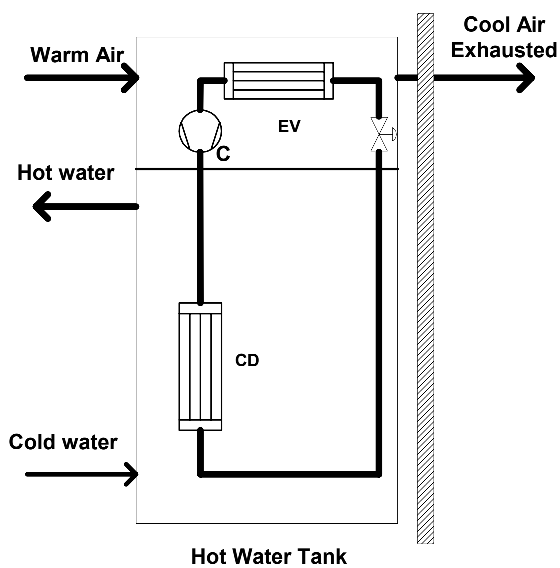

Figure 2.2 – Residential exhaust-air hot water heat pump
EV – evaporator; CD - ceondenser

2.2.2.2 Combined space heating or air conditioning, and hot water heating

The combined heat pumps for space heating or air conditioning, and hot water heating (**Figure 2.3**) are offered by a few manufacturers North-America. They imply that the space heating, cooling, water heating and, sometimes, ventilation functions are combined in the same unit, and are ideal when the heat source is other than that surrounding the domestic water tank (outdoor air). These units contain an outdoor unit with compressor and evaporator, and a refrigerant-to-water heat exchanger (desuperheater or condenser) intended to be connected to the inlet and outlet of gas, oil or electric water heaters and/or storage tanks. Water heating is thus either by desuperheating only, or desuperheating and full-condensing of refrigerant. The latter permits water heating when no space heating or cooling is required.

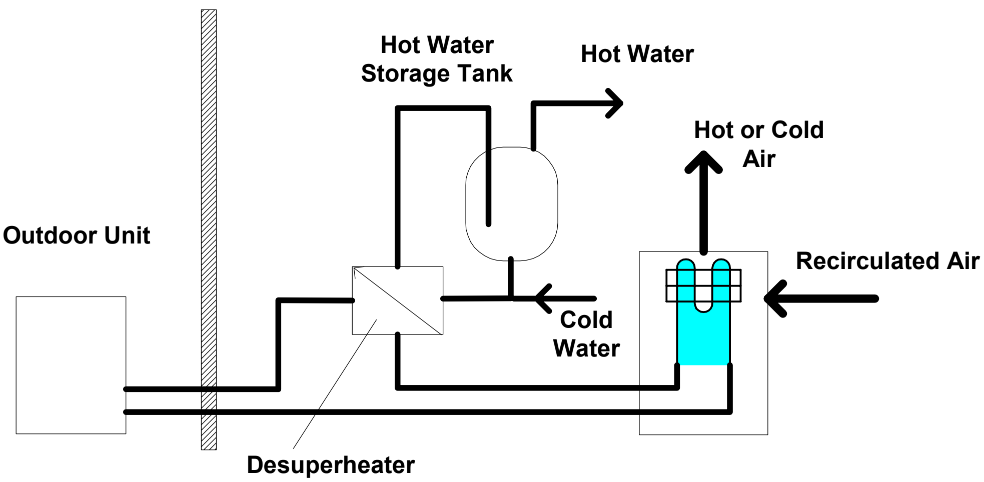

Figure 2.3 - Combined Space Heating or Air Cooling, and Hot Water Heating Heat Pump

Desuperheating Hot Water Heat Pumps

As noted, a desuperheating hot water heat pump is a refrigerant hot gas-to-water heat exchanger which is installed between the compressor and the reversing valve of a conventional heat pump. It is a heat exchanger that removes the high-temperature superheat available in the refrigerant gases exiting the heat pump compressor prior to entry into the refrigerant condenser, and can be used on both air-to-air and water-to-air heat pumps. Domestic hot water heating using a heat pump can significantly decrease energy costs during both heating and cooling seasons. Depending on the heat pump design and operating conditions, superheat temperatures of 93°C (200°F) and greater may be reached. By removing the superheat, desuperheaters can provide a source of water typically at a temperature of 71°C (160°F).

In the United States, for many years, desuperheating systems have been successfully retrofitted on *central air conditioning systems*. For example, some 50,000 units have been installed during

1985 on central air conditioning to provide hot water. Frequently, such systems were reported to pay for themselves in only one year. Heating of hot water is not "free" but is provided at a reduced cost based on the COP of the heat pump.

On *ground-source heat pumps*, desuperheaters may provide most of the domestic hot water required for a typical residence. During summer cooling cycles, it is not uncommon for the homeowner to turn off the power serving the hot water heater's electric heating elements. During the summer, the entering water temperatures are higher, resulting in higher superheat temperatures. In winter, domestic hot water production will be reduced because of lower entering water temperatures from both the domestic water supply and the ground heat exchanger.

The benefits of a desuperheater are totally dependent upon the *water use profile* of the occupant. A typical middle-class North American home might have a 3-ton heat pump. The heat rejected to the desuperheater would be on the order of 2.9 kW to 3.5 kW (10,000 to 12,000 Btu/h). This compares very unfavourably with the 14.65 kW to 17.6 kW (50,000 to 60,000 Btu/h) of a conventional gas water heater, but more favourably with the typical electric water heater. As a consequence, the percentage contribution of the desuperheater to the total water heating load may be quite small. On the other hand, the best application of a desuperheater may be for pool heating.

System Layout and Installation

The desuperheater must always be installed as close to the compressor discharge as possible and upstream of the reversing valve (**Figure 2.4**).

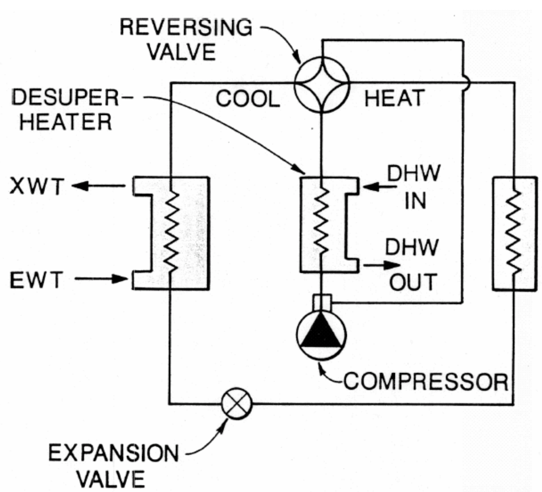

Figure 2.4 – Recommended Desuperheater Placement
DHW – Domestic Hot Water; EWT – Entering Water Temperature;
XWT – Leaving Water Temperature

The compressor discharge temperature have to be at least 10°C (20°F) higher than the desired end use water temperature. In residential applications, the desuperheater should generate enough hot water to satisfy nearly all domestic needs while the heat pump is operating. It is acceptable to circulate water directly from the desuperheater to the existing domestic hot water tank (**Figure 2.5**). The aquastat on the lower hot water tank heating element should be lowered 5.5° to 11°C (10° to 20°F) below the normal setting. Auxiliary heating in the tank will activate only when water usage is high and the temperature falls below the set point on the aquastat. The desuperheater circulating pump (circulator) is cycled with the compressor.

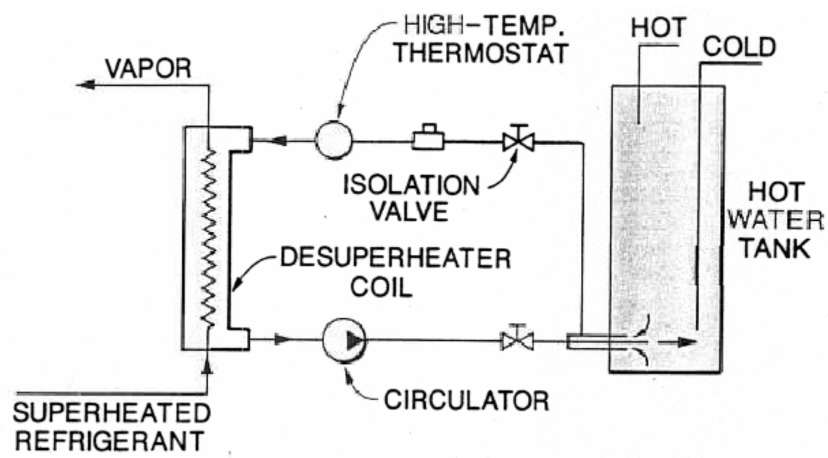

Figure 2.5 - Desuperheater System Configuration

A high-water temperature thermostat should be used to prevent overheating of the hot water. If the desuperheater is unable to meet the hot water demand, it is suggested an intermediate tank (**Figure 2.6**). Preheated water stored in the intermediate tank is then fed directly to the existing hot water heater. A good rule of thumb for water storage is 37.85 L/ton (10 gallons/ton) for HCFC-22.

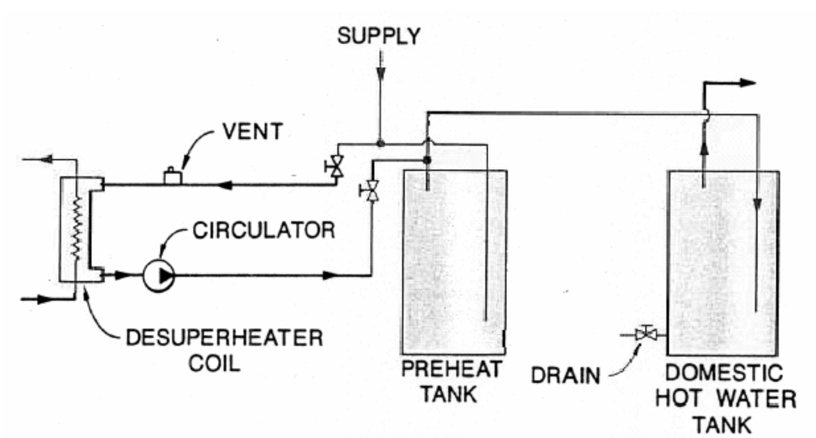

Figure 2.6 – Desuperheater Two-Tank System

Since the energy savings achievable by a de-superheating system are related to the amount of operating time of the heat pump and its capacity, the energy savings have to be correlated to space heating and cooling loads [**3**]. The results shown in **Figure 2.7** for desuperheaters on air-source heat pumps units assume a daily hot water consumption of 285 litres at 60°C, 20 % of heat pump capacity is available to heat water and the storage capacity of the hot water tank is large enough to offset any effect of non-simultaneous heat pump operation and hot water demand. The simulations included the effects of cycling losses and diminished space heating capacity when recovering superheat. The percent energy savings shown can be adjusted to account for hot water usage different than the assumed 285 litres. This adjustment is given by:

Water heating savings (%) = (0.00254*Daily water usage in Litres) + 0.28)*(% savings from **Figure 2.7**)

From **Figure 2.7** it is evident that desuperheaters achieve increasing energy savings with increasing cooling load. However, the impact of heating load on energy savings is not so clear. Winter space heating loads between 10,000 kWh per year and 12,000 kWh per year appear to allow for the greatest water heating energy savings. At this range of load, heat pumps will be operating for many hours above the balance point, thus able to meet space and water heating loads. Below this range, the number of running hours diminishes aver-all, resulting in less opportunity for energy savings. Above this range, the heat pump is operating for more hours below the balance point and, as a result, is not able to meet the loads, resulting in the use of the back-up heating system and no capacity available for water heating.

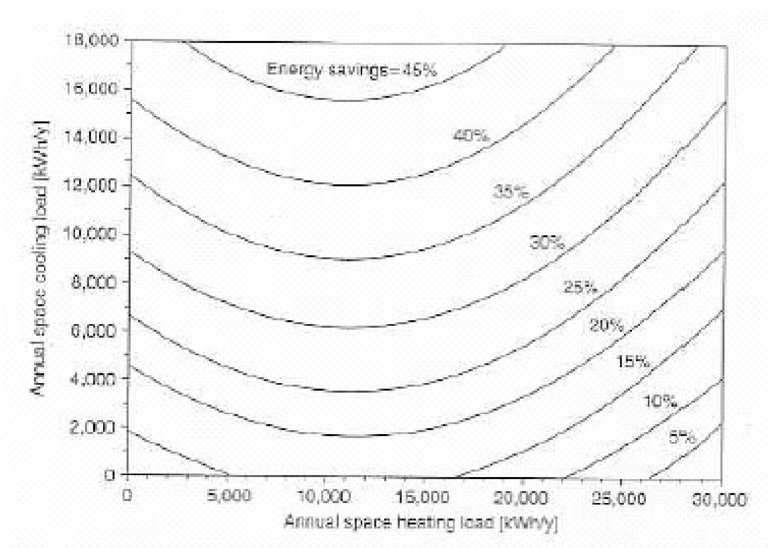

Figure 2.7 – Water heating energy savings with a desuperheater

Desuperheaters can also be installed on *ground-source heat pump* systems. From an efficiency standpoint, the use of a desuperheater increases the heat pump cycle time, which increases the coefficient of performance (COP). Summer heat rejection loads to the ground heat exchanger are reduced by using a desuperheater, which makes its use and cost easily justified.

Desuperheaters perform better on ground-source systems during the heating season than on air-source systems. As outdoor air temperature decreases, the heating capacity of an air-source heat pump decreases reducing the desuperheater capacity. Since ground temperatures do not experience the same decrease, desuperheater capacity on ground-source systems remains relatively constant.

Carrier Corporation has field tested 19 of their "*Hydrotech"* systems throughout the United States [**4**] and determined that a desuperheater was able to supply, on average, 36 % of the hot water load. Utilizing the full-condensing capabilities of the "*Hydrotech*" system led to the heat pump system supplying a further 50 % of the hot water requirements. Unfortunately, data is not available to quantify energy savings based on space heating and cooling loads. But, as space loads decrease, more time is available for full-condensing water heating. Therefore, it is expected that water heating energy savings from full-condensing will increase as space loads decrease, a trend contrary to that experienced with de-superheaters.

Manufacturer's Performance Data

Some North-American manufacturers provide desuperheater capacity as a function of heat pump entering water temperature. The example given in **Table 2.2** is based on a factory-installed de-superheating unit. Heat pump size and performance are as follows: Cooling Capacity = 14.8 kW (50,500 Btu/h) at 21°C entering water temperature (70°F EWT); Nominal Capacity = 4.2 tons; Heat Pump Flow Rate = 0.567 kg/s (9 gpm); Desuperheater Flow Rate = 6.358 L/min (1.68 gpm) or 1.5 L/min/ton (0.4 gpm/ton). Based on a nominal capacity of 4.2 tons, desuperheater capacity ranges from 0.32 to 0.84 kW/ton (1,095 to 2,881 Btu/h/ton) for the ranges of entering water temperatures for heating and cooling seasons. These capacities are based on the manufacturer's desuperheater flow rate of 1.5 L/min/ton (0.4 gpm/ton).

Table 2.2 – Example: Measured desuperheater performances

Entering Water Temperature	Heat Pump Capacity	Desuperheater Capacity
°F	**Btu/h**	**Btu/h (Btyu/h/ton)**
Heating : Entering Air temperature 70°F		
30	34,100	4,600 (1,095)
50	45,800	6,900 (1,643)
70	54,700	8,500 (2,023)
Cooling: Entering Air Temperature 80°F Dry Bulb; 67°F Wet Bulb		
70	52,200	7,500 (1,786)
90	46,900	9,800 (2,333)
110	41,600	12,100 (2,881)

Design and Selection

There are many factors that influence the proper design and selection of desuperheaters. The most important are compactness, cost, and an acceptable level of heat transfer. Other factors include:

(i) *Vented double wall*: many local and/or state codes require that the desuperheater be constructed with a “vented double wall”. In the event of failure in one wall, either refrigerant or water is vented to the atmosphere from the heat exchanger. This allows a leak to be detected prior to any mixing of domestic water with refrigerant and/or oil.

(ii) *Field or factory installed*: generally, a factory-installed unit on a heat pump is preferred. Factory installation is normally under more controlled conditions.

(iii) *Ratings and sizing*: desuperheater ratings are based on the nominal cooling capacity of the system. For example, a 5-ton desuperheater is usually appropriate for an application on any 5-ton cooling system. However, a 5-ton desuperheater can also be installed on a smaller system, but the reverse is not true. Desuperheaters should not be undersized. This can result in an excessively high pressure drop in the refrigerant circuit which in turn results in poor performance of the heat pump circuit.

(iv) *Retrofit applications*: if the circulating pump is purchased separately, a nominal water flow rate of about 0.75 USgpm (0.0437 L/s) per ton of cooling in recommended by manufacturers of de-superheating equipment.

(v) *Ratings*: As a rule of thumb, a desuperheater will transfer approximately 1.025 kW/ton (3,500 Btu/h/ton) or 22.7 L/h/ton (6 USgallon/h/ton) at an entering water temperature of 21°C (70°F) and an exit water temperature of 60°C (140°F) for HCFC-22 systems (**Figure 2.8**). This design value assumes specific compressor discharge temperatures and will vary for heating and cooling cycles.

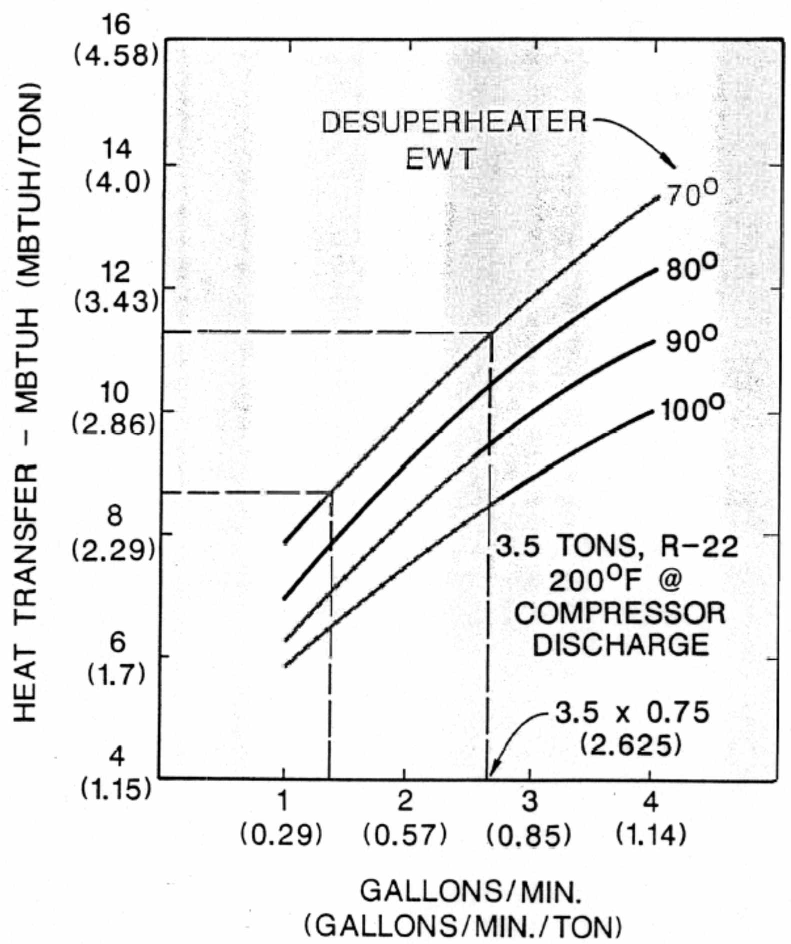

Figure 2.8 – Desuperheater Performance

The standard *ARI 470-1987* [**6**] establish for desuperheaters/water heaters procedures for testing and rating, safety provisions and requirements for marking. Ratings shall be determined in accordance with cleaned tube ratings and fouled ratings. **Table 2.3** indicates initially cleaned tubes conditions with a fouling factor of 0 $m^2.°C/W$.

Table 2.3 – Standard rating conditions

Type System	**Saturated Temperature of Entering Refrigerant Vapour**	**Actual Temperature of Entering Refrigerant Vapour**	**Temperature of Entering Water**	**Temperature of Leaving Water**
-	**°C**	**°C**	**°C**	**°C**
Air Cooled	51.7	104.4	32.2 and 48.2	60
Water Cooled	40.6	82.2	32.2 and 48.2	60

From the results of the cleaned tube test, it can be calculate *the cleaned tube overall heat transfer coefficient* (U_C):

$$U_C = \frac{\dot{Q}}{A*(LMTD)_C},$$

where $(LMTD)_C$ is defined by:

- For counter flow desuperheater: $(LMTD)_C = \dfrac{(T_{RE} - T_{WL}) - (T_{RL} - TE)}{\ln(\dfrac{T_{RE} - T_{WL})}{T_{RL} - T_{WE})}}$

- For parallel flow: $(LMTD)_C = \dfrac{(T_{RL} - T_{WL}) - (T_{RE} - T_{WE})}{\ln(\dfrac{T_{RL} - T_{WL})}{T_{RE} - T_{WE})}}$

where:

A is the total heat transfer area (m^2),

$(LMTD)_C$ - logarithm mean temperature difference (°C),

$\dot{Q}$ – total thermal power transferred (W),

$\mathrm{T_{RE}}$ and $\mathrm{T_{RL}}$ - refrigerant entering and leaving temperatures (°C),

$\mathrm{T_{WE}}$ and $\mathrm{T_{WL}}$ - water entering and leaving temperatures (°C).

The total thermal resistance is equal to the reciprocal of the overall coefficient of heat transfer:

$$R_C = \frac{1}{U_C} = \frac{A*(LMTD)_C}{q}.$$

The next step in determination of fouled ratings is the calculation of the total thermal resistance including fouling. This is found by adding the specified fouling resistance to the cleaned-tube overall resistance.

- Refrigerant outside tubes (fouling inside)

  (a) Basing calculations on outside surface area: $R_{FO} = R_{CO} + \left(\dfrac{A_O}{A_I}\right)$

(b) Basing calculations on inside surface area: $R_{FI} = R_{CI} + r_{FI}$

- Refrigerant inside tubes (fouling outside):

(a) Basing calculations on outside surface area: $R_{FO} = R_{CO} + r_{FO}$

(b) Basing calculations on inside surface area: $R_{FI} = R_{CI} + r_{FO}\left(\frac{A_I}{A_O}\right)$

The actual heat transfer under fouled conditions ($\dot{Q}_F$) is calculated from:

$$\dot{Q}_F = \varepsilon \dot{Q}_{\max} \ ,$$

where:

$\dot{Q}_{\max} = C_{\min}(T_{RE} - T_{WE})$, in W,

$C_{\min}$ - the smaller of :

- $C_{hot} = \dot{m}_R . c_{pR}$ - hot fluid (refrigerant) capacity rate at rating conditions (W/°C).
- $C_{cold} = \dot{m}_W . c_{pW}$ - cold fluid (water) capacity rate at rating conditions (W/°C),

where $\dot{m}_R$ and $\dot{m}_W$ respectively represent refrigerant and water mass flow rate (kg/s), and c_{pR} and c_{pW} - respectively, refrigerant and water specific mass heat at constant pressure (kJ/kg.°C).

ε is determined for either counter flow or parallel flow arrangements:

Counter flow: $$\varepsilon = \frac{1 - e^{-N_{tu}(1 - C_{\min}/C_{\max})}}{1 - (C_{\min}/C_{\max})e^{-N_{tu}(1 - C_{\min}/C_{\max})}}$$

Parallel flow: $$\varepsilon = \frac{1 - e^{-N_{tu}(1 + C_{\min}/C_{\max})}}{1 + C_{\min}/C_{\max})} \ ,$$

Where $C_{\max}$ is the larger of C_{hot} or C_{cold},

N_{tu} - the number of exchanger heat transfer units.

Now, having the actual heat transfer under fouled conditions, the leaving water and refrigerant temperatures can be calculated from:

$$T_{WL} = T_{WE} + \frac{\dot{Q}_F}{C_{cold}}$$

$$T_{RL} = T_{RE} - \frac{\dot{Q}_F}{C_{hot}}$$

The combination of $\dot{Q}_F$, T_{WE} and T_{RE} define *the fouled ratings at the given operating conditions*.

2.2.3 Available Equipment

Some of the North American manufacturers of hot water heat pumps (HWHP) and de-superheating systems, and the water heating capacities offered by each manufacturer, are listed in **Table 2.4** [**2**]. The offered water heating capacities are based on the hot water loads that in North-America vary between 5,000 and 6,000 kWh per year. Most systems are dedicated hot water heat pumps with a few manufacturers of systems integrated with space conditioning.

Table 2.4 – North-American Manufacturers of HWHP Equipment

Type of HWHP and manufacturer	**Country**	**Water Heating Capacity**
-	**-**	**kW**
Ambient Air-Source HP Systems (with or without tank)		
Therma-Stor Products Group	US	Up to 18.1
Reliance Water Heater Company	US	3.3
State Industries Inc.	US	3.3
Crispaire Company	US	> 3.3
Energy Utilization Systems	US	3.5
Florida Heat Pump Inc.	US	7.9
Exhaust-Air HWHP Systems		
Dec International - *Therma-Stor* Product Group	US	2
Heatrade Inc.	Canada	Up to 22
Carrier Corporation	US	6
DEC International - *Therma-Stor* Product Group	US	n/a
Nordyne Inc.	US	n/a
Desuperheating Water Heaters		
Addison Products Company	US	Up to 17.6
Enro Manufacturing Inc.	US	5.6
Climate Master Inc.	US	2.6

Cost

An EPRI handbook [Friedman, 1989] indicates that the installed cost for a residential ambient air-source hot water heat pump with a heating capacity of 0 to 5.3 kW, can vary from 340 to 680 US$/kW, and the annual maintenance costs, between 100 and 400 US$/kW.

Performance and Pay-back Period

For ambient air-source hot water heat pumps, the coefficient of performance increases with increasing capacities. Performance characteristics of HWHP indicate that, depending if the relative prise of the electricity and fossil fuels, they consume 40 % to 70 % less energy than conventional electric and gas or oil water heaters under similar heating conditions. A monitoring program of the Canadian Electrical Association (CEA) showed that HWHP systems reduce water heating costs by up to 50 %, when compared to conventional electric and oil water heaters. HWHPs have the added benefits of dehumidifying and air conditioning with the option of using waste heat from various sources. Payback periods are typically 10 to 15 years (not accounting for the air conditioning feature) for residential applications.

The graph shown in **Figure 2.9** plot simple pay-back period in years versus fuel price ratio for several conditions. The annual COP of the hot water heat pump was assumed to be equal to 2.0, and the annual hot water load on the water heater is 2,500 kWh. The cost ratio (CR) is defined as the capital cost of the heat pump system minus the base system cost in thousands of monetary units (i.e. in thousands of US$), divided by the customer cost for electricity expressed in the same monetary units as incremental cost (i.e. US$/kWh). For hot water loads and COPs that differ from the assumed values, the fallowing equation can be used to calculate simple payback periods:

$$\text{Payback period} = \frac{\text{Cost ratio} * 1000}{\left(\frac{1}{\text{Fuel price ratio}} * \frac{\text{Hot Water Load}}{\text{COP}_{\text{BASE}}} - \frac{\text{Hot Water Load}}{\text{COP}_{\text{HP}}}\right)}$$

The *fuel price ratio* is defined as the customer cost for electrical energy divided by the customer cost for energy for the competing base technology, in similar units. The base technology considered is a fuel-fired hot water heater with an annual efficiency (COP) of 0.55. The energy for the base technology could be natural gas or light oil. The *fuel price ratio* is a dimensionless ratio, i.e. independent of monetary unit. The graph do not account for annual maintenance costs, a factor which may become significant especially for larger systems.

Refrigerants

The refrigerant HCFC-22 (ODP = 0.05) has been used in the majority of models offered by North American companies (up to 120 kW capacities), but HFC-134 (ODP = 0) is today currently used.

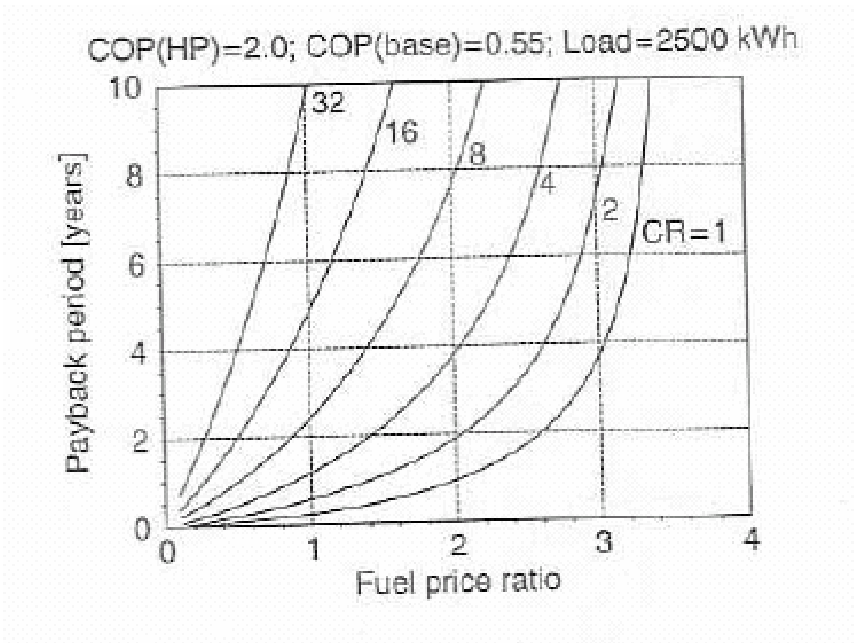

Figure 2.9 – Residential hot water heat pump simple payback period versus fuel price ratio [**2**]
CR – Cost Ratio

2.2.4 Benefits, Market Drivers and Barriers

Benefits

An economic cost comparison was performed for a residential hot water heating system, from a societal perspective [**2**]. This represents an attempt to quantify the benefit of hot water heat pumps and desuperheaters, versus conventional water heaters, to society as a whole. A society includes energy utilities as well as their rate payers. The societal cost differs from the simple payback period. The simple payback period anly account for capital cost and operating and maintenance costs. The societal cost, which is a life cycle cost, includes the capital and maintenance costs plus the utility costs to generate and supply the power and energy to the water heating system (**Table 2.5**).

Table 2.5 - Comparison of societal costs – residential application (1993) [**2**]

Technology	**HWHP**	**Desuperheater**	**Electrical Resistance**	**Gas-fired Heater**	**Calculation**
COP	2.0	-	0.9	0.55	cop
Energy used for a 5,500 kWh annual hot water load (u)					
a. Electricity (kWh)	2750	4610	6110	0	(e = u/cop)
b. Natural gas (m^3)*	0	0	0	957	(g = u/cop/hhv)
Cost to electric utility					
1. Power – Generation station ($/kW)	70	70	70	0	(y)
2. Cost of energy ($/kWh)	0.05	0.05	0.05	0	(z)
3. Power cost ($)	1433	2866	2866	0	(pc = p*x*y)
4. Energy cost ($)	1251	2098	2780	0	(ec = e*x*z)
TOTAL	2684	4964	5646	0	(uc = pc + ec)
Present value cost of gas ($/ m^3)	0	0	0	1742	(gc = 0.2*g*x)
Capital cost ($)	1500 - 3000	1000	500	500	(i)
Incremental maintenance cost ($)	50	-	-	-	(m)
Present value of maintenance cost ($)	455	-	-	-	(mc = m*x)
Total capital, O&M, gas ($)	4639 - 6139	5964	6146	2242	(tc = I+uc+gc+mc)

* The energy content of gas is assumed to be 37.6 MJ/ m^3 (hhv)

The cost of power generation (70 $/kW) represents an annualized value of the cost to provide new incremental generation capacity. The cost of transmission and distribution systems are not included in this calculation. The cost of energy (0.05 $/kWh) is assumed to be a composite value for seasonal and on-peak/off-peak variations. The annuity factor is based on a fifteen year life of equipment with 7 % per year discount rate. The assumed cost of gas was 0.02 $/ m^3 (1993), and the escalation of energy prices was assumed equal to inflation. Capital cost of the hot water heat pump system is based on a 4.5 kW heating capacity unit. It was assumed that the desuperheater had electric resistance for backup and that the gas-fired water heater shared a vent with a gas furnace. The desuperheater was assumed to save 1500 kWh per year versus an electric resistance water heater while having no impact on the peak electric power required. Maintenance costs were estimated to be 100 $/year for the hot water heat pump system and 50 $/year for the competing systems.

Table 2.5 shows that the total societal cost of the hot water heat pump system was estimated to be within the range of 4639 – 6139 $. This range is lower than the estimate of 6146 $ for the electric resistance water heater. Therefore, for the assumptions used in this example, society benefits from the installation of a hot water heat pump rather than an electric resistance heater. However, when compared to a gas-fired water heater, society does not benefit. The estimated societal cost of the gas-fired water heater is 2242 $, substantially below the range of cost of the hot water heat pump. In this example, the space cooling effect of the hot water heat pump was not taken into account. Rather, it was assumed that any benefit during the cooling season is offset by a penalty during the heating season.

The desuperheater, with a societal cost of 5964 $, shows a benefit versus an electric resistance water heater. To facilitate this comparison, both the desuperheater and the electric resitance water heater were assumed to be installed where there are heat pumps providing space conditioning. The desuperheater case should only be compared versus the electric resistance water heater as this is the ususl situation where a desuperheater is considered for installation.

In conclusion, the example showed that the hot water heat pump has a lower societal cost than electric resistance water heating in residential applications when owner capital and maintenance and utility costs to supply power and energy are taken in account. However, natural gas-fired water heating was found to have a much lower societal cost than the hot water heat pump in North America (1993).

Competitiveness

The economic attractiveness of residential heat pumps water heating depends on electricity costs relative to competing fuels, hot water requirements, incremental capital cost over conventional water heating equipment and hot water heat pump coefficients of performance. In Canada, the natural gas price has substantially been increased since 2000, and the use of hot water heat pumps could be more interesting for residential market.

Utility Studies

Most of the utility studies concerned with the demand reduction achievable by hot water heat pumps versus electric resistance water heaters. Peak water heating demand reductions of up to 3.0 kW were reported in Canada and the United States. However, a report of the "*Florida Solar Energy Centre*" identified that hot water heat pumps measured a 200 Watt reduction and de-superheating systems a 720 Watts reduction on power at the time of the utility peak demand.

Factors Affecting Market Penetration

Market penetration of hydronic heat pump heating systems in North America is dependent on technical and institutional factors and utility rate structure. There are not major technical factors that will hinder market penetration of hydronic cost-optimized heat pump heating systems. They can be designed and installed to provide acceptable reliability and comfort. Traditional complaints about cool circulation air with air-to-air heat pump systems should not occur with hydronic heat pump systems. Institutional factors affecting market penetration are building codes, zoning ordinances, distribution/marketing channels, and decision makers. Existing codes make no specific references to hydronic (air-to-water or water-to-water) heat pumps, but hydronic home heating systems are covered by mechanical codes.

Marketing Programmes

Many North-American utilities have adopted in the past marketing or incentive programmes to promote hot water heat pumps and/or desuperheaters (**Table 2.6**).

Table 2.6 – Example of North-American utility incentive and promotion programmes

Utility	Country	Status
Ontario Hydro	Canada	"*Saving by Design*" program allowed incentives of up to CD$ 500/kW or 50 % of the incremental cost
Alabama Power Company	USA	Incentive of US$ 400/kW for commercial units
Hawaiian Electric Company	USA	State tax credit of 20 %
Potomac Electric Power Co.	USA	Incentive of US$ 500/HWHP under 190 Litres; US$ 1000/HWHP over 190 Litres
Wisconsin Public Service Corp.	USA	Offers an incentive of US$ 300/kW for commercial customers (1993)
Hydro-Quebec	Canada	Incentives for geothermal heat pump high efficiency systems (2004 - 2007)

Market Drivers and Barriers

Hot water heat pumps have had difficulty in becoming an established technology in North America due to their high incremental costs and low cost of energy. In certain regions of the United States, however, sales of hot water heat pumps do very well. In the states of Hawaii and Florida, both hot climates with little or no heating degree days, hot water heat pumps and de-superheaters are very competitive. The maximum technical increase in efficiency for water heating was estimated by EPRI [**5**] at 30 % to 60 % if every water heater was replaced by a hot water heat pump. The efficiency improvement in residential water heating was estimated to be attained primarily through changes in the marketplace and by regulatory mandates such as minimum energy standards. A similar situation exists in Canada where energy prices, both electricity and natural gas are still low resulting in long payback periods for hot water heat pumps. The hot water heat pump have not marketed aggressively in Canada, and such a result awareness of product was very low or non-existent before 1995. However, the residential exhaust-air heat pump market has increased since 1995 when regulations for mechanical ventilation became mandatory.

3. Boundary Conditions for Calculation

3.1 Meteorological Data

The actual heating load hours, regional heating load hours, and distribution of actual cooling load hours throughout the United States are given for six climatic regions. Generally, region IV is the climatic region which is the basis for the published HSPF ratings in the United States, and region V for standard HSPF ratings in Canada. Some standards (i.e., *ASHRAE Standard 116-1995*) give meteorological data required directly in the calculation procedure for cooling seasonal energy efficiency ratios and heating seasonal performance factors [**7**].

3.1.1 Air

Most of the recommended outdoor design conditions (temperatures, wind, solar radiation, underground data, etc.) for calculation of the seasonal performance are based on data from the US National Climatic Data Centre and Canadian Atmospheric Environment Service [**11**]. Design temperatures are based on the assumption that the frequency level of a specific temperature over a suitable time period will repeat in the future. The selected winter and summer temperature frequencies enable to match the risk level desired. For **winter** calculations, two frequency levels are offered representing temperatures that have been equalled or exceeded by 99% or 97% of the total hours in the months of December, January and February (a total of 2169 hours). In a normal winter there would be approximately 22 hours at or below the 99% value and 54 hours at or below 97.5% value. In Canada, the two design values are based on only the month of January. The Canadian design temperatures are a few degrees lower than those based on three winter months. For **summer**, the dry-bulb temperatures represent values that have been equalled or exceeded by 1%, 2.5% and 5% of the total hours during the months of June through September (a total of 2928 hours). The coincident wet-bulb temperature is the mean of all wet-temperatures occurring at the specific dry-bulb temperature. For Canada, The three values are based on the month of July only, and the corresponding design values are a few degrees higher that those based on four summer months.

3.1.2 Soil and Groundwater

The performances of the ground-source heat pumps depend on the ground's thermal properties. The ground temperature near the surface cycles with the time of the year. These variations disappear at lower depths, where the ground remains at its mean temperature throughout the year. An equation for calculating the ground temperature at any time of the year was developed based on the mean ground temperature (T_{mean}), the surface amplitude (A_S), and the time of minimum surface temperature (t_0). The annual surface amplitude os one-half the temperature range ($T_{surface,\max} - T_{surface,\min}$). The environmental agencies provide the maximum and minimum temperatures at very shallow depths (i.e., 4 in.), and the date at which $T_{surface,\min}$ occurs (t_0) is also available.

The thermal conductivity of the soil (k_s) and the thermal diffusivity – a measure of the ground's ability to conduct energy relative to its ability to store thermal energy ($\alpha = k_s / C_p$) are two soil and rock properties that most affect the design of the heat pump systems (**Table 3.1**).

Table 3.1 – Thermal properties of Common Ground Types [**12**]

Material	**Conductivity**	**Diffusivity**	**Density**	**Heat Capacity**
	Btu/hr.ft.°F	ft^2 / hr	lb / ft^3	Btu/lb.°F
Dense Rock (Granite)	2.0	0.05	200	0.20
Average Rock (Limestone)	1.4	0.04	175	0.20
Heavy Soil – Damp	0.75	0.025	131	0.23
Heavy Soil – Dry	0.50	0.020	125	0.20
Light Soil – Damp	0.50	0.020	100	0.25
Light Soil - Dry	0.20	0.011	90	0.20

1 Btu/hr.ft.°F = 0.1442 W/m.K; Btu/lb.°F = 4.184 kJ/kg.K

The moisture content of soil has a considerable impact on its thermal properties. The thermal conductivity of soil is relatively constant above a specific moisture threshold, called critical moisture content (CMC) (**Table 3.2**). Below the CMC, the conductivity drops rapidly. For Canadian and many northern U.S. locations, thermal instability is not a significant concern. The relatively high tables and smaller cooling loads prevent the moisture content from dropping below critical levels.

Table 3.2 – Critical Moisture Content [**12**]

Soil	**Critical Moisture Content (%)**
Granular	< 12
Silts	12 – 16
Clays	16 – 22
Organic and Peaty Soils	18
Organic and Expansive Clays	> 22

Water movement has also a significant impact on heat transfer through the ground because the heat transfer by conduction is reinforced by convection due to the moving water.

For groundwater systems, the entering water temperatures are approximated by the mean ground temperatures. In Canada, the groundwater temperatures approximately vary from 6°C (northern regions) to 10°C (South), and maps with isothermal curbs are available for geothermal systems designers.

3.1.3 Exhaust Air

For residential applications, small-scale packaged ventilators with built-in heat recovery components known as heat recovery ventilators (HRV) are available. These air-to-air recovery devices may be sensible heat devices (i.e., transferring sensible energy only) or total heat devices (i.e., transferring both sensible energy and moisture). The most popular devices use fixed plate heat exchangers, rotary wheels, heat pipes and integrated *heat pumps*. *ASHRAE Standard 84* (Method of Testing Air-to-Air Heat Exchangers) establishes rating and testing procedures for commercial air-to-air heat recovery equipment. The Canadian standard *CAN/CSA-C439* (Methods of test for Rating the Performance of Heat Recovery Ventilators) is used to rate small (under 200 L/s air flow rate) packaged ventilators with heat recovery.

3.2 Energy Requirements

3.2.1 Heating Requirement

Residential heating load calculation is performed in accordance with *ASHRAE Handbook – Fundamentals*. For single-family detached homes (having exposed walls in four directions, more than one store and a roof), multifamily buildings, duplexes, town houses or condominiums, the general procedure for calculation of design heat losses involves the selection of outdoor and indoor design conditions, select the transmission coefficients, compute the heat losses for walls, floors, ceilings, windows, and floor slabs, the heat load due to infiltration and outdoor ventilation air.

3.3.2 Domestic Water Requirement

Annual variation of the *cold supply water temperature* vary in function of the season. **Figure 3.1** shows, as an example, the annual variation of the cold water supply temperature in Toronto (Canada). This temperature have to be used for designing the electric heating elements or other heat source devices. Representative hot water temperature requirements for some services are shown in **Table 3.3.**

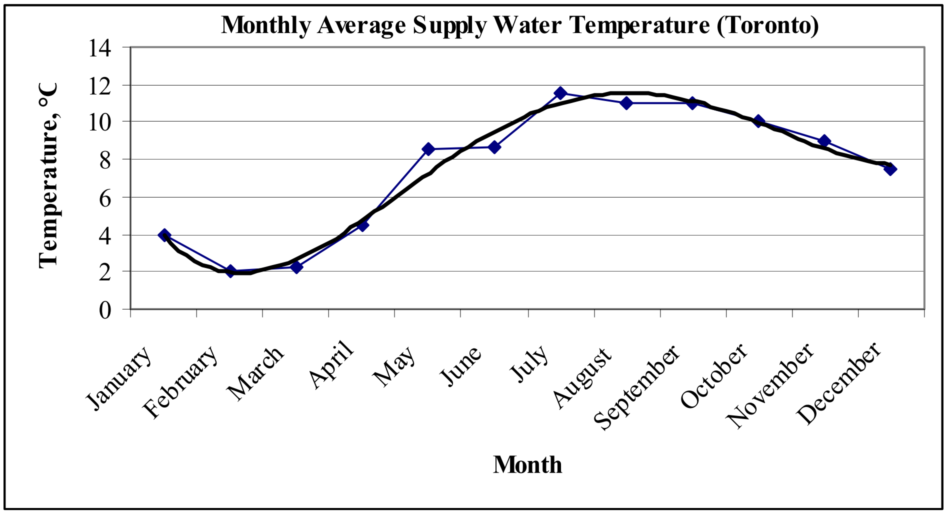

Figure 3.1 – Annual Variation of the Cold Water Supply Temperature in Toronto

Table 3.3 – Representative Hot Water Temperatures [**12**]

Use	Temperature, °C
Hand washing	40
Showers	43
Residential dish washing and laundry	60

The **Table 3.4** shows typical hot water usage in a North American residence.

Table 3.4 - Typical residential Usage of Hot Water per Task [**12**]

Use	High Flow	Low Flow (Water Savers Used)
-	L	L
Food preparation	19	11
Hand dish washing	15	15
Automatic dishwater	57	57
Clothes washer	121	80
Shower or bath	76	57
Face and hand washing	15	8

To determine ***hot water demands***, generally manufacturer's specifications for fixtures and appliances are used (**table 3.5**).

Table 3.5 – Hot water demand per fixture for residential (60°C water final temperature) [**1**]

	Fixture	Hot water demand (L/h)
1	Lavatory	7.6
2	Bathtub	76
3	Dishwater	57
4	Foot basin	11
5	Kitchen sink	38
6	Laundry	76
7	Pantry sink	19
8	Shower	114
9	Service sink	76

The code of HUD-FHA (Minimum Property Standards for One- and Two-Family Living Units – 4990.1-1982), establishes minimum permissible water heater sizes (**Table 3.6**). Storage water heaters may vary from the sizes shown in the table if combinations of recovery and storage are used that produce the required 1-hour draw.

Table 3.6 – Minimum Water (Electric) Heater Capacities for One- and Two-Family Living Units

Number of Baths	**1 to 1.5**		
Number of Bedrooms	**1**	**2**	**3**
Storage, L	76	114	150
KW input	2.5	3.5	4.5
One-hour draw, L	114	167	220
Recovery, mL/s	10	15	19

The *first hour rating* (FHR) is the amount of hot water that the water can supply in one hour of operation (DOE 1990). It is gaining recognition as a *sizing criterion* for selecting water heaters. The procedure for determining the first hour rating specifies that the tank initially be full of heated water (DOE 1993). The linear regression line for electric heaters is showed in **Figure 3.2**. Regression lines for *heat pump* heaters are not available because of limited data. The first hour rating represents water heater performance characteristics that are similar to those represented by the one-hour draw values.

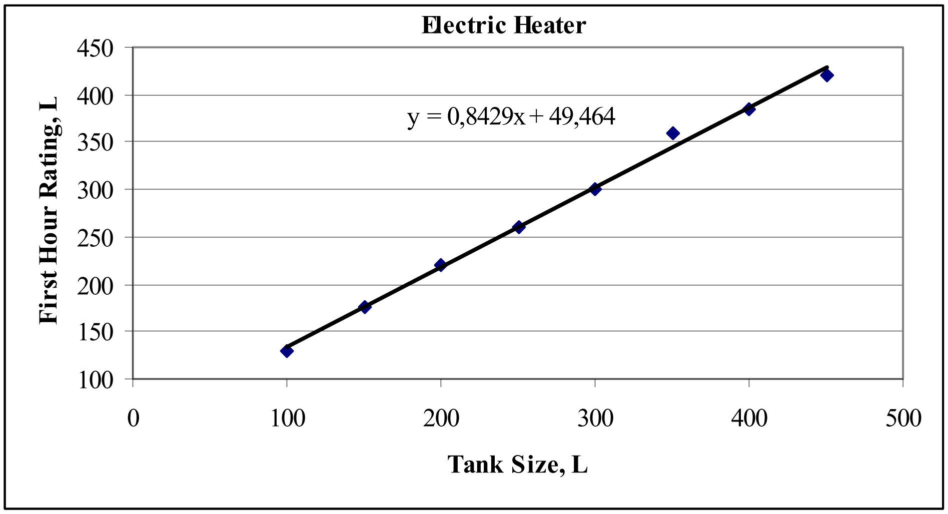

Figure 3.2 – First Hour Rating Relationship for Residential Water Heaters

Another factor to consider when sizing water heaters is *set point temperature*, as lower hot water temperatures may increase the volume of hot water used. Currently, manufacturers are shipping residential water heaters with a recommendation that the initial set point be about 49°C to minimize the potential for scalding. Reduced set points generally lower standbay losses and increase the water heater's efficiency and recovery capacity.

The *overall and peak average hot water use volumes* are shown in **Table 3.7**, and *average hourly pattern* is illustrated in **Figure 3.3**.

Table 3.7 – Overall (OVL) and Peak Average Hot Water Use Volumes for a Typical Family

	Average Hot Water Use, L							
	Hourly		**Daily**		**Weekly**		**Monthly**	
Group	OVL	Peak	OVL	Peak	OVL	Peak	OVL	Peak
Typical Family	9.9	21.9	239	252	1673	1981	7270	7866

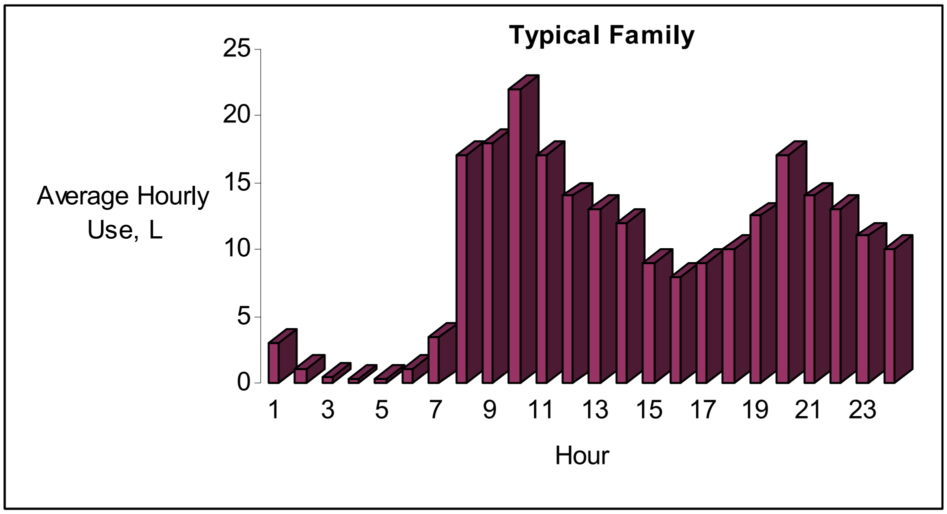

Figure 3.3 – Residential Average Hourly Hot Water Use [**1**]

Water quality, scale and corrosion

Hard water may cause scale (fouling or liming of heat transfer), while soft water may aggrave corrosion problems. Scale formation is also affected by system requirements and equipment. The rate of scaling increases with temperature and usage because calcium carbonate and other scaling compounds lose solubility at higher temperatures. Generally, when water hardness is over 140 mg/L, water softening or other treatments are often recommended.

Corrosion problems increase with the temperature because corrosive oxygen and carbon dioxide gases are released from the water. Electrical conductivity also increases with temperature, enhancing electrochemical reactions such as rusting. A deposit of scale provides some protection from corrosion. However, this deposit also reduces the heat transfer rate. Typically, one or more rods of magnesium alloy (the anode) are installed in the vessel. This electrochemically active material sacrifice itself to reduce or prevent corrosion of the tank (the cathode). To prolong the life of the vessel, periodic replacement of the rods is necessary.

Safety Devices

Temperature and pressure limiting devices that prevent water temperatures from exceeding 99°C by stopping the flow of fuel or energy, should be listed and labelled by a recognized certifying agency, as *Underwriters's Laboratories* (UL) or the *American Gas Association* (AGA) in United States.

Legionnaire's Disease

The bacteria that causes Legionnaire's disease when inhaled has been discovered in the service water systems of various buildings in the United States and Canada. Infection has often been traced in shower heads. Studies showed that *Legionelle pneumophila* can colonize in hot water systems maintained at 46°C or lower. Domestic water in the 60°C range are recommended to limit the potential for *Legionella* growth, but this high temperature increases the potential for scalding.

3.4 Design

Several design guidelines and standards for safety for residential reversible (heating and cooling) heat pumps systems are available in North America. For example, the Air Conditioning Contractors of America (ACCA) publishes the *ACCA manual H* (Heat Pump Systems, Principles and Applications – Commercial and Residential), and *ACCA Manual S* (Residential Equipment Selection). The Electric Power Research Institute (EPRI) published a "*Heat Pump Manual*" (1985), and Underwriters Laboratories (UL) – the *UL 559-1985 Standard* (Safety for Heat Pumps) and *Heating and Cooling Equipment* (1995). The Canadian national standard *CAN/CSA-C445-M92* (Design and Installation of Earth Energy Heat Pump Systems for Residential and Other Small Buildings), and the *ASHRAE Engineering Manual* [**12**] cover the ground-source systems design area for residential and commercial applications. For storage tank design [**1**], 60 to 90 % of the hot water in a tank is assumed to be used before dilution by cold water lowers the temperature below an acceptable level. Thus, the hot water available from a self-contained storage heater is considered to be:

$$V_T = Rd + MS_T$$

where :

V_T is the available hot water volume, L;

R – recovery rate at the required temperature, L/s;

d – duration of peak hot water demand, s;

M – ratio of usable water to storage tank capacity;

S_T - storage capacity of the heater tank, L.

4. North American Test Procedures and Seasonal Performance Calculations

Standards are intended for guidance of the heat pump industry, including manufacturers, installers, purchasers/consumers, and the government. The purpose of a standard is to establish definitions and product classifications, and requirements for testing, rating and performance. This information allow manufacturers to rate heat pumps on an uniform basis which enables the buyers and users to properly make selections of equipment. North America uses as standard references ASHRAE (*American Society of Heating, Refrigerating and Air-Conditioning Engineers*), ARI (*Air Conditioning and Refrigeration Institute*), and CSA (*Canadian Standards Association*) (**Table 4.1**). These standards provide test conditions and test methods for Seasonal Performance Energy Efficiency Ratio (SEER – the total cooling output in *Btu* during an annual usage period for cooling, divided by the total electric energy in *watt-hours* during the same period) and the Heating Seasonal Performance Factor (HSPF – the total heating output of a heat pump in Btu – including supplementary electric heat – necessary to meet the building heating requirements during its normal annual usage period for heating, divided by the total electric energy in watt-hours consumed during the same period). In addition, ARI standards provide rating procedures for part-load rating, and require a voltage tolerance test. The CSA standards contain minimum energy efficiency levels for air-to-air and water-to-air heat pumps. Efforts have been and continue to be made in harmonizing North American various standards (ASHRAE, ARI and CSA).

The current DOE (Department of Energy) procedure determine air-source heat pump heating capacity and performances at three outdoor conditions: High Temperature Heating (+8,3°C), Frost Accumulation (1,6°C) and Low Temperature Heating (-8,3°C). Similarly, cooling capacity and performance are determined at 35°C and 27,7°C. These points are sufficient to permit interpolation or extrapolation to determine air-source heat pump performance over the full-range of conditions in different climatic regions by using a *bin* method procedure.

The rule for air-source heat pumps takes into account that HSPF for a given heat pump and climate region varies depending on the match of building load to heat pump capacity. For the purposes of “standard” rating, the rule defines minimum design heating requirements (building loads for each region. In the United States, *Region IV* is the climatic region which is basis for the published HSPF ratings. *Region V* is the climatic region for standard HSPF ratings in Canada. In both regions, it is assumed, for the published rating, that the air-source heat pump is installed in a house with a design heating requirement equal to the air-source heat pump’s capacity obtained in the high temperature heating test at 8,3°C (47°F).

Tableau 4.1 – Existing North American Standards and Codes for Water Heaters and Heat Pumps

Title	Publisher	Reference
Canada		
Construction and Test of Electric Storage-Tank Water Heaters	CSA	CAN/CSA-C22.1-M90
Performance of Electric Storage Tank Water Heaters	CSA	CAN/CSA-C191-M90
Oil Burning Stove and Water Heaters	CSA	B140.3-1991
Oil-Fired Service water Heaters and Swimming Pool Heaters	CSA	B140.12-1991
United- States		
Methods of Testing for Rating Unitary Air-Conditioning and heat Pump Equipment	ASHRAE	ANSI/ASHRAE 37
Unitary Air-conditioning and Air-source Heat Pump Equipment	ANSI/ARI	Standard 210/240 - 1989
Desuperheater Water Heaters	ARI	Standard 470-1987
Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps	ASHRAE	ANSI/ASHRAE 116-1995
Method of Testing for Rating Residential Water Heaters	ASHRAE	ANSi/ASHRAE 118.2-2002
Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances	ASHRAE	ANSI/ASHRAE 124-2002
Methods of Testing for Efficiency of Space-Conditioning/Water Heating Appliances that Include a Desuperheater Water Heater	ASHRAE	ANSI/ASHRAE 137-2001
Water Heaters, Hot Water Supply Boilers and Heat Recovery Equipment	NSF	NSF-5
Household Electric Storage Tank Water Heaters	UL	ANSI/UL 174-1989
Others		
Heat Pump Systems : Principles and Applications	ACCA	Manual H
Residential Equipment Selection	ACCA	Manual S
Ground-Source Heat Pumps	ARI	ARI 325
Water-Source Heat Pumps	ARI	ANSi/ARI 320

CSA – Canadian Standard Association; ASHRAE – American Society of heating, Refrigeration and Air-Conditioning Engineers; ANSI – American National Standard Institute; ARI – Air-Conditioning and Refrigeration Institute; NSF – National Sanitation Foundation; UL – Underwriters Laboratories; ACCA – Air-Conditioning Contractors of America.

4.1 Space Cooling or Space Heating ONLY

Step-by-step test procedures for steady-state, cyclic, and part-load performance, data analysis and calculation methods for establishing seasonal efficiency ratios for unitary air conditioning and heat pumps are provided by ANSI/ASHRAE Standard 116-1995. This standard covers electrically driven, air-cooled air conditioners and *heat pumps* used in *residential applications* with cooling capacity of 19 kW (65,000 Btu/h) and less or, in the case of heating-only heat pumps, heating capacity of 19 kW (65,000 Btu/h) and less.

Test procedures are indicated for several applicable test methods [**7**]:

- Indoor air enthalpy method for (a) steady-state cooling and high-temperature heating tests; (b) frost accumulation, heating test; (c) low-temperature, heating test; (d) cyclic dry-coil, cooling and heating test.
- Outdoor air enthalpy method, cooling and heating test.
- Compressor calibration method, steady-state performance, cooling and heating.
- Refrigerant flow enthalpy method, steady-state, cooling and heating test.

For each method, are indicated the indoor and the outdoor, dry and wet bulb temperatures, room apparatus and equipment, equilibrium conditions, intervals for data recording, ON/OFF cycles and defrost operating conditions, number of required test methods, etc. Data to be recorded for different test methods include air side parameters (dry and wet temperatures, pressure, velocity and temperature at nozzle throat, etc.), refrigerant side parameters (evaporator and condensing pressure and temperature, refrigerant temperatures and flow rate, rate of condensate collection, etc.), and other general information (barometric pressure, power input, voltage, frequency, etc.).

Tables 4.2 and **4.3** summarize the calculation procedure for single-speed, two-speed or two compressor systems, and variable-speed units (heating mode). The calculation methods depend on outlining system capacity and power profiles over different temperature bins. The seasonal efficiency, SEER, is calculated by weighting system performance at individual bins with bin hours (number of hours a given temperature occurs over the season). The algorithm accounts for the cyclic losses during part-load operating conditions. The heating seasonal performance factor, HSPF, is strongly dependent upon the climatic region in which the heat pump operates, the type of heat pump system (e.g., single-speed or variable-speed compressor), and the heating requirement of the building. The basic approach for determining the HSPF is the temperature (dry-bulb) bin analysis. In addition to the cyclic losses, performance penalty due to frosting of the outdoor coil is incorporated. Since the HSPF is a function of climate and sizing of the unit, the procedure prescribes calculations for six climatic regions (Unites States) and different building loads. Distribution of fractional heating hours in temperature bins, heating load hours and outdoor design temperature for different climatic regions are given in the standard.

Table 4.2 – Calculation for Cooling Seasonal Energy-Efficiency Ratios (SEER)

	Cooling Seasonal Energy-Efficiency Ratio (SEER)	**Observations**
Single-speed, Single-Compressor Units	$SEER = PLF(0.5)\dfrac{\dot{q}_{ss}(t_{a8})}{\dot{E}_{ss}(t_{a8})}$	
Two-Speed or Two (Dual)-Compressor Units	$SEER = \sum_{j=1}^{8} q(t_j) / \sum_{j=1}^{8} E(t_j)$	$q(t_j)$ and $E(t_j)$, summed over temperatures bins, are evaluated at each temperature bin according to four cases (**Note 1)**)
Variable-Speed Units	$SEER = \sum_{j=1}^{8} q(t_j) / \sum_{j=1}^{8} E(t_j)$	$q(t_j)$ and $E(t_j)$ are evaluated at each temperature bin according to three possible cases (identified in terms of three outdoor temperature ranges or two outdoor temperature which separate them **(Note 2)**)

Note 1: Case I – low speed compressor (single compressor cycling). Case II – Unit runs continuously and alternates between high compressor speed (two compressors) and low compressor speed (one compressor). Case III – Unit cycles on and off at high compressor speed (2 compressors cycling on and off). Case IV – Continuous operation, high compressor speed (two compressors). **Note 2:** Case I - Continuous operation at maximum speed. Case II – Continuous operation at intermediate speed. Case III – Cycling at the minimum speed.

In **table 4.2** :

$PLF(0.5) = 1 - 0.5 * C_D$ is part-load factor where cooling load factor:

$$CLF = q_{cyc} / (\dot{q}_{tci} * \Theta_{cyc}) = 0.5\,,$$

where $\dot{q}_{tci}$ is the total dry-coil cooling capacity from a steady-state dry-coil test (Btu/h), q_{cyc} - total integrated capacity for dry-coil cycling test (Btu), and Θ_{cyc} - duration of time for one complete cycle consisting of one compressor ON time and one compressor OFF time (hours).

C_D - degradation coefficient for cooling cyclic operation determined by

$$C_D = \left\{1 - \frac{EER_{cyc}}{EER_{ss}}\right\} / (1 - CLF)$$

where EER_{ss} is the energy-efficiency ratio from a steady-state dry-coil test.

Table 4.3 – Calculation for Heating Seasonal Performance Factor (HSPF)

	Heating Seasonal Performance Factor (HSPF)	Discussion
Single-speed, Single-Compressor Units	$HSPF = \dfrac{\sum_{j=1}^{18} n_j BL(t_j) * F_{def}}{\sum_{j=1}^{18} n_j HLF(t_j)\delta(t_j)\dot{E}(t_j)/PLF(t_j) + \sum_{j=1}^{18} RH(t_j)}$	Note A
Two-Speed or Two (Dual)-Compressor Units	$HSPF = \dfrac{\sum_{j=1}^{18} n_j BL(t_j) F_{def}}{\sum_{j=1}^{18} E(t_j) + \sum_{j=1}^{18} RH(t_j)}$	Note B
Variable-Speed Units	$HSPF = \dfrac{\sum_{j=1}^{18} n_j BL(t_j) F_{def}}{\sum_{j=1}^{18} E(t_j) + \sum_{j=1}^{18} RH(t_j)}$	Note C

In **table 4.3** :

$RH(t_j)$ is the supplementary resistance heat term at temperature t_j required in those cases where the heat pump automatically turns OFF or when it is needed to meet the balance of the building heating requirements (kW).

n_j - fractional bin hours in the jth temperature bin.

$BL(t_j)$ - building load at temperature t_j, Btu/h/1000.

F_{def} - demand-defrost enhancement factor.

$\delta(t_j)$ - heat pump low-temperature cut-out factor.

$HLF(t_j)$ - heat pump heating load factor.

$PLF(t_j)$ - heat pump part-load factor.

All these quantities, and also the minimum and the maximum design heating requirements (DHR) for a residence in which a heat pump is likely to be installed, are defined by appropriate equations [**7**].

Note A: The numerator is the sum of the total building load weighted with fractional bin hours over the entire season. The building load equals exactly the heat pump system output. The denominator is a sum of the summation of the compressor power input (amended for part-load effects) and the summation of the power input into the resistance heater, both also weighted with fractional bin hours.

Note B: The terms $RH(t_j), \mathrm{E}(\mathrm{t_j})$ and F_{def} are evaluated according to three possible cases of heat pump operation:

- Case 1: Unit cycling on and off at low speed
- Case II: Unit cycling between high speed and low speed.
- Case III: Unit operating at high speed continuously with supplemental resistance heat.

Note C: The terms $BL(t_j), E(t_j)$ and $RH(t_j)$ are evaluated at each temperature bin according to three possible cases:

- Case I: Cycling on and off at minimum speed.
- Case II: Continuous operation at intermediate speed.
- Case II: Continuous operation at maximum speed

Evaluation of SEER and HSPF may be performed by hand, calculator ar computer methods. A step-by-step procedure and a worksheet for summarizing intermediate calculations and results are provided. The final determination of SEER and HSPF is made by the equations as indicated beneath the table. The *bin temperatures* and *fractional bin hours* applicable for the selected climatic region are given in standard for the calculation of SEER and HSPF respectively.

4.2 Domestic Hot Water – Directly Heating ONLY

The test procedures for rating the efficiency and hot water delivery capabilities of residential directly heated electrical water heaters not requiring (Type I) or requiring (Type II) circulation of water for heating (**Table 4.4**), with rated input no greater than 12 kW, as for single-phase air-source heat pump water heaters (Types IV and V), with rated input no greater than 6 kW, are provided by ASHRAE Standard 118.2-2002 [**10**].

Table 4.4 - Types of Directly Heated Residential Water Heaters

Heater	Type	Characteristics	Rated Input
Gas-Fired	I	Self-contained temperature-activated control Not requiring circulation of water for heating	No greater than 22 kW
Oil-Fired	I	Self-contained temperature-activated control Not requiring circulation of water for heating	No greater than 30,8 kW
Electric Resistance	I	Self-contained temperature-activated control Not requiring circulation of water for heating	No greater than 12 kW
Gas-Fired	II	Self-contained temperature-activated control Requiring water flow for heating	No greater than 58,6 kW
Oil-Fired	II	Self-contained temperature-activated control Requiring water flow for heating	No greater than 61,5 kW
Electric Resistance	II	Self-contained temperature-activated control Requiring water flow for heating	No greater than 12 kW
Gas-Fired	III	Remote temperature-activated control Requiring water flow for heating	No greater than 22 kW
Oil-Fired	III	Remote temperature-activated control Requiring water flow for heating	No greater than 30,8 kW
Electric Resistance	III	Remote temperature-activated control Requiring water flow for heating	No greater than 12 kW
Air-source Heat Pump	IV and V	Single-phase	No greater than 6 kW

The *test procedures* involve:

(i) A detailed *first-hour rating* water draw test for *Type I* water heaters, and

(ii) A *simulated use test* for both *Type I* and *Type II* water heaters.

The *calculation of results* comprises:

(a) Calculation of the *first-hour rating* for Type **I** and Type **II** water heaters (**Table 4.5**).

(b) Recovery efficiency for thermostatically controlled water heaters (**Table 4.6**).

(c) Daily water heater energy consumption (**Table 4.7**).

(d) Daily hot water energy content, and

(e) Energy factor.

Table 4.5 – First-hour calculation for rating the efficiency of directly heated water heaters

Device	Equation	Definitions
Type I Heaters	First-hour rating: $F = \frac{G[T_m - (T_o - (T_t - T_c))]}{(T_t - T_c)}$ T_m - mean outlet water temperature, °C T_o - maximum outlet temperature, °C T_t - nominal mean tank temperature, °C T_c - nominal cold water supply temperature, °C	T_m - mean outlet temperature $G = W * V_m (1000)$, W – weight of water (kg) collected during first-hour rating V_m - specific volume of water (m^3 / kg).
Type II Heaters	First-hour rating: $F = \frac{(FR)3600 * \Delta T_a}{(T_t - T_{c)}}$	FR – water flow at burner minimum input rate (gpm) ΔT_a - actual ΔT established in determining the flow rate.

Table 4.6 – Recovery efficiency for thermostatically controlled water heaters calculation of directly heated water heaters

Parameter	Device	Equation	Definitions
Recovery Efficiency for Thermostatically Controlled Water Heaters *	Electric water heaters	Energy consumed (kJ): $Q_r = (Z)/(1000)$	Z – electric energy used (MJ)
	Gas-fired heaters	Energy consumed (kJ): $Q_r = (Vol)(H)(C_s) + (Z_{aux}(1000)$ C_s - correction factor if the gas is not at standard temperature Z_{aux} - electrical energy used by auxiliary electric equipment (MJ)	Vol – quantity of gas metered during first recovery (m^3) H – higher heating value of gas (kJ/m^3)
	Oil-fired heaters	Energy consumed (kJ): $Q_r = (H_o)(W_f) + (Z_{aux})(1000)$	H_o - heating value of the fuel oil (kJ/kg) W_f - weight of fuel used (kg)

*The recovery efficiency for thermostatically controlled water heaters, a dimensionless quantity, is expressed with equation :

$$E_r = \frac{\lfloor (T_r - T_s)(U_1)(d)(C_p) + (T_{md1} - T_{mi})(d_1)(V)(C_p) \rfloor}{Q_r}$$

where U_1 is the quantity of water withdrawn during the first draw (liters).

C_p - specific heat of water (4.19 kJ/kg.K).

T_{md1} - maximum mean tank temperature recorded after the recovery following the first draw (°C).

T_{mi} - maximum mean tank temperature recorded prior the first draw in order to correspond with T_{md1} (°C).

V – storage tank volume (liters).

Table 4.7 – Daily water heater energy consumption, hot water energy content and energy factor

Parameter	Device	Equations	Definitions
Daily Water Heater Power	Heat-pump water heater	$P = \frac{Z_1}{t_1}$ (kW)	Z_1 - electrical energy consumed during first recovery cycle of simulated use test (kWh) t_1 - duration of first recovery cycle of the simulated use test (h).
Recovery time (h)	Gas and oil water heaters	$t_r = \frac{d_2(C_p)U_t(T_r - T_s)}{P * E_r}$	d_2 - density of water at T_t T_r - mean of the outlet water temperatures during the draws (°C) T_s - mean of the inlet temperatures during draws (°C) E_r - heat recovery efficiency P – kW input.
	Electric water heaters	$t_r = \frac{d_2(C_p)U_t(T_r - T_s)}{P * E_r(3600)}$	
Daily Hot Water Energy Content	-	$C_c = (d_2 C_p)(U_t)(T_t - T_c)$ T_t - nominal mean tank temperature (°C)	U_t - total amount of water withdrawn during the simulated use test (L)
Energy Factor	-	$EF = \frac{C_c}{C_y}$ C_y - daily water heating energy consumption	$C_y = Q_1 + Q_2$ (correct factors for energy consumed during the draw portion of the test to the reference inlet and outlet water conditions, and respectively, during the standby portion of the test to the reference ambient and mean tank temperature conditions.

4.3 Combined Space and Domestic Hot Water Heating

4.3.1 Appliances without Heat Pump

4.3.1.1 Primary Function – *Space Heating*

For electric, gas-fired or oil fired combination space heating and water heating appliances (without heat pump), a method of test to rate the performance is provided by ASHRAE Standard 124-2002 [**9**] that defines the *Type I* appliance as an appliance whose **primary design function** is *space heating* and which has a secondary function of domestic water heating, with a space heating capacity of 3.9 kW (13,500 Btu/h) or more. For space heating, parameters shall be calculated and tests conducted in accordance with ANSI/ASHRAE Standard 103-1993 ("*Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces*").

For ***water heating function***, the ASHRAE Standard 124-2002 indicates procedures for :

- Tank storage capacity determination,
- First-hour rating for integrated heaters and maximum GPM rating for tankless heaters, and
- Simulated-use test.

For ***water heating function***, the following parameters shall be calculated.

(i) First-hour rating for storage-type and integrated water heaters:

$$F = \sum_{i-1}^{n} V_i^* ,$$

where n is the number of draws that are completed during the first-hour rating test,

V_i^* - volume of water removed during the *i*th draw of the first-hour rating test (L). If the mass of water is being measured:

$$V_i^* = 1000 \frac{W_i^*}{d}$$

where W_i^* is the mass of water removed during the ***i***th draw of the first-hour rating test (kg),

d – density of water at the average outlet temperature, (kg/m^3),

1000 – conversion factor from m^3 to litre.

(ii) Maximum L/min rating for instantaneous and tankless water heaters:

$$F_{\max} = 1000 \frac{W_{10m}(T_{del} - T_{in})}{10(d)(42.8°C)} ,$$

where W_{10m} is the mass of water collected during the 10-minute test (kg),

T_{del} - average delivery temperature, °C.

T_{in} - average inlet temperature, °C,

d – density of water at the average delivery temperature, kg/m^3 .

(iii) Energy factor

The standby energy input for heaters with storage shall be corrected for the variation of test mean tank temperature and room air temperature from the nominal values. The corrected standby energy is:

$$Q_1 = (Q_{sb} + Q_{r1})(T_t - T_r)(24 - t_{run})/\left[18(T_s - T_a)\right]$$

where Q_{sb} is standby period measured energy consumption during the 18-hr standby period (kJ).

Q_{r1} - energy consumption at end of 18-hr standby period (kJ).

$(T_t - T_r)$ - nominal temperature difference between mean tank and air (°C).

$(T_s - T_a)$ - test temperature difference between mean tank and air (°C).

t_{run} - burner operating time recorded during the draw period.

The draw period energy due to the 6 draws is corrected for the variation of the average test inlet and outlet and outlet temperatures from nominal values. The corrected draw period energy is:

$$Q_2 = \left[(Q_d + Q_{r2})(T_t - T_c)/(T_o - T_i)\right] - \left[Q_1(6 - t_{run})/(24 - t_{run})\right]$$

where $Q_d = \left[Q_{rs} - (Q_{sb} + Q_{r1} + Q_{r2})\right]$ is draw period energy consumption (kJ).

Q_{r2} - energy consumption at end of draw period (kJ).

$(T_t - T_c)$ - nominal temperature difference between mean tank and inlet water (°C).

$(T_o - T_i)$ - test temperature difference between average outlet and average inlet water temperatures during draws (°C).

The energy factor for *heaters with storage* is calculated by dividing the nominal energy delivered in the hot water draws during 6-hour draw period by the corrected energy consumption in the total 24-hour test period:

$$EF = k(d)(U_s)(T_t - T_c)/\left[1000(Q_1 + Q_2 + 1000C_{aux})\right]$$

where ***k*** is the specific heat of water (4.187 kJ/kg°C).

U_s - amount of water drawn during simulated use test (L).

C_{aux} - auxiliary electrical input for entire test (MJ).

Recovery efficiency for *heaters with storage* is calculated as follows:

$$E_r = k(d)(U_s)(T_t - T_c)/\left[(1000)(Q_2 + 1000(C_{aux-d} + C_{aux-r2}))\right]$$

where C_{aux-d} is the auxiliary energy during draw period (MJ),

C_{aux-r2} - auxiliary energy at recovery following draw period (MJ).

Finally, the corrected standby loss (in $hour^{-1}$) for *heaters with storage* is defined as:

$$S = 1000Q_1 / \left[k(24 - t_{run})(d)(V)(T_t - T_r)\right]$$

where V is measured storage volume (L).

4.3.1.2 Primary Function – *Domestic Hot Water Heating*

The *ASHRAE Standard 124-2002* defines the *Type II* appliance (without heat pump) as an appliance whose *primary design function is domestic water heating* and which has a *secondary function of space heating* of less than 3.9 kW. The first-hour rating, energy factor, recovery efficiency and standby loss for such appliances are tested and calculated in accordance with the *US Federal Regulations*.

Table 4.8 get together the mean parameters to be calculated and their respective equations.

Table 4.8 – Parameters to be Calculated for Appliances with Domestic Hot Water Heating as Primary function

Parameter	Equation
Heating-Season Space Heating Factor	$SHF = 0.225(Effy_{ss} / EFFy_{hs})$ 0.225 – a constant; $Effy_{ss}$ - steady-state, space-heating efficiency (%). $Effy_{hs}$ - space-heating seasonal efficiency (%).
Heating-Season Water Heating Factor	$WHF = (U)(d)(k)(T_t - T_c)/\left[24(1000)(Q_{in})(Effy_{ss}/100)\right]$ U - daily hot water consumption (L); $(T_t - T_c)$ - temperature rise of water, °C; Q_{in} - input rate (kJ/h).
Non-Heating-Season Water Heating Factor	$NHF = (U)(d)(k)(T_t - T_c)/\left[24(Q_{in})(EF)\right]$
Combined Annual Efficiency	$CAE = \left[SHF)Effy_{hs}/100) + WHF(Effy_{ss}/100) + (R * NHF * EF)\right]/$ $\left[(SHF) + (WHF) + (R * NHF)\right]$ where R is the ratio of non-heating-season days to heating-season days
Combined Heating-Season Efficiency	$CE_{HS} = \left[SHF(Effy_{HS}/100) + WHF(Effy_{SS}/100)\right]/(SHF + WHF)$
Non-Heating-Season Efficiency	$CE_{NS} = (R)(NHF)(EF)/\left[(R)(NHF)\right] = EF$

A methodology for comparison of combined appliance efficiency with efficiencies of single-function appliances that are covered by the US Energy Policy and Conservation Act is included in an **Appendix** of the standard [**9**].

4.3.2 Appliances with Heat Pump (Desuperheaters)

Test methods and calculation procedures for establishing the efficiency ratings and for estimating annual energy consumption of space-conditioning/water-heating appliances having refrigerant-to-water desuperheaters are provided by ASHRAE Standard 137-2001 for electric, air-to-air, space-conditioning appliances having rated cooling capacities of less than 19 kW (residential applications) [**8**].

The *test procedure* firstly define the *test requirements* for combined appliances including an air conditioner and a heat pump, and the tests tolerances and energy balance requirements.

Combined appliances that include an air conditioner or a heat pump shall be tested in accordance with the test requirements specified in ANSI/ASHRAE 116-1995. **Table 4.9** Indicates, as an example, some test conditions associated with space-cooling and space-heating seasons respectively.

Table 4.9 – Test Conditions Associated with the Space-Cooling and Space-Heating Seasons

Operating Mode	**Outdoor Temperature**		**Indoor Temperature**	
-	**Dry-bulb (°F)**	**Wet-bulb (°F)**	**Dry-bulb (°F)**	**Wet-bulb (°F)**
Space Cooling Only	95	75	80	67
Combined Mode (COOL&WH)	82	65	80	67
High-Temperature Heating, Space Heating Only	47	43	70	60
Cyclic Heating, Space Heating Only or Combined Mode (HEAT&WH)	47	43	70	60
Frost Accumulation, Space heating Only	35	33	70	60
Low-Temperature Heating, Space Heating Only or Combined Mode (HEAT&WH)	17	15	70	60

Data to be recorded during the simulated use tests include:

- the energy removed during the three hot water draws,
- electrical energy used by the water heater,
- desuperheater water inlet and outlet temperatures,
- internal tank temperatures,
- desuperheater water flow rate,
- ambient air temperature,
- test room conditions,
- minimum water heater outlet temperature, and
- electrical power requirement for desuperheater water pump.

There are also defined procedures for testing when space-conditioning-only-mode, and for water-heating-modes (COOL&WH and HEAT&WH). **Figure 4.5** summary represents the boundary energy conditions for combined space and DWH heat pump with desuperheater calculations.

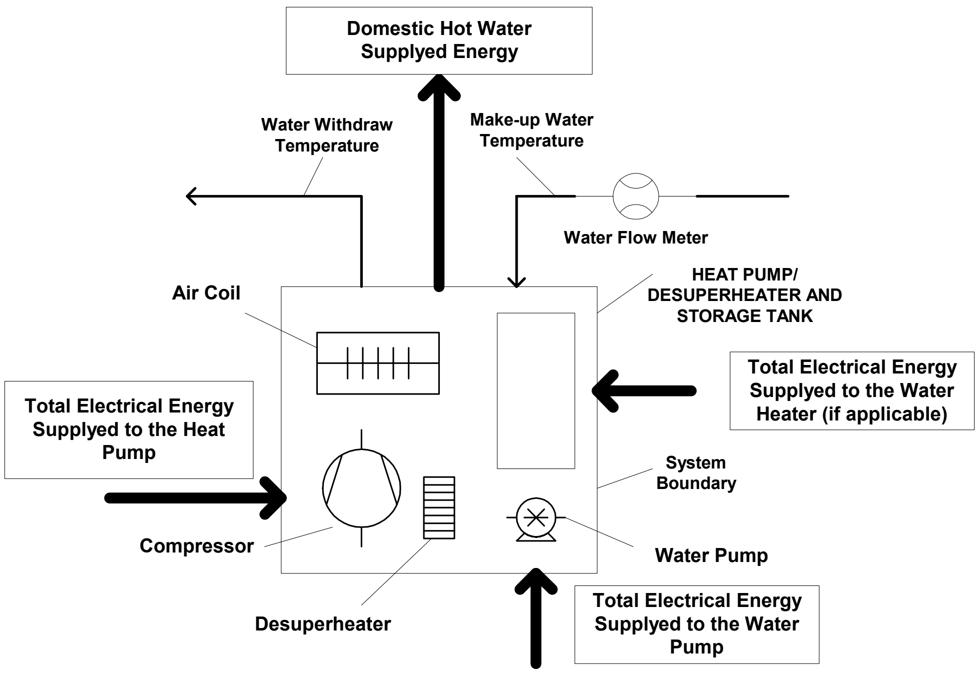

Figure 4.6 – Boundary Energy Conditions for a Combined Space and DWH Heat Pump with Desuperheater

Calculation methods (**Table 4.10**) allow to determine:

(i) Seasonal energy efficiency ratio (SEER) of the combined appliance when used for space cooling;

(ii) Heating seasonal performance factors (HSPF) of combined appliance when used only for space heating;

(iii) Combined performance factor for the space-cooling and water heating season (CPF_{cs});

(iv) Combined performance factor for the space heating and water heating season (CPF_{hs});

(v) Combined performance factor for the water heating-only season (CPF_{ws}).

Table 4.10 - Seasonal Performance Calculation s for COMBIND Appliances [**8**]

	SEER	**HSPF**	**Combined Performance Factor**
Space Cooling Only	ASHRAE 116-1995	-	-
Space Heating Only	-	ASHRAE 116-1995	-
COMBINED Space-Cooling and Water-Heating Season	-	-	$CPF_{cs} = \dfrac{\sum_{j=1}^{8}\left[\dfrac{q(t_j)}{N} + \dfrac{qw(t_j)}{N}\right]}{3.413\dfrac{Btu/h}{W} * \sum_{j=1}^{8}\left[\dfrac{E(t_j)}{N} + \dfrac{ER(t_j)}{N}\right]}$
COMBINED Space-Heating and Water-Heating Season	-	-	$CPF_{hs} = \dfrac{\sum_{j=1}^{8}\left[\dfrac{q(t_j)}{N} + \dfrac{qw(t_j)}{N}\right]}{3.413\dfrac{Btu/h}{W} * \sum_{j=1}^{8}\left[\dfrac{E(t_j)}{N} + \dfrac{ER(t_j)}{N} + \dfrac{RH(t_j)}{N}\right]}$
Water-Heating Only Season			$CPF_{ws} = EF$ (Energy Factor "EF" - determined as specified in ANSI/ASHRAE 118.2-2002)

SEER – Seasonal Energy Efficiency Ratio; HSPF - Heating Seasonal Performance Factor. Conversion factor: 1 Watt = 3.413 Btu/h

In **Table 4.10**:

CPF_{cs} is – for combined space-cooling and water-heating season, the sum of the total space cooling provided and the useful portion of the total water-heating kload divided by the total electrical energy consumed by the combined appliance (desuperheater heat pump).

$\dfrac{q(t_j)}{N}$ - for temperature bin j, the ratio of the total space ***cooling*** (or ***heating***) provided to the total number of temperature bin hours in the space-cooling (or space-heating) and water-heating season.

$\dfrac{qw(t_j)}{N}$ - for temperature bin j, the ratio of the total thermal energy associated with ***domestic hot water*** that is delivered to the consumer to the total number of temperature bin hours (tank standby losses are not included in this quantity).

$\frac{E(t_j)}{N}$ - for temperature bin j, the ratio of the total electrical energy supplied to the heat pump (or air conditioner) and, if applicable, to the desuperheater water pump to the total number of temperature bin hours.

$\frac{ER(t_j)}{N}$ - for temperature bin j, the ratio of the total electrical energy supplied to the electric water heater to the total number of temperature bin hours.

CPF_{hs} - for the combined space-heating and water-heating season, the sum of the total space-heating load and the useful portion of the total water-heating load divided by the total electrical energy consumed by the combined appliance (heat pump, electric water heater and, if applicable, desuperheater water pump).

$\frac{RH(t_j)}{N}$ - for temperature bin j, the ratio of the total electrical energy used for resistive space heating to the total number of temperature bin hours (resistive heating is required when operating below the space-heating balance point).

j – for each climatic region, the total number of 5°F outdoor temperature bins having a nonzero entry for the fractional bin hours (i.e., $n_j / N > 0$) are given in the standard.

N – total number of temperature bin hours in the space-cooling or space-heating and water-heating season.

This section shows that almost all combinations of the combined space-cooling/heating and hot water heating calculations are available in North American standardisation. The available equations for seasonal energy efficiency ratios and heating seasonal performance factors are reliable and based on bin method. They could be used as basis for developing similar calculation methods for Europe and Japan standards.

5. Conclusions

References

1. ASHRAE Handbook - Fundamentals, 1995
2. IEA/OECD – Heat Pump Centre – “*Domestic Hot Water Heat Pumps in Residential and Commercial Buildings*”, Analysis Report, HPC-AR2, by *Caneta Research Inc*. (Canada), April 1993
3. Olszewsky and Fontana – “The development of equipment for retrofit of air conditioning, heat pump and refrigeration systems”, ORNL
4. Kesselring, J.; Lannus, A. – “Field testing of the Hydrotech 2000 heat pump”, EPRI Journasl, December 1991, pp. 33-36
5. EPRI – “End-use energy effiency”, report no. CU-3032, 1990
6. ARI Standard 470 –1987
7. ANSI/ASHRAE Standard 116-1995 – “Methods of testing Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps”.
8. ANSI/ASHRAE Standard 137-1995 – “Methods of Testing for Efficiency of Space-Conditioning/Water-Heating Appliances that Include a Desuperheater Water Heater”.
9. ANSI/ASHRAE Standard 124-2002 – “Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances”.
10. ANSI/ASHRAE Standard 118.2-2002 – “Method of Testing for Rating Residential Water Heaters”.
11. ASHRAE – “Commercial/Institutional Ground-Source Heat Pump – Engineering Manual”, 1995.
12. ASHRAE Handbook 1995 – “HVAC Applications.
13. Minea, V. – “Ground-Source Heat Pump Systems – Technical Specifications and Practical Installation Guidelines in North America”, Hydro-Quebec, LTEE-RT-0150, October 30, 1999
14. Minea, V. – “Air-to-Air Heat Pumps – Technical Specifications and Practical Installation Guidelines in North America”, Hydro-Quebec, LTEE-RT-0148, November 11, 1999