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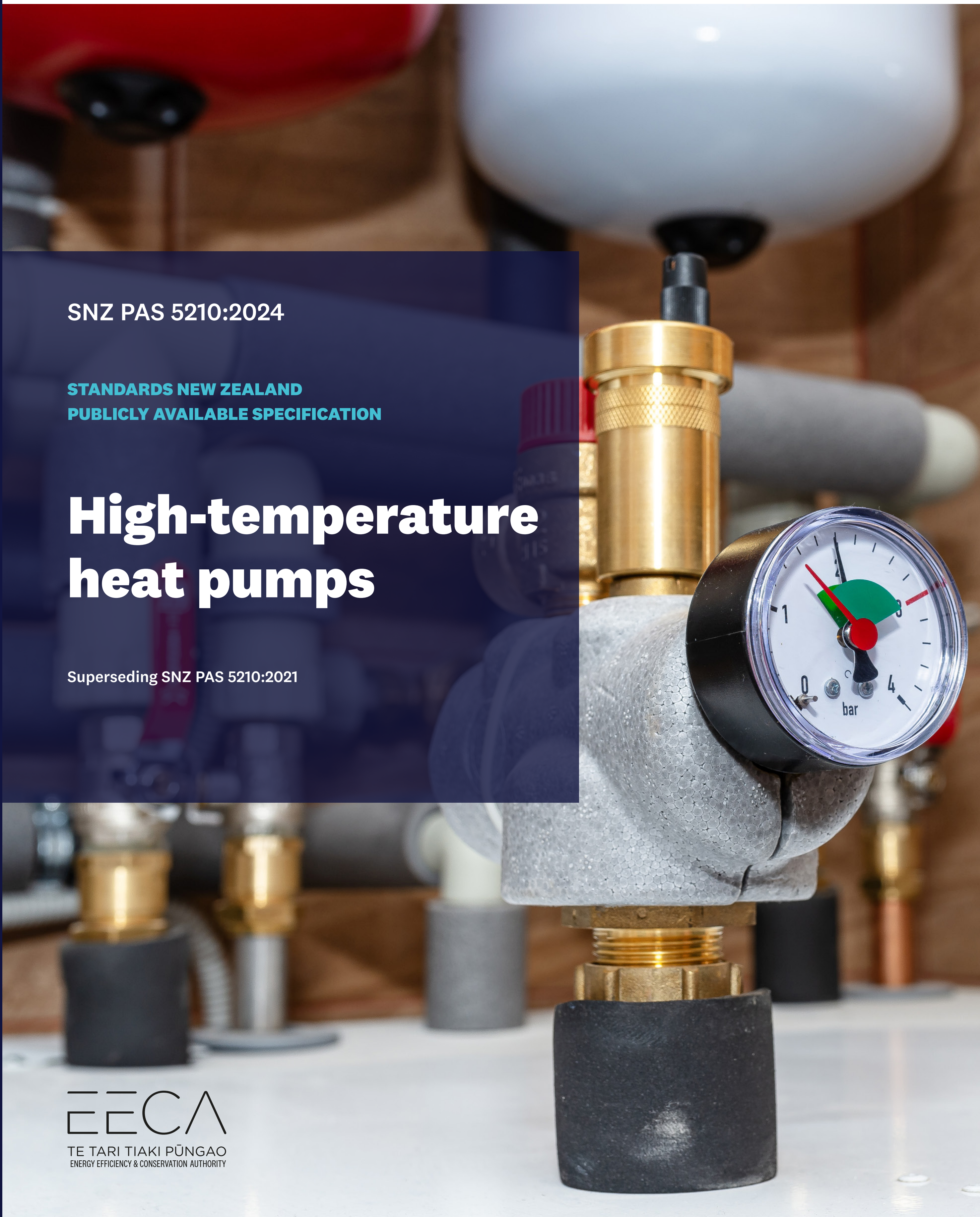

SNZ PAS 5210:2024

STANDARDS NEW ZEALAND
PUBLICLY AVAILABLE SPECIFICATION

High-temperature heat pumps

Superseding SNZ PAS 5210:2021

EECA
TE TARI TIAKI PŪNGAO
ENERGY EFFICIENCY & CONSERVATION AUTHORITY

SNZ PAS 5210:2024

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TECHNICAL ADVISORY GROUP REPRESENTATION

This publicly available specification was prepared by the P5210 High-Temperature Heat Pumps Technical Advisory Group (TAG). Members of the TAG were appointed by the New Zealand Standards Executive under the Standards and Accreditation Act 2015.

The TAG consisted of representatives of the following nominating organisations:

Carbon and Energy Professionals New Zealand

Energy Efficiency and Conservation Authority

Energy New Zealand

Institute of Refrigeration Heating and Air Conditioning Engineers of New Zealand

Massey University

Ministry for the Environment

New Zealand Green Building Council

New Zealand National Committee of the International Institute of Refrigeration

Solar Association of New Zealand

Western Institute of Technology

WorkSafe New Zealand

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REFERENCED DOCUMENTS

Reference is made in this document to the following:

New Zealand standards

NZS 4219:2009	Seismic performance of engineering systems in buildings

Joint Australian/New Zealand standards

AS/NZS 3500:- - - -	Plumbing and drainage
Part 4:2021	Heated water services
AS/NZS 4020:2018	Testing of products for use in contact with drinking water
AS/NZS 4234:2021	Heated water systems – Calculation of energy consumption
AS/NZS 4692:- - - -	Electric water heaters
Part 1:2005	Energy consumption, performance and general requirements
AS/NZS 5125.1:2014	Heat pump water heaters – Performance assessment
AS/NZS 5149:- - - -	Refrigerating systems and heat pumps – Safety and environmental requirements
Part 1:2016	Definitions, classification and selection criteria
Part 2:2016	Design, construction, testing, marking and documentation
Part 3:2016	Installation site
Part 4:2016	Operation, maintenance, repair and recovery
AS/NZS 60079:- - - -	Explosive atmospheres
Part 0:2019	Equipment – General requirements
AS/NZS 60335:- - - -	Household and similar electrical appliances – Safety
Part 2.40:2023	Particular requirements for electrical heat pumps, air-conditioners and dehumidifiers
AS/NZS ISO 817:2016	Refrigerants – Designation and safety classification

Australian standards

AS 1210:2010	Pressure vessels
AS 2971:2007	Serially produced pressure vessels

British standards

BS EN 13445:- - - -	Unfired pressure vessels
Part 1:2014	General
EN 14511:- - - -	Air conditioners, liquid chilling packages and heat pumps for space heating and cooling and process chillers, with electrically driven compressors
Part 1:2022	Terms and definitions
EN 16147:2017+A1:2022	Heat pumps with electrically driven compressors – Testing, performance rating and requirements for marking of domestic hot water units
PD 5500:2024	Specification for unfired fusion welded pressure vessels

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ASHRAE 34:2019	Designation and safety classification of refrigerants
ISO 19967:- - - -	Heat pump water heaters – Testing and rating for performance
Part 1:2019	Heat pump water heater for hot water supply
Part 2:2019	Space heating and/or space cooling
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Building Regulations 1992 (New Zealand Building Code [NZBC])

Climate Change Response (Zero Carbon) Amendment Act 2019

Electrical (Safety) Regulations 2010

Energy Efficiency (Energy Using Products) Regulations 2002

Health and Safety in Employment (Pressure Equipment, Cranes, and Passenger Ropeways) Regulations 1999

Ozone Layer Protection Act 1996

Websites

https://www.ashrae.org/

www.legislation.govt.nz

www.state.gov

LATEST REVISIONS

The users of this publicly available specification should ensure that their copies of the above-mentioned New Zealand standards are the latest revisions. Amendments to referenced New Zealand and joint Australian/New Zealand standards can be found on www.standards.govt.nz.

REVIEW OF STANDARDS

Suggestions for improvement of this standard are welcomed. They should be sent to the National Manager, Standards New Zealand, PO Box 1473, Wellington 6140.

FOREWORD

Heat pumps have been available for many years; however, recent advances in the technology have made them better suited to high-temperature applications where heat is delivered at a temperature greater than 50°C.

The objective of this publicly available specification (PAS) is to provide good practice advice and information to support the adoption of energy-efficient heat pump systems for domestic, commercial, and industrial applications. This PAS is limited to the most common systems available, which are those that utilise a closed mechanical vapour compression cycle using electricity.

Document navigation

This document has been written with three main audiences in mind: domestic, commercial, and industrial. Different sections of the document are aimed at different readers:

(a) Sections 1 and 2 provide information about the scope of the PAS, key concepts and definitions, and a general introduction to high-temperature heat pumps (HTHPs) and their benefits. These sections are intended for all readers.

(b) Sections 3, 4, 5, and 6 are aimed at domestic HTHPs. The key application is the provision of potable hot water. Section 3 gives a general overview, whereas Sections 4, 5, and 6 provide technical information of use to contractors, designers, and manufacturers of domestic HTHP systems.

(c) Sections 7, 8, 9, 10, 11, and 12 are aimed at commercial HTHPs. Section 7 is general design and configuration information. Section 8 includes requirements for commissioning and ongoing performance assessment. Purchasers of commercial HTHPs may have their own contractual commissioning and performance requirements and it is recommended that the requirements here are used to inform those requirements. Sections 9, 10, 11, and 12 cover the main commercial applications: commercial-scale potable-water heaters, space heating, pool heating, and fresh-air tempering.

(d) Section 13 is aimed at the use of HTHPs for industrial-process heat applications.

(e) Appendix A contains additional information on refrigerants that is applicable to all applications. The domestic-, commercial-, and industrial-specific sections also contain application-focused information on refrigerants.

(f) Appendix B contains information on a modification of the AS/NZS 5125.1 test methodology to apply for HTHPs with a heating capacity greater than 15 kW.

(g) Appendix C contains calculation methods for the Carnot coefficient of performance (COP). Comparing actual performance to the Carnot COP is an important performance metric in all HTHP applications.

(h) Appendix D contains calculation methods for the energy use, energy cost, greenhouse gas emissions, emissions trading scheme (ETS) savings, and simple payback for an HTHP and compares it to traditional fossil-fuel-based heating systems.

Interpretation

This document, SNZ PAS 5210:2024 *High-temperature heat pumps*, has been prepared as a guidance document and published as a PAS.

PAS are an ISO-recognised category for documents that are not national standards but are produced by a national standards body to respond to a particular market need. They represent either consensus in an organisation or industry, or consensus of the experts within a specific working group.

For the purposes of this PAS, the word ‘shall’ refers to requirements that are essential for compliance with this specification, while the word ‘should’ refers to practices that are advised or recommended.

SNZ PAS 5210:2021

SNZ PAS 5210:2024 supersedes SNZ PAS 5210:2021. The 2021 version of this publicly available specification (PAS) was focused on high-temperature heat pumps used to provide reticulated hot water for heating in institutional settings – for example, radiators in schools or office buildings. The revision extends the applications to domestic hot-water heaters, commercial users of hot water, and process heating (water, air, or other substances) in industry. The field of high-temperature heat pumps is an area of active research and innovation, so the PAS has been updated with the latest information.

Publicly Available Specification

High-temperature heat pumps

1 GENERAL

1.1 Scope

1.1.1 Inclusions

This PAS covers these areas:

(a) Electrically driven, packaged heat pumps that deliver a temperature greater than 50°C;

(b) Issues for integrating commercial and industrial applications into sites (such as connectivity, advanced controls, thermal storage and heating distribution, parallel provision of cooling, electrical supply, and performance monitoring);

(c) The range of available refrigerants;

(d) WorkSafe New Zealand safety requirements; and

(e) New installations and retrofit applications.

1.1.2 Exclusions

This PAS does not cover these areas:

(a) Standard heating, ventilation, and air conditioning (HVAC) systems and other heat pump applications that deliver temperatures lower than 50°C;

(b) Detailed installation advice; and

(c) Heat-driven heat pumps (such as absorption heat transformers or absorption heat pumps).

1.2 Key concepts

It is important to understand these concepts:

(a) The coefficient of performance (COP) measures the efficiency of a high-temperature heat pump (HTHP). An HTHP system with a COP of 4 will provide 4 units of heat energy (kWh) for every unit of electricity consumed (kWh). The actual COP depends on the application and operating conditions;

(b) Refrigerant is the fluid used in a sealed HTHP to absorb and reject heat. Some refrigerants have a greater environmental impact than others in terms of their contribution to ozone depletion, global heating, or accumulated heat in the environment. Some refrigerants that have less impact on the environment are flammable or toxic. The type of refrigerant an HTHP uses is an important consideration when purchasing a system. See 3.3 for details of common HTHP refrigerants used in New Zealand; and

(c) Site integration is the process of assessing commercial and industrial facilities to establish the best HTHP system for the situation. Site integration should include a site visit and a report that thoroughly investigates the best HTHP options.

1.3 Defined terms

For the purposes of this PAS, the following definitions apply:

Carnot cycle A thermodynamically ideal heat-pump cycle with no losses. It has the highest theoretically possible COP, given the heat source and sink temperatures. The Carnot cycle COP cannot be achieved in practice, but it sets a useful benchmark to measure actual heat-pump systems against

Cascade system A configuration that combines two or more heat pumps, with the heat sink of one heat pump acting as the heat source for the next in the cascade. The component heat-pump systems are sealed and may use different refrigerants. When used correctly, the final heat pump in the cascade can deliver a high temperature at its heat sink more efficiently, and with a lower temperature lift, than a single heat pump. A cascade system is not the same as a multi-stage system, which achieves a similar benefit in a different manner. A cascade system can also be used in environments where it is easier to isolate a refrigerant that has high safety requirements

Coefficient of performance (COP) The ratio of the amount of useful heating (plus useful cooling, in some cases) to the amount of input energy. System COPs, which are used in this PAS, are more comprehensive, but compressor-only COPs (these include only compressor-energy input) are often quoted to allow comparisons with alternative heating systems when the rest of the system is not fully specified. Compressor-only COPs are higher than system COPs

Design condition(s) An operating condition for which an HTHP is designed or selected. Design conditions include the amount of heating demand and the temperatures of the heat sink and heat source.

The designer should know the most extreme operating condition. For an HTHP with an ambient-air heat source, this is typically the heating demand at the coldest expected ambient temperature. The HTHP should be able to operate at this extreme condition and deliver the required amount of heat.

The designer should also ensure that the HTHP can perform with high energy efficiency (high COP) at normal operating conditions, which is how the HTHP will operate most often (normal operating conditions are additional design conditions). COP is less important at extreme operating conditions, as the HTHP is not operated in this configuration for long periods.

The seasonal coefficient of performance (SCOP) is a method of aggregating performance over different operating conditions rather than at a single design condition.

For some applications (such as some industrial process-heating applications), extreme and normal operating conditions may be similar

Heat-exchanger	A device that exchanges heat between two fluids. In a heat pump, the evaporator and condenser, or equivalent components, are both heat-exchangers
Heating demand	The heating demand is the amount of heat (kW) required for the heating application. The instantaneous, average, or peak heat demand can all be referred to depending on the context. Heat load is an equivalent term that is sometimes used
Heat rejected	The heat transferred by the heat pump into the heat sink. The heat is rejected from the heat pump in the condenser, or equivalent
Heat sink	The medium that receives heat from the heat pump, via its condenser or equivalent
Heat source	The medium that supplies heat to the heat pump, via its evaporator or equivalent
Multi-pass heat pump	A heat-pump water heater that is coupled to a tank. It heats water from the base of the tank in small increments (typically 5K) and returns the heated water to the lower part of the tank. The water-flow rate through the condenser is high, and the temperature difference across the heat-exchanger is low
Part load	When a heat-pump system needs to deliver less heat to the heat-sink than its actual capacity at the actual operating condition, then it is controlled to a part load state to deliver only the required heat. There are various technologies and configurations for the control systems to do this. The selection and configuration of these controls can affect the overall energy efficiency of the system. Understanding how and when the heat-pump operates in part load states can be critical to calculating an accurate seasonal coefficient of performance (SCOP) for the heat-pump. 'Full load' refers to the heat pump operating at its full capacity for the operating conditions
Pinch analysis	A technique to determine the most energy-efficient configuration of heat exchange between heating and cooling operations. It is most relevant for complex industrial sites that have multiple simultaneous heating and cooling needs

Primary and secondary fluid	Some systems distinguish between primary fluid (the refrigerant) and secondary fluid (another utility fluid that is heated at the heat sink). The secondary fluid is pumped to the application that it is used to heat. An example is a heat pump that is used to heat hot water that is then reticulated to radiators to heat the air space. In this example, the primary fluid is the refrigerant in the heat pump, the secondary fluid is the hot water, and the application is the air in the room NOTE – Do not confuse 'primary' and 'secondary' fluid with the use of 'primary' and 'secondary' in a hydronic context.
Primary and secondary hydronic system	In many heating systems, the utility fluid (usually hot water) can be configured in primary and secondary hydronic systems (loops). In this configuration, the primary hydronic system is a fixed flow rate pumped through the heat generator (such as an HTHP) and returned – hot – to the hot-water tank. The secondary hydronic system is the hot water that is pumped from the hot-water tank, on a separate distribution circuit, to meet the heating demand. In some cases, the fluid heated in the primary hydronic system is separated from the secondary hydronic system by a heat-exchanger. In other cases, the fluids in the primary and secondary circuits mix NOTE – Do not confuse a primary-secondary hydronic system with the use of 'primary' and 'secondary' in a refrigerant context.
Refrigerant	The fluid that is circulated within a heat pump to convey heat from the heat source to the heat sink. It can be a liquid or a vapour at different points in the heat-pump system
Seasonal coefficient of performance (SCOP)	The annual average COP for a heat pump. The SCOP accounts for the COP varying throughout the year as heating demand and operating conditions change. SCOP can be calculated as the sum of heat provided (kWh) over the year divided by the sum of power consumption (kWh) over the year. The seasonal energy-efficiency ratio (SEER) is the same concept as the SCOP, but SEER is used in the refrigeration industry for cooling applications. The heating seasonal performance factor is also sometimes used to quantify how much a heat-pump system's performance varies over a year. The heating seasonal performance factor uses different units to the SCOP, and the two should not be directly compared
Single-pass heat pump	A type of heat pump that heats the incoming water to the target temperature by passing it through the heat-exchanger(s) once. The heat pump uses pump speed or compressor speed, or a combination of both, to maintain a near-constant supply-water temperature. It returns the heated water to near the outlet of the storage tank. If the heat pump is part of a closed loop, it can be tankless

Single stage/ multi-stage

The configuration of a heat pump's mechanical vapour-compression cycle. The simplest configuration is a single compression and single expansion stage.

When the temperature lift is very large, it can be more energy efficient to perform the compression or expansion, or both, in two or more stages (multi-stage). Multi-stage systems can be configured in many ways; this is beyond the scope of this PAS. Consult your equipment supplier or consultant for site-specific advice on multi-stage units

Space heating

Heating rooms for human occupancy

Subcritical heat pump

A conventional heat-pump cycle where the refrigerant is operated below its critical point at all points in the compression cycle

NOTE – Do not confuse a subcritical heat pump with a transcritical heat pump.

Temperature increase

The increase in heat-sink temperature due to heat transferred by the heat pump. For example, if a heat pump draws heat from a heat source (such as ambient air) at 20°C and rejects it to heat water from 50°C to 85°C, the temperature increase is 35°C

NOTE – Do not confuse temperature increase with temperature lift.

Temperature lift

The difference between the heat-sink temperature and the heat-source temperature.

For heat sinks and heat sources that change temperature, the log mean temperature of the heat sink or heat source should be used (see Appendix C3 for equations).

Temperature lifts can be defined for the system (for example, the temperature difference between the space or water being heated and the ambient air). They can also be defined for the HTHP (for example, the temperature difference between refrigerant condensing and evaporating). For example, if a heat pump draws heat from a constant temperature heat source (such as ambient air) at 20°C and rejects it to heat water from 50°C to 85°C, the log mean temperature of the heat sink is 67°C and the temperature lift defined for the system is 47°C

NOTE – Do not confuse temperature lift with temperature increase.

Thermal stratification	When the liquid used for thermal storage has different temperatures that do not mix. Liquid density changes with temperature. Normally, higher-temperature liquid rises to the top of the tank and lower-temperature liquid sinks to the bottom. Stratification can be stable if, for example, hot water enters and leaves the top of the tank and cold water enters and leaves the bottom of the tank, and flow distributors are used to prevent mixing. Such a system, if configured correctly, can increase the connected heat pump's operating efficiency and help ensure it delivers a consistently high temperature to the application
Transcritical heat pump	A heat-pump cycle where the high-pressure part of the system is operated above the refrigerant's critical point. When refrigerant is above its critical point, it cannot condense. Instead, it cools as a supercritical fluid, so the 'condenser' is referred to instead as a 'gas cooler'. As the refrigerant will be a supercritical fluid on the high-pressure side of the system, the heat pump – especially the gas cooler – shall be appropriately designed. A transcritical system allows the heat pump to generate a higher temperature with its refrigerant than a subcritical (conventional) system could with the same refrigerant

1.4 Abbreviations

The following abbreviations are used in this PAS:

AHU	Air-handling unit
BLDC	Brushless, direct-current motor
BMS	Building management system
CC	Capital cost
CFC	Chlorofluorocarbon
CO_2eq	Carbon dioxide equivalent
COP	Coefficient of performance
DB	Dry bulb
dBA	A-weighted decibels (measurement of sound loudness weighted to mimic the perception of the human ear)
EC	Electronically commutated
ETS	Emissions trading scheme
EVI	Enhanced vapour injection

EXV	Electronic expansion valve
GHG	Greenhouse gas
GWP	Global warming potential
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbon
HCFO	Hydrochlorofluoroolefin
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HPWH	Heat-pump water heater
HTHP	High-temperature heat pump
HVAC	Heating, ventilation, and air conditioning
LAN	Local area network
MEPS	Minimum energy performance standards
MVR	Mechanical vapour recompression
OC	Operating costs
ODP	Ozone depletion potential
PAS	Publicly available specification
PCB	Printed circuit board
PFAS	Poly- or perfluorinated alkyl substance
PLC	Programmable logic controller
SCOP	Seasonal coefficient of performance
SEER	Seasonal energy-efficiency ratio
SG	Safety group
SPL	Sound pressure level
STC	Small-scale technology certificate
TXV	Thermostatic expansion valve
UV	Ultraviolet
VPN	Virtual private network
VSD	Variable-speed drive
WB	Wet bulb

1.5 Notation

This PAS uses the following notation:

COP_{Carnot}	Carnot coefficient of performance
P	Electrical power consumption of the heat pump (kW)
P_c	Critical pressure
Q_{cooling}	Amount of cooling provided by the heat pump (kW)
Q_{heating}	Amount of heating provided by the heat pump (kW)
T_{BP}	Boiling point at standard atmospheric pressure
T_c	Critical temperature
$\bar{T}_{\text{sink}}$	Log mean temperature of the heat sink (°C)
$\bar{T}_{\text{source}}$	Log mean temperature of the heat source (°C)
η_{Carnot}	Thermodynamic efficiency of the heat pump relative to the Carnot cycle

2 INTRODUCTION TO HEAT PUMPS

2.1 What a high-temperature heat pump is

A high-temperature heat pump (HTHP) is a device that absorbs heat energy from a heat source at a low temperature and transfers or ‘rejects’ the heat energy to a heat sink at a high temperature. (In this PAS, ‘high temperature’ means a heat sink between 50°C and 160°C.) This has the effect of providing heat at the heat sink. Another energy source (usually electrical energy) is required to operate the heat pump.

An HTHP can be very energy efficient because it transfers more energy from the heat source to the heat sink (this means the amount of heating it does) than the electrical energy needed to operate it. Typically, it is also more energy efficient to use an HTHP than an appliance that directly uses electrical energy for resistance heating (such as a conventional electric hot-water tank). The general flow of energy is illustrated in Figure 1.

In a typical application, the heat source is ambient air and the heat sink is hot water. Industrial applications commonly use process low-grade waste heat as their heat source. As low-grade waste heat usually has a higher temperature than ambient air, it allows an HTHP to achieve higher temperatures with greater efficiency than air-source systems can typically attain.

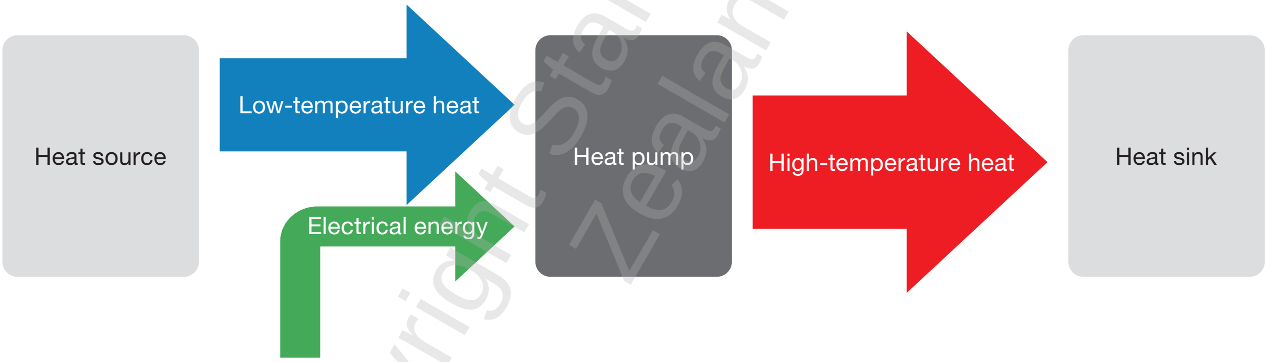

Figure 1 – Energy flows in an HTHP

There are several methods to implement an HTHP. This PAS considers the most common, best-developed, and generally most efficient method, which is the mechanical vapour-compression cycle (see Figure 2). In this type of HTHP, heat is absorbed from the heat source by evaporating a refrigerant at a low pressure and transferring heat to the heat sink by condensing, or cooling, the refrigerant at a high temperature. Most of the additional energy for operating the HTHP is used to compress the refrigerant, but also to operate fans, pumps, and controls.

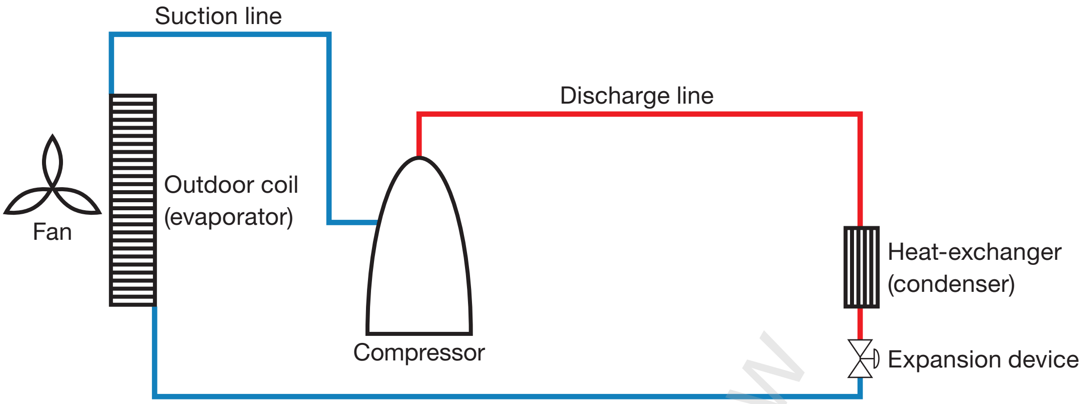

Figure 2 – Schematic diagram showing typical components in a single-stage, air-source HTHP

2.2 Reasons to consider a high-temperature heat pump

Under the Climate Change Response (Zero Carbon) Amendment Act 2019, the New Zealand Government has set a target to achieve net zero carbon emissions by 2050. The Government has also indicated it will align with the global pledge, agreed at the 2023 United Nations Climate Change Summit, to double energy efficiency and treble renewable energy.[1] Reducing the use of fossil fuels across New Zealand is an obvious step towards these targets.

Fossil fuels continue to be used in many heating applications in hospitals, prisons, and industry. As New Zealand's electrical energy supply has a high renewable component, HTHPs have excellent potential to replace coal and other fossil fuels for heating, which can cost effectively mitigate the impact of heating fuel use on climate carbon.

In domestic applications, HTHPs can be a cost-effective and sustainable method of providing hot water by replacing gas burners and direct electrical heating.

2.3 Advantages of high-temperature heat pumps

There are several general advantages of using HTHP technology:

(a) HTHP technology is **electrically powered**, so it results in lower carbon emissions than heating systems that use fossil fuels. This is especially true in New Zealand, where the electrical grid is dominated by renewable sources and connections to the grid are widely available and reliable;

(b) HTHP technology is **energy efficient**, as HTHPs typically transfer more heat energy than they consume in electrical energy. They are typically more efficient and have lower operating costs than other heating options;

(c) HTHPs typically have **lower maintenance costs** than equivalent fossil-fuel- and combustion-based heating systems;

(d) HTHPs can efficiently meet a **variable heating demand** when they are integrated with sophisticated control systems;

[1] COP28 National Statement for New Zealand, 9 December 2023, https://www.beehive.govt.nz/speech/cop28-national-statement-new-zealand

(e) HTHPs have a **relatively small physical and environmental footprint**, so they do not normally need a separate resource consent. They do not require fuel to be stored on site and they do not produce flue gases (such as sulphur oxide or nitrogen oxide), ash, or other pollutants;

(f) In sophisticated installations, HTHPs can be used to **recover and upgrade waste heat** and **provide cooling capacity** as well as heat; and

(g) HTHPs are **viable for a wide range of capacities** and can be selected to suit the specific application. It is possible to install multiple small HTHPs, each close to an application, which may reduce the need for vast hot-water reticulation systems and enable them to be gradually deployed across a site as a retrofit or as demand grows.

Some application-specific advantages of HTHPs are discussed in other sections of this PAS.

2.4 Limitations of high-temperature heat pumps

HTHP technology has several limitations:

(a) HTHPs are usually **more expensive to purchase** than other heating systems. Therefore, it is critical that they have a relatively high COP to ensure that running costs are significantly lower than other heating options. Potential purchasers should assess the purchase price, installation costs, and running costs to understand the lifetime cost of an HTHP;

(b) HTHPs that use ambient heat sources have **lower heating capacity and are less efficient as the ambient temperature reduces**. HTHPs typically require a defrosting cycle below about 7°C because frosting (ice) can form on the air-source heat-exchanger. A well-designed air-source HTHP will have an effective defrost cycle. Other technologies (such as vapour injection) can enhance the performance of systems (particularly non-stratified or closed-loop systems) at low ambient temperatures. Apply caution with an HTHP designed to operate without a defrost cycle if the HTHP is being operated at an ambient temperature below 5°C;

(c) HTHPs can be **grossly oversized and expensive** because they are often designed for the worst-case winter morning. This means they may operate part-loaded and less efficiently for most of the year. Initial costs can by greatly reduced by mitigating underperformance during extreme weather conditions by having resistive heating or thermal storage. Alternatively, newer, inverter-based systems can be designed to operate at an elevated capacity to meet peak demand for short periods at reduced efficiency, while offering enhanced efficiency when they operate in part-load conditions for the rest of the time. Systems with advanced controls, which are designed to operate across a wide flow rate and capacity range, can operate very efficiently in part-load conditions;

(d) Large commercial or industrial HTHPs will **demand significant electricity**, which may require upgrading electrical supply lines within the site and in the distribution network that supplies the site from the national grid. Electrical upgrade costs can be significant, especially for industrial sites, and shall be included in feasibility analyses. Sites are advised to engage early with the retailer, electricity network company, and, where relevant, Transpower, to understand any upgrade requirements and their impact so they can review the pros and cons of accommodating a large HTHP;

(e) HTHPs use refrigerants to convey heat around the system. **Refrigerants have various risks** associated with them, such as toxicity, flammability, and high global warming potential (GWP). When HTHPs are being installed, operated, maintained, and disposed of, it may be necessary to use appropriate technical and management controls to reduce refrigerant leakage and mitigate the risks of leaks that do occur; and

(f) HTHPs are fitted with compressors and, often, fans. These can be a source of noise that may be intrusive to some people, particularly in domestic and residential settings. This should be considered when installing an HTHP (and the supplier should be able to provide details of noise levels). Ideally, HTHPs in residential settings would be located such that they minimise transmission of noise to neighbouring properties or bedroom windows.

2.5 Core technologies used in heat pumps

2.5.1 Capacity and speed control

The compressors used in HTHPs are fixed speed or variable speed. The type of compressor used determines how the HTHP's effective heat output is controlled.

Fixed-speed compressors are widely used in HTHPs that have a single compressor. When the heat-pump output exceeds the heating demand, the system can be cycled on and off. Connecting to a storage tank reduces the cycling frequency. When there is a need for a heat pump with a variable output, multiple fixed-speed compressors can be used as part of the same refrigeration circuit, or via individual refrigeration circuits.

Some fixed-speed compressor-based systems control capacity by allowing refrigerant to bypass either within the compression chamber or between the discharge and suction ports (for example, slide valves are a bypass technology for screw compressors). Refrigerant bypass decreases the efficiency of an HTHP system.

A compressor's output can also be controlled by varying the compressor speed. Typically, this is achieved by using a variable-speed drive to reduce the compressor speed or a brushless, direct-current (BLDC) compressor motor and inverter drive (this is referred to as an inverter compressor system). An inverter drive can allow the compressor to rotate at higher-than-standard speeds, which increases compressor output. It can also reduce compressor speed, which makes the system more efficient under part-load conditions.

2.5.2 Compressor type

Compressors used in HTHPs can also be classified by how they compress the refrigerant. Five types of compressors are commonly used:

(a) **Rotary compressors** are often used in HTHPs with lower capacity. Rotary compressors contain a large cylinder with a smaller vane that rotates eccentrically within a casing;

(b) **Scroll compressors** are widely used in HTHPs with medium capacity. Scroll compressors contain two interleaving scroll plates (these resemble spirals); one is stationary and the other orbits eccentrically. The capacity can be controlled by changing the speed of the scroll plates, or allowing more bypass to occur. Compliant scroll compressors can allow small amounts of liquid to pass through the compressor without damaging the compression head, which makes the compressor more durable;

(c) **Reciprocating compressors** resemble a piston within a cylinder. Multiple compressors can be connected to a common drive shaft. The HTHP's output can be varied by changing the speed of the drive shaft. Reciprocating compressors are particularly susceptible to damage from liquid entering the compression chamber;

(d) **Screw compressors** are often used in HTHPs with higher capacity that use lower pressure refrigerants. Their capacity can be controlled by changing the rotating speed of the screws, which allows more bypass to occur, or via a slide valve, or both; and

(e) **Centrifugal compressors** are used in HTHPs with high capacity that use lower-pressure refrigerants. Their capacity is controlled by the compressor's rotating speed. Centrifugal compressors with magnetic bearings do not need oil as a lubricant in the system.

2.5.3 Liquid-injection cycle

A liquid-injection cycle is used to control the temperature of gas discharged from the compressor when it would otherwise be too high. A small amount of liquid refrigerant is injected into a second inlet port on the compressor.

2.5.4 Vapour-injection (economised) cycle

A vapour-injection or 'economised' cycle also uses the compressor's second inlet port. In industrial-scale screw compressors, this port is often called an 'economiser' port. In a vapour-injection cycle, liquid refrigerant is sub-cooled after the condenser, using another heat-exchanger or flash vessel (see Figure 3). Some of the refrigerant boils or 'flashes' to sub-cool the rest of the refrigerant at an intermediate pressure (this is known as 'two-stage expansion'). The flash vapour is compressed via the compressor's vapour-injection port. This is called enhanced vapour injection (EVI).

The vapour-injection cycle increases the heat pump's capacity and energy efficiency because sub-cooled refrigerant means a lower refrigerant flow rate can deliver the same amount of heating and the sub-cooling flash vapour is compressed only from intermediate pressure to discharge pressure (rather than from the lower suction pressure to the discharge pressure).

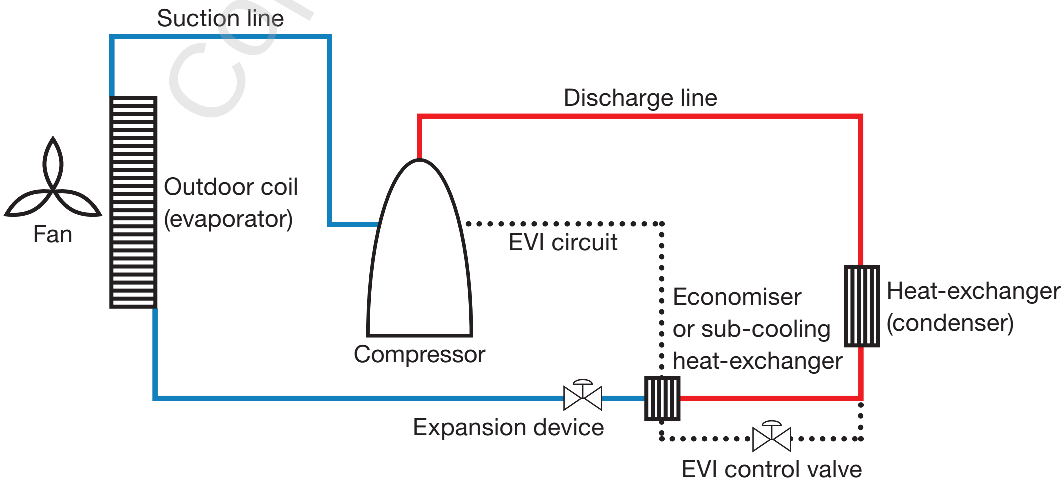

Figure 3 – Schematic illustrating the addition of EVI to a single-stage, air-source HTHP

R744 (carbon dioxide) systems commonly use a small, separate compressor to compress the flash gas from the sub-cooler, instead of a vapour-injection port. This configuration is often referred to as 'parallel compression'. It has the same benefits as vapour injection.

2.5.5 Metering or throttling devices

The flow of refrigerant through the cycle is controlled by a metering or throttling device. This device creates a pressure drop, which causes the fluid to expand before it enters the evaporator. There are four main types of metering devices.

(a) **Electronic expansion valves (EXV)** are more expensive than thermostatic expansion values, but they allow more precise metering of refrigerant flow through a heat-pump system. An EXV can keep the refrigerant superheat at the evaporator exit stable at lower superheat values, which provides efficiency benefits.

(b) **Thermostatic expansion valves (TXV)** have a small opening that is opened or closed by a diaphragm. The diaphragm is controlled by fluid that expands or contracts according to its temperature. TXV are a cost-effective way of controlling the flow of refrigerant. They perform reasonably well but have a limited operating range and often require manual adjustment. To achieve stable operation, the superheat value is often higher than EXV, which can be less efficient.

(c) **Capillary tubes** are the most basic way to control the flow of refrigerant. They consist of a fixed length of tube with a very small diameter, which creates a fixed-pressure drop. Capillary tubes are inexpensive, but they have a very small operating range and provide limited stability of superheat. This compromises their performance and efficiency, so they should not be used.

(d) **On/off or modulating valves** control the refrigerant-liquid level in downstream, low-pressure vessels. This type of valve is used in large systems that are not configured as direct expansion.

2.5.6 Motor types

Electronically commutated (EC) motors, BLDC motors, and permanent magnet motors are supplied with DC electricity and have an artificially rotating field around them. This technology gives the motor a larger speed range than is normally possible with an AC motor. Small- and medium-scale heat pumps that use an inverter compressor and variable-speed fans and pumps are commonly based on this technology. These motors are usually much more efficient than small-scale, shaded pole motors, which are often found in evaporator fans and pumps associated with HTHPs.

2.5.7 Defrost methods

Heat pumps need water heaters to defrost the evaporator coils when they operate in ambient temperatures below about 7°C, as frost or ice can accumulate. There are two main methods to defrost coils:

(a) **Reverse-cycle operation** allows the unit to operate temporarily in cooling mode, and uses the heat generated to defrost the coil; and

(b) **Hot-gas bypass systems** essentially short-circuit the compressor discharge through the evaporator coil, which allows the heat of the compressor motor to defrost the coil.

Reverse-cycle operation is generally more efficient than hot-gas bypass systems, but it requires the refrigerant volume in the evaporator and condenser to be approximately the same.

3 DOMESTIC HEAT-PUMP WATER HEATERS

3.1 General

Domestic heat-pump water heaters (HPWH) tend to be smaller-capacity (3 kW to 5 kW) air-source units. In general, they can replace a household's existing hot-water heater, but, in some cases, they can supplement an existing hot-water system.

They are designed for light-to-medium duty and should have a design life of well over 10 years. These units are not generally covered by a service contract, so significant faults are likely to result in replacing a unit rather than repairing it.

The air source for the evaporator can be outside air or, in some instances, air ducted from within the building envelope. It is important that manufacturer guidelines are followed to ensure the unit has sufficient air flow.

Some models may also be used to provide space heating or cooling, although these models typically have higher upfront costs than dedicated HPWH.

3.2 Technologies available

3.2.1 Integrated 'all in one' units

These units consist of a tank with a small refrigeration system mounted at the top or base. These units can source air from the immediate surroundings or have it ducted to, from, or to and from the unit. The condensers can be wrapped around the outside of the tank or located within either the tank or the unit's housing. Units with a wrap-around condenser generally need a larger tank (more than 270 L) to allow sufficient refrigerant sub-cooling while delivering an acceptable quantity of hot water.

3.2.2 Split units

3.2.2.1 General

Typically, split units offer greater heating capacity than integrated units as they separate the refrigeration system from the storage tank. Split units can heat the water outside of the tank and pipe it to the tank, or they can pipe refrigerant to a heat-exchanger within the tank. In units with an external heat-exchanger, the heated water can be delivered to the top of the tank at the required temperature and thermally layered from the top of the tank down (a single-pass heat pump unit). Alternatively, the heated water can be supplied near the bottom of the tank, which heats the water in the tank from the bottom up (a multi-pass heat pump unit).

3.2.2.2 Single-pass heat pump units

Single-pass heat pump units can be used to produce domestic hot water, particularly when the demand exceeds the capabilities of integrated units (see Figure 4). Single-pass heat pump units have these main advantages:

(a) They rapidly recover hot water once the tank is depleted;

(b) They take advantage of the natural thermal stratification that occurs in 'push-through' heated-water storage tanks. This increases the sub-cooling and, therefore, the efficiency of the heat-pump system;

(c) They significantly reduce the peak refrigerant pressure for a given target water temperature, which increases the efficiency and durability of the compressor; and

(d) They integrate more effectively into systems that demand flexibility.

When single-pass heat pump units are applied to reticulated-return (ring-main) systems, care is required to maintain the thermal stratification within the storage tank.

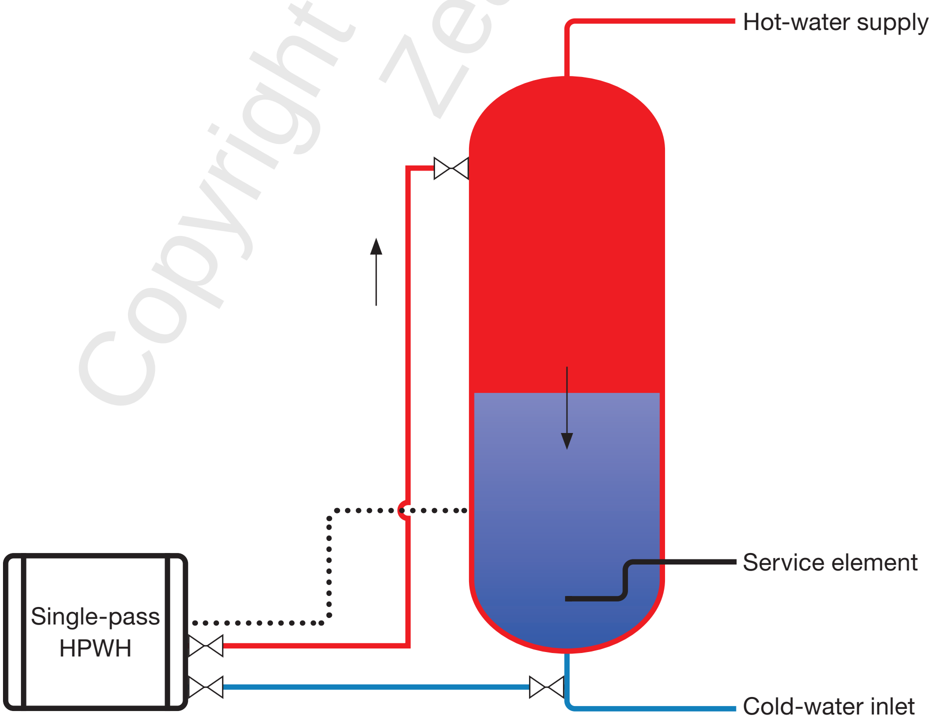

Figure 4 – Schematic showing the integration of a single-pass HPWH with a domestic hot-water system

NOTE – The illustrated tank is thermally stratified.

3.2.2.3 Multi-pass heat pump units

Multi-pass heat pump units need to be carefully designed and applied to be effective domestic HPWH (see Figure 5). This is primarily because a high flow rate is required between the HPWH and the storage tank, which generally results in the tank becoming 'mixed'. 'Mixing' the storage tank means that, for a large proportion of the time, the HPWH operates with a high inlet water temperature (top-up heating after relatively small hot-water draws). HPWHs operate less efficiently with high-temperature inlet water than with low-temperature inlet water.

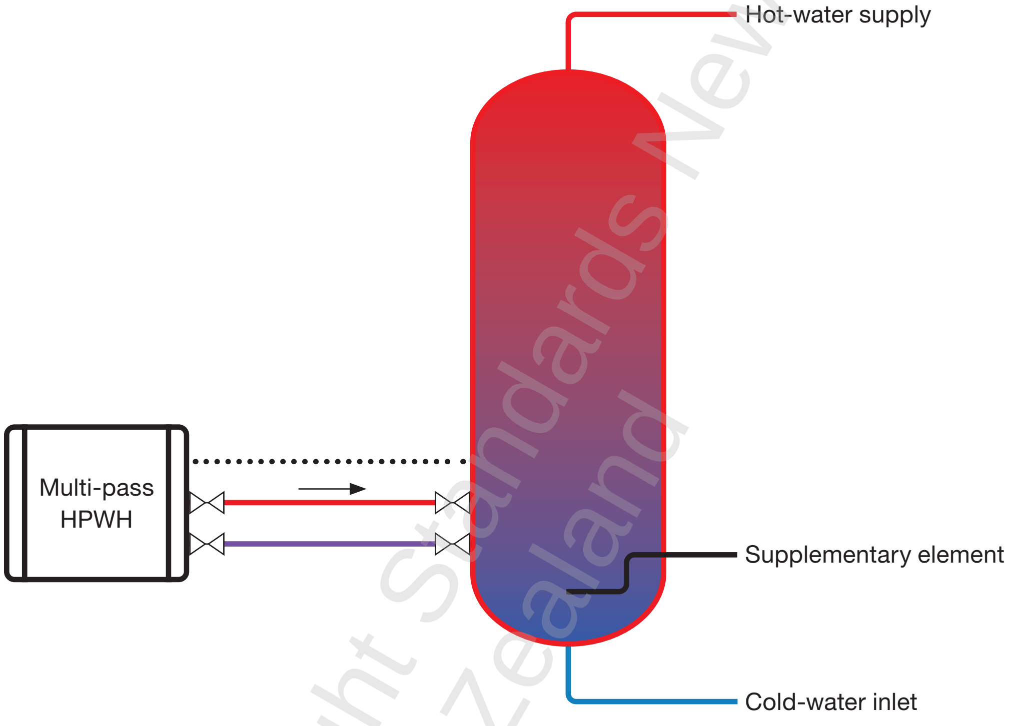

Figure 5 – Schematic illustrating integration of a multi-pass HPWH with a domestic hot-water system

3.2.2.4 Combined space-and-water-heating heat pumps

Although these units are available, they should be carefully considered and applied because they have some disadvantages.

(a) In general, space heating needs more capacity than hot-water heating, which leads to a mismatch outside the winter space-heating season.

(b) In general, the temperatures required for water heating are higher than for space heating. Systems that store water at the temperature required for hot water, but use it for space heating, are inefficient for two reasons:

   (i) The heat for space heating is generated at the same high temperature lift required for the hot water, whereas a separate system could generate it more efficiently with a lower temperature lift

   (ii) The thermal-storage system leads to larger heat losses; and

(c) Inverter units with separate refrigerant-to-water heat-exchangers for the space- and water-heating applications are more efficient because the target water temperature can be adjusted. However, their water heating is less efficient than an integrated or single-pass HPWH.

3.3 Heat-exchangers for potable-water systems

For domestic HPWHs, the choice of heat-exchanger between the refrigerant system and potable water is an important consideration. This is especially true in regions that have higher levels of dissolved minerals in the water. In general, plate heat-exchangers, or other compact types with small flow channels, initially offer good heat-transfer features. However, their performance is prone to deteriorate through fouling (when the passages between the plates become blocked). These blockages reduce the heat-exchanger's performance. In addition, the refrigerant pressure and discharge temperatures of the refrigeration system increase, which can reduce the compressor's service life.

In this respect, HPWHs with 'wrap-around', 'co-axial', and 'tube-in-shell' condensers, which are less prone to fouling, are more likely to provide a longer service life.

4 REFRIGERANTS FOR DOMESTIC APPLICATIONS

4.1 General

Refrigerant is a critical element of a domestic heat pump. A refrigerant is referred to by an alphanumeric code (for example, R32), which indicates its chemical composition. Appendix A lists some potential refrigerants (this is not an exhaustive list). Talk to your supplier about which refrigerant is appropriate for your specific requirements.

Domestic HPWHs use a range of refrigerants. Older systems often contain refrigerants with high GWP, such as R134a, R407c, or R410a. More recent products often contain refrigerants with much lower GWP, such as R32, R290 (propane), R600a (iso-butane), and R744 (carbon dioxide).

Single-pass heat pump units are available as R32 or carbon dioxide. R32 products are more cost effective than carbon dioxide products, but carbon dioxide products allow higher water temperatures to be achieved. The range of propane and iso-butane options is expected to increase.

4.2 Environmental and safety issues

Many current refrigerants are hydrofluorocarbons (HFCs). HFCs are potent greenhouse gases (GHGs) that contribute to climate change if released to the atmosphere. It is anticipated that these refrigerants will become less readily available owing to international efforts to reduce their impact on the climate.

Some refrigerants are also hazardous owing to their toxicity or flammability. The refrigerant installed in a heat pump shall be appropriately labelled, with correct safety measures applied for the specific hazard level. See AS/NZS ISO 817:2016 for definitions of safety levels.

It is generally not advised to charge an HTHP with a refrigerant different from the one it was designed to use. Refrigerants of different safety classifications require different engineering controls, so charging an HTHP with a refrigerant that has a different safety class could be hazardous to people or property. Consult an appropriately qualified technician before doing this.

'Charge minimisation' means designing a heat-pump system to use as little refrigerant as possible. This practice can be cost efficient and can mitigate some of the risks associated with using hazardous refrigerants.

When consumers select a refrigerant for their application, it is recommended to select the refrigerant with the lowest GWP practicable (at least GWP < 750), given cost and efficiencies. If they use a refrigerant with hazardous qualities, they shall implement all the appropriate safety measures.

4.3 Refrigerant leaks

In a sealed HTHP unit, the refrigerant is designed to remain within the system. However, refrigerant leaks will occur if pipes or other components fracture, for example, due to corrosion or metal fatigue from vibration.

Consumers should ensure they service their HTHP unit as specified by the manufacturer. They should also carefully observe any change in the home's water temperature or increased power consumption by the HTHP, as these may indicate a refrigerant leak.

4.4 Temperature constraints

To achieve high energy efficiency, most HTHPs operate with the refrigerant's condensing temperature well below the refrigerant's critical temperature. The critical temperature (see Table A1) is the upper temperature limit that most systems can provide at the heat sink. The exception to this is systems designed to operate using a transcritical cycle, such as carbon dioxide systems.

5 OPERATIONAL CONSIDERATIONS FOR DOMESTIC HEAT PUMPS

5.1 General

The temperature of some heat sources, such as ambient air, vary according to the time of day and season. The COP of an installed HTHP will, therefore, vary according to these conditions. However, a properly specified and configured HTHP should deliver the desired temperature despite changes in the heat source.

Integration of the heat pump with the hot-water system is an important consideration for an HPWH. A residential HPWH has the potential to be more efficient when the water-inlet temperature is colder. This is especially the case with carbon dioxide heat pumps, which require a relatively cool water-inlet temperature to be efficient and maintain heating capacity.

5.2 Thermal stratification

In domestic applications, HTHPs are most often used for water heating. Therefore, they frequently incorporate some form of thermal storage (typically an insulated water tank). If the thermal storage is sufficiently large, it may allow the HTHP operation to be reduced or shut off at times when there is a high demand on the electricity network. Depending on consumers' electricity tariffs, this may mean they avoid significant demand charges for electricity and allow them to use cheaper, off-peak electricity.

The design and implementation of thermal-storage-and-distribution systems should be carefully considered as they can affect the temperature supplied to, and by, the HTHP. Heat pumps can perform less well, and be less efficient and durable, if they are operated with a much lower temperature lift across the heat-pump condenser than they are designed for. This should be considered in the design of any thermal storage system.

To maximise the performance of an HTHP, it is good practice to maintain the thermal storage as a stratified tank. In a stratified tank, higher-temperature (lower-density) water will be at the top of the tank, while lower-temperature (higher-density) water will be in the lower portion. This results in water of different temperatures forming natural layers in an unmixed vessel. Figure 4 illustrates a stratified tank.

Maintaining 'stratification' allows the heat pump to draw water from the coldest area of the tank (that is, the bottom) to cool the condenser and sub-cool the refrigerant. The heated water is then returned to the hottest area of the tank (that is, the top). The heat-distribution network would draw from the hottest part of the tank to satisfy the heating load, while adding make-up water to the coldest part of the tank. This allows the heat pump to supply the hottest possible temperature to the application. It is important that the water in the storage tank is not allowed to mix to maintain the stratification effect, so the tank inflow and outflow shall be set up with this in mind (for example, a distribution plate is often included in the storage tank).

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In most cases, domestic HTHP potable-water heaters need a storage tank. In most installations, the hot-water outlets are directly supplied from the storage tank without a reticulated pumped circuit. The water heater should be designed to maintain natural thermal stratification in the tank by:

(a) Avoiding the use of electric elements in the base of the tank as booster heaters. If the water heater requires an electric element, ensure it is positioned above the tank's temperature sensor;

(b) Controlling the flow rate from the HTHP into the tank, based on the manufacturer's specifications. Single-pass heat pump units have an inherently controlled flow;

(c) Ensuring that the size of the port that connects heated water from the heat pump to the tank is sufficient for the design velocity into the tank to be no greater than 0.5 m/s, unless there are internal baffles in the tank;

(d) Ensuring that the hot-water return from the heat pump is to the top of the tank (single-pass heat pump system) or to the bottom of the tank below the thermostat (multi-pass heat pump system); and

(e) Avoiding connecting reticulated-return connections to the main storage tank unless the system is specifically designed for this.

Thermal conduction down the walls of a tank is a key factor in heat-energy transfer between stratified layers. Tanks made from lower-thermal-conductivity materials (such as stainless steel) are generally preferred for thermally stratified systems but should still be insulated.

5.3 Legionella control

Legionella pneumophila bacteria cause Legionnaires' disease, which is a severe form of pneumonia. Heated-water systems can support *L. pneumophila* to grow. Installations that supply reticulated hot water that is tempered to 45°C adjacent to the tank are particularly vulnerable to *L. pneumophila*.

The design of any thermal-storage system should consider this risk and take appropriate measures to prevent it. For example, regularly heating the tank to at least 60°C or using ultraviolet (UV) sterilisers are common ways to prevent *L. pneumophila* growing. Clause G12 of the New Zealand Building Code contains a range of other options for controlling the growth.

When installing heated-water systems at places that serve people with compromised immune systems, such as hospitals and care facilities, particular care is needed.

5.4 Controls

In a residential situation, an HTHP should be automatically controlled, with minimal input from the user.

The tank sensor shall be raised sufficiently from the base of the tank so that the HTHP operates for at least 20 minutes under all hot-water draw-off conditions. Setting the thermostat dead band sufficiently wide is one way to achieve this.

An HTHP shall be sized and configured to meet the premises' hot-water demand without needing a supplementary (booster) heater. Products shall also have an active defrost method. If the lower region of the tank has a backup element, this shall be activated only if a fault occurs with the HTHP. Some units incorporate the installed element as a supplementary heater for higher heating capacity or have a higher temperature setpoint. However, this feature significantly reduces the water heater's energy efficiency and should be avoided.

A few products are designed to be integrated with a reticulated (ring-main) hot-water system. Unless the manufacturer specifically supports this function, using a heat pump for a ring-main system is not recommended.

Typically, residential HTHPs are not connected to load-management-control tariffs. In future, heat pumps with external connectivity (such as Wi-Fi) will be more common and may allow owners or demand-flexibility aggregators to manage the load.

Some units have a timer function, allowing them to operate more efficiently when they are set to preferentially operate during the warmer periods of the day.

Ducted units produce significant quantities of cold, dry air, which can help to cool and dehumidify the building envelope when configured. In parts of New Zealand, this is a useful feature during the warmer seasons. However, caution is needed to ensure that ducting and dampers do not violate the minimum air flow requirements determined by the manufacturer.

5.5 Factors that affect performance and durability

For water heating, HTHPs are most effective when they can operate continuously for sustained periods when heating is required. If the heating capacity is 'oversized' relative to the thermal load or demand for hot water, the HTHP will 'short cycle' or repeatedly turn on and off. Short cycling can accelerate the degradation of the system's components and reduce its performance.

To prevent short cycling, a hot-water storage tank is recommended. The storage tank and controls should be configured so that the HTHP turns on once the hot water in the tank is significantly depleted, then turns off when the tank is fully recharged with hot water (see 5.4).

If the heating capacity is 'undersized', the HTHP may fail to meet the required heating demand and need 'boosting' (this often involves an electrical heating element), which may reduce the system's annual energy savings.

A well-designed HTHP will not need an electric resistance heater to meet the required water temperature unless the environmental conditions are extreme. Routinely using resistance heaters in the tank will significantly reduce the HTHP's efficiency.

Using resistance heaters to maintain reticulation losses in a thermally isolated zone as part of a thermally stratified system can be an appropriate solution for commercial systems.

5.6 Compliance and standards

Air-source HTHPs for domestic water-heating systems shall be tested to AS/NZS 5125.1 and subsequently modelled with AS/NZS 4234.

AS/NZS 5125.1 characterises the energy performance of an HPWH by measuring the power input and the thermal energy produced by the HPWH under a range of water inlet temperatures and ambient conditions within a mixed tank system. The test data allow correlation to a mathematical model of the HPWH, which can then be extrapolated to other system configurations, for example, stratified tank. A set of performance coefficients are derived, representing how the COP and heating capacity of the water heater adjust to changing operational conditions.

The performance coefficients derived from testing to AS/NZS 5125.1 can then be used as part of a thermal simulation model, allowing a wide range of system configurations, climatic conditions, and heating demands to be assessed for energy performance.

AS/NZS 4234 describes a methodology for undertaking thermal simulations of a water heating system using the performance coefficients derived in AS/NZS 5125.1. Factors that influence the performance of an HPWH (such as thermal stratification of the storage tank) and various controls strategies can be configured within the simulation software. AS/NZS 4234 provides an estimate of the annual energy consumption by the water-heating system to deliver a thermal load for a specified climate location. A SCOP for the modelled system can also be determined.

By comparing the modelled energy usage with a reference heating system (either that specified in the AS/NZS 4234, or an existing system being replaced) an estimate of the difference in the energy use between the two systems can be made. By applying the carbon-emission factor for the energy consumed by the system, the difference in GHG emissions between two or more systems can be calculated.

The tapping load test, described in Appendix G of AS/NZS 5125.1, provides an alternative approach to assess the performance of an HTHP used for heating water in a domestic context. The main advantage of the tapping load test is that it tests the water heater on the basis of its energy content versus the electrical energy required to produce the energy content and does not consider the technology involved. ISO 19967-1 and EN 16147 describe similar test methods.

5.7 In-situ performance verification

The approximate in-situ performance of a domestic HPWH is best determined by measuring the volume of hot water delivered from a tempering valve installed on the outlet of the water heater, the electricity used, and the location's monthly cold-water temperature. The set temperature of the tempering valve shall be at least 5°C lower than the HPWH setpoint.

More precise analysis of energy performance requires measuring at least every 5 seconds, in-situ, the temperature at the storage tank's hot-water outlet and cold-water inlet, the flow rate from or to the storage tank, and electricity usage.

Determining in-situ energy performance for a multi-pass HTHP by measuring the flow rate and changes in temperature across the HTHP results in excessive errors. This is because these units have a high flow rate and low temperature differential. This approach is more suitable for single-pass heat pump units. For all HTHP, the energy used for defrosting needs to be accounted for. It is preferable to measure performance based on the total hot water delivered from the tank to the household and total energy use. In-situ performance monitoring of an HTHP should be undertaken for at least a week.

5.8 Replacement of an instantaneous gas hot-water heater

Instantaneous gas hot-water heaters (sometimes known as on-demand hot-water heaters, gas califonts, or instant hot-water heaters) heat hot water as the water is needed, without using a hot-water tank for thermal storage. In many instances, these systems are mounted on the external wall of a house.

Many instantaneous gas hot-water heaters will have sufficient capacity at the electrical outlet to power an HPWH, particularly models without an internal resistance boost heater. The most appropriate options for external installation are integrated HPWHs or single-pass split units.

5.9 Replacement of an internal electric water heater

In many cases, there is not enough space where an existing tank is located to fit an integrated HPWH. When an existing tank is suitable and in good condition, a split single-pass HPWH located outside the building envelope can be fitted to the existing tank, provided the pipe lengths do not exceed the manufacturer's recommendation.

5.10 Building consent requirements for HPWH

Schedule 1 of the Building Act 2004 provides a list of exemptions to the requirements for a Building Consent for a range of water heater installations. In all cases, even where a Building Consent is not required, it is mandatory that an Authorised Person complete the work.

In general, where an existing water heater of any type is replaced or repositioned, a Building Consent is not required provided all the heat sources are controlled. A controlled heat source is one which ensures the tank cannot be heated to greater than 90°C.

Typical installations of HPWHs that would be exempt under this provision are:

(a) Replacing a water storage heater with a heat-pump water-storage heater;

(b) Replacing and repositioning an internal water storage heater with an external heat-pump water-storage heater;

(c) Replacing an external water-storage heater with an external heat-pump water-storage heater;

(d) Replacing an external gas instantaneous heater with an external heat-pump water-storage heater.

NOTE – Connecting a single-pass water heater to an existing storage tank requires a Building Consent since the existing water heater is being neither replaced nor repositioned. It is suggested that the council be approached for a discretionary exemption in this instance.

6 SPECIFICATION FOR DOMESTIC HEAT-PUMP WATER HEATERS

The HPWH shall have a SCOP greater than 3.2, as determined by AS/NZS 4234 modelling when delivering water of at least 60°C. It is also recommended that domestic HPWHs should:

(a) Typically provide the hot-water storage specified in Table 1 for the number of occupants in the household;

(b) Draw at least one tank volume through the tank each day to maintain stratification;

(c) In the case of integral condenser units, have the condenser concentrated towards the lower section of the storage vessel for best thermal performance;

(d) Deliver a water temperature of at least 60°C without needing a booster element;

(e) Have external installation to maintain a sound pressure level (SPL) below 55 dBA at the property boundary (or 3 m from the unit, whichever is less), and;

(f) For units installed within the building envelope, have ducting for the inlet and exhaust air.

NOTE – Observe the manufacturer's limits on duct length.

Table 1 – Determining the demand for a domestic water heater

Number of occupants	1–3	4–6	6–10
Medium/mains pressure (L)	150–180	180–300	> 300
Capacity (kW)	2	3	4

7 COMMERCIAL HIGH-TEMPERATURE HEAT PUMPS

7.1 General

Commercial HTHPs generally have greater capacity than those used for residential applications. They are produced in moderate quantities to a fully engineered design. They are seldom mass-produced by semi-automated production systems.

Commercial HTHPs can be used for a wide range of applications related to commercial buildings and to support industrial processes. It is important to select the most suitable product for the specific application. Commercial HTHPs often require professional design services to be successfully implemented.

7.2 Refrigerants for commercial applications

7.2.1 General

Refrigerant is a critical element of a commercial heat pump. A refrigerant is referred to by an alphanumeric code (for example, R32), which indicates its chemical composition. Appendix A contains a list of some potential refrigerants, but this is not an exhaustive list. Talk to your supplier about which refrigerant is appropriate for your specific requirements.

7.2.2 Environmental and safety issues

Many current refrigerants are hydrofluorocarbons (HFCs). HFCs are potent greenhouse gases (GHGs) that contribute to climate change if released to the atmosphere. It is anticipated that these refrigerants will become less readily available owing to international efforts to reduce their impact on the climate.

Some refrigerants are also hazardous owing to their toxicity or flammability. The refrigerant installed in a heat pump shall be appropriately labelled, with correct safety measures applied for the specific hazard level. See AS/NZS ISO 817:2016 for definitions of safety levels.

It is generally not advised to charge an HTHP with a refrigerant different from the one it was designed to use. Refrigerants of different safety classifications require different engineering controls, so charging an HTHP with a refrigerant that has a different safety class could be hazardous to people or property. Consult an appropriately qualified technician before doing this.

'Charge minimisation' means designing a heat-pump system to use as little refrigerant as possible. This practice can be cost efficient and can mitigate some of the risks associated with using hazardous refrigerants.

When consumers select a refrigerant for their application, it is recommended to select the refrigerant with the lowest GWP practicable (at least GWP < 750), given cost and efficiencies. If they use a refrigerant with hazardous qualities, they shall implement all the appropriate safety measures.

7.2.3 Refrigerant leaks

In a sealed HTHP unit, the refrigerant is designed to remain within the system. However, refrigerant leaks will occur if pipes or other components fracture, for example, due to corrosion or metal fatigue from vibration.

The HTHP unit should be serviced regularly, in accordance with the manufacturer's requirements. The service should include checking for potential refrigerant leaks. The owner or operator should also carefully observe any change in the available water temperature, any increased power consumption by the HTHP, or any unexplained, significant increase in the time it takes to recover the tank, as these may indicate a refrigerant leak. If the unit controller data can be accessed (locally or remotely), check the recorded position of the electronic expansion valves. If they are regularly in the fully open position, this indicates a refrigerant leak.

7.3 Temperature

7.3.1 Temperature of the heat source and heat sink, and the return-water temperature

HTHPs operate most efficiently when the temperature lift between the heat sink and heat source is minimised. Where possible, the system should deliver the coolest possible water temperature while still meeting the heating requirement. For example, in a radiator system, if 60°C water satisfies the heating requirement, a heat pump that delivers 60°C water is more efficient than one that delivers 70°C water. Similarly, the system should use the heat source with the warmest possible temperature (for example, warm exhaust air from the building rather than colder ambient air).

The return-water temperature back to the heat pump is also an important design consideration for many HPWH. In general, a heat pump is more efficient when the heat-sink-temperature increase (that is, the water temperature increase across the heat-exchanger) is greater. This is especially the case for carbon dioxide heat pumps, which require a relatively cool return-water temperature to maintain high heating capacity and high efficiency.

The following strategy can be used to minimise the temperature lift while also maximising the temperature increase across the heating side of the heat pump:

(a) Determine the warmest heat source available. While this is often the ambient air, if waste heat sources are available, choose the warmest;

(b) Investigate the heating requirement at the application and select the coldest temperature that will meet this requirement (this is the heat sink). Together, steps (a) and (b) minimise the temperature lift; and

(c) Configure the flow of the fluid to be heated on the HTHP's heat-sink side so that it is heated to the required temperature – determined in step (b) – from the coldest possible starting temperature without wasting the heat stored in a reticulated system. For a reticulated hot-water system, thermal storage with stratification achieves this. It is configured so that the coldest water in the thermal storage is sent to the heat pump to be heated to the required temperature for the application. Step (c) maximises the temperature increase across the heat-sink side of the heat pump.

The temperature of some heat sources (such as ambient air) varies according to the season and time of day. Therefore, the energy efficiency of an installed HTHP may vary according to the conditions. However, a properly specified and configured HTHP should provide heating at the desired temperature despite changes to the heat-source temperature.

HTHPs that use ambient air as the heat source for space heating will be most heavily constrained on cold days when the heating demand is also high. Overall, the system should be designed to perform in these worst-case conditions. However, rather than sizing the HTHP for these conditions, it can sometimes be more cost effective to use a smaller HTHP with an auxiliary heating system (such as an electric resistance heater) or thermal storage (that is, a hot-water tank). This solution means using the HTHP alone on most days but supplementing it with the auxiliary heating system on extreme-temperature days. Alternatively, an inverter-based HTHP can support limited periods of higher-than-standard speed operation to meet the heating demand during extreme-temperature events.

7.3.2 Temperature constraints

To be energy efficient, most HTHPs operate with the refrigerant condensing temperature well below the refrigerant critical point. In most systems, the refrigerant critical point (see Table A1) sets an upper limit for the temperature that can be provided at the heat sink. In practice, most heat pumps operate at least 10°C below the critical point.

Specially designed transcritical heat-pump systems can operate efficiently when the heat-sink temperature is at, or above, the critical point. These system configurations are common when carbon dioxide (R744) is the refrigerant, but seldom when other refrigerants are used. Transcritical systems may become more common in the future for HTHPs whose heat sinks have a very high or variable temperature.

The refrigerant boiling point at atmospheric pressure (see Table A1) constrains the temperature at the heat source to avoid the HTHP operating in a vacuum condition at the compressor suction. The refrigerant selected should have a lower boiling point, at atmospheric pressure, than the usual heat-source temperature. In most systems, the refrigerant boiling point sets the lower temperature constraint, based on what is practical for the heat source. Refrigerants with higher boiling points (these are sometimes called ‘low-pressure refrigerants’) are less dense as vapour at the heat-source temperature. They will, therefore, need compressors, heat-exchangers, and piping that are physically larger than what high-pressure refrigerants would need at the same temperatures. Requiring larger equipment typically increases the capital costs associated with low-pressure refrigerants.

7.4 Direction of technological development

Current trends in commercial HTHP development include:

(a) A transition towards refrigerants with low GWP, such as R32, R454b, and their equivalents. Products with GWP > 750 are not recommended because alternatives are available;

(b) An increase in the use of natural refrigerants, such as carbon dioxide and hydrocarbons (HC);

(c) An increase in the adoption of inverter compressors for applications that can use the improved part-load efficiency;

(d) The adoption of HTHPs that support variable-flow hydronic space-heating systems; and

(e) An increase in the use of mixed-mode hydronic systems that allow simultaneous heating and cooling.

For safety and cost reasons, secondary reticulation systems are also being used more to physically isolate the primary refrigerant circuit and have lower charge. For example, they allow hazardous refrigerants with lower GWP to be used safely. In HTHPs, the secondary reticulation fluid is often water, and, in commercial applications, the secondary refrigerant circuit is often referred to as a ‘hydronic loop’.

Secondary systems can be less efficient due to the primary circuit operating at more extreme conditions and the secondary circuit needing extra pumping power. However, they have the advantage of decoupling the outdoor and indoor units, which allows the outdoor unit to be replaced without needing to simultaneously replace the indoor unit(s).

7.5 Electrical supply and controls

7.5.1 Electrical supply

Smaller HTHPs may be connected to an existing electrical installation relatively easily. However, the installation should still be investigated to check it has sufficient capacity and capability to operate the HTHP in the proposed location.

Larger HTHPs have significant electricity demands. A proposed site may need to be extensively investigated to ensure that the following requirements can be met:

(a) When the HTHP is running in the desired location, there is sufficient capacity in the electrical system to start and operate it satisfactorily across all expected conditions;

(b) When the HTHP is used in controlled or uncontrolled modes, the electricity purchasing contracts for the electrical system are not affected; and

(c) The HTHP can provide at least one of these benefits:

   (i) Thermal storage, which provides additional flexibility about when the HTHP operates to minimise its connection and operating costs

   (ii) Peak-demand or time-controlled systems that reduce electricity consumption to minimise capacity investment or purchasing electricity at high-cost periods. 'Time-of-use' electricity pricing is becoming increasingly common in New Zealand as 'demand-flexibility' as a service becomes available. When an HTHP is correctly sized and configured, thermal storage can ensure heat is always available while also making use of lower electricity prices.

The site investigation should include consulting with the electricity retailer and the owner of the network about these matters:

(d) Any potential impact the HTHP will have on the current electricity purchasing and connection contracts;

(e) Any alternative purchasing contracts that may offer greater benefits, or incur penalties, particularly if control is enabled on the HTHP; and

(f) Any costs associated with connecting the HTHP within the electrical system and the local supply network.

7.5.2 Controls

A modern HTHP is controlled by a microprocessor, which is typically part of a printed circuit board (PCB). A PCB has sensor inputs to measure various parts of the refrigeration system. It can also have electromechanical outputs (such as relays), analogue outputs (such as 0–10 V control signal), or communication-based outputs. A PCB controls the compressor cycling or inverter speed, the fan speed, and the metering device (that is, the electronic expansion valve). A PCB can also control the operation of pumps and valves.

Some systems use a programmable logic controller (PLC) instead of a PCB. A PLC has many of the same functions as a PCB, but it can also be part of linking the HTHP to the site's controls.

A building automation specialist may be needed to connect a commercial HTHP to a building management system (BMS) as part of integrating the HTHP with the heating system. This is an important aspect of successfully applying HTHP technology, especially in retrofit projects, and requires specialist knowledge of the application and product, particularly when the BMS system will control the HTHP.

7.6 Factors affecting durability and lifetime

An HTHP system shall be designed to operate reliably and efficiently for at least 15 years in the environment where it is installed. Ensure that an effective maintenance schedule is adhered to, as provided by the manufacturer or supplier. Also ensure that the manufacturer will maintain a supply of key parts, including control boards, for the unit's expected lifetime. Several factors reduce the lifetime of the system:

(a) Condensing pressure and discharge temperatures that exceed the operating envelope supplied by the compressor manufacturer;

(b) Insufficient suction superheat, especially when combined with insufficient discharge superheat;

(c) Corrosion of the heat pump's cabinetry and base. This can be prevented by ensuring that water does not pool but can drain freely at the base of the unit;

(d) Corrosion of coils, especially in marine and geothermal environments. HTHPs use aluminium fins, which need to be suitably coated to prevent corrosion. In some environments, some systems may need more protection than stated by the equipment manufacturer;

(e) Blocked water-side flow channels in the water-heating heat-exchanger, if the water quality is poor or has high levels of minerals that form scale. This can be a particular risk for heat-exchangers that have smaller flow channels (such as compact heat-exchangers like plate heat-exchangers). Even if the flow is not completely blocked, scaling on the heat-transfer surfaces can reduce the effectiveness of heat transfer, which reduces the system's heating capacity and COP over time. It also increases the refrigerant pressure and discharge temperatures, which could reduce the service life of the compressor;

(f) An operating system that does not comply with the manufacturer's stated operating limits. Consumers should pay attention to the manufacturer's test conditions when they select a unit. They should ensure the application's design condition is within the manufacturer's operating envelope for the compressor;

(g) An HTHP system that is sized to meet an extreme design condition but usually operates at part-load conditions. Sizing and controlling the system can prevent it starting more frequently than recommended by the manufacturer. This can be achieved by using either a rack of multiple compressors, variable-speed compressors, or thermal storage; and

(h) Systems whose circuits have split evaporator coils need each coil to have equal access to air in order to prevent the circuits flooding and running the risk of liquid returning to the compressor. Ensure air flow is balanced by paying attention to the system's proximity to walls and other equipment.

7.7 Compliance and standards

7.7.1 Electrical safety

All HTHP systems shall be electrically safe and shall comply with the Electrical (Safety) Regulations that are current when the system is installed.

Residential and light-commercial HPWHs shall comply with AS/NZS 60335.2.40:2019 or later.

Products for sale in New Zealand shall be registered in the Electrical Equipment Safety System (EESS) administered by the Australian Government.

Commercial HPWH that are not marketed as suitable for household use shall be assessed to ensure they comply with AS/NZS 60335.2.40 :2019 or later, or the AS/NZS 5149 series.

Systems that incorporate flammable refrigerants (A2, A2L, or A3) shall comply with AS/NZS 60335.2.40:2019 or later, or AS/NZS 60079.0:2019 or AS/NZS 60335.2.40, and AS/NZS 5149.

7.7.2 Refrigerants

Refrigerants in AS/NZS ISO 817 and AS/NZS 5149, Parts 1 to 4 are classified on the basis of flammability and toxicity. The HTHP shall comply with the safety requirements for the relevant classification.

Many common refrigerants are A1. A1 is not flammable or toxic but often has a high GWP rating. As the industry starts to use refrigerants with lower GWP ratings, it needs to make a trade-off with flammability. The A2L class was introduced to identify refrigerants that have a low GWP rating and low flammability, and which therefore require suitable safety controls. HC refrigerants are A3 class. They are highly flammable and require additional safety controls.

AS/NZS 60335.2.40 provides a compliance pathway for HTHPs that use flammable refrigerants (including A3) and have similar site-specific safety controls to A1 and A2L refrigerants.

Pressure vessels for refrigerants shall comply with the requirements of AS/NZS 5149, AS 1210, or an international code approved under AS 1210 (such as section VIII of the ASME's boiler and pressure vessel code; BS EN 13445-1; or PD 5500). Alternatively, pressure vessels that comply with AS 2971 can be used.

If the refrigerant is substituted, the contractor undertaking the change shall ensure that the system complies with the requirements for the substituted refrigerant. HPWHs are deemed to be electrically unsafe if the refrigerant is substituted, unless the HPWH is designed, or has been adapted, to operate safely with the new refrigerant.

7.7.3 Heat-transfer medium

If a heat-transfer medium (such as hot water) is used, the pressure equipment used to contain the heat-transfer medium shall meet the *Approved code of practice for pressure equipment (excluding boilers)*, if required by the Health and Safety in Employment (Pressure Equipment, Cranes, and Passenger Ropeways) Regulations.

7.7.4 Physical protection

HPWHs that comply with AS/NZS 60335.2.40, including products that incorporate A2L and A3 refrigerant, shall be installed so that the public cannot access them (they are usually protected by a cage or secured fence). Installations for which compliance with AS/NZS 60335.2.40 cannot be demonstrated shall comply with the site requirements of the AS/NZS 5149 series.

7.7.5 Contamination of potable-water supplies

Heated-water systems shall be installed in accordance with AS/NZS 3500.4. Potable-heated-water heaters shall comply with AS/NZS 4020.

7.7.6 Seismic design

Where appropriate, heat-pump installations, indoor heating and cooling fan coils, and radiators shall comply with NZS 4219.

8 COMMISSIONING

8.1 General

The equipment installer shall provide a commissioning report for the purchaser as part of their installation contract. For total installed capacity > 50 kW, the commissioning report should be by an independent person.

The commissioning report measures capacity and COP over a range of conditions. It records the temperature and flow rate through the heat-exchangers, including any heat-exchangers associated with thermal storage. These values should match the system's design flow rates, which are specified in the equipment supply contract. Commissioning the system should capture a range of operating conditions as wide as the prevailing climate allows. Here is an example of commissioning the system for space heating:

(a) Cool conditions (an early morning start-up on a cold day and running the system for at least 3 hours, or until equilibrium conditions are reached);

(b) Warm conditions (a warm day and minimal heating demand); and

(c) Typical conditions (mild climate and moderate heating demand).

Complete commissioning and system tuning can take several months to complete after the system's initial start-up and handoff. This process can be facilitated by remote monitoring and control.

The commissioning report fulfils several functions:

(d) It determines that the flow rate through the heat-exchangers matches the system's design flow rates specified by the manufacturer;

(e) It records the flow rate through the heat-exchangers when the system is commissioned;

(f) It confirms that all flow-control valves and variable-speed pumps in the distribution system are correctly configured;

(g) It determines that the flow rates through the heating system are within the system's design specifications;

(h) It records the temperature differential across each circuit in installations with a flow/return manifold;

(i) It records the condensing temperature (the discharge pressure), evaporating temperature (the suction pressure), and compressor's capacity after 3 hours of operation, and references these recordings against the compressor's operating envelope (this is provided by the compressor manufacturer). The system's operating conditions shall be within the compressor's operating envelope and include conditional limits, such as compressor speed limits;

(j) It measures the HTHP's capacity, power input, and COP after 3 hours of operation, and references these recordings against data supplied by the manufacturer;

(k) It observes and reports on the compressor's operations under part-load heating conditions, in which multiple compressor systems are installed for a single stage. The manufacturer's stated minimum and maximum compressor speeds shall be observed during commissioning;

➤

(l) It observes the completion of a defrost cycle during conditions that require it, or by replicating the conditions by reducing the air flow through the coil. Ensure there is sufficient thermal energy in the system to complete a defrost cycle;

(m) It uses data from the monitoring system to demonstrate that the system's control logic is operating to the stated design during the first month of the heating season;

(n) It ensures that the operation of a potable-water-heating system, if present, is separate from the operation of the space-heating system;

(o) It ensures that the operation of a potable-water-heating system, if present, does not compromise the efficiency of the space-heating system, or vice versa; and

(p) It ensures that the potable-hot-water-system, if present, meets *Legionella pneumophila* control requirements, such as those in AS2 for G12 of the NZBC (MBIE, 2023).

A report on the system's operations at 1 month, 3 months, and 6 months shall be provided. These reports can be generated using remote monitoring, if it is enabled. These reports include the following content:

(q) The maximum and average supply-water temperature achieved while the HTHP and heating system are operated;

(r) The average daily run hours per month, average weekly electricity consumption, and average monthly COP while the HTHP is operated;

(s) Analysis of the suction superheat achieved by the system, especially for events where low-suction superheat coincides with low-discharge superheat;

(t) Analysis of the system's condensing pressure and discharge temperatures, referenced to the compressor envelope, and the temperature limits for the oil used in the lubrication system;

(u) A fault report with details of what caused the faults and how they were resolved; and

(v) Details of any post-commissioning adjustments that have been made to the HTHP or heating system.

8.2 Compliance and standards

Air-source HTHPs for water-heating systems should be tested using the methodology in AS/NZS 5125.1. This methodology can be adapted to commercial air- and water-source HTHPs by increasing the size of the water tank in proportion to the unit's heating capacity, or by selecting water-source temperatures to match the application conditions. For testing purposes, the storage tank shall contain at least 50 L/kW of rated unit capacity and shall be insulated with a known heat-loss coefficient that has been tested in accordance with AS/NZS 4692.1. Other systems should be tested using an equivalent methodology.

AS/NZS 5125.1 involves measuring, under a range of standardised test conditions, the power input and thermal energy produced by the unit when coupled to a storage tank.

AS/NZS 4234 should be used to estimate the unit's annual energy consumption and SCOP. Once the performance coefficients have been determined, by analysing the test results from the methodology in AS/NZS 5125.1 or an equivalent methodology, they can be entered into the heat-pump model of a thermal-simulation software package using

AS/NZS 4234. The unit's energy use can then be compared with a reference heating system (for example, a boiler). The difference between the GHG emissions of two or more systems can be calculated using the carbon-emission factor for the energy that the system consumes. This information can be valuable when applying for a consent.

Variable-speed HTHPs shall be tested in accordance with AS/NZS 5125.1, or an equivalent methodology for non-air-source systems, using fixed compressor speeds of at least 100%, 75%, 50%, and 30%.

8.3 Remote monitoring

8.3.1 Capability to connect

The HTHP shall have the technical capability to communicate with an external device, such as a BMS. It shall allow this device to have the read and write parameters required to operate and monitor the HTHP. This PAS does not define which communication method and protocol to use; however, Modbus RTU, Modbus TCP/IP, and BACnet are commonly used by the industry. This PAS also does not define what level of access should be available to specific users, but this is an important consideration (see 8.3.5).

When the HTHP is purchased or installed, this technical capability to communicate does not need to be enabled or active. However, it shall be fitted, available, and capable of being activated without the HTHP or controller needing to be significantly changed.

8.3.2 Remote switching

The HTHP's control system shall allow the HTHP operations to be safely switched on and off via the remote connectivity system ('remote switching'). At a minimum, this may be implemented as voltage-free ('dry') contacts but may also include more comprehensive integration with the system.

The remote-switching capability should allow the heat pump to be stopped remotely if it is running, and be started remotely if it is idle. This PAS does not define who should have access to remote switching, but this is an important consideration (see 8.3.5).

The availability of remote switching may affect which electricity-supply contracts the consumer can negotiate.

8.3.3 Alarm reporting

The HTHP controller shall provide fault reporting or alarms through an external monitoring system (such as a BMS) or remote-monitoring service, or through audible or visual alerts.

8.3.4 Monitoring

HTHP systems shall have the capability to connect with a monitoring service that provides service support and diagnostics, or – if the consumer requires it – access to time-of-use pricing signals. The connectivity speed should be sufficient for monitoring, and the connection should have an up-time reliability of at least 95% over the HTHP's lifetime.

Monitoring that connects to the local area network (LAN) by an ethernet or Wi-Fi connection can encounter security restrictions, especially when two-way communication is needed to allow settings to be changed or faults to be resolved. An extra challenge with Wi-Fi is maintaining the connection over the lifetime of the unit. Wi-Fi connections ➤

are more appropriate in a residential context, where the user can access the connection easily (for example, by using a mobile phone application) and can re-establish the connection if it fails.

Remote access via the mobile network (such as the 4G or 5G networks) enables two-way communication independent of a site's security restrictions. This also avoids the cost of establishing a connection where the HTHP is located. However, the connection cost needs to be maintained for the duration of the remote-access connection.

Remote access shall not rely on the 3G network, as this network has limited time left in service.

8.3.5 Connectivity security and access

The system that provides connectivity and remote access should allow access to authorised users only. It shall use accepted good practice that maintains privacy, integrity, confidentiality, and cybersecurity. This PAS does not define who authorised users are, but this is an important consideration.

To maintain an audit trail, it is necessary to assign an individual user account to every user and record the edits made by each user on any system. Only authorised users shall be able to access specific monitored systems.

All communications between the site IT equipment and external IT infrastructure shall use predefined encryption techniques and keys, or a secure virtual private network (VPN) or structure that provides an equivalent level of security. The system shall also use a VPN, or a structure that provides an equivalent level of security, to change software and settings.

The company that monitors and controls the system remotely shall use accepted good practice and comply with relevant security frameworks, guidance, and standards. The company should have these plans and policies:

(a) A current business-continuity plan;

(b) A relevant incident-response policy and plan; and

(c) A disaster-recovery policy and plan.

Some systems that control HTHPs have the capability to measure performance and energy-usage data and to communicate or log the data. This can be useful to help quantify the benefits and cost savings of an HTHP.

The system that controls the HTHP may also have the capability to measure and record certain values related to the heat pump's operations that may help detect and diagnose faults. Alternatively, the system may be able to communicate these values to an external device (such as a BMS) for logging.

8.4 Performance assessment and verification

8.4.1 Modelling and simulation

Annual energy use and SCOP can be estimated by modelling the water-heating system (this includes the HTHP) using an adapted version of the methodology described in AS/NZS 4234 (see Appendix B).

Transient thermal-simulation software (such as TRNSYS or HTHPSimNZ) can be used to model the performance of a commercial HTHP application as well as determine what size components a proposed system will need and what its energy savings will be. Pre-monitoring the existing system to determine the peak demand and demand profile will enable the consumer to create an accurate simulation of the proposed system for the specific site. Thermal-simulation software requires characterising the performance of the HTHP according to the methodology described in AS/NZS 5125.1.

The difference in energy consumption between the existing and proposed systems can be calculated by modelling both systems. The impact of the proposed HTHP system on GHG emissions can be determined by applying emission factors to both estimates of annual energy consumption. The emission factors published by the Ministry for the Environment should be used for this purpose.

For each commercial HTHP a modelling report shall be produced, in accordance with AS/NZS 4234. The modelled system shall account for pumping energy used in the heated-water-distribution system, and operational controls logic employed by the system.

It is possible to undertake a representative simulation without expertise in thermal-simulation software.

8.4.2 Daily heating demand

When AS/NZS 4234 is applied to a commercial HTHP system, rather than using the domestic water-draw profiles defined in this standard, the daily heating-demand profile of the application shall be estimated as a function of the time of day, climate conditions, and range of ways that the system will be used.

8.5 In-situ performance assessment

8.5.1 General

Ongoing performance monitoring is important. It allows deviation from the HTHP's expected performance to be identified and diagnosed, and helps evaluate the site's, organisation's, or nation's progress towards goals such as decarbonisation.

A system's energy performance can be determined in-situ through data logging or monitoring the energy usage and heat produced over time. In general, this requires power meters, flow meters, and temperature sensors to be installed on the individual HTHP. Some HTHPs can report power usage and flow rates from their internal controls.

To ensure the desired level of accuracy is achieved, it is important to consider what timesteps the data capture requires. This is particularly important when measuring potable-hot-water draw-offs from storage tanks when the duration of draw-offs is relatively short.

Remote monitoring allows data to be frequently checked for continuity without accessing the site. This makes the data gathered more complete and, therefore, more useable. See 8.3 for more details about requirements for remote monitoring.

In-situ performance assessment allows degradation in the HTHP's expected performance over time to be readily identified and addressed through remedial measures.

The amount of performance monitoring that is stored and accessible to the system's owner should be sufficient to make meaningful performance comparisons between years, seasons, and sites. These performance comparisons shall include the system's power consumption and heat-energy production at least once every 5 minutes (time-averaged data).

8.5.2 Carbon accounting

The impact of an HTHP on a site's or business's GHG emissions should be assessed to compare the HTHP's performance with the other heating option (for example, the existing system). Both assessments should use the same weather data and the latest emissions factors published by the Ministry for the Environment.

9 APPLICATION: COMMERCIAL POTABLE-WATER HEATERS

9.1 General

Commercial HPWH for potable-water-heating applications are normally split units with a capacity of more than 15 kW and are designed for medium-to-high operating hours. They are manufactured in lower volumes than domestic units, and to a higher price point. Their design life is more than 15 years, and they are generally repairable. Units with larger capacity may be covered by a service contract, especially when a facility needs a continuous hot-water service to operate. A backup heating system is commonly part of the installation.

Most commercial HPWH are air-source units; however, water-source units are available.

Commercial hot-water loads can also be delivered by connecting several integrated, low-capacity split units together in parallel. This approach provides increased redundancy but, in general, the design life of each unit is shorter than that of a commercial HPWH.

Most systems involve a reticulated ring-main, so this shall be considered when the installation is designed, especially when multiple units are connected.

9.2 Technologies available

9.2.1 Single-pass heat pump units

Single-pass heat pump units use R32 or carbon dioxide as their refrigerant.

Single-pass heat pump units are particularly suited to commercial applications (see Figure 6). One of their main advantages is the storage tank heating from the top down. This allows the tank to function as a buffer tank, producing hot water in real time to help meet the peak demand load. This reduces the volume of hot-water storage required and provides rapid recovery if the water supply is fully depleted.

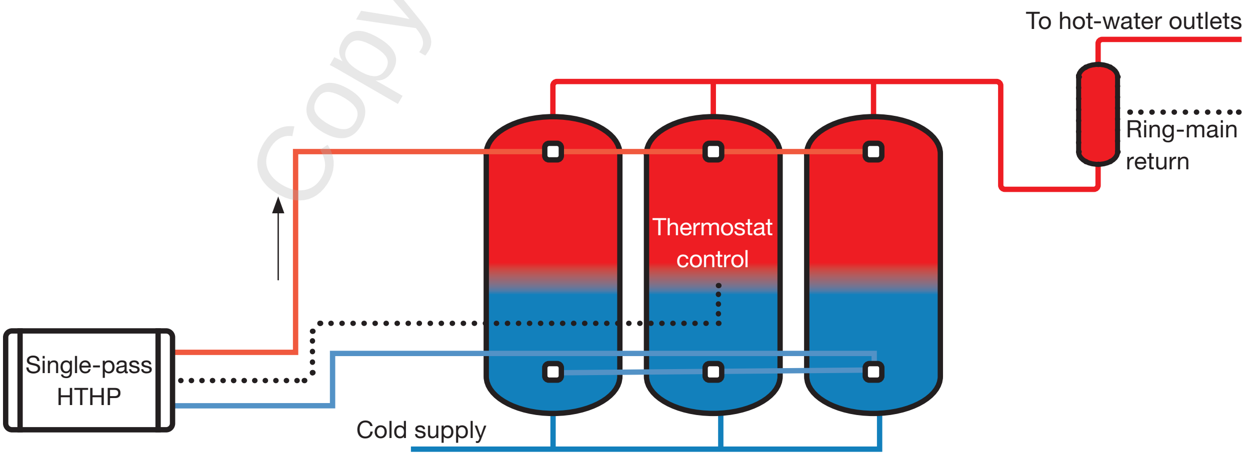

Figure 6 – Schematic of a stratified storage tank and single-pass HTHP system

9.2.2 Multi-pass heat pump units

Multi-pass heat pump units come in a range of configurations and use a range of refrigerants.

Multi-pass heat pump units that have a larger capacity require a high flow rate between the heat pump and the tank (see Figure 7). This often results in these systems having poor thermal stratification and slower recovery if the water supply is fully depleted.

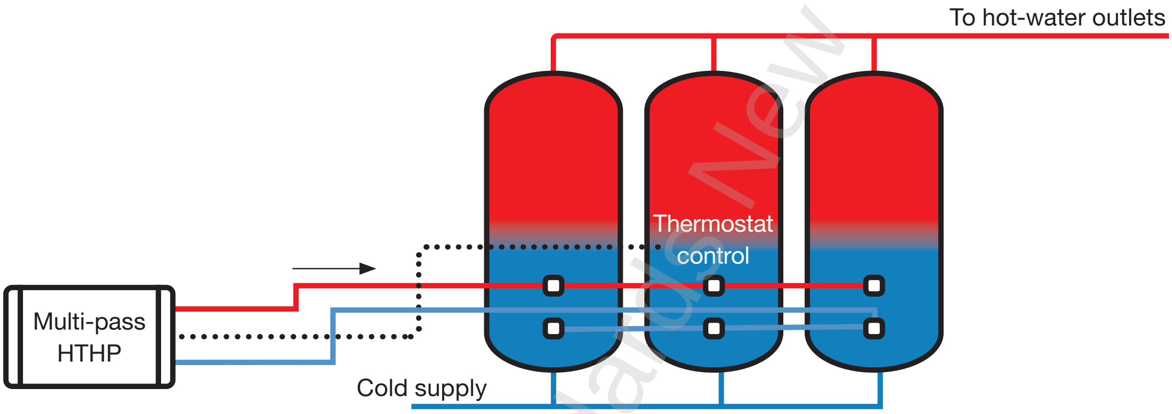

Figure 7 – Schematic of a storage tank and multi-pass HTHP system

9.3 Tank connections

Commercial systems often need multiple tanks to be connected in order to meet the required volume of stored hot water. The tanks can be connected in parallel or in series. Parallel connection is the preferred connection for single-pass heat pump systems. It reduces the flow velocity through each tank, which establishes and maintains more stable thermal stratification. With a parallel connection, it is essential to observe the 'first in, last out' principle for hot and cold water, heat-pump flow, and return connections.

The 'first in, last out' principle allows the water flow through multiple tanks to be balanced by ensuring the flow path through each tank is the same length. In practice, this often requires an additional length of pipe to be added on the return path back to the HPWH to ensure balanced flow. Flow-balancing valves should not be relied on. The same approach is required for the cold-water inlet/hot-water outlet piping, as shown in Figure 7.

If the layout of the tanks prevents the flows being properly balanced through the tanks, connect the tanks in series. Tanks connected in series have less stable thermal stratification due to higher flow rates through each tank.

9.4 Reticulated systems

Reticulated (ring-main) systems should be carefully integrated into the hot-water system (see Figure 8) to maintain thermal stratification in the main storage tank(s). The water returning from the reticulation system shall not be connected to the base of the main storage tank. Instead, a separate zone within the main tank, or a standalone tank, should be used to reheat this water, as this allows the main storage tank to remain stratified.

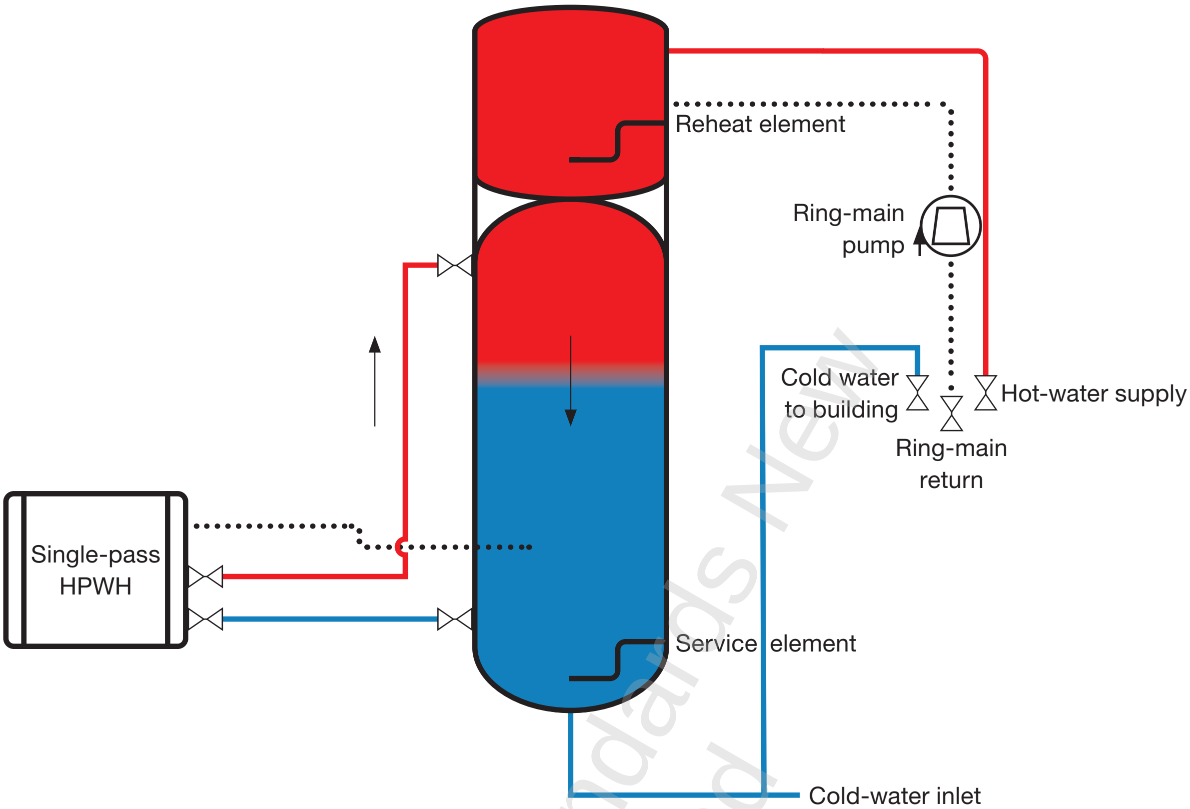

Figure 8 – Schematic of a single-pass HTHP with an integrated ring-main reheat system

9.5 Heat-exchangers

Commercials systems produce a considerable volume of hot water, and the choice of heat-exchanger will affect the unit's lifespan. Plate heat-exchangers are very effective initially, but they are prone to fouling over time. Alternatives (such as tube-in-shell and co-axial heat-exchangers) are less prone to fouling and more likely to maintain the unit's efficiency for longer periods. When plate heat-exchangers are used, they should be readily removable if they need to be replaced. They can be removed by using refrigerant isolation valves on either side of the heat-exchanger and mac unions on the water connections.

It is particularly important to choose a heat-exchanger carefully if it will be used in an area where the water contains dissolved calcium carbonate or other minerals. The process of heating water decreases the solubility of dissolved minerals, which leads to them being deposited on the heating surfaces.

Single-pass HPWHs that operate with R32 and carbon dioxide refrigerants are readily available for commercial use. It is likely that HC- and hydrofluoroolefin- (HFO)-based units will become more available in future.

9.6 Multi-residential heated-water systems

Multi-residential buildings suit a centralised heated-water system that distributes hot water to each residential unit. This system takes advantage of the diverse hot-water demand from multiple residences, which makes the HTHP system more cost effective than would be possible otherwise.

In a multi-tenant residential development, a centralised potable-hot-water system has significant benefits over individual water-heating systems:

(a) It is easier and more convenient to maintain a centralised plant;

(b) It increases the amount of useable space in each unit;

(c) It reduces the risk of water leaks and equipment failure in each unit;

(d) The operating costs are significantly lower than those of individual systems; and

(e) The capital costs are significantly lower than installing an individual HTHP in each unit.

This system suits a centralised billing system with remote monitoring. The hot-water usage can be determined by installing a water meter in each unit. The electricity usage of the ring-main reheat is metered separately.

The advantages of single-pass HTHPs also apply to these systems. The quicker heat recovery means that less water-storage volume is needed to meet peak demand.

It is important that reticulated systems are carefully designed to:

(f) Balance hot water delivery times to the outlets, with thermal losses through the distribution system;

(g) Maintain the thermal stratification of the storage tank;

(h) Avoid noise from the pipework being audible within the occupied spaces;

(i) Maintain compliance with Legionella control requirements.

The key disadvantage of a centralised heated water system is the removal of individual choice as to service provider.

9.7 Performance assessment

There are two main ways to test the performance of HTHPs for potable-water-heating applications.

(a) **Tapping load test**: This method, described in AS/NZS 5125.1, EN 16147, and ISO 19967-1, tests an integrated HTHP and storage tank. In the test, a water heater is installed to operate in stated environmental conditions, and hot water is drawn from the tank on a fixed schedule. The increased energy content in the drawn-off water is compared with the energy used to produce it. This provides a COP for the test period. The tapping load test includes the interaction between the HTHP and storage tank in the results; and

(b) **AS/NZS 5125.1 test method**: This method, described in AS/NZS 5125.1, measures the COP and capacity of an HTHP when it heats a mixed tank under different test conditions to determine coefficients for a performance curve using a quasi-steady-state methodology. (Each test at a constant ambient condition occurs over a range of heat pump water inlet conditions as the mixed tank heats slowly, such that performance at a number of water temperatures can be determined from measurements for quasi-steady-state periods in the trial where the change in water inlet temperature is relatively small [less than 5.5 K].) The coefficients from the performance curve are fed into a thermal-simulation software package (such as TRNSYS) to provide SCOP estimates.

It is inappropriate to use space-heating HTHP standards (such as ISO 19967-2 or EN 14511-1) because they do not take into account how the tank interacts with the unit's performance.

Currently, few commercial HTHPs have been specifically tested for potable-water-heating applications, as their capacity exceeds the small-scale technology certificate requirements in Australia. There are also no test facilities commercially available in Australasia that have the capacity to test these units. Units sold in Europe may have been tested using the tapping load methodologies detailed in EN 16147 or ISO19967-1.

The preferred alternative approach to the tapping load methodology is to use a modified version of AS/NZS 5125.1 and AS/NZS 4234, as outlined in Appendix B.

9.8 Determining demand

Potable-hot-water systems shall be tested as described in AS/NZS 5125.1 and AS/NZS 4234, which choose a water-draw profile that is a close match to the actual draw pattern. Commercial buildings are likely to have a higher draw flow and different timings than domestic residences. The heated-water demand from a centralised heated-water system for a multi-residential building is typically flatter than normal, due to the diversity of water usage by residents.

When using published hot-water-demand tables, take care that the demand is not overstated. This can happen when the published demand is intended for calculating system size, rather than representing likely actual demand. The outlet type and the installation's water pressures also need to be considered.

10 APPLICATION: SPACE HEATING

10.1 Commercial space-heating systems

Many commercial space-heating systems use a boiler to provide heating via hydronic circuits in the building. These systems are generally configured as a primary-secondary hydronic system. Piping networks distribute the heated water around the building and radiators, fan coils, air-handling units (AHU), or underfloor heating pipes deliver heating into the building's spaces.

Some systems, particularly recent installations, are designed for condensing boilers to operate at moderate temperatures. However, most are designed to operate at the conventional-boiler return-water temperature of approximately 70°C. Non-condensing boilers require this return-water temperature to prevent corrosion caused by exhaust gases condensing.

In general, air-source HTHPs cannot match the boiler's existing return-water temperature, unless they are specifically designed to do so. To successfully apply an HTHP to these applications, the system as a whole should be considered.

10.2 Refrigerants

Commercial air-source space-heating systems generally require the HTHP to deliver the required water temperature during the coldest time of the year and remain efficient, despite relatively high return-water temperatures. Refrigerants that allow higher supply-water temperatures are less effective during low ambient conditions. Currently, the most appropriate refrigerants are R32 and similar blends. R32 supports a supply-water temperature of 55°C in typical New Zealand conditions.

Commercial propane HTHPs are likely to become more readily available. They provide a supply-water temperature between 65°C and 70°C. Carbon dioxide HTHPs are generally not suited to space-heating applications unless the return-water temperature can be controlled to stay below roughly 35°C.

10.3 Temperature

10.3.1 Temperature of the heat source and heat sink, and the return-water temperature

The heat source with the warmest possible temperature should be used.

The return-water temperature (or the temperature of an alternative heat-transfer fluid, such as glycol) back to the heat pump is an important design consideration for many HPWHs. A heat pump is generally more efficient when the temperature differential (the difference between the supply and return temperatures) across the heat-exchanger is greater. This is especially the case for carbon dioxide heat pumps, which require a relatively cool return-water temperature to maintain high heating capacity and high efficiency.

The temperature of some heat sources (such as ambient air) varies according to the season and time of day. Therefore, the energy efficiency of an installed HTHP may vary, according to the conditions. However, a properly specified and configured HTHP should provide heating at the desired temperature even when there are changes to the heat-source temperature.

For some commercial and industrial space-heating situations, other heat sources (such as wastewater, geothermal activity, or other processes) may be available. Typically, these sources may be warmer than ambient air and vary less between seasons. Using these sources as the heat source may provide higher annual performance (SCOP) than using ambient air.

10.4 Operational considerations

HTHPs that use ambient air as the heat source for space heating will be most heavily constrained on cold days when the heating demand is also high. Overall, the system should be designed to perform in these worst-case conditions. However, rather than sizing the HTHP for these conditions (this would result in an oversized heat pump for most of the time), it can sometimes be more cost effective to use a smaller HTHP with an auxiliary heating system (such as an electric resistance heater) or thermal storage (that is, a hot-water tank). This solution means using the HTHP alone on most days and supplementing it with the auxiliary heating system on extreme-temperature days.

10.5 Pipe size restrictions

When replacing a central-heating plant, the sizes of the existing secondary-system pipes may constrain the design options for a heat pump. Increasing the temperature differential across the secondary system may be a way to re-use existing pipework (this reduces the circulation flow rates). Another alternative would be to use a variable-flow-rate system design. This approach increases the flow and pump power during high heating-demand periods but reduces the pump power for the rest of the time.

10.6 Whole-of-life analysis

A whole-of-life analysis shall be completed to compare different HTHP options and configurations with alternative systems. At a minimum, this should include analysing and comparing direct and indirect emissions, capital costs, energy use, maintenance costs, and end-of-life costs (see Appendix D). Ideally, it also considers the environmental impact of different options and configurations.

10.7 Performance standards

For space-heating applications, the HTHP unit's SCOP shall be greater than 3.2 for all climate zones. For heated-water applications, the SCOP shall be estimated by modelling, using AS/NZS 4234 or a similar methodology.

10.8 Temperature differential

The system modelled using AS/NZS 4234 shall achieve an appropriate temperature differential across the HTHP, when it is operating. The system design should maximise the opportunities for refrigerant sub-cooling.

10.9 Thermal storage

When thermal storage is used, it shall be thermally stratified to achieve an appropriate temperature differential across the HTHP, when it is operating.

The thermal storage shall have sufficient capacity to meet the heating demand for the application for at least 30 minutes in the 5% worst weather conditions experienced at the location, and at the times when the heating is most likely to be used.

When thermally stratified storage is integrated into a heating system, the storage tanks can be distributed or centralised. All thermal storage that operates at more than 50°C shall be insulated with an insulation R value of at least 1.67 $m^2 K^{-1} W^{-1}$ (equivalent to 50 mm of high-density polyurethane foam)[2] for tanks whose capacity is less than 5000 L, or an R value of at least 3.33 $m^2 K^{-1} W^{-1}$ (equivalent to 100 mm of high-density polyurethane foam) for tanks whose capacity is 5000 L or more.

10.10 Fluid distribution

If present, the fluid-distribution system shall be a variable-flow distribution system. Where possible, all fluid (such as water) that returns to the HTHP should pass through a heat-exchanger (such as a radiator or fan coil) before it returns to be reheated.

10.11 Minimum temperature for space heating

When the application is space heating for occupancy, the room temperature in the 5% worst weather conditions experienced at the location shall be at least 20°C within 15 minutes of being occupied.

10.12 Pump efficiency

When pumps are needed with a power-consumption rate of more than 200 W at maximum capacity, their motor efficiency shall be more than 70%. EC motors are an example technology to achieve this motor efficiency. This requirement also applies to pumps replaced as part of a retrofit or upgrade project. Pumps whose power-consumption rate is more than 200 W shall operate under constant pressure control, when centralised capacity control is not enabled. When it is practical to do so, temperature compensation shall be applied in mild heating conditions.

10.13 Fluid-distribution pipe insulation

All fluid-distribution pipes shall be insulated to the requirements of H1 of the NZBC, where applicable. Where H1 of the NZBC does not apply, then a minimum of 30 mm of closed-cell, foil-faced (or equivalent) pipe insulation shall be used. Pipe insulation that will be degraded by sunlight or weathering shall not be used in exterior locations.

[2] This assumes the polyurethane foam has thermal conductivity of 0.03 $W m^{-1} K^{-1}$.

10.14 Retrofit constraints

When an HTHP is retrofitted to an existing system (such as radiators), the existing system might not have sufficient capacity to maintain room temperature during low ambient conditions at the lower water temperature that the HTHP provides. This problem can be overcome with additional air-heating heat-exchangers in the space (such as a water-to-air fan-coil unit). The preferred energy source for these extra heat-exchangers is the return water line back to the HTHP or storage tank in the water-heating system (see Figure 9). These extra heat-exchangers shall be controlled to operate only when required.

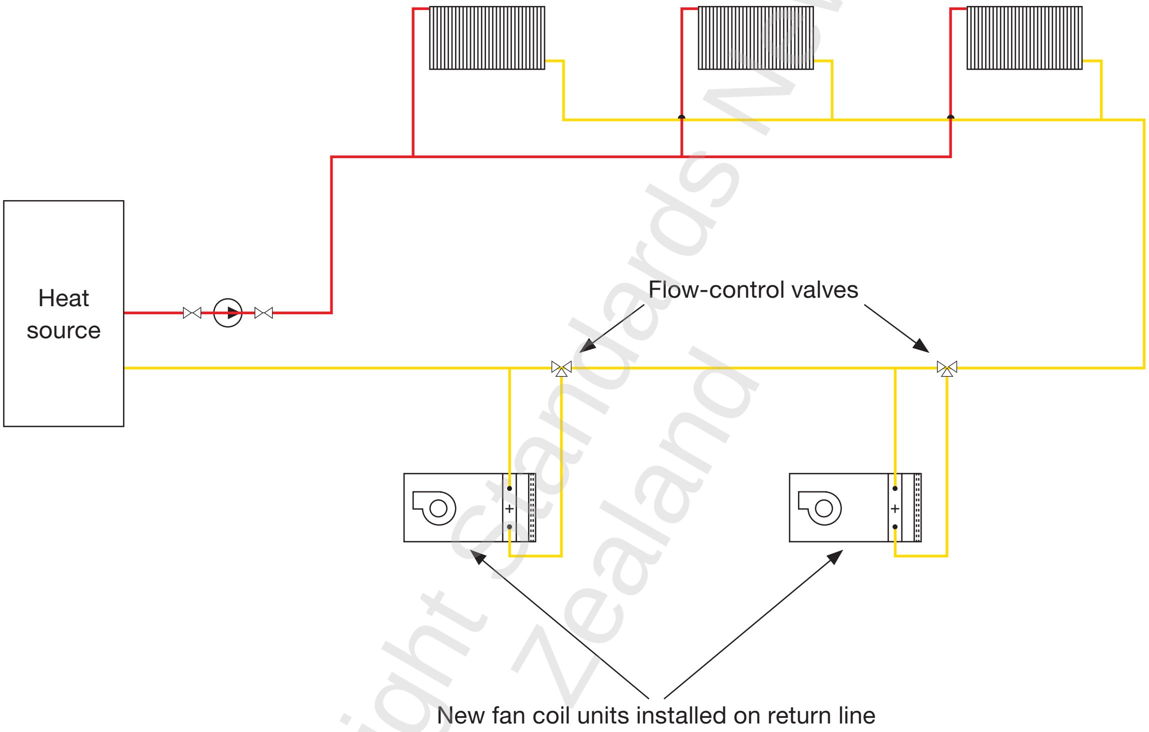

Figure 9 – Schematic showing use of additional heat-exchangers for space-heating

10.15 Provision of cooling

When an HTHP removes heat from the heat source (to deliver it to the heat sink), cooling happens at the heat source. This can be used by the heat pump to provide cooling. In general, providing cooling increases an HTHP's capital cost. It may also reduce the energy efficiency of its heating if the heat source is operated at a colder temperature to provide cooling. However, it removes the need for a separate refrigeration system.

When the cooling and heating demands are very different, the choice of HTHP should be based on delivering the lower demand. The remaining demand should be met by a separate HTHP, for heating, or a refrigeration system, for cooling.

Cold pipework should be insulated. If cooling occurs at temperatures below the dew point, condensation on cold pipework and cold parts of the heat pump should be prevented from causing damage.

Three main cases occur that involve different relative timings of the cooling and heating applications:

(a) When **cooling and heating occur simultaneously**, the fluid cooled in the heat source is simply the cooling application. Alternatively, a cold secondary fluid (such as chilled water) can be pumped to the cooling application;

(b) When **cooling and heating occur within hours of each other** (for example, in alternating shifts), thermal storage should be provided on the heat-source side, or heat-sink side, or both. This allows cooling produced during the 'heating shift' to be stored for use during the 'cooling shift', and vice versa. If cold storage (such as a chilled water tank) is used, it should be stratified; and

(c) When **cooling and heating occur in different seasons** (for example, summer and winter), in the cooling season the heat-source side of the heat pump can be configured to cool a fluid, and the heat-sink side can be configured to dump the rejected heat into ambient air or another heat sink. This is achieved by configuring the heat pump so that the evaporator and condenser heat-exchangers switch roles (that is, the heat-source heat-exchanger is switched to the heat-sink heat-exchanger, and vice versa), or by configuring the external plumbing so that different fluids are sent to the heat pump during different seasons.

When heating and cooling occur simultaneously, or when thermal storage allows heating or cooling to be moved between time periods, an effective combined COP ($COP_{combined}$) can be calculated using Equation 1.

$$COP_{combined} = \frac{Q_{heating}+Q_{cooling}}{P} \quad \text{.........(Eq. 1)}$$

10.16 Scope and benefits of advanced controls

10.16.1 Considering the installation as a complete system

It is often possible to significantly improve the efficiency and reliability of an HPWH by considering the installation as a complete system instead of a standalone heat source. When the temperature of water supplied by the heat pump is lower than that of a previous space-heating system, converting the distribution system from a primary-secondary system to one with a variable flow rate that matches the heating demand can more effectively use the HTHP's lower water temperature. It may also be necessary to install additional heat-exchangers to meet the required room temperatures.

10.16.2 Capacity control

Variable-capacity installations allow an HTHP's heat output to be matched to demand. The heat pump's capacity is adjusted by changing compressor motor speed (this is usually done via an inverter) or adjusting the number of compressors operating (this applies to installations with several compressors).

During part-load operation, speed-control systems are more efficient than most other systems. Other ways of controlling the capacity of fixed-speed compressor systems include digital scroll compressors, hot-gas bypass systems, and slide valves on screw compressors. These methods are effective at controlling capacity but are less efficient than speed-control systems.

Instead of variable-speed control, capacity can be controlled by limiting the maximum compressor speed or reducing the water flow through a variable-flow heating system.

Unlike boiler-based systems, the efficiency and operating temperatures of an HTHP are affected by the return-water temperature into the unit. In general, the lower the temperature of the heat sink entering the heat-pump-to-heat-sink heat-exchanger, the lower the operating pressure of the unit and the more efficient it will be. Reduced operating pressure also results in the refrigeration system being less stressed and more reliable. In periods of low heating demand, the return-water temperature can be warmer. This reduces efficiency, so the system should be designed to avoid this.

Using buffer tanks to prevent compressor short-cycling and provide a source of energy for defrost cycles should be carefully considered. In highly dynamic installations, a buffer tank can be installed on the return heat-sink fluid loop to the heat pump in order to smooth temperature changes and improve control.

When thermal storage needs to be included in the system, consider the hot-water temperature when the system is designed. Thermal stratification (see 5.2) is effective and recommended; however, other configurations may be considered. Under part-load conditions, the thermal store should not result in small temperature differentials across the HTHP.

10.16.3 Pump control

In conventional-boiler installations, any excess water flow through the heating system, beyond what is needed to meet the heating demand, is diverted past the heat-exchanger and back to the boiler (this is often called a ‘bypass’). When an HTHP is retrofitted to an existing system that was operated at a higher circulation temperature than the new system will be, the system shall be converted to a variable-flow distribution system.

A variable-flow distribution system is designed to deliver the required amount of heated water to the heating outlets to meet demand. All water circulating within the distribution system is passed through a heat-exchanger (such as a radiator, fan coil, or underfloor heating circuit) before it returns to the HTHP or thermal-storage tank.

Diverting valves in a fixed-flow-distribution system should be set in the fully open position (that is, no diversion), and the flow rate should be controlled with variable-speed pumps. Alternatively, when existing fixed-speed pumps are retained, variable-speed drives (VSDs) can be installed to control the water flow through the system. Pump speeds will either be centrally controlled to meet the building’s heating and cooling demand or controlled to a constant pressure. When flow-control valves are deployed, constant pressure control allows the pump speed to be varied according to the total amount of water flow delivered to the heating or cooling outlets. Flow-control valves can be binary on/off valves (such as those on a radiator) or variable-flow valves (such as a motorised ball valve).

10.17 Estimation of required heating capacity

It is important to estimate the required heating demand accurately so that the system that is designed and installed is fit for purpose. Heating-demand requirements should be calculated by an independent professional (this is not usually the installation contractor). Online calculators are available for space heating.

Consider these parameters to get an accurate estimate of the heating demand for space heating:

(a) The size and number of spaces to be heated. Note the floor areas and room volumes;

(b) The construction of the floors, walls, and ceilings, particularly the overall insulation values;

(c) The area of windows for each space. Note whether the windows are double or triple glazed, what type of frame material is used, and how they are constructed (this includes any thermal breaks), as these features affect the windows' R values; and

(d) The estimated air infiltration or leakage for each room, or ventilation quantity ($m^3\ s^{-1}$) for each space if the space has a separate ventilation system.

For a retrofit application, existing monitoring data of the heating demand may be used, if the data are reliable. Monitoring equipment may be installed before any proposed new HTHP is considered. So that distribution heat losses can be estimated, monitoring of an existing installation should separate the energy supplied from the heating unit from the energy supplied to occupied spaces or applications. When a proposed new installation includes changing the distribution system, the effects of these changes should also be measured by the monitoring data collected.

10.18 Compliance and standards

Regulators are currently assessing HTHP units for inclusion in the New Zealand Government minimum energy performance standards (MEPS) regulatory framework. The MEPS regulatory framework has been used for 20 technologies since the early 2000s and has saved New Zealand consumers more than $1.3 billion in avoided electricity costs.

Although HTHP technology is acknowledged as being very energy efficient, including it in the MEPS regulatory framework will set a minimum efficiency 'floor' in the market. This will ensure the technology meets the claimed performance levels, giving consumers confidence that their investment will perform to an acceptable level.

MEPS are typically based on local, trans-Tasman, or international standards. When technology has been tested to the standard approved by the national body (Standards New Zealand), it can be relied on to perform as claimed.

The Energy Efficiency and Conservation Authority operates a comprehensive programme that checks technology meets the minimum requirements in the MEPS regulatory framework.

10.19 Seasonal coefficient of performance

A heat pump's COP depends on the operating conditions. Therefore, the COP can be quite different at different times of the day or in different seasons. Different operating conditions affect the heating demand, the heat pump's heating capacity, and the heat pump's power requirement.

The SCOP measures the annual average COP. It is weighted by heating capacity, heating requirements, and operating hours. When calculating SCOP, make sure consistent units are used for the quantities in the SCOP equation, and that they come from the same period.

To size the capacity of an HTHP's design, the COP and thermodynamic efficiency relative to the Carnot cycle should be evaluated for the application's typical operating conditions and for more extreme conditions.

Typical operating conditions should be used to evaluate the thermodynamic efficiency relative to Carnot (see C1). If the typical heating requirement is less than the installed capacity, the COP should account for performance at a part load, or the effect of other control strategies (such as thermal storage).

10.20 Performance monitoring

Ongoing performance monitoring of a system identifies and diagnoses any deviation from its expected performance. It also provides empirical evidence to evaluate the site's, organisation's, or nation's progress towards goals such as decarbonisation.

The amount of performance monitoring that is stored and accessible to the system's owner should be sufficient to make meaningful performance comparisons between years, seasons, and sites. These performance comparisons shall include the system's power consumption and heat-energy production at least once every 5 minutes (time-averaged data).

Depending on the application, other variables should be recorded to characterise the system's performance. These variables may include the supply and return temperatures and delivered flow rate for a hot-water-heating application.

The connectivity system should have the capability for remote monitoring (see 8.3).

10.21 Case study: George Manning Lifecare

George Manning Lifecare and Village is an 89-bed facility in Christchurch that provides hospital care. Its domestic hot-water system comprised an 8000 L thermal store heated to 80°C, using 100 kW of resistance heating controlled to night-rate electricity tariffs. The original ring-main had a 'push pump' on the original reticulated loop. A newer wing of the facility had a separate reticulation circuit with a circulator on the ring-main return. The facility experienced an erratic hot-water supply to the outlets, which was resolved by replacing the 'push pump' with a circulator pump on the reticulation return pipe of that circuit.

The facility's space heating was provided by a 400-kW diesel boiler in a conventional primary-secondary heating system. Throughout the building it had wall-mounted radiators operating on two circuits. It also had fan coils in the ceiling that were connected to a third heating circuit, which provided tempered fresh air. The primary circuit operated at 70°C to 80°C throughout the year. In the previous 12 months of operation, the site had consumed $168,000 of diesel. In the previous 2 years, the boiler had incurred service and maintenance costs of $80,000.

George Manning Lifecare undertook a decarbonisation project, which involved two stages. In the first stage, two 1000 L thermally stratified potable hot-water cylinders were installed. A 50-kW single-pass HTHP was installed to provide the hot water. The reticulation-circuit return lines were joined together and reheated outside the main storage tanks. This meant the thermal store could be decommissioned, which freed up 85 kW of electrical capacity. A 30 $kW_{(p)}$ solar PV system was also installed that generated 31.3 MWh of electricity in the first year.

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In the second stage, four 90-kW variable-flow HTHPs were installed, controlled by a multi-compressor load-management controller. The heating-distribution system was converted to a variable-flow system that operates at 55°C. Variable-speed EC pumps were installed to replace the existing fixed-speed pumps, and all bypass paths were closed. Upgraded fan coils were installed in the existing ventilation-system ductwork.

The In-Control remote-monitoring system allows the system to be tuned in-situ and can determine the system's real-time energy performance.

In the 12 months since the project was completed, the electricity savings from removing the thermal store and reconfiguring the domestic hot-water system have fully offset the cost of additional electricity usage from the space-heating system, with a net reduction in electricity use of 55 MWh. The project has reduced the site's annual energy costs by more than $186,000. Excluding the maintenance costs of the diesel boiler, the payback period is 2.9 years. Reducing the heating water temperature from 80°C within the primary circuit, to 55°C as a variable-flow system, has had no noticeable effect on the site's indoor temperature.

11 APPLICATION: POOL HEATING (COMMERCIAL)

Commercial pools typically contain multiple pools within the complex, each heated to a different temperature, and of different sizes. The water within each pool is kept separate from the other pools, requiring separated plantroom equipment. The pool water is usually chlorinated, which is corrosive to some metals typically used for heat-exchangers.

Where previously heated via a boiler, the heating system typically incorporates a primary circuit supplying heat to each of the individual pool heat-exchangers. Water from each pool is pumped through the secondary side of each heat-exchanger. Control of the individual pool temperatures is achieved by controlling the primary circuit flow rate through each individual heat-exchanger.

Indoor pools also require that the indoor temperature and humidity are controlled to prevent condensation occurring within the building structure by high levels of fresh-air ventilation. An Air Handling Unit (AHU) with heat recovery preheats the incoming air using the energy contained within the exhaust air. A common alternative approach is the run-around system, which uses a water loop to extract heat from the exhaust air to preheat the incoming fresh air. Both AHU and run-around systems require a heating coil to supplement the energy recovered to maintain the required indoor temperatures. Smaller pools may use supplementary resistance heating. Most larger pools use a heating coil connected to a water heater to supply this energy.

HTHPs are ideally suited to replacing a resistance or fossil-fuel heating system for commercial pools as the water temperatures required are well within the range for efficient operation. There are two main approaches.

(a) Direct heating. This approach directly heats the pool water within the heat-pump unit. Each pool requires a separate heat-pump water heater. A titanium-based heat-exchanger is used within the heat-pump unit (noting that due to the temperatures, this is not usually a HTHP).

(b) Indirect heating. This approach retains the primary circuit, and heats the water via the individual heat-exchangers external to the unit. The separation of the refrigeration system from the chlorinated water from the pool decreases the risk of contamination of the refrigerant circuit with water (noting that this may be an HTHP depending on the heat-sink temperature).

Traditionally, the direct heating approach provides the lowest operating cost, but requires a higher installed capacity as each heat-pump unit can only heat its associated pool. A separate unit is also required for providing heating to the AHU or heating coil.

A more recent development of variable-capacity multi-compressor HTHPs overcomes the efficiency limitation of the indirect approach by controlling the flow rate through each heat-exchanger while eliminating the bypass path. A higher efficiency is achieved by lowering the return-water temperature to the HTHP, particularly during part-load operation. This approach also has the advantage of greater redundancy and diversity, allowing a significant reduction in total installed heat-pump capacity and electrical capacity.

12 APPLICATION: FRESH-AIR TEMPERING

Fresh-air tempering systems are well suited for conversion to HTHPs. It is important to ensure that there is sufficient energy available to complete a coil defrost cycle during low ambient conditions.

13 INDUSTRIAL PROCESS HEAT

13.1 Process heating

Process heating refers to heating uses on an industrial manufacturing site, including food-industry sites. Process heating often involves heating utility fluids (such as hot water, steam, or hot air) that are used as a secondary fluid to transfer heat to an application. However, it can also involve directly heating a process fluid.

Understanding the heating-demand load profile is critical to designing the right size of HTHP system. Compared with the heating demand on residential or commercial systems, the process-heat demand on industrial sites is often larger in scale, and involves higher temperatures and significantly longer run hours. Industrial process-heating demands are influenced by the process's production levels, rather than the ambient temperature. As well as being large scale, process-heating loads can be relatively steady across an operating shift(s).

HTHPs for industrial use are typically much larger than commercial systems, and usually integrate the heating demand across several processes. Consequently, industrial HTHPs often use a bespoke design rather than pre-configured equipment. The process of designing an HTHP for industrial use should be overseen by a qualified process engineer.

13.2 Space heating

Industrial settings sometimes require space heating. This can be delivered by a separate HTHP system, following the guidelines for commercial space heating (see 10.1). Alternatively, industrial space heating can be treated as another heat load for the application and delivered by the HTHP system that provides process heating. The choice depends on three factors:

(a) **Coincidence**: Are space heating and process heating required at the same time;

(b) **Locality**: Are space heating and process heating required in the same area of the manufacturing site, or will a secondary fluid be pumped a long distance; and

(c) **Temperature requirements**: Are the heat-sink temperatures for the space heating and process heating similar?

13.3 Heat recovery

HTHPs are generally more expensive than heat-exchangers, so heat recovery from heat-exchangers should be maximised before HTHPs are considered. At complex industrial sites, this usually involves conducting a pinch analysis (process integration) of heating and cooling needs. The HTHP is designed only to provide residual heating, and possibly meet cooling needs, after economic heat-exchangers are maximised. For a building, this may simply involve recovering heat from exhaust air or wastewater streams, to preheat incoming cold air or water.

13.4 Technologies available

13.4.1 Ammonia heat pumps

Ammonia heat pumps can operate up to heat sink temperatures of about 95°C, using equipment rated to 60 bar. They are commonly configured as a high-stage system, using the heat rejected from an ammonia refrigeration plant (this heat is about 30°C) as their heat source.

The heat pump can be fully integrated with the refrigeration system, to avoid needing a heat-source heat-exchanger (that is, to use a multi-stage system). However, this approach can make it difficult to find lubricants that can operate across the full temperature of the refrigeration system and the heat pump.

An alternative option is having the heat pump as a separate system that is cascaded to the refrigeration system by a cascade heat-exchanger (that is, to use a cascade system). This avoids the problem of finding lubricants. However, it makes the heat pump less efficient, due to lower evaporation temperatures for the same refrigeration heat-rejection temperatures. Ammonia heat pumps can be used with the cascade approach, even if the refrigeration system does not use ammonia as the refrigerant.

Ammonia heat pumps can be configured to provide top-up heating (such as for a hot-water ring-main) or to heat the heat sink across a wide temperature range.

13.4.2 Carbon dioxide heat pumps

Carbon dioxide heat pumps commonly provide heating up to about 95°C, but can provide heating up to about 120°C. They use equipment rated to 150 bar. Carbon dioxide heat pumps operate in a transcritical cycle and can simultaneously provide refrigeration at temperatures as low as –15°C (that is, the heat source can be an application that needs cooling by refrigeration).

Due to the transcritical cycle, to be efficient, carbon dioxide heat pumps should preferably be used to heat the heat sink across a large temperature difference (such as heating cold water to hot water in a single pass). The heating COP may not be as high as it is for subcritical heat-pump systems. However, if the refrigeration benefits are used, the combined COP will often exceed that of separate refrigeration and heat-pump stages or systems.

13.4.3 Mechanical vapour recompression systems

Mechanical vapour recompression (MVR) systems are subcritical heat-pump systems that use water as the refrigerant. In New Zealand, they are most used for milk evaporators in an open cycle. In this application, the heat source is water evaporating from milk at 60°C to 65°C. The MVR produces steam at 70°C to 76°C, to heat the evaporator. Given the open cycle and small temperature lifts, an MVR COP can be higher than 20.

The main disadvantages of MVRs are that they operate at vacuum pressures (water evaporates and condenses at less than 100°C), and the compressors have a very large swept volume capacity (turbo-fan compressors are commonly used). MVRs can operate at much higher temperatures and pressures, but systems with this capability have only recently become commercially available. A multi-stage system with intercooling is probably needed to get high temperature lifts. This is because of water's refrigerant characteristics, as well as the availability of compressor technologies that are cost-effective to use with water as the refrigerant.

An MVR can be designed as a closed system, where the refrigerant – water – indirectly heats another fluid. However, for steam-generating heat pumps, open-cycle, water heat-pump systems are more likely to be used. In these systems, the discharge from the compressor is used directly as the heat sink.

13.4.4 Technology direction

Heat pumps for temperatures above 100°C are starting to emerge, but they are not yet common commercially. Achieving these high temperatures with standard refrigeration and heat-pump equipment is likely to need different refrigerants to those used for temperatures below 100°C. At higher temperatures, refrigerants have to operate at very high pressures. Therefore, lower pressure refrigerants are likely to be used to achieve high temperatures. These refrigerants include water (R718), butane (R600), iso-butane (R600a), hydrochlorofluoroolefins (HCFOs) (R1224yd(E) and R1233zd(E)), and HFOs (R1336mzz(E) and R1336mzz(Z)).

For large-scale heating, using several smaller units in parallel could be more feasible than using one or two large-scale units. This approach can take advantage of lower-cost equipment, reduce the refrigerant charge per unit (for example, if A1 refrigerants cannot be used), and capacity control can be simply and effectively provided by only running the number of units required.

13.5 Refrigerants

13.5.1 General

Refrigerant is a critical element of an industrial heat pump. A refrigerant is referred to by an alphanumeric code (for example, R32), which indicates its chemical composition. Appendix A contains a list of some potential refrigerants, but this is not an exhaustive list. Talk to your supplier about which refrigerant is appropriate for your specific requirements.

13.5.2 Environmental and safety issues

Many current refrigerants are hydrofluorocarbons (HFCs). HFCs are potent greenhouse gases (GHGs) that contribute to climate change if released to the atmosphere. It is anticipated that these refrigerants will become less readily available owing to international efforts to reduce their impact on the climate.

Some refrigerants are also hazardous owing to their toxicity or flammability. The refrigerant installed in a heat pump shall be appropriately labelled, with correct safety measures applied for the specific hazard level. See AS/NZS ISO 817:2016 for definitions of safety levels.

It is generally not advised to charge an HTHP with a refrigerant different from the one it was designed to use. Refrigerants of different safety classifications require different engineering controls, so charging an HTHP with a refrigerant that has a different safety class could be hazardous to people or property. Consult an appropriately qualified technician before doing this.

'Charge minimisation' means designing a heat-pump system to use as little refrigerant as possible. This practice can be cost efficient and can mitigate some of the risks associated with using hazardous refrigerants.

When consumers select a refrigerant for their application, it is recommended to select the refrigerant with the lowest GWP practicable (at least GWP $<$ 750), given cost and efficiencies. If they use a refrigerant with hazardous qualities, they shall implement all the appropriate safety measures.

13.5.3 Refrigerant leaks

In a sealed HTHP unit, the refrigerant is designed to remain within the system. However, refrigerant leaks will occur if pipes or other components fracture, due to corrosion or metal fatigue from vibration.

Consumers should ensure they service their HTHP unit as specified by the manufacturer, and check that the service includes checking for potential refrigerant leaks. They should also carefully observe any change in the HTHP's delivery temperature or power consumption, as such changes may indicate a refrigerant leak.

13.6 Temperature

13.6.1 Temperature of the heat source and heat sink, and the return temperature

HTHPs operate most efficiently when the temperature lift between the heat sink and the heat source is minimised. Where possible, the system should deliver the coolest possible heat-sink temperature while still meeting the heating requirement. For example, if 60°C water satisfies the heating requirement, a heat pump that delivers 60°C water is more efficient than one that delivers 70°C water. Similarly, the system should use the heat source with the warmest possible temperature (for example, warm exhaust air from a dryer rather than colder ambient air).

The return heat-sink temperature back to the heat pump is also an important design consideration for many HPWH. In general, a heat pump is more efficient when the heat-sink temperature increase (that is, the water temperature increase across the heat-exchanger) is greater. This is especially the case for carbon dioxide heat pumps, which require a relatively cool return heat-sink temperature to maintain high heating capacity and high efficiency.

The following strategy can be used to minimise the temperature lift while also maximising the temperature increase across the heating side of the heat pump:

(a) Determine the warmest heat source available. While this is often the ambient air, if waste heat sources are available, choose the warmest;

(b) Investigate the heating requirement at the application and select the coldest temperature that will meet this requirement (this is the heat sink). Together, steps (a) and (b) minimise the temperature lift; and

(c) Configure the flow of the fluid to be heated on the HTHP's heat-sink side so that it is heated to the required temperature – determined in step (b) – from the coldest possible starting temperature without wasting the heat stored in reticulated systems. For reticulated hot-water systems, thermal storage with stratification achieves this; the configuration is such that the coldest water in the thermal storage is sent to the heat pump to be heated to the required temperature for the application. Step (c) maximises the temperature increase across the heat-sink side of the heat pump.

Process-heating systems are most heavily constrained when the heat source is at its coldest, and the heating demand is at its greatest. Carefully consider the schedule of the application's heat loads to determine the scenarios when the system will be most constrained.

13.6.2 Temperature constraints

To achieve high energy efficiency, most HTHPs operate with the refrigerant condensing temperature well below the refrigerant's critical temperature. The critical temperature (see Table A1) is the upper temperature limit that most systems can provide at the heat sink.

Specially designed transcritical heat-pump systems can operate efficiently when the heat-sink temperature is at, or above, the critical point. These system configurations are common when carbon dioxide (R744) is the refrigerant, but less so when other refrigerants are used. Transcritical systems may become more common in future for HTHPs whose heat sinks have a very high or variable temperature.

The refrigerant boiling point at atmospheric pressure (see Table A1) constrains the temperature at the heat source. The refrigerant selected should have a lower boiling point, at atmospheric pressure, than the usual heat-source temperature. In most systems, the refrigerant boiling point sets the lower temperature constraint, based on what is practical for the heat source. Refrigerants with higher boiling points (these are sometimes called 'low-pressure refrigerants') are less dense as vapour at the heat-source temperature. They will, therefore, need compressors, heat-exchangers, and pipes that are physically larger than high-pressure refrigerants would need at the same temperatures. Requiring larger equipment typically increases the capital costs associated with low-pressure refrigerants.

13.7 Determining demand

The demand from a process-heating application is determined by considering these factors:

(a) The final (target) and initial temperature and heat load (kW) for each application; and

(b) The expected schedule of heat load for each application, including any variation during operating shifts and between seasons.

The total demand during each shift, season, or other relevant period is the sum of simultaneous heat loads that cannot be met by heat-recovery options (see 13.3). The HTHP system should be designed to meet the greatest demand.

Calculating the total demand should take account of thermal storage used to generate heat in low-usage periods for using later, which spreads the HTHP's operational period over a longer period than the heating demand. Therefore, when thermal storage is used, the total demand equals the average heating demand over the HTHP's operational period, rather than the peak demand. This is a feasible option when, for example, shifts with high heating demand are followed by shifts with low, or no, heating demand. This option can mean investing less capital in an HTHP to meet a larger demand.

If the demand during different shifts, seasons, or other relevant periods varies by more than 30%, consider installing multiple HTHPs, or one HTHP with multiple compressors that are fed by a common suction manifold. The size of the HTHP system should allow some compressors to be switched off during periods of low demand.

If some of the application's heat loads need a much higher heat-sink temperature (target temperature) than others, it may be more cost-effective to deliver the different temperatures with different levels of a multi-stage or cascade system. Alternatively, the high-temperature heat loads could be delivered by a separate HTHP.

When a site has simultaneous cooling *and* heating demands, consider using heat pumps to provide cooling where possible (see 10.15).

Heat-pump systems that use subcritical cycles are generally best suited to applications where the heat-source and heat-sink temperature changes are small as they pass through the heat pump. Examples of suitable applications include generating steam or topping up the heat of a ring-main.

Heat pumps that use transcritical cycles are best suited to applications where the heat-sink temperature change is large. Examples of suitable applications include heating potable hot water from cold water for sanitary purposes or heating air for process dryers.

A heat pump's heat source should be as hot as possible to increase the heat pump's capacity and efficiency.

13.8 Factors affecting performance and durability

Process-heating systems often have a required duty, or level of service, that means they consistently run for long hours at relatively high load. This level of service can be higher than, for example, some heating, ventilation, and air conditioning (HVAC) equipment required to run at near-peak output for only a few hours each year. Therefore, equipment for process-heating systems may need to be designed specifically for their level of service.

Industrial equipment may be designed with heavier-duty motors, bearings, and gaskets to ensure parts provide the expected level of service and avoid them failing prematurely. Discuss the system's design loads and expected run hours with equipment suppliers to ensure the equipment they recommend will be fit for purpose.

The design for a system with multiple units or a system that needs some redundancy could include the ability to remove one unit from service in order to repair or replace it without affecting the operating ability of the remaining units.

The choice of lubricant used with compressors is critical because of the high temperatures that HTHPs experience. To avoid the lubricant breaking down at the high temperatures, a different lubricant might need to be used with the compressor than would be used with a similar refrigeration compressor. Equipment suppliers are generally responsible for selecting the right oil types and designing the right oil-cooling system for the agreed level of service. Oil-free compressor options avoid the need for lubricants; however, they can be less efficient than equivalent oil-lubricated compressors. Discuss lubricant options with the equipment supplier before selecting equipment.

An HTHP's performance at an industrial site can also be affected by the temperature lift, inlet and outlet fluid temperatures, heat-exchanger (design and size), fluid (type, cleanliness, and fouling), fluid flow rates, and heat source (temperature consistency).

It is usually best to design an HTHP system where the HTHP can run continuously to meet the production heat loads and avoid it starting and stopping frequently.

13.9 Compliance and standards

Before installing a system, using performance-verification spreadsheet models and thermal-design tools (such as TRNSYS) can validate a proposed design and its likely energy savings. These tools are beneficial for commercial- and industrial-scale systems; they can optimise the installed capacity, configuration of thermal storage, and pump and electrical requirements.

13.10 Performance monitoring

Once a heat pump is commissioned, performance monitoring reviews and optimises its operations over a period, to establish its baseline performance. Ongoing performance monitoring of a system identifies and diagnoses any deviation from its expected performance. It also provides empirical evidence to evaluate the site's, organisation's, or nation's progress towards goals such as decarbonisation. Performance monitoring should include regularly reviewing how equipment is performing and whether relevant performance indicators are being met.

The amount of performance monitoring that is stored and accessible to the system's owner should be sufficient to make meaningful performance comparisons between years, seasons, and sites. These performance comparisons shall include the system's power consumption and heat-energy production at least once every 5 minutes (time-averaged data).

Depending on the application, other variables should be recorded to characterise the system's performance. These variables may include the supply and return temperatures and delivered flow rate for a hot-water-heating application.

Capability for remote monitoring should also be a component of the connectivity system (see 8.3.4).

13.11 Performance benchmarking

Process heating can be conducted in a wide range of operating conditions, so it is difficult to specify an absolute COP benchmark. When a high-temperature heat source is available, the COP can be very high. Conversely, when a large temperature lift is needed, the COP can be quite low. Therefore, the recommended performance benchmark is the Carnot cycle COP for the same heat-source and heat-sink temperatures. Appendix C explains how to calculate the Carnot cycle COP and thermodynamic efficiency relative to the Carnot cycle.

At all design and operating conditions, the recommended performance benchmark for the COP is a thermodynamic efficiency relative to the Carnot cycle of at least 0.5.

When the heat source for a process-heating application is not ambient, the COP might not vary much throughout the year.

When there are several HTHP configurations or technologies to choose from, and they all comply with the recommended performance benchmarks, comparing each option's thermodynamic efficiency relative to the Carnot cycle is a reasonable way to compare their energy efficiency. However, there may be other factors that will guide the choice, such as capital cost, maintenance cost and requirements, and refrigerant selection.

APPENDIX A – ADDITIONAL INFORMATION ON REFRIGERANTS

(Normative)

A1 Environmental issues

A1.1 General

There are several environmental issues associated with releasing refrigerants into the atmosphere.

A1.2 Ozone depletion

The previous generation of refrigerants consisted of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). These have been phased out, as they deplete the ozone layer. This phase-out was facilitated by the Montreal Protocol on Substances That Deplete the Ozone Layer, which is an international agreement that New Zealand is party to.

NOTE – The Montreal Protocol, finalised in 1987, is a global agreement to protect the stratospheric ozone layer by phasing out the production and consumption of ozone-depleting substances. See https://www.state.gov/key-topics-office-of-environmental-quality-and-transboundary-issues/ for more information.

A refrigerant's ability to deplete the ozone is represented by its ozone depletion potential (ODP). A refrigerant's ODP is measured relative to the ODP of R11. In general, newer refrigerants do not deplete the ozone.

A1.3 Climate change

Hydrofluorocarbons (HFCs) are the most common type of refrigerant currently used, but they are being phased out as they are potent greenhouse gases (GHGs). A refrigerant's ability to act as a GHG is measured by its global warming potential (GWP). GWP is measured relative to carbon dioxide in mass equivalents of carbon dioxide (CO_2eq). The Kigali Amendment (adopted in 2016) to the Montreal Protocol requires party nations to limit their consumption of certain HFCs. This means they are likely to become more expensive and less readily available.

A1.4 Environmental contamination

Many refrigerants, such as HFCs and hydrofluoroolefins (HFOs), can be classified as poly- or per-fluorinated alkyl substances (PFAS), or they degrade into PFAS. These substances are a concern because they are chemically stable and, therefore, persist in the environment.

A2 Refrigerant safety

When a refrigerant is being selected, some safety concerns need to be considered. Refrigerant safety classifications have two components:

(a) A letter component denotes whether the refrigerant is toxic (A = low or no toxicity; B = high toxicity); and

(b) A number component denotes how flammable the refrigerant is (1 = no flame propagation at standard pressure and 60°C; 2 = flammable; 3 = highly flammable).

Most common refrigerants are A1 (not flammable or toxic), but they generally have high GWP. Most refrigerants with low GWP are flammable to some degree.

The A2L class was introduced to identify refrigerants with lower toxicity and lower flammability than A2. Although their flammability is lower, they still require more safety controls than A1 refrigerants.

Hydrocarbon (HC) refrigerants are class A3, which means they are highly flammable and need more safety controls than A2L refrigerants. These controls include lower charge limits.

Some refrigerants (such as carbon dioxide refrigerants) also need a high-pressure system to operate. Pressure vessels for refrigerants shall comply with AS 1210 or an international code approved under AS 1210 (such as section VIII of the ASME's boiler and pressure vessel code; BS EN 13445-1; or PD 5500).

A3 Refrigerant types

A3.1 Hydrofluorocarbons

HFCs are the most common type of refrigerant currently used and generally have high GWPs. The Kigali Amendment to the Montreal Protocol requires party nations to limit their consumption of certain HFCs. This means they are likely to become more expensive and less readily available.

A3.2 Hydrocarbons

HCs are derived from common natural gases, like propane and butane. HCs have very low GWP. However, they are highly flammable, so they present a safety risk that shall be managed.

A3.3 Natural refrigerants

Natural refrigerants are a broad class that covers HCs, carbon dioxide, and ammonia. These refrigerants have very low GWP, but each has other safety hazards. HCs are highly flammable; carbon dioxide requires a high-pressure system; and ammonia is toxic.

A3.4 Hydrofluoroolefins

HFOs are emerging as a replacement for HFCs. In general, HFOs have very low GWP, but they are generally moderately flammable (A2L). HFOs are also suspected of indirectly contaminating the environment, as they can degrade into HFCs and polyfluorinated acids.

A3.5 Chlorofluorocarbons and hydrochlorofluorocarbons

CFCs and HCFCs are a previous generation of refrigerant. The Montreal Protocol has phased out their use because of their propensity to deplete the ozone layer.

A4 Refrigerant leaks

A4.1 General

In a sealed HTHP unit, the refrigerant stays in the system and is not consumed as the unit is used. Refrigerants may escape to the atmosphere if the system leaks or when the product reaches the end of its life.

A4.2 Leak minimisation

Refrigerant can leak if pipes, or other components, fracture or corrode. Follow these steps to minimise the risk of leaks:

(a) So that abnormal losses can be identified, investigated, and fixed, keep records of refrigerant levels and any top-ups;

(b) Regularly inspect the physical integrity of the system; and

(c) Check and top up refrigerant levels during ongoing maintenance.

A4.3 Leakage by handling

Follow these steps to ensure that refrigerants are not released into the atmosphere when a system is being built, maintained, or decommissioned:

(a) Ensure an appropriately qualified technician carefully handles the refrigerant, using industry-agreed practices, to minimise the risk of it being accidentally released into the atmosphere; and

(b) Ensure an appropriately qualified technician recovers the refrigerant when a product reaches the end of its life and that they give the refrigerant to a specialist organisation to be destroyed or reused.

A5 Charge minimisation

'Charge minimisation' is the practice of designing a heat-pump system to use as little refrigerant as possible while maintaining its efficiency. The practice also reduces the risks posed by hazardous refrigerants.

The refrigerant charge of two systems can be compared by the ratio of kilograms of refrigerant charge to kilowatts of heating provided ($kg_{refrigerant}/kW_{heating}$).

Another way to measure the environmental impact of the refrigerant charge for the HTHP unit is with the ratio of kilograms of CO_2eq of refrigerant charge to kilowatts of heating provided ($kg_{CO_2eq}/kW_{heating}$). 'Kilograms of CO_2eq' is found by multiplying the refrigerant's GWP by the unit's charge size of the unit.

With both measures, take care when comparing systems that use refrigerants with different safety requirements (see ASHRAE [the American Society of Heating, Refrigerating and Air-Conditioning Engineers] safety groups and Table A1).

Table A1 – HTHP refrigerant types as listed in AS/NZS ISO 817

Type	Refrigerant	Chemical formula	Tc[e] (°C)	Pc[f] (bar)	TBP[g] (°C)	ODP[h]	GWP[i]	SG[j]
HFC[a]	R32	CH_2F_2	78.1	57.8	–52	0	677	A2L
	R152a	CH_3CHF_2	113.3	45.2	–24	0	138	A2
	R227ea	CF_3CHFCF_3	101.8	29.3	–16	0	3350	A1
	R245fa	$CF_3CH_2CHF_2$	154.1	36.4	15	0	858	B1
	R134a	CH_2FCF_3	101.1	40.6	–26	0	1300	A1
	R410a	(50%) CH_2F_2 (50%) CHF_2CF_3	72.6	49.0	–51	0	1924	A1
HFO[b]	R1234ze(E)	$CF_3CH{=}CHF$	109.4	36.4	–19	0	<1	A2L
	R1234yf	$CF_3CF{=}CH_2$	94.7	33.8	–30	0	<1	A2L
	R1336mzz(E)	$CF_3CH{=}CHCF_3$	137.7	31.5	8	0	18	A1
	R1336mzz(Z)	$CF_3CH{=}CHCF_3$	171.3	29.0	33	0	9	A1
HCFO[c]	R1224yd(Z)	$CHCl{=}CFCF_3$	156.0	33.4	15	0.00012	4	A1
	R1233zd(E)	$CHCl{=}CHCF_3$	165.6	35.7	18	0.000134	5	A1
HC[d]	R600	$CH_3CH_2CH_2CH_3$ (butane)	152.0	38.0	0	0	4	A3
	R600a	$CH(CH_3)_2CH_3$ (iso-butane)	134.7	36.3	–12	0	3	A3
	R290	$CH_3CH_2CH_3$ (propane)	96.7	42.5	–42	0	3	A3
	R601	$CH_3(CH_2)CH$ (pentane)	196.6	33.6	36	0	5	A3
Natural	R718	H_2O (water)	373.9	220.6	100	0	0	A1
	R717	NH_3 (ammonia)	132.3	113.3	–33	0	0	B2L
	R744	CO_2 (carbon dioxide)	31.0	73.8	–78	0	1	A1

[a] HFC = hydrofluorocarbons

[b] HFO = hydrofluoroolefins

[c] HCFO = hydrochlorofluoroolefins

[d] HC = hydrocarbons

[e] T_c = critical temperature (°C)

[f] P_c = critical pressure (bar)

[g] T_{BP} = boiling point at standard atmospheric pressure (°C)

[h] ODP = ozone depletion potential (calculated on the basis that R11 = 1.0)

[i] GWP = global warming potential (calculated for a 100-year time horizon on the basis that CO_2 = 1.0, AR4 and AR5 data where available)

[j] SG = safety group (ASHRAE 34) (A = low/no toxicity; B = highly toxic; 1 = no flame propagation at standard pressure and 60°C; 2L = lower flammability; 2 = flammable; 3 = highly flammable)

APPENDIX B – MODIFICATION OF AS/NZS 5125.1 METHODOLOGY FOR HTHP GREATER THAN 15 KW

(Normative)

B1 Lab testing

The methodology described in AS/NZS 5125.1 for commercial air- and water-source HTHPs can be modified by increasing the size of the water tank in proportion to the unit's heating capacity, or by selecting water-source temperatures that match the application's conditions. The storage tank shall contain at least 50 L/kW of rated unit capacity and shall be insulated with a known heat-loss coefficient.

AS/NZS 5125.1 involves measuring, under a range of standardised test conditions, the power input and thermal energy produced by the unit when it is coupled to a storage tank.

AS/NZS 4234 shall be used to estimate the unit's annual energy consumption and SCOP. Once the performance coefficients have been determined, by analysing the test results from the methodology in AS/NZS 5125.1 or an equivalent methodology, they can be entered into the heat-pump model of a thermal-simulation software package using AS/NZS 4234. The unit's energy use can then be compared with a reference heating system (for example, a boiler). The difference between the GHG emissions of two or more systems can be calculated using the carbon-emission factor for the energy that the system consumes.

Variable-speed HTHPs shall be tested in accordance with AS/NZS 5125.1, or an equivalent methodology for non-air-source systems, using a locked compressor speed.

B2 Ambient environmental testing

Units whose rated capacity is more than 15 kW at ambient condition 7°C/6°C (DB/WB), and whose water-outlet temperature is more than 50°C, can adopt these modifications to AS/NZS 5125.1 Appendix A:

(a) Data collected from outdoor testing can be used, if no direct or indirect radiation on the unit occurs that raises the surface temperature more than 5°C above the ambient temperature. The surface temperature shall be measured with a calibrated infrared thermometer from a 1 m distance;

(b) The ambient condition shall remain within 1°C of the nominal dry-bulb (DB) temperature during the test;

(c) The maximum ambient test condition shall be within 3°C of the maximum modelled ambient test conditions for the location;

(d) The minimum ambient test condition shall be no more than 10°C;

(e) At least four test conditions are tested, including a test with a relative humidity difference greater than 25% at the warmest condition tested. There is at least a 5°C ambient temperature difference between each of the four test conditions;

(f) Immediately before and after the test, and during the test, the maximum wind speed at the unit shall be no more than 0.5 m/s;

(g) The tank may be unpressurised; and

(h) The minimum number of tank sensors shall be 10.

The requirements for measurement accuracy in AS/NZS 5125.1 clause A4 still apply. The low ambient test condition shall be applied to HTHPs, and the frost penalty shall be determined in accordance with AS/NZS 5125.1.

The low ambient test condition includes reducing the average heating capacity due to accumulated frost on the evaporator coils. A frosting penalty can be determined by simulating the system's energy performance under non-frosting conditions and comparing it with the system's actual performance once defrosting cycles are considered. It can be difficult for a test laboratory to maintain these low ambient test conditions throughout the test, especially with large-capacity units.

When the frosting penalty cannot be determined, a standardised penalty of 25% shall be applied to the lowest DB temperature that does not result in frost that persists for more than 10 minutes with an inlet water temperature no higher than 15°C for potable-water-heating systems or 30°C for space-heating systems.

Variable-speed products may apply the frosting penalty separately for different compressor speeds, since the lower the compressor speed, the lower the ambient condition can be for the suction line temperature to remain above –2°C. The frosting penalty shall be applied proportionally for intermediate speeds tested.

APPENDIX C – CALCULATION OF CARNOT COP

(Normative)

C1 Carnot COP

The Carnot coefficient of performance (COP_{Carnot}) is a theoretical limit to the COP that depends on the log mean temperature of the heat source ($\bar{T}_{\text{source}}$) (°C) and the log mean temperature of the heat sink ($\bar{T}_{\text{sink}}$) (°C) (see Equation C1). The Carnot COP cannot be achieved in practice but is a useful benchmark to measure actual heat-pump systems against.

$$COP_{\text{Carnot}} = \frac{\bar{T}_{\text{sink}}+273.15}{\bar{T}_{\text{sink}}-\bar{T}_{\text{source}}} \quad \text{(Eq. C1)}$$

Technically, the Carnot COP is used when the heat-source and heat-sink temperatures are constant. However, Equation C1 (with log mean temperatures for the heat source and heat sink) can be used when the temperature of the heat source or heat sink changes, which is common in the real world when using HTHP units.

The log mean temperature of a heat sink ($\bar{T}_{\text{sink}}$) is calculated using Equation C2 when the heat sink enters the heat pump at an inlet temperature ($T_{\text{sink,inlet}}$) (°C) and is heated to an exit temperature ($T_{\text{sink,exit}}$) (°C).

$$\bar{T}_{sink} = \frac{T_{\text{sink,exit}}-T_{\text{sink,inlet}}}{\ln\left(\frac{T_{\text{sink,exit}}+273.15}{T_{\text{sink,inlet}}+273.15}\right)} - 273.15 \quad \text{(Eq. C2)}$$

The log mean temperature of a heat source ($\bar{T}_{\text{source}}$) is calculated using Equation C3 when the heat source enters the heat pump at an inlet temperature ($T_{\text{source,inlet}}$) (°C) and is cooled to an exit temperature ($T_{\text{source,exit}}$) (°C).

$$\bar{T}_{source} = \frac{T_{\text{source,exit}}-T_{\text{source,inlet}}}{\ln\left(\frac{T_{\text{source,exit}}+273.15}{T_{\text{source,inlet}}+273.15}\right)} - 273.15 \quad \text{(Eq. C3)}$$

In this example, a heat pump heats hot water from 50°C to 85°C. Its heat source is ambient air, with an inlet temperature of 20°C. The flow rate is sufficient for the air temperature change to be small (≤ 1°C). The log mean temperature of the hot water (the heat sink) is calculated using Equation C4.

$$\bar{T}_{\text{sink}} = \frac{T_{\text{sink,exit}}-T_{\text{sink,inlet}}}{\ln\left(\frac{T_{\text{sink,exit}}+273.15}{T_{\text{sink,inlet}}+273.15}\right)} - 273.15 = \frac{85-50}{\ln\left(\frac{85+273.15}{50+273.15}\right)} - 273.15 = \frac{35}{0.10284} - 273.15 = 67.2°\text{C} \quad \text{(Eq. C4)}$$

As there is only a small change in the ambient-air (the heat source) temperature, the value of $\bar{T}_{\text{source}}$ is taken to be 20°C (see Equation C5).

$$\bar{T}_{\text{source}} = 20.0°\text{C} \quad \text{(Eq. C5)}$$

The Carnot COP for these conditions is calculated as shown in Equation C6.

$$COP_{\text{Carnot}} = \frac{\bar{T}_{\text{sink}}+273.15}{\bar{T}_{\text{sink}}-\bar{T}_{\text{source}}} = \frac{67.2+273.15}{67.2-20.0} = 7.2 \quad \text{(Eq. C6)}$$

C2 Thermodynamic efficiency relative to Carnot

The thermodynamic efficiency relative to Carnot (η_{Carnot}) is the ratio of the COP of the real system to the Carnot COP (COP_{Carnot}) (see Equation C7). It measures how closely the real system is performing to the theoretical maximum for the conditions.

$$\eta_{\text{Carnot}} = \frac{COP}{COP_{\text{Carnot}}} \quad \text{(Eq. C7)}$$

C3 Temperature lift

The temperature lift (ΔT_{lift}, °C) is the difference between the temperature of the heat sink and the temperature of the heat source.

$$\Delta T_{lift} = \bar{T}_{sink} - \bar{T}_{source} \quad \text{(Eq. C8)}$$

Sometimes, the temperature lift is expressed in terms of the maximum sink temperature and the maximum source temperature rather than the log mean temperatures, so care is needed when comparing values that the same definition has been used.

APPENDIX D – ESTIMATION OF ENERGY, COST, AND GHG EMISSIONS SAVINGS

(Normative)

D1 General

The following methods allow the annual energy use, operating cost, and greenhouse gas (GHG) emissions of an HTHP to be calculated, and for these to be compared with those for a traditional heating system (for example, a boiler) using combustion of fossil fuels.

The calculations are presented on an annual basis using SCOP but could be calculated for an alternative period (for example, daily) using the COP for the operating conditions for that period. To do so, replace SCOP with COP in the formula below, and use the operating hours appropriate to the alternative period selected.

Note the units for the quantities in the equations below and ensure that the values used are in the correct units.

D2 Energy use

$$D_{annual} = Q_{average} \times h_{annual} \quad \text{(Eq. D1)}$$

$$E_{HTHP} = \frac{D_{annual}}{SCOP} \quad \text{(Eq. D2)}$$

$$E_{Boiler} = \frac{D_{annual}}{\eta_b} \quad \text{(Eq. D3)}$$

$$E_{Savings} = E_{Boiler} - E_{HTHP} \quad \text{(Eq. D4)}$$

Where: D_{annual} = annual heating demand (kWh), $Q_{average}$ = average heating demand (kW), h_{annual} = annual hours of operation (hours), E_{HTHP} = annual energy use for the HTHP (kWh), $SCOP$ = seasonal COP for the HTHP (see 1.3), E_{Boiler} = annual energy use for the competitor technology (for example, boiler) (kWh), η_b = boiler efficiency (as a fraction), $E_{Savings}$ = annual energy savings realised (kWh).

D3 Cost

$$AC_{HTHP} = \frac{D_{annual}}{SCOP} \times C_{elect} \quad \text{(Eq. D5)}$$

$$AC_{Boiler} = \frac{D_{annual}}{\eta_b} \times C_{fuel} \quad \text{(Eq. D6)}$$

$$AC_{Savings} = AC_{Boiler} - AC_{HTHP} \quad \text{(Eq. D7)}$$

Where: AC_{HTHP} = annual energy cost of the HTHP ($), AC_{Boiler} = annual energy cost of the competitor technology (for example, boiler) ($), $AC_{Savings}$ = annual energy cost savings ($), C_{elect} = electricity cost ($/kWh), C_{fuel} = fuel cost (for coal, gas, and so on) ($/kWh).

D4 GHG emissions

$$GHG_{HTHP} = \frac{D_{annual}}{SCOP} \times EF_{elect} \quad \text{(Eq. D8)}$$

$$GHG_{Boiler} = \frac{D_{annual}}{\eta_b} \times EF_{fuel} \quad \text{(Eq. D9)}$$

$$GHG_{Savings} = GHG_{Boiler} - GHG_{HTHP} \quad \text{(Eq. D10)}$$

Where: GHG_{HTHP} = annual greenhouse gas emissions of the HTHP (kg CO_2eq), GHG_{Boiler} = annual greenhouse gas emissions of the competitor technology (for example, boiler) (kg CO_2eq), $GHG_{Savings}$ = annual savings in greenhouse gas emissions (kg CO_2eq), EF_{elect} = emission factor of the electricity supply (kg CO_2eq_q/kWh), EF_{fuel} = emission factor of the fuel supply (kg $CO_{2,eq}$/kWh).

D5 ETS savings

Some users of HTHP systems (for example, some industrial users) may need to purchase emissions trading scheme (ETS) units to offset GHG emissions. If this is the case, then the GHG savings can result in additional dollar savings due to reducing the number of ETS units that must be purchased. This can be calculated as follows:

$$ETS_{Savings} = \frac{GHG_{Savings}}{1000} \times ETS_{Cost} \quad \text{(Eq. D11)}$$

Where: $ETS_{Savings}$ = annual savings due to reduced ETS unit purchases ($), ETS_{Cost} = ETS unit cost ($/tonne CO_2eq). Note the factor of 1000 is due to the ETS unit cost being per tonne, whereas the GHG emissions savings in kg CO_2eq.

D6 Payback

The simple payback time (in years) for installing an HTHP versus a traditional heating system can be calculated using the equations below. This assumes that the HTHP has a higher capital cost (CC) than the traditional heating system and that the extra capital expenditure can be justified by lower operating costs (OC). The operating costs ($/year) excluding power (for the HTHP) and the fuel (for the boiler) are included in the OC variables below. These include items such as labour, repairs, and maintenance. The $OC_{Marginal}$ value will be a negative number if the HTHP has other operating costs that are lower than the traditional, fossil-fuel-fired heating system. The fuel and electricity costs should include carbon charges (for example, ETS charges).

$$CC_{Marginal} = CC_{HTHP} - CC_{Boiler} \quad \text{(Eq. D12)}$$

$$OC_{Marginal} = OC_{HTHP} - OC_{Boiler} \quad \text{(Eq. D13)}$$

$$PB = \frac{CC_{Marginal}}{D_{annual} \times \left(\frac{C_{fuel}}{\eta_b} - \frac{C_{elect}}{SCOP}\right) - OC_{Marginal}} \quad \text{(Eq. D14)}$$

Where: $CC_{Marginal}$ = marginal capital cost of installing the HTHP rather than a competitor technology ($), CC_{HTHP} = capital cost of the HTHP ($), CC_{Boiler} = capital cost of the competitor technology (for example, boiler) ($), $OC_{Marginal}$ = marginal operating cost of installing the HTHP rather than a competitor technology ($/year), OC_{HTHP} = annual operating cost of the HTHP excluding power ($/year), OC_{Boiler} = annual operating cost of the competitor technology (for example, boiler), excluding fuel ($/year), PB = simple payback time (years).

D7 Calculation example

An HTHP is considered for an application with a heating demand of 55 kW on average for 5000 hours per year. The price of electricity is $0.12/kWh, and the SCOP for the proposed HTHP is 3.5. Alternatively, the heating demand can be met with a natural gas-fired boiler with an efficiency of 0.8 (in other words, 80%). The cost of natural gas is $11.1/GJ (which is $0.04/kWh). The electricity emission factor is 0.108 kg CO_2/kWh (30 kg CO_2/GJ), and the natural gas emission factor is 54 kg CO_2/GJ (0.194 kg CO_2/kWh).

Using equations D1 to D10:

$$D_{annual} = Q_{average} \times h_{annual} = 55 \times 5000 = 275{,}000\ kWh \quad \text{(Eq. D1)}$$

$$E_{HTHP} = \frac{D_{annual}}{SCOP} = \frac{275{,}000}{3.5} = 78{,}571\ kWh \quad \text{(Eq. D2)}$$

$$E_{Boiler} = \frac{D_{annual}}{\eta_b} = \frac{275{,}000}{0.8} = 343{,}750\ kWh \quad \text{(Eq. D3)}$$

$$E_{Savings} = E_{Boiler} - E_{HTHP} = 343{,}750 - 78{,}571 = 265{,}179\ kWh \quad \text{(Eq. D4)}$$

$$AC_{HTHP} = \frac{D_{annual}}{SCOP} \times C_{elect} = \frac{275.000}{3.5} \times 0.12 = \$9{,}429 \quad \text{(Eq. D5)}$$

$$AC_{Boiler} = \frac{D_{annual}}{\eta_b} \times C_{fuel} = \frac{275{,}000}{0.8} \times 0.04 = \$13{,}750 \quad \text{(Eq. D6)}$$

$$AC_{Savings} = AC_{Boiler} - AC_{HTHP} = 13{,}750 - 9{,}429 = \$4{,}321 \quad \text{(Eq. D7)}$$

$$GHG_{HTHP} = \frac{D_{annual}}{SCOP} \times EF_{elect} = \frac{275{,}000}{3.5} \times 0.108 = 8486\ kg\ CO_2eq \quad \text{(Eq. D8)}$$

$$GHG_{Boiler} = \frac{D_{annual}}{\eta_b} \times EF_{fuel} = \frac{275{,}000}{0.8} \times 0.194 = 66{,}688\ kg\ CO_2eq \quad \text{(Eq. D9)}$$

$$GHG_{Savings} = GHG_{Boiler} - GHG_{HTHP} = 66{,}688 - 8486 = 58{,}202\ kg\ CO_2eq \quad \text{(Eq. D10)}$$

If the ETS unit price is $62/tonne CO_2eq then the $ value of the emissions savings can also be calculated:

$$ETS_{Savings} = \frac{GHG_{Savings}}{1000} \times ETS_{Cost} = \frac{58{,}202}{1000} \times 62 = \$3{,}609 \quad \text{(Eq. D11)}$$

NOTES

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