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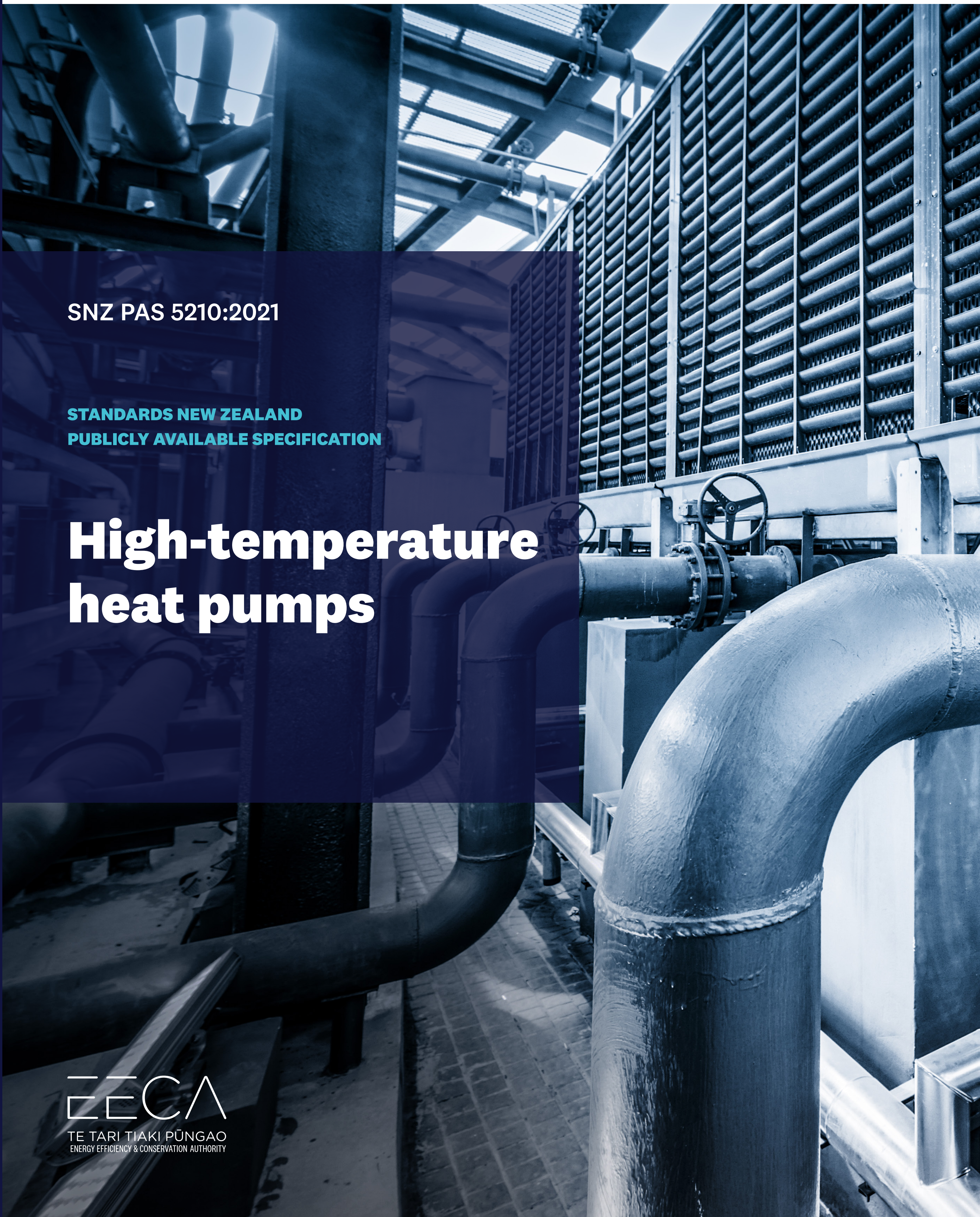

SNZ PAS 5210:2021

STANDARDS NEW ZEALAND
PUBLICLY AVAILABLE SPECIFICATION

High-temperature heat pumps

TE TARI TIAKI PŪNGAO
ENERGY EFFICIENCY & CONSERVATION AUTHORITY

SNZ PAS 5210:2021

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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). The membership of the TAG was approved by the New Zealand Standards Executive.

The TAG consisted of representatives of the following nominating organisations:

Carbon and Energy Professionals New Zealand

Energy Efficiency and Conservation Authority

Institute of Refrigeration, Heating and Air Conditioning Engineers of New Zealand Inc.

Massey University

New Zealand Green Building Council

New Zealand National Committee of the International Institute of Refrigeration

Solar Association of New Zealand

WorkSafe New Zealand

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High-temperature heat pumps

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NOTES

CONTENTS

Table

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 4:2018	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 (draft at time of writing)
AS/NZ 5125:2014	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 60335:----	Household and similar electrical appliances – Safety
Part 2.40:2019	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

British standards

BS EN 13445:----	Unfired pressure vessels
Part 1:2014	General
PD 5500	Specification for unfired fusion welded pressure vessels

Other publications

ASME. Boiler and Pressure Vessel Code (BPVC), Section VIII-Rules for Construction of Pressure Vessels Division 1. 2019

New Zealand legislation

Climate Change Response Act 2002

Climate Change Response (Zero Carbon) Amendment Act 2019

Electrical (Safety) Regulations 2010

Ozone Layer Protection Act 1996

Energy Efficiency (Energy Using Products) Regulations 2002

Websites

www.legislation.govt.nz

LATEST REVISIONS

The users of this 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

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

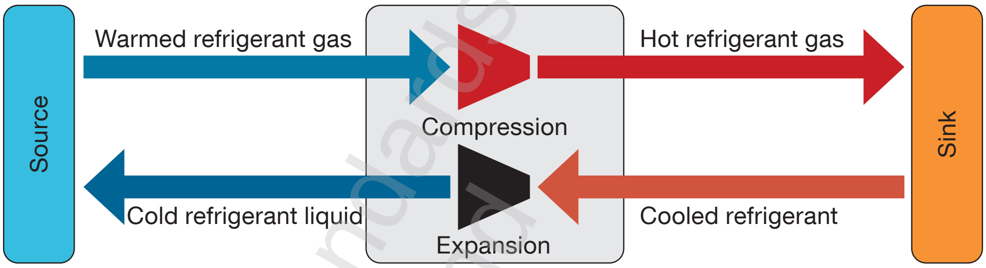
FOREWORD

The objective of this publicly available specification (PAS) is to provide advice and information to support the adoption of energy-efficient heat pump systems to deliver hot water in New Zealand commercial space heating applications (for example, schools or hospitals), and industrial process heat applications.

What is a high-temperature heat pump?

A high temperature heat pump (HTHP) is a device that absorbs heat from a heat 'source' at a low temperature and then transfers (rejects) that heat to a heat 'sink' at a high temperature. The effect is to increase the temperature of the heat sink.

In a typical application, the heat source is ambient air and the heat sink is hot water used for space heating. Process waste heat is a common heat source in industrial applications.



Internally within the heat pump, heat is absorbed from the heat source by evaporating a refrigerant at a low pressure, and transferred to the heat sink by condensing (or cooling) the refrigerant at a high temperature. Additional energy must also be added to operate the HTHP, mostly to compress the refrigerant, but also for fans, pumps, and controls.

A HTHP can be very energy efficient because the amount of energy transferred from the temperature source to the temperature sink (that is, the amount of heating done) is typically greater than the electrical energy supplied to operate the device. A HTHP is also more energy efficient than using electrical energy directly for resistance heating.

Why consider a high-temperature heat pump?

The New Zealand Government has set ambitious targets under the Climate Change Response (Zero Carbon) Amendment Act 2019 to achieve net zero carbon emissions by 2050. Reducing coal use across New Zealand is an obvious step towards these targets. Coal continues to be used in many heating applications in schools, hospitals, prisons, and much of industry. HTHPs have excellent potential to replace coal and mitigate climate carbon impact in a cost-effective way.

NOTES

Publicly Available Specification

High-temperature heat pumps

1 BASIC OVERVIEW

1.1 General

This section provides a basic overview of HTHP system terminology. Please refer to section 3.3 for a technical description of HTHP system operation.

Heat pumps have been available for many years, however recent advances in the technology have made them better suited to high-temperature applications. This publicly available specification (PAS) is limited to the most common systems available, which are those that utilise a closed mechanical vapour compression cycle using electricity.

The primary focus of this PAS is packaged electrically driven high-temperature heat pumps that provide greater than 10 kW of heating.

Very large heating requirements can be met with larger individual packaged units, by smaller multiple packaged units, or by custom engineered systems. At the time of writing, the largest individual packaged units available provide up to about 1000 kW of heating.

1.2 Terms to understand

Important terms to understand include the following:

(a) COP is the coefficient of performance and is how the efficiency of a HTHP is measured:

   (i) In simple terms, a COP of 1 means that for every 1 unit of electrical power consumed (kW), 1 unit of heating power is delivered (kW)

   (ii) A well-designed HTHP system will typically provide a minimum COP of between 2 and 3 meaning that for every 1 unit of electrical power consumed, between 2 and 3 units of heating power is delivered;

(b) A refrigerant is the fluid used within a sealed HTHP to absorb and reject heat. Some refrigerants are more flammable than others, and many refrigerants can damage the ozone layer or contribute to global warming if released to the atmosphere. The type of refrigerant a HTHP uses is an important consideration in the purchase process. Please refer to the technical overview section 3.3 for an examination of common refrigerants used in HTHPs in NZ;

(c) Site integration is the process where your facility is assessed by a qualified inspector to establish the best HTHP system to suits your needs. This will include a site visit and a report that thoroughly investigates the best options to either install a new HTHP or integrate a HTHP into your existing heating system.

1.3 Checklist

At the end of section 3 is a checklist of questions to work through when seeking quotes for a HTHP system, or when assessing quotes provided by suppliers. The checklist is detailed and includes the following key elements:

(a) Heating requirements – establishing your heating load;

(b) Site integration – how and where does a HTHP fit?

(c) HTHP design and performance – sizing a system that will deliver the right result;

(d) Maintenance and monitoring – optimising HTHP performance over time;

(e) Safety;

(f) Connectivity – reacting to remote price signals to minimise costs.

2 GENERAL

2.1 Interpretation and document navigation

Sections 1 to 3 of this document are written to be broadly accessible to a non-technical audience who may be making decisions about the purchase and use of a HTHP system. Section 3 also considers site integration, operation, and maintenance and also a checklist of key questions to ask potential suppliers and installers, and site-specific elements to consider. Section 4 includes a detailed performance specification and is written primarily for a technical audience (for example, specifiers, consultants, or installers).

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.

2.2 Definitions

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

Carnot cycle	The Carnot cycle is the thermodynamically ideal heat pump cycle with no losses (and has the highest theoretically possible coefficient of performance (COP) given the heat source and sink temperatures). The Carnot cycle COP cannot be achieved in practice but sets a useful benchmark that actual heat pump systems can be measured against
Coefficient of performance (COP)	The coefficient of performance is the ratio of the amount of useful heating (plus useful cooling in some cases) to the amount of input energy. System COPs are more comprehensive (and used in this PAS) but compressor-only COPs are often quoted to allow comparisons when the rest of the system is not fully specified
Heat exchanger	A heat exchanger is a device that exchanges heat between two fluids. In a heat pump, the evaporator and condenser components are both heat exchangers
Heat rejected	The heat transferred by the heat pump into the heat sink is commonly referred to as the heat rejected. It is rejected from the heat pump
Heat sink	The fluid receiving heat from the heat pump via its condenser or equivalent
Heat source	The medium supplying heat to the heat pump via its evaporator or equivalent
Pinch analysis	Pinch analysis is an analysis 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

Refrigerant	The refrigerant is the fluid internally circulated within the heat pump to convey heat from the heat source to the heat sink. It can be either a liquid or a vapour at different points in the heat pump system
Seasonal energy efficiency ratio	The seasonal energy efficiency ratio (SEER) is an annual average COP for the heat pump. This accounts for COP varying throughout the year as the heating demand and operating conditions change
Space heating	Space heating refers to heating of rooms for human occupancy
Stage/multi-stage	The simplest configuration of a mechanical vapour compression cycle for a heat pump has a single compression and a single expansion stage. When the temperature lift is very large it can be more energy efficient to perform either or both the compression and expansion in two or more steps or stages. There are many possible configurations of multi-stage systems the detail of which is beyond the scope of this PAS
Temperature lift	The difference in temperature between the heat sink temperature and the heat source temperature. Temperature lifts can be defined for both the system (for example, difference between the temperature of the space or water being heated and the ambient air) and the high-temperature heat pump itself (for example, difference between refrigerant condensing and evaporating temperatures)
Thermal stratification	When the liquid used for thermal storage is not mixed it will stratify according to temperature, as the liquid density changes with temperature. That is, normally the higher temperature liquid will rise to the top of the tank and the lower temperature liquid will sink 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. Configured correctly, such a system can increase the operating efficiency of the connected heat pump and help to ensure a high temperature is delivered to the application

Transcritical heat pump	In a transcritical heat pump the high-pressure part of the system is operated above the refrigerant's critical point. Above the critical point the refrigerant cannot condense, it cools as a supercritical fluid instead, and the 'condenser' is instead referred to as a gas cooler. A transcritical system allows a higher temperature to be generated by the heat pump with that refrigerant than would be possible in a conventional (subcritical) system with the same refrigerant. In such systems, the refrigerant will be a supercritical fluid on the high-pressure side and the heat pump, particularly the gas cooler, must be designed appropriately

2.3 Abbreviations

This PAS uses the following abbreviations:

BMS	Building management system
COP	Coefficient of performance
CO_2-e	Carbon dioxide equivalent
ETS	Emissions Trading Scheme
GWP	Global warming potential
HC	Hydrocarbons
HCFO	Hydrochlorofluoroolefins
HFC	Hydrofluorocarbons
HFO	Hydrofluoroolefins
HTHP	High-temperature heat pump
HVAC	Heating, ventilation, and air conditioning
HXs	Heat exchangers
LAN	Local area network
ODP	Ozone depletion potential
OLPA	Ozone Layer Protection Act 1996
PAS	Publicly available specification
P_c	Critical pressure
ppm	Parts per million
SEER	Seasonal energy efficiency ratio
SG	Safety group

$\mathbf{T_{BP}}$	Boiling point at standard atmospheric pressure
$\mathbf{T_c}$	Critical temperature
VPN	Virtual private network
VSDs	Variable speed drives

2.4 Notations

This PAS uses the following notations:

COP_{Carnot}	= Carnot coefficient of performance
$EF_{electricity}$	= Emission factor for electricity (kg CO_2 kWh^{-1}; or equivalents)
$EF_{heating\ fuel}$	= Emission factor for heating fuel (for comparison; kg CO_2 kWh^{-1}; or equivalents)
P	= Electrical power consumption of the heat pump (kW)
$Q_{cooling}$	= Amount of cooling provided by the heat pump (kW)
$Q_{heating}$	= Amount of heating provided by the heat pump (kW)
$\bar{T}_{sink}$	= Log mean temperature of the heat sink (°C)
$\bar{T}_{source}$	= Log mean temperature of the heat source (°C)
ΔT_{lift}	= Temperature lift (difference between heat sink and heat source temperatures) (°C)
η_b	= Efficiency of a combustion based system (boiler) for comparison
η_{Carnot}	= Thermodynamic efficiency of the real heat pump relative to the Carnot cycle

2.5 Scope

2.5.1 Inclusions

The areas covered in the scope of this PAS are:

(a) Electrically driven packaged heat pump systems with a heating capacity greater than 10 kW delivering a temperature between 50°C and 160°C;

(b) Issues of site integration (including connectivity, advanced controls, thermal storage and heating distribution, provision of cooling, electrical supply, and performance monitoring);

(c) The range of available refrigerants;

(d) WorkSafe New Zealand safety requirements;

(e) New installations and retrofit applications.

2.5.2 Exclusions

The areas excluded from the scope of this PAS are:

(a) Standard HVAC systems;

(b) Installation advice;

(c) Heat-driven heat pumps (such as absorption refrigeration);

(d) Small-scale single-phase heat pumps (such as residential units and systems).

3 HTHP SYSTEMS

3.1 Advantages of the technology

Key advantages of HTHP technology in New Zealand include the following:

(a) HTHPs use largely renewable electricity (with low emissions factor) rather than fossil-fuel fired heating systems (with high emissions factor) thereby reducing carbon emissions for the same heating utility (often by more than 80%). Electricity is also nearly universally available in New Zealand;

(b) Electrically driven HTHPs are often quieter and more reliable (lower maintenance) than boiler-based heating systems;

(c) On site fuel supply or fuel storage is not required. This may greatly reduce the physical size of the heating system compared to a fossil fuel-based alternative;

(d) Multiple HTHPs can be distributed around a site (if the site is suitable) close to the heating demand, reducing distribution network heat-losses and improving heating system efficiency;

(e) Electrically driven HTHPs have no emissions (for example, no products of combustion to dispose of such as ash or flue gases including NO_x and SO_x, and no coal dust) and are cleaner to operate than a boiler-based heating system. Therefore, HTHPs typically do not require resource consent;

(f) Due to the thermodynamic cycle a HTHP implements, one unit of input electrical energy can deliver more than one unit of useful heat (quantified via the coefficient of performance, COP). This means that they often have reduced operating energy costs for a given heat demand;

(g) HTHPs can upgrade low temperature ambient or waste heat to higher and more useful temperatures (often called temperature lift);

(h) HTHPs are electronically controlled, and therefore have potential for more precise and advanced control methods including electrical load management. Control methods can include remote access and monitoring capabilities that are easily integrated with a building management system (BMS) or similar;

(i) HTHPs can have high part-load and variable-load efficiency, especially if variable speed control is used for the compressors, pumps, and fans. Variable speed control can also allow the electrical demand of the HTHP to be temporarily lowered in times of high site or network electrical demand, to respond to electricity tariff rates or participate in demand-response programmes;

(j) The economic feasibility of HTHPs is likely to improve if carbon charges (for example, Emissions Trading Scheme (ETS) charges) increase because the impact will be higher on the supply cost of fossil fuels for combustion than on renewable electricity;

(k) Life-cycle costs for HTHPs are often lower than for fossil fuel heating systems or electric resistance heating, due to their generally lower operating costs;

(l) Some HTHPs can be used to efficiently meet both heating and cooling demands, either simultaneously or at different times. This can improve the economic case for installation as the same capital outlay provides two benefits. This can be an advantage on complex industrial sites, or in space conditioning applications where heating and cooling are both required at different times of the year.

3.2 Limitations of the technology

Limitations of HTHP technology include the following:

(a) HTHPs are generally more expensive to purchase than electric resistance heating and are often more expensive than boiler systems. While they often have a lower life-cycle cost compared to other heating systems, a high COP is critical for lower operating costs (particularly lower energy costs);

(b) For a HTHP to have lower operating energy costs than a combustion-based heating system, the minimum performance is that:

$$COP > \frac{cost\ of\ electricity}{cost\ of\ fuel} \times \eta_b \quad \text{(Eq. 1)}$$

Where:

COP = HTHP coefficient of performance,

η_b = efficiency of the combustion system (boiler) in converting fuel into useful heat as a fraction.

For example, if electricity is \$0.12/kWh and natural gas for an 80% efficient boiler is \$11.1/GJ (\$0.04/kWh), then the COP or seasonal energy efficiency ratio (SEER, see 4.3) needs to be greater than (0.12/0.04 × 0.8 =) 2.4 for the energy costs to be lower for the HTHP;

(c) For a HTHP to have lower operating carbon emissions than a combustion-based heating system, the minimum performance is that:

$$COP > \frac{EF_{electricity}}{EF_{heating\ fuel}} \times \eta_b \quad \text{(Eq. 2)}$$

Where:

EF = emission factor (kt CO_2-e/PJ; g CO_2-e/MJ; g CO_2-e/kWh).

For example, if 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 (assuming the boiler efficiency remains at 80%), then the COP or SEER needs to be greater than (30/54 × 0.8 =) 0.44 for the emissions to be lower for the HTHP;

(d) HTHPs using ambient heat sources reduce in heating capacity and efficiency as the ambient temperature reduces, and when the heating demand increases. HTHP units will typically require defrosting cycles below about 7°C due to frosting (icing) on the heat source heat exchanger. A well-designed air-source HTHP will have an effective defrost cycle or other technologies such as vapour injection to minimise the impact of frosting and frost removal on system performance. Some HTHPs are not designed to operate under frosting conditions and so should not be used if ambient temperature are likely to be below about 5°C for long periods of time when the HTHP is operating;

(e) HTHPs are often designed for the worst-case winter morning but this can result in a grossly oversized and expensive HTHP that might operate part-loaded and potentially less efficiently for most of the year. Mitigation (such as resistive heating) for ‘underperformance’ at extreme weather conditions or installation of thermal storage can greatly reduce initial costs, and a well-designed system with good control strategies can avoid penalties during part-load operation;

(f) HTHPs will operate more reliably and efficiently if temperature conditions are steady and do not vary rapidly nor by large magnitudes. If heat sinks or sources are highly variable or not coincident, then thermal storage may be required for the HTHP to run effectively, at an additional capital cost (if not already installed as part of an existing system);

(g) Large HTHPs will have significant electricity demands and may require electrical supply lines to be upgraded (both internally within the site and in the distribution network supplying the site from the national grid). Particularly for industrial sites, the costs of electrical upgrades can be significant and must be included in the feasibility analysis. The extra electricity demand of the HTHP can also have a significant impact on the ongoing electricity supply charges, such as electricity retailer charges, network demand charges, and grid interconnection charges (if these apply). Engaging early with the retailer, electricity network company, and, where relevant, Transpower, is advised to understand these impacts and to review the constraints and opportunities for accommodating a large HTHP on the site;

(h) There are constraints on some of the refrigerants used in HTHPs. In particular:

  (i) Many HFC refrigerants have a very high global warming potential (GWP) and are being phased out

  (ii) Low-GWP refrigerants (GWP < 150), tend to be flammable (except R718, which is water, and R744, which is CO_2) and some are also toxic (for example, R717, which is ammonia), which means that HTHPs need to be designed to meet safety regulations and to ensure safe operation. Usually, all safety concerns will be isolated to the HTHP's location by use of a secondary transfer fluid (for example, hot water) to distribute the heat around the site

  (iii) Refrigerants with intermediate GWP (between 500 and 1500) that are non-flammable and non-toxic are likely to be used in HTHPs until lower GWP refrigerants become available and more experience is gained in their use. If such a refrigerant is used, then there should be a clear plan to minimise risks related to refrigerant phase-out over the life of the HTHP. For example, the HTHP might be designed so that a future retrofit with a flammable low-GWP refrigerant is likely to be possible and safe, designed to minimise the refrigerant charge, or by securing a future supply of the original refrigerant subject to phase-out;

(i) The temperatures at which HTHPs can provide heat are limited by a number of factors including:

  (i) Reduced COP at high delivery temperatures (including reduced operating efficiency of HTHP components, particularly compressors)

  (ii) High temperature lifts often require a HTHP to be a multi-stage design to maintain acceptable efficiencies (that is, COP > 0.4 COP_{Carnot}) but this significantly increases the capital cost

  (iii) Lubrication of HTHP compressors becomes more difficult at high temperatures due to the wider range of temperature the oil experiences in the HTHP (for example, too viscous at source temperatures or too 'thin' at sink temperatures) and the reduced chemical stability of oil at high temperatures. Oil-free compressors minimise lubrication issues but may be less efficient than lubricated compressors

(iv) Increased maintenance and reduced economic life due to the effect of high temperature on mechanical integrity of HTHP componentry, particularly compressors and heat exchangers;

(v) Limited availability or high cost of HTHP components that operate at high pressure. As temperature increases, the pressure of the refrigerant increases. HTHPs normally use refrigerants that have lower pressure at high temperature but this restricts choice and may compromise the capacity or efficiency of the HTHP and have safety implications (for instance the use of lower pressure hydrocarbons refrigerants such as butane);

(j) HTHPs heating a large thermal mass (for example, a large hot water tank) can require a period of time for the hot water to reach its operating temperature. This is a normal function of the thermal mass being heated rather than the heat pump itself, but in such systems the heating system must be started early in order to effectively provide heating. Under particularly cold conditions, it may be necessary to allow a longer period of pre-heating time before the heat pump can produce the desired temperature, or to include thermal storage with enough capacity to supply to residual heating during the start-up period. HTHPs designed to operate with stratified thermal storage systems can reduce the start-up time for the system to be ready to deliver at the required operating temperature;

(k) The electricity demand of HTHPs is variable depending on a range of factors, primarily operating parameters and ambient conditions. Upon starting, heat pumps typically draw a higher current than during normal operation. The electricity supply of the heat pump should allow for this peak draw;

(l) The noise produced by a HTHP during operation can be significant, particularly for large commercial heat pumps (compressor noise) or those with large fans located outside (such as on heat-source heat exchangers). Insulated enclosures for indoor equipment, careful placement of units, and selection of low-noise fans can minimise noise impact;

(m) Air-source heat pumps can require large volumes of air to operate effectively and efficiently. The installation of such heat pumps should be done in a way that prevents air recirculation between the fan discharge and air inlet locations as this can significantly reduce performance;

(n) Heating at different temperatures (for example heating water from 60°C to 70°C for space heating plus heating potable water from 20°C to 60°C for sanitary use, or for low-pressure steam at 120°C for sterilisation) can be difficult to achieve cost-effectively with a single HTHP. Such cases might require a special HTHP design, or an alternative system (such as an electric resistance or boiler-based heating system) for the low-capacity or high-temperature duty.

3.3 Technical overview

3.3.1 Metrics of efficiency and performance

The fundamental metric of performance for a heat pump is the coefficient of performance (COP). This is the ratio of the amount of heating done ($Q_{heating}$, kW) to the electrical power to operate the system (P, kW).

$$COP = \frac{Q_{heating}}{P} \qquad \text{(Eq. 3)}$$

The expected COP for a system is usually significantly greater than 1.0, in order for the system to be effective, but the value depends upon the technology, system configuration, and the operating conditions.

The COP is equivalent to the heating efficiency of a boiler. For example, a boiler efficiency of 80% corresponds to a COP of 0.8. For more information on the Carnot coefficient of performance see 4.1.

3.3.2 Estimation of required heating capacity – Installing the right system for the space

An accurate estimation of required heating demand is important to ensure that the designed and installed system is fit for purpose. Heating demand requirements should be calculated by an independent professional (not the installing contractor). Online calculators are available for space heating.

The following parameters need to be considered to obtain an accurate required heating demand:

(a) For space heating applications:

   (i) The size of and number of spaces to be heated, noting detailed floor areas, and room volumes

   (ii) The construction of the space's floors, walls and ceilings, particularly noting overall insulation values

   (iii) The area of windows for each space and if the windows are double or triple glazed

   (iv) An estimate of the 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;

(b) For process heating applications:

   (i) The temperature and heat load of each application

   (ii) The expected schedule of application heat loads, including variations throughout an operating shift, from season to season, and so forth as applicable;

(c) For a retrofit application, existing monitored data of required heating demand can be used, if the data is reliable. Monitoring equipment can be installed in advance of considering a prospective HTHP proposal. Monitoring of an existing installation should be designed to separate the energy supplied from the heating unit from the energy supplied to the occupied spaces or applications so distribution heat losses can be estimated. Where changes to the distribution system are proposed as part of the installation, the effect of these changes should also be measured by the data collected;

(d) For heating demands that are influenced by the ambient temperature (such as space heating) then the location of the best source of local weather data for setting the design condition should be agreed, along with an agreement upon the allowance for weather extremes.

3.3.3 System selection

The selection of a HTHP solution shall consider the following:

(a) The life-cycle cost of the system. This shall be defined as the amortised sum of the capital cost, the operating cost, the maintenance cost, and the installation cost for an operating period of 15 years;

(b) The availability of local support capability;

(c) On sites with buildings spread out into blocks, a distributed HTHP system. This places smaller HTHP units around the site so the supply of heated water can be delivered more directly to the areas of heating demand;

(d) The use of thermal storage (typically in the form of an insulated hot water tank) to smooth out heating demand and to minimise delay in providing heating.

In addition to the above, when considering a retrofit or replacement of an existing system, the energy expected to be saved from the operation of the unit, and from the total installed system compared with the existing system shall be determined to the methodology described in AS/NZS 4234.

3.4 Refrigerant selection

3.4.1 Refrigerant basics

The refrigerant is a critical component of the heat pump. It is used to convey heat energy around the system, and it can be either a liquid or a vapour at different points of the system.

Refrigerants are referred to by alphanumeric codes (for example, R32). Some refrigerants suitable for high-temperature heat pumps are given in Table 1. Note that this table is of informative examples only and is not an exhaustive list of suitable refrigerants.

3.4.2 Refrigerant safety

Some refrigerants are hazardous substances due to toxicity or flammability. The refrigerant installed in the HTHP must be appropriately labelled and correct safety measures applied for the hazard level specific to the refrigerant. See AS/NZS ISO 817 for definitions of safety levels.

3.4.3 Refrigerant leaks

In a sealed HTHP unit, the refrigerant remains within the system. However, refrigerant leaks can occur if corrosion or fracturing of pipes or other components occurs. The following steps should be taken:

(a) Records of refrigerant levels and any additions should be kept so that abnormal losses can be identified, investigated, and remediated;

(b) System physical integrity should be regularly checked by inspection;

(c) At the end of life and during maintenance the refrigerant shall be carefully handled to ensure accidental release to the atmosphere does not occur;

(d) Refrigerant levels should be checked and topped up as part of ongoing maintenance.

3.4.4 Environmental impact of refrigerant selection

Some refrigerants cause environmental damage when released into the atmosphere. The principle issues are ozone depletion and contribution to global warming. Refrigerants can have an ozone depletion potential (ODP) value to assess their potential impact on the ozone layer if released to atmosphere.

Refrigerants can also have a global warming potential (GWP) value to assess their impact on climate change. The GWP value is the impact relative to carbon dioxide (CO_2).

The import and manufacture of some refrigerants with a high GWP is controlled, which will affect current and future availability. The purchase price of refrigerants with a high GWP is also strongly influenced by the pricing structure of the Emissions Trading Scheme and will be subject to future carbon-based taxes, levies, or charges.

Refrigerant selection should reflect these factors and, where possible, low GWP refrigerants should be favoured. Refrigerants with a GWP of less than 150 are the most likely to remain viable long-term.

3.4.5 Charge minimisation

Charge minimisation is the practice of designing the heat pump system to use as little refrigerant as possible for cost efficiency. This practice also mitigates risk for HTHPs using hazardous refrigerants.

A suitable metric to compare the refrigerant charge of two systems is the ratio of kilograms of refrigerant charge to kilowatts of heating provided (kg refrigerant/kW heating). Another potential metric is the ratio of the GWP of the total refrigerant charge to the kilowatts of heating provided (kg GWP/kW heating). However, with both metrics, care must be taken when comparing systems that utilise refrigerants with very different safety requirements (see ASHRAE safety groups, Table 1).

Table 1 – Refrigerant types for HTHPs

Type	Refrigerant	Chemical Formula	T_c (°C)	P_c (bar)	T_{BP} (°C)	ODP	GWP	SG
HFC	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	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	R1224yd(Z)	$CHCl{=}CFCF_3$	156	33.4	15	0.00012	4	A1
	R1233zd(E)	$CHCl{=}CHCF_3$	165.6	35.7	18	0.000134	5	A1
HC	R600	$CH_3CH_2CH_2CH_3$ (butane)	152.0	38.0	0	0	4	A3
	R600a	$CH(CH_3)_2CH_3$ (isobutane)	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

NOTE –

(1) HFC: hydrofluorocarbons.

(2) HFO: hydrofluoroolefins.

(3) HCFO: hydrochlorofluoroolefins.

(4) HC: hydrocarbons.

(5) T_c: critical temperature (°C).

(6) P_c: critical pressure (bar).

(7) T_{BP}: boiling point at standard atmospheric pressure (°C).

(8) ODP: ozone depletion potential (basis R11 = 1.0).

(9) GWP: global warming potential (for 100 year time horizon, basis CO_2 = 1.0, AR4 and AR5 data where available).

(10) SG: safety group (ASHRAE standard 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.

3.5 Temperature

3.5.1 Temperature of the heat source and heat sink

HTHPs operate most efficiently when the temperature lift is minimised. Where possible, the system should deliver the coolest possible temperature at the heat sink, while still meeting the heating load requirement. Similarly, the warmest possible temperature heat source should be utilised.

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

For HTHPs providing space heating and utilising ambient air for the heat source, the system is generally most heavily constrained on a cold winter's day when heating demand is also high. The overall system should be designed to perform in such worst-case conditions. However, rather than sizing the HTHP for this extreme condition, it can sometimes be more cost effective to use a smaller HTHP in combination with another auxiliary heating system (such as an electric resistance heater) or thermal storage. The HTHP can be used alone on the vast majority of days and supplemented by the auxiliary heating system on rare extreme days.

3.5.2 Temperature constraints

To achieve high energy efficiency, most HTHPs operate with the refrigerant condensing temperature well below the critical point of the refrigerant used in the system. The critical point of the refrigerant (see Table 1) sets an upper limit to the temperature that can be provided at the heat sink in most systems.

However, specially designed transcritical heat pump systems can operate efficiently with the heat sink temperature at or above the critical point. Such system configurations are common when carbon dioxide is used as the refrigerant (CO_2, R744), but is not common for other refrigerants. Transcritical systems may become more common for HTHPs operating with very high-temperature heat sinks and when the temperature change of the heat sink is large.

The boiling point of the refrigerant, at atmospheric pressure (see Table 1) provides a temperature constraint at the heat source. The refrigerant should be selected so that the boiling point of the refrigerant, at atmospheric pressure, is lower than the usual temperature of the heat source. In most systems, the boiling point of the refrigerant sets a lower limit to the temperature that is practical for the heat source. Furthermore, refrigerants with higher boiling points (sometimes called low-pressure refrigerants) have lower density as vapour at the heat source temperature and will therefore require physically larger compressors, heat exchangers, and piping than a higher-pressure refrigerant at the same temperatures. This need for larger equipment can increase capital costs.

3.5.3 Time to achieve temperature

A HTHP usually requires only a few minutes to come up to temperature and can be cycled on and off, or controlled, to meet heating demand. Delays to delivering heating are more likely due to the time required to heat hot water (or other heating medium) from a cold initial temperature.

For systems operating on a daily heating cycle (such as space heating for daytime occupancy), scheduling the heat pump to start ahead of first occupancy is usually a simple but effective strategy to deliver timely heating.

In applications with highly variable heating demand, variable speed motor control (or banks of smaller heat pumps that can cycle on and off to match demand) should also be considered.

3.5.4 Thermal storage

Thermal storage can be used to smooth out heating demand and to minimise delay in providing heating. Thermal storage is the practice of using the HTHP to heat a medium and store that heat for later use. This is similar to using a battery to store power. A typical approach to thermal storage is to use an insulated hot water tank.

Thermal storage can serve several purposes including:

(a) Peak heating demand reduction to reduce HTHP capital;

(b) Heating demand smoothing to simplify HTHP control and improve efficiency;

(c) Matching heat demand to heat source availability;

(d) Electrical load shifting to minimise exposure to high tariffs or to defer installation costs;

(e) Potential to participate in demand response programmes to gain revenue.

3.6 Advanced controls scope and benefits

3.6.1 Considering the installation as a complete system

Significant improvements in the efficiency and reliability of heat-pump water heaters can often be achieved through considering the installation as a complete system, rather than as a standalone heat-source. Where the supply water temperature from the heat-pump is lower than that of the previous space heating system, additional heat-exchangers may need to be installed to meet the required room temperatures.

3.6.2 Capacity control

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

Speed control systems provide better efficiency during part-load operation than most alternatives. Alternative methods for controlling the capacity of fixed-speed compressor systems include digital compressors, hot gas bypass systems, and slide valves on screw compressors. While such methods do provide effective capacity control, they do so at the cost of efficiency.

As an alternative to variable speed control, capacity control can be enabled through the limiting of maximum compressor speed, or by reducing the water flow through the system in an in-line heating system.

Unlike boiler-based systems, the efficiency and the operating temperatures of HTHPs is impacted by the return water temperature into the unit. In general, the lower the entry temperature of the heat sink to the heat-pump heat-sink heat exchanger the more efficient and lower the operating pressure of the unit will be. The reduction in operating pressure also results in a less stressed, more reliable refrigeration system. In periods of low heating demand, the return water temperature can be warmer, reducing efficiency, and the system should be designed to avoid this.

Caution should be used when considering the use of buffer tanks designed to prevent compressor short cycling and to provide a source of energy for defrost cycles. In highly dynamic installations, a buffer tank can be installed on the return heat sink fluid loop to the heat-pump to smooth the changes in temperature for improved control.

Where thermal storage is required as part of the system, the temperature of the hot water to be heated should be considered when the thermal storage system is designed. A recommended effective approach is thermal stratification (see 3.7.3), however other configurations can be considered.

3.6.3 Pump control

In conventional boiler installations, water flow through the heating system in excess of that required to meet the heating demand is diverted past the heat exchanger, and back to the boiler (often called a bypass). Where a HTHP is retrofitted to an existing system previously operated at a higher circulation temperature than that proposed by the new system, the system shall be converted to a variable flow distribution system.

A variable flow distribution system is designed to deliver to the heating outlets the required amount of heated (or chilled) water 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) prior to returning back to the HTHP or thermal storage tank.

Diverting valves should be set in the fully open position (no diversion), and the flow rate controlled with variable-speed pumps. Alternatively, where existing fixed speed pumps are to be 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 building heating and cooling demand or be controlled to a constant pressure. Constant-pressure control allows the pump speed to be varied according to the total amount of water flow being delivered to the heating or cooling outlets where flow control valves are deployed. Flow control valves can be either binary on/off valves such as on a radiator, or variable flow valves such as a 0 V to 10 V controlled motorised ball valve. In systems where supplementary heating is required to meet peak heating loads, the additional heating should be sourced from the return water loop, where feasible, to further reduce return water temperatures back to the heat-pump water heater. Supplementary heating capacity should be based on room or application temperatures.

3.6.4 Ventilation

Where practical, CO_2 levels in buildings for occupancy shall be controlled to a maximum of 1500 ppm. Control of ventilation shall be considered in installations where supplementary heating is being added to the pre-existing system. Suitable methods for incorporating ventilation control include ducted fan coil and water-sourced air-conditioning units. Where significant fresh air ventilation is required, the thermal load on the room being heated or cooled shall be factored into unit selection.

3.6.5 Heat recovery

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

3.7 Thermal storage and distribution

3.7.1 Size

Heating plants including HTHPs are sized to provide a specific amount of useful heat to meet the required peak (maximum) demand for heating in a system. By carefully designing a system with thermal storage, the heating demand can be somewhat decoupled from the heat pump operation. This means the overall size and cost of a heating plant can be minimised while improving utilisation, and depending on the configuration there can also be control and efficiency benefits.

Heating plants should be sized so the total system, including the heat plant and any thermal storage, can efficiently supply the expected range of heating loads. This may include consideration of multiple smaller heat pumps, or compressor stages, instead of a single larger heat pump; an arrangement that can be controlled to efficiently meet a wide range of heating loads. This may also provide some system redundancy, which can be a design requirement in some situations.

HTHPs usually work most efficiently when both the heat-source (low temperature) side and the heat-sink (high temperature) side remain close to steady-state conditions. In addition, systems can usually operate with improved efficiency when the thermal load on the heating plant itself (the HTHP) is relatively steady, although the system can be specifically designed to accommodate dynamic heat loads instead. In some systems with a variable demand for heat, or a variable heat source supplying a HTHP, thermal storage on one or both sides of the HTHP (heat sink or heat source) may be recommended. Thermal storage can also help heat pump systems that are designed to capture and upgrade waste heat to maximise the amount of waste heat that is captured and used especially when the availability of the waste heat is not always fully coincident with the heating demand.

3.7.2 Maximising efficient operation

Careful consideration should be made of how thermal storage and distribution system affect the temperature supplied to the HTHP. Heat pumps can suffer reduced performance and efficiency if they are operated with a very low temperature lift across the heat-pump heat exchanger compared to the design temperature lift. In general, this means that the fluid being heated can have a maximum return temperature (inlet to the HTHP) for efficient operation of the heat pump. This requirement may affect the design of thermal storage. The most common configuration to maximise efficient operation is to operate the thermal storage as a stratified tank. Thermal stratification takes advantage of the reduction in density of water (or other thermal storage fluids) as it increases in temperature. This results in the natural layering of water at different temperatures in an unmixed vessel, separating the hotter supply water from the colder return water. Similar effects can be achieved in other configurations.

3.7.3 Thermal stratification

The purpose of stratification is to allow the heat pump to draw water from the coldest area of the tank (the bottom) and return the heated fluid to the hottest area of the tank (the top). The heat distribution network would draw from the hottest part of the tank. Make-up fluid is usually cold and would be added to the coldest area of the tank. This allows the heat pump to operate effectively while supplying the hottest possible temperature to the application. It is important that the storage tank is not allowed to mix to maintain the stratification effect, so the tank inflow and outflows must be set up with this in mind.

Note that thermal storage for cooling applications can also utilise a stratified configuration, but the opposite configuration is applied, with the supply of chilled fluid from the bottom of the tank, and fluid returning from the distribution system to the top of the tank. Systems that swing between providing heating and cooling should consider this.

It may be advisable to consider a return line, or bypass from the distribution system, of a specific minimum size and flow capacity to ensure the fluid in the distribution system is continuously replaced by heated fluid from the thermal storage tank. This will maintain a suitable high temperature in the distribution system to overcome heat losses. However, care must be taken that this flow is not so high that it causes the storage tank to mix and lose the stratification effect. Where a return line is implemented, the flow rate through the distribution system shall be controlled to maintain a maximum return temperature no less than 10°C cooler than the supply fluid temperature. Chilled fluid systems shall maintain no less than 5°C warmer return fluid temperature to the thermal store.

3.7.4 Microbial growth

Storing liquids at elevated temperatures can provide conditions for microbial growth, which can present hazards to human health (such as *Legionella*). The design of any thermal storage system should consider such risks and take appropriate measures to prevent this from occurring. For example, one or more of the following may be required: chemical treatment, a regular cleaning procedure, or more typically a suitably high storage temperature.

Thermal storage of sufficiently large volume or capacity may allow HTHPs to be reduced or shut off during periods of high demand on the electricity network. Depending on the customer's electricity tariffs, this may avoid significant demand charges for electricity and allow use of cheaper, off-peak electricity rates.

3.8 Electrical supply

While smaller HTHPs may be connected within an electrical installation relatively easily, an investigation should still be carried out to ensure that there is sufficient capacity and capability to operate the HTHP in the proposed location.

Larger HTHPs will have significant electricity demands and can require a more extensive investigation to ensure that the HTHP:

(a) Can be connected in the desired location, that is, there is sufficient capacity in the electrical installation to start and operate the HTHP satisfactorily; and

(b) The addition of the HTHP in either a controlled or uncontrolled mode will not impact the electricity purchasing contracts for the electrical installation; and

(c) Can provide installation and operating benefits by including either or both of the following:

  (i) Thermal storage as a buffer against connection and operating costs;

  (ii) Peak demand or time-controlled systems to reduce electricity consumption to minimise either capacity investment, or high electricity purchasing cost periods.

Investigations should include consulting with the electricity retailer and the owner of the network regarding:

(d) Any potential impacts to the current electricity purchasing and connection contracts; and

(e) Any alternative purchasing contracts that might be of greater benefit (or will incur a penalty), particularly if control is enabled on the HTHP; and

(f) Associated costs of the HTHP connection both within the electrical installation, and within the supply network.

3.9 Provision of cooling

3.9.1 Seasonal cooling

Some reverse cycle heat-pump water heaters are capable of providing cooling as well as heating. The provision of cooling is increasingly important in parts of New Zealand during summer months. Cooling can be achieved by direct use of the heat pump refrigerant in an air cooling coil, the cooling of chilled water, and the use of water in a heat exchanger such as a fan coil. Alternatively, the water loop can be reconditioned in installations incorporating water source HTHPs. In some installations, the existing water loop can be used to provide partial cooling. Pipework and heat exchangers operating below the dew point require pipe insulation and drainage to prevent water damage from condensation.

3.9.2 Simultaneous heating and cooling

Some HTHPs can also provide useful cooling simultaneously with useful heating (for example, four-pipe chillers or a transcritical CO_2 HTHP can often operate relatively efficiently with the CO_2 evaporating as low as –10°C at the same time as heating cold water to as high as 100°C). In such cases, the cooling COP is often lower than a traditional cooling-only refrigeration system, but the marginal benefit can be very high (a combined heating and cooling system often has low extra capital cost and minimal extra energy use compared with two separate cooling and heating systems). The COP for a system that provides simultaneous heating and cooling is calculated from both the heating ($Q_{heating}$, kW) and cooling ($Q_{cooling}$, kW) effects, and the electrical power (P, kW) to operate the system.

$$COP = \frac{Q_{heating}+Q_{cooling}}{P} \quad \text{(Eq. 4)}$$

3.9.3 Nearly simultaneous heating and cooling

If the demand for cooling and heating is not well matched quantitatively then normally the combined refrigeration and HTHP system should be sized for the smaller of the demands (so that it is fully utilised and provides simultaneous heating and cooling) and a separate (refrigeration-only or HTHP-only) system is used to provide the residual cooling or heating required to meet the larger of the demands.

3.10 Performance monitoring

Ongoing performance monitoring is important as it allows identification and diagnosis of deviation from expected performance and helps to evaluate site, organisational, or national progress towards goals such as decarbonisation.

At a minimum, sufficient performance monitoring shall be stored and accessible to the owner of the system so that meaningful performance comparisons can be made from year to year, season to season, and site to site. This shall include the heat energy produced and the power consumption of the system recorded, at a frequency no less than once every five minutes.

Depending upon the application, other variables should be recorded sufficient to characterise performance. For example, supply and return temperature and delivered flow rate for a hot water heating application.

Capability for remote monitoring should also be a component of the connectivity system (see 3.19).

3.11 Operation instructions

All personnel charged with the operation, supervision, and maintenance of the equipment shall be adequately instructed and competent at their tasks. The installer of the plant shall draw attention to the necessity for adequate instruction of operating and supervising personnel.

The operating instructions should include sufficient manufacturer performance data to allow the expected performance of the system to be calculated for the full range of expected operating conditions, and allow deviations in the system's performance to be monitored and identified.

3.12 Instruction of operating personnel

Before a new plant is put into service, the person responsible for making the system operational shall ensure that the operating personnel are instructed, on the basis of the instruction manual, about the construction, supervision, operation, and maintenance of the plant, as well as the safety measures to be observed, including the properties and correct handling of the refrigerant used.

A record of the fact of this instruction shall be made, along with the subsequent instruction of new operating personnel.

3.13 Documentation

A logbook for the system shall be maintained and updated as required. It shall record:

(a) Details of all maintenance and repair work;

(b) Changes and replacements of components in the system;

(c) Results of all periodic routine tests;

(d) Significant periods of non-use.

The logbook shall either be kept in the same room as the HTHP, or the data stored in a computer system belonging to the party concerned, with a printout made available in the machinery room to allow maintenance personnel access to the information during servicing or testing. When the logbook is kept in the same room as the HTHP an archive copy should also be kept in a secure location or electronically.

Cloud-based storage of documentation is permissible providing access is also available as detailed above.

3.14 Time to heat up and cool down

The start-up time for HTHP units to reach peak output and efficiency is generally less than five minutes. Once operating, multi-stage or variable speed systems can also rapidly and automatically adjust to the required heating demand. At the end of the heating period, units can immediately shut down and be in a state of operational readiness, if necessary. This operational flexibility provides the potential to integrate smart energy features into installed systems, such as capacity response to changing electricity tariffs or peak demand electricity supply constraints, especially when the system has integrated thermal storage.

If integrated thermal storage is present, the time required for the heated water loop (or a similar distribution and/or thermal storage media) to reach operating temperature is dependent upon the thermal mass of the heated water loop and the delivered rate of heating provided by the HTHP. The heated water loop can be configured so that hot water is delivered as soon the HTHP is started, especially if the initial heating requirement is small relative to the installed heating capacity. However, in many applications it can be necessary to start a cold system some time prior to the time of first demand (one or more hours). Remote start-up capabilities, initiated by trained operating personnel, can be utilised and managed in a number of ways including a simple timer, or a smart monitoring system triggered by ambient conditions and other factors.

3.15 Expected lifetime

A HTHP system shall be designed to have a reliable efficient operating life of a minimum of 15 years.

3.16 Maintenance requirements for continued safe and efficient operation

Maintenance personnel shall have qualifications appropriate to the type of plant installed. This often includes manufacturer training and this should be included in the purchase price of the system.

The extent and time schedule for maintenance shall be fully described in the instruction manual supplied with the unit. At a minimum, routine maintenance should be performed annually and maintenance actions taken when equipment faults are indicated.

The exact requirements for maintenance are site and installation-specific. Manufacturer instructions should be complied with.

3.17 Checklist of questions

3.17.1 Key questions and elements to consider

The following information outlines the key questions to ask and elements to consider before installing a HTHP.

3.17.2 Heating requirements

The following questions should be addressed to determine heating requirements:

(a) Can the heat source temperature be raised or the heat sink temperature lowered without compromising the heating load requirements (for example, the room temperature in space heating applications)?

(b) Are the heat sources and sinks coincident and steady (not fluctuating) or will thermal storage be required or (if fitted) can existing thermal storage be reused?

(c) Can the site tolerate slightly lower space and potable water temperatures for short periods on the coldest winter mornings so that the HTHP is not oversized for normal use?

(d) Can the HTHP model being considered do useful cooling at the same time as useful heating?

(e) Is there a need for a back-up system for the HTHP? What level of redundancy is required to ensure continuity of heating?

(f) Is there a requirement for heating to more than one temperature? If so, is the HTHP specially designed to allow effective and efficient design for all temperature levels? Where two temperatures are required, is the higher temperature the main heating load?

(g) What are the estimated heat losses from the distribution system and thermal storage? Can the distribution system be reconfigured to a distributed heat-pump system to minimise heat-losses, and improve heating times for all areas of the site?

3.17.3 Site integration

The performance and suitability of a HTHP depends upon the wider heating system it is a part of. The opportunities for optimal performance are site specific, so a thorough site inspection and analysis by a suitably qualified supplier, installer or consultant should be part of the purchasing process. The analysis should cover:

(a) Safety considerations;

(b) Determining the heating requirement and the optimal total system for delivering heating, including consideration of thermal storage and (potentially) control of the water distribution system;

(c) Determining whether a distributed air-to-air lower temperature heat pump is more cost-effective for the space heating;

(d) Sizing the HTHP to meet the heating requirement;

(e) Estimated efficiency and performance of system options;

(f) Retrofit options when an existing heating system is already in place;

(g) Data connectivity to allow performance monitoring and remote control;

(h) Balancing capital investment and operating cost to achieve lowest life-cycle costs;

(i) Connection of the HTHP system to the electrical installation and any consequential changes to connection to the electricity network;

(j) Changes to electricity purchasing contract(s).

3.17.4 HTHP design and performance

The following questions should be considered to determine appropriate HTHP design and performance:

(a) What refrigerant does the HTHP use? What is the ODP and GWP of the refrigerant used in the heat pump? Is use of the refrigerant restricted, or likely to become restricted, under the Ozone Layer Protection Act (OLPA) for ozone-depleting substances or the Kigali Agreement amendment (for refrigerants with high GWP)? Does the refrigerant have any particular safety concerns (such as flammability, toxicity, or very high pressure and operating requirements)?

(b) Has the safety of the HTHP, including its refrigerant, been accounted for in the design?

(c) What is the rated heating capacity and efficiency (COP) of the heat pump? Has the capacity and performance been defined at a range of appropriate operating conditions? What is the SEER (see 4.3) for the heat pump system?

(d) Are there operating constraints on the flow rate and inlet and outlet temperatures of the heat sink and heat source? Are these appropriate for the intended purpose of the heat pump?

(e) How is the HTHP controlled and will the part-load efficiency be significantly lower than at full load? Due to control (and redundancy provisions) should the option of a configuration with several smaller compressor units rather than a single large compressor be considered?

(f) What is the likely maximum electricity demand for the HTHP, and will this require an upgrade of the retailer contract volumes, site or electricity networks? Will operation of the HTHP affect electricity purchase costs, maximum electricity demand, and grid interconnection charges (if they apply)?

(g) How can the additional electricity requirement of the heat pump be offset through reduction in electrical demand from other areas of the site? Can resistance load be decommissioned? Can demand management be used to provide sufficient electricity supply or reduced operating costs for the HTHP when operating?

3.17.5 Maintenance and monitoring

The following questions relating to maintenance and monitoring should be considered:

(a) Will the HTHP operate within its normal operating range for the majority of the time? (Constant operation at the upper end of its design range can impact the HTHP's efficiency and longevity);

(b) What are the HTHP's maintenance and supervision requirements?

(c) Does the HTHP have a guarantee? What is the availability of spare parts and qualified service technicians? How quickly will most repairs take?

(d) What capabilities does the heat pump have for remote access or remote monitoring? ➤

(e) Can the HTHP be used for electricity load management and demand response? If so, how quickly can the HTHP be taken offline, how long can it be turned off, and how many starts per hour can it safely tolerate?

3.18 Safety

3.18.1 Electrical safety

All HTHP systems shall be electrically safe and comply with the Electrical (Safety) Regulations that are current when the system is installed. In most instances, heat-pump water heaters shall comply with AS/NZS 60335.2.40 with a test report available on request.

3.18.2 Refrigerants

Refrigerants in AS/NZS ISO 817 (also AS/NZS 5149, Parts 1 to 4) are classified on the basis of both flammability and toxicity. The safety requirements for the relevant classification shall be complied with.

Many currently common refrigerants are A1 (neither flammable nor toxic), but generally have a high GWP rating. As the industry moves to lower GWP refrigerants, a trade-off is made with flammability. The A2L class has been recently introduced to identify refrigerants with lower GWP, and low flammability. Hydrocarbon refrigerants are A3 class refrigerants, which are highly flammable and require additional safety controls.

Pressure vessels for refrigerants shall comply with AS 1210, or an international code approved under AS 1210, such as ASME BPV-VIII, BS EN 13445, or PD 5500.

Heat-pump water heater installations are deemed to be electrically unsafe if the refrigerant is changed, unless the heat-pump water heater has been designed or adapted to operate safely with the new refrigerant.

When a refrigerant is substituted, the system shall comply with the requirements for the substituted refrigerant.

3.18.3 Heat-transfer medium (for example hot water)

If a heat-transfer medium such as hot water is utilised, pressure equipment used to contain the heat transfer medium shall meet the requirements of the Approved Code of Practice for the Design, Safe Operation, Maintenance and Servicing of Boilers, and the Approved Code of Practice for Pressure Equipment (Excluding Boilers).

3.18.4 Physical protection

HTHP installations shall be inaccessible to the public and generally be protected by a barrier such as a cage or secured fence.

3.18.5 Contamination of potable water supplies

Heated water systems shall be installed in accordance with AS/NZS 3500.4. Potable heated water systems shall comply with AS/NZS 4020, and are required to comply with one of the *Legionella* control options in G12/AS2 of the New Zealand Building Code.

3.18.6 Seismic design

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

3.19 Connectivity

3.19.1 Capability to connect

The HTHP shall have the technical capability to communicate with an external device, such as a building management system, and allow such a device to read and write parameters required for operation and monitoring. This PAS does not define the communication method and protocol that must be used; however, common industry examples are Modbus RTU, Modbus TCP/IP, and BACnet. This PAS also does not define the level of access available to specific users, but this is an important consideration (see 3.19.5).

This technical capability to communicate is not required to be enabled or active at the time of purchase or installation. However, the capability shall be fitted and available and able to be activated without need for significant changes to be made to either the HTHP or the controller.

3.19.2 Remote switching

The control system of the HTHP shall allow for safe remote switching (on and off) of the HTHP operation via the remote connectivity system. This may be implemented, at a minimum, as dry contacts, but may also include more comprehensive system integration. The remote switching capabilities should allow for the heat pump operation to be stopped remotely if it is running, or for the heat pump operation to be started if it is idle. This PAS does not define who should have access to remote switching, but this is an important consideration (see 3.19.5). The availability of remote switching may impact the electricity supply contracts the customer can negotiate.

3.19.3 Alarm reporting

The HTHP controller shall provide fault reporting or alarms to either an external monitoring system, such as a BMS, or to provide an audible or visual alert.

3.19.4 Remote monitoring

Heat-pump water heating systems shall support connection to remote monitoring services to allow for off-site service support and diagnostics. Connectivity should be of sufficient speed to allow the monitoring and have an uptime reliability of at least 95% over the lifetime of the HTHP. Ethernet and 4G (or later) platforms are likely to be sufficient, whereas a concern is that some Wi-Fi systems may not remain in place for the operating lifetime of the HTHP.

3.19.5 Connectivity security and access

Connectivity and remote access should be supported with a system that allows access to authorised users only. The system shall operate in accordance with accepted good practices to ensure that privacy, integrity, confidentiality, and security is maintained. This PAS does not define whom an authorised person might be, but this is an important consideration.

One possible implementation is to assign all users an individual user account and the edits made by each user on any system shall be recorded and retrievable by authorised persons. All communication between IT equipment on site and external IT infrastructure shall use predefined encryption techniques and keys or utilise a secure virtual private network (VPN) or structure providing an equivalent level of security. The system shall utilise a VPN or structure that provides an equivalent level of security for accessing the change of software and settings.

The company providing the remote monitoring and control system shall operate in accordance with good practice and in compliance with relevant security frameworks, guidance, and standards and have in place:

(a) A current business-continuity plan;

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

(c) A disaster-recovery plan and policy.

If data is transferred via mobile networks, the mobile data plan in place shall not expire and shall have adequate and accepted management in place during the contracted term. It is preferred that devices do not utilise the client's local area network (LAN) such that they provide a potential security door into the client's LAN. If devices use the client's LAN, clients shall be provided with guidance outlining the risks of such an approach.

Some HTHP control systems incorporate the capabilities to measure and either communicate or log performance data and energy usage data. This can be useful for assisting in quantifying the benefits of installing a HTHP.

The HTHP control system might also have the capability to measure and record certain values regarding the heat-pump operation that might assist with fault detection and diagnosis. Alternatively, the HTHP control system may have the capability to communicate such values to an external device such as a BMS for logging.

4 SPECIFICATION

4.1 Energy efficiency

4.1.1 Carnot coefficient of performance (COP_{Carnot})

The Carnot coefficient of performance (COP_{Carnot}) is a theoretical limit to the COP that depends upon the logarithmic mean temperature of the heat source ($\bar{T}_{source}$, °C) and the log mean temperature of the heat sink ($\bar{T}_{sink}$, °C). The Carnot COP cannot be achieved in practice but sets a useful benchmark that actual heat pump systems can be measured against.

$$COP_{Carnot} = \frac{\bar{T}_{sink}+273.15}{\bar{T}_{sink}-\bar{T}_{source}} \quad \text{(Eq. 5)}$$

Technically, the Carnot COP is for the special case when both the heat source and heat sink temperatures are constant. However, Equation 3 (with logarithmic mean temperatures for the heat source and heat sink) can be used for general applications when either the heat source or heat sink do change temperature, which is common in real world use of HTHP units.

The logarithmic mean temperature of a heat sink ($\bar{T}_{sink}$) is calculated as follows, when the heat sink enters the heat pump at an inlet temperature ($T_{sink,inlet}$, °C) and is heated to an exit temperature ($T_{sink,exit}$, °C):

$$\bar{T}_{sink} = \frac{T_{sink,exit}-T_{sink,inlet}}{\ln\left(\frac{T_{sink,exit}+273.15}{T_{sink,inlet}+273.15}\right)} - 273.15 \quad \text{(Eq. 6)}$$

The logarithmic mean temperature of a heat source ($\bar{T}_{source}$) is calculated as follows, when the heat source enters the heat pump at an inlet temperature ($T_{source,inlet}$, °C) and is cooled to an exit temperature ($T_{source,exit}$, °C):

$$\bar{T}_{source} = \frac{T_{source,exit}-T_{source,inlet}}{\ln\left(\frac{T_{source,exit}+273.15}{T_{source,inlet}+273.15}\right)} - 273.15 \quad \text{(Eq. 7)}$$

For example, to heat hot water from a temperature of 50°C to a temperature of 85°C using a heat pump with a heat source of ambient air with an inlet temperature of 20°C and of sufficient flow rate that the air temperature change is small (≤ 1°C).

The heat sink is the hot water. The log mean temperature of the hot water is:

$$\bar{T}_{sink} = \frac{T_{sink,exit}-T_{sink,inlet}}{\ln\left(\frac{T_{sink,exit}+273.15}{T_{sink,inlet}+273.15}\right)} - 273.15 = \frac{85-50}{\ln\left(\frac{85+273.15}{50+273.15}\right)} - 273.15 = \frac{35}{0.10284} - 273.15 = 67.2\text{°C} \quad \text{(Eq. 8)}$$

The heat source is the ambient air. As there is only a small change in the ambient air temperature the value of $\bar{T}_{source}$ is taken to be 20°C:

$$\bar{T}_{source} = 20.0\text{°C} \quad \text{(Eq. 9)}$$

So, the Carnot COP for these conditions is:

$$COP_{Carnot} = \frac{\bar{T}_{sink}+273.15}{\bar{T}_{sink}-\bar{T}_{source}} = \frac{67.2+273.15}{67.2-20.0} = 7.2 \quad \text{(Eq. 10)}$$

4.1.2 Thermodynamic efficiency relative to Carnot (η_{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}). It is a measure of how near to the theoretical maximum the real system is performing for the conditions.

$$\eta_{Carnot} = \frac{COP}{COP_{Carnot}} \qquad \text{(Eq. 11)}$$

The thermodynamic efficiency relative to Carnot should exceed 0.35 at all normal operating conditions.

4.2 Temperature lift (ΔT_{lift})

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

$$\Delta T_{lift} = \bar{T}_{sink} - \bar{T}_{source} \qquad \text{(Eq. 12)}$$

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

4.3 Seasonal energy efficiency ratio (SEER)

As the COP of a heat pump depends upon the operating conditions, the COP can be quite different for different times of the day or different times of the year. The required heating demand, the heating capacity of the installed heat pump, and the power requirement of the heat pump can all change with different operating conditions.

The seasonal energy efficiency ratio (SEER) is a measure of the annual average COP, weighted by heating capacity, heating requirements, and operating hours.

$$\mathrm{SEER} = \frac{annual\ E_{heating}}{annual\ E_{electrical}} \qquad \text{(Eq. 13)}$$

Where:

$E_{heating}$ = the total energy supplied to the heat sink by the high-temperature heat pump (kJ), and

$E_{electrical}$ = the total electrical energy consumed by the heat pump (kJ).

SEER values change with the units, so only compare values with the same units.

As a minimum, the COP and the thermodynamic efficiency relative to Carnot should be evaluated at the typical operating conditions for the application, as well as at the more extreme conditions used to size the capacity of the design.

The typical operating conditions should be used to evaluate the thermodynamic efficiency relative to Carnot (see 4.1.1). If the typical heating requirement is lower than the installed capacity, the COP should include accounting for performance at a part load, or the effect of other control strategies such as thermal storage.

4.4 Payback and reduction in emissions calculations

The payback and carbon emissions reduction relative to a fossil-fuel based heating system can be calculated.

The simple payback for a HTHP relative to a fossil-fuel fired heating system is given by:

$$\textit{simple payback (years)} = \frac{\textit{marginal capital cost } (\$)}{\textit{annual heating demand } (MJ)\left(\frac{\textit{cost of fuel}\left(\frac{\$}{MJ}\right)}{\eta_b} - \frac{\textit{cost of electricity}\left(\frac{\$}{MJ}\right)}{SEER}\right) - \textit{marginal other costs}\left(\frac{\$}{year}\right)} \quad \text{....(Eq. 14)}$$

Where the fuel and electricity costs include any carbon charges (for example, ETS charges). Other costs include labour, repairs, and maintenance. They may be either higher (positive) or lower (negative) for a HTHP relative to a fossil-fuel fired heating system.

The annual reduction in carbon emissions for a HTHP relative to a fossil-fuel fired heating system is given by:

$$\textit{annual } CO_2 \textit{ emission savings } \left(\frac{kg\ CO_2}{year}\right) = \textit{annual heating demand } (MJ)\left(\frac{\textit{fuel emission factor}\left(\frac{kg\ CO_2}{MJ}\right)}{\eta_b} - \frac{\textit{electricity emission factor}\left(\frac{kg\ CO_2}{MJ}\right)}{SEER}\right) \quad \text{...(Eq. 15)}$$

4.5 General information

4.5.1 Pipe size restrictions

The replacement of a central plant will constrain selection options due to existing secondary system piping sizes.

4.5.2 Whole-of-life analysis

A whole-of-life analysis shall be completed.

4.5.3 Performance standards

For space heating applications, the SEER of the HTHP unit shall be greater than 3.0 for all climate zones. For heated water applications the SEER shall be estimated by modelling using AS/NZS 4234.

4.5.4 Temperature differential

The modelled system under AS/NZS 4234 shall achieve an appropriate temperature differential across the HTHP when operating. The system design should maximise the opportunities for refrigerant subcooling.

4.5.5 Thermal storage

Thermal storage, where used, shall be thermally stratified to achieve an appropriate temperature differential across the HTHP when operating.

The capacity of thermal storage shall be sufficient to meet the heating demand for the application for a minimum of 30 minutes at the fifth percentile ambient condition for the location, during likely usage hours.

Where thermally stratified storage is integrated into a heating system, the storage tanks can be either distributed or centralised. All thermal storage shall be insulated with an insulation *k* value at least the equivalent of 50 mm of high-density polyurethane foam ➤

for tanks under 5000 L, and 100 mm of high density polyurethane foam for tanks greater than 5000 L in total capacity.

Where thermal storage is not integrated into a space heating system, heated water shall be prevented from passing through in-room heat-exchangers more than 60 minutes prior to occupation.

4.5.6 Fluid distribution

If present, the fluid (for example, hot water) distribution system shall be of the variable flow distribution system design. All fluid returning to the HTHP shall pass through a heat-exchanger (radiator, fan coil) prior to returning to be reheated except as noted in section 4.

4.5.7 Space heating minimum

When the application is space heating for occupancy, room temperatures during the fifth percentile ambient condition during likely occupied hours shall maintain a minimum of 20°C within 15 minutes of occupation.

4.5.8 Pump efficiency

Pumps, where needed, of greater than 200 W rate power consumption at maximum capacity shall have motor efficiencies greater than 70% (for example, EC motors). This also applies to pumps replaced as part of a retrofit or upgrade project. Pumps rated greater than 200 W shall operate under constant pressure control where centralised capacity control is not enabled.

4.5.9 Distribution fluid pipe insulation

All distribution fluid pipes shall be insulated with a minimum of 25 mm of closed cell, foil-faced (or equivalent) pipe insulation. Pipe insulation that is subject to degradation by sunlight shall not be used in exterior locations.

4.5.10 Supplementary heating

Where supplementary heating is installed to maintain room temperature during low ambient conditions, it is preferred that the energy source is from the water heating system. Where the heat source is the water heating system, the supplementary heating shall be sourced from the return water line back to the HTHP or storage tank. Supplementary heating shall not be connected to the supply water heating line. Supplementary heating shall be controlled to an in-room temperature sensor, and no less than one sensor per four rooms heated. Temperature averaging sensors are recommended.

4.5.11 Potable water heating

Potable water heating shall be separated from space heating so the space heating distribution network is not heated to provide localised DHW demand.

Potable water heating systems shall be of the single-pass design for capacities greater than 5 kW. If a ring-main is present, then water flow from the ring-main shall be heated separately to the water in the main storage tanks. Variable speed ring-main pumps shall be installed to control to a maximum water temperature of 50°C. Installations must still comply with *Legionella* control requirements stated in the New Zealand Building Code acceptable solution G12/AS2.

4.5.12 Load efficiency

Ensure that the efficiency of heating the largest heating demand is not compromised by operating at a higher temperature to meet a smaller heating load.

4.5.13 Standard requirements

Units shall be compliant with AS/NZS 60335.2.40 with a test report available on request.

Installations shall be compliant to NZS 4219. This includes fan coils within occupied areas, storage tanks, and roof-mounted HTHPs. Ground-mounted HTHPs shall be securely mounted. Condensate from evaporator coils shall be collected and drained to a suitable waste point with a minimum of 19 mm of tubing.

Modelling reports to AS/NZS 4234 shall be produced for each commercial application of HTHP. The modelled system shall take into account pumping energy used in the heated water distribution system, and operational controls logic employed within the system.

4.5.14 Test standards

Air-source HTHPs for water heating systems shall be tested to AS/NZS 5125. The methodology in AS/NZS 5125 can be adapted to commercial air and water-source HTHPs by increasing the size of the water tank in proportion to the heating capacity of the unit or by selecting water-source temperatures to match the application 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. Other systems should be tested to an equivalent testing methodology.

AS/NZS 5125 involves the measurement of the power input, and thermal energy produced by the unit when coupled to a storage tank, under a range of standardised test conditions.

AS/NZS 4234 shall be used to estimate the annual energy consumption and SEER. The performance coefficients determined by the analysis of the test results from AS/NZS 5125 (or equivalent test methodology) can be entered into a thermal simulation software package heat-pump model using AS/NZS 4234. When compared to a reference heating system (for example, a boiler), a comparison of energy use can be determined. By applying the carbon emission factor for the energy consumed by the system, the difference in greenhouse gas (GHG) emissions between two or more systems can be provided.

Variable speed HTHPs shall be tested in accordance with AS/NZS 5125 (or an equivalent methodology for non-air-source systems) with a locked compressor speed.

4.5.15 AS/NZS 5125 test modifications

For units with a rated capacity greater than 15 kW at ambient condition 7°C/6°C (DB/WB), and a water outlet temperature of greater than 50°C, the following modifications to Appendix A of AS/NZS 5125 may be used:

(a) Data collected from outdoor testing can be used providing that there is no direct or indirect radiation on the unit with a surface temperature greater than 5°C above the ambient temperature. Surface temperatures are determined using a calibrated infrared thermometer at a 1 m measurement distance;

(b) The ambient condition is to remain stable within 1°C of the nominal dry bulb during the test condition;

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

(d) The minimum ambient test condition tested is no greater 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) The maximum wind speed at the unit during the test shall be no more than 0.5 m/s both immediately preceding and following the test;

(g) The tank may be unpressurised;

(h) The minimum number of tank sensors is increased to ten.

The requirements on measurement accuracy in clause A4 (Appendix A) of AS/NZS 5125 still apply. The low ambient test condition shall be applied to HTHP, and the frost penalty determined as described in AS/NZS 5125.

The low ambient test condition includes the reduction in average heating capacity due to the accumulation of frost on the evaporator coils. Through simulating the energy performance of the system under non-frosting conditions and comparing it with the actual performance once defrosting cycles are considered, a frosting penalty can be determined. These low ambient test condition can present difficulties for a test laboratory to maintain throughout the test, especially with larger capacity units.

Where the frosting penalty cannot be determined, then a standardised penalty of 25% shall be applied from the lowest dry bulb temperature that doesn't result in frost that forms and persists for greater than 10 minutes with an inlet water temperature no greater than 15°C for potable water heating systems, and 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 that the suction line temperature will remain below –2°C. Intermediate speeds from those tested shall have the frosting penalty applied proportionally.

4.5.16 Metrics of refrigerant charge

Metrics of refrigerant charge shall be determined. This should be kilograms of total refrigerant charge per kilowatt of heating delivered and GWP of total refrigerant charge per kilowatt of heating delivered.

4.6 Commissioning

The equipment installer shall complete a commissioning report for the purchaser as part of the installation contract.

A plant performance report (by an independent person for plants > 50 kW) measuring capacity and COP over a range of conditions shall be provided as part of the equipment supply contract.

The commissioning report should record the temperature and flow rate through the heat exchangers, including any heat exchangers associated with thermal storage. These values should match the design flow rates of the system, as specified in the equipment supply contract. Commissioning of the system should cover as wide a range of operating conditions as the prevailing climate allows. Commissioning should be across a suitable range of conditions. For example for space heating this should include:

(a) An early morning start-up on a cold day for a minimum period of 3 hours or until equilibrium conditions are reached;

(b) A warm day with minimal heating demand; and

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

Complete commissioning and system tuning may be a process that takes several months to be completed after initial start-up and handoff. Remote monitoring can facilitate this process.

The commissioning report should:

(d) Determine that the flow rate through the heat-exchangers matches the design flow rates of the system, as specified by the manufacturer;

(e) Record the flow rate through the heat-exchangers at the time of system commissioning;

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

(g) Determine that the flow rates through the heating system are within the specifications of the system design;

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

(i) Record the condensing temperature (or discharge pressure), evaporating temperature (or suction pressure), and capacity of the compressor after three hours of operation and reference it against the operating envelope of the compressor provided by the compressor manufacturer. The operating conditions of the system shall be within the compressor envelope provided, and including conditional limits, such as hours of operation;

(j) Determine that the temperature differential across the storage tank is at the minimum of the design value after 3 hours of full capacity operation of the heating system. Where the system design allows the unit to operate simultaneously with heating demand, the tank shall maintain the design thermal differential during both unit operation and full capacity heating operation;

(k) Measure the capacity, power input, and COP of the HTHP during unit commissioning after 3 hours of heat system operation, and reference against data supplied by the manufacturer;

(l) Observe and report on the staging of the compressors under part-load heating conditions where multiple compressor systems are installed. The minimum and maximum stated compressor speeds stated by the manufacturer shall be observed during commissioning;

(m) Data from the monitoring system shall be used to demonstrate that the control logic for the system is operating to the stated design during the first month of the heating season;

(n) Ensure the operation of the potable water heating system, if present, is separated from the operation of the space heating system;

(o) Ensure operation of the potable water heating system, if present, does not compromise the efficiency of the space heating system;

(p) Ensure potable hot water systems, if present, meet at least one of the *Legionella* control options described in G12/AS2.

A 1-month, 3-month, and 6-monthly report on the operation of the system shall be provided. This can be generated using remote connectivity if enabled and shall include:

(q) The maximum and average supply water temperature achieved during HTHP operation, and during operation of the heating system;

(r) The average monthly daily run hours, weekly electricity consumption and average monthly COP during HTHP operation;

(s) A fault report.

If post-commissioning adjustments are made to the operation of the HTHP or heating system, they shall be detailed in the following report.

4.7 Carbon emissions

The annual energy use and the SEER can be estimated by modelling the water heating system, including the HTHP, using an adaption of the methodology described in AS/NZS 4234. Through the modelling of the reference system, a calculation of the difference in energy consumption can be determined. Applying emission factors to both estimates of annual energy use allows the impact on carbon emissions to be determined (emission factors published by the Ministry for the Environment should be used).

4.8 Daily heating demand

A key difference in applying AS/NZS 4234 to HTHP system is that the daily heating demand profile needs to be estimated as a function of time of day and climate conditions (rather than the water-draw profiles defined in AS/NZS 4234). The pump energy and energy losses in any thermal storage and distribution system shall be taken into account, and the actual control logic that the HTHP employs shall be included.

4.9 Potable water

Potable hot water systems shall be tested to AS/NZS 5125 and AS/NZS 4234 where the water-draw profile chosen is a close match to the actual draw pattern.

NOTE – There is likely to be higher draw flow and different timings for commercial building than for domestic residences.

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