REPORT

- ***Preliminary final version –***

Can refrigerants with a GWP below 150 be used for Heat Pumps in Europe?

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Brief summary

The viability of applying alternative refrigerants with a GWP < 150 to domestic heat pumps (DHPs) with nominal heating capacities below 30 kW is considered. The assessment reflects the proposed revision to the European F-gas regulation. DHPs currently use a variety of refrigerants, primarily R410A, but also R32, R407C, R404A, R417A, R454B, R454C, R452B, R448A, R449A, amongst others, all of which are medium or high GWP. A number of models already use R290, whilst other alternative refrigerants with GWP < 150 under consideration are R1270, R1234yf and R152a; all of which are flammable. These low GWP alternatives are assessed, relative to R410A as baseline and R32 as leading medium GWP refrigerant.

Using the Eurovent and Keymark databases and extending them to include additional parameters such as refrigerant charge, physical size and weight and retail price, general characteristics of these products are mapped out with particular attention to the parameters that may be affected by refrigerant choice. To assist with the analysis and interpretation of requirements, the various types of DHPs are grouped according to two main categories: air-to-water (ATW) and liquid-to-water (LTW), which may also be installed indoors or outdoors. Domestic hot water (only) and split air-to-air type DHPs are not considered (see air conditioner report) and for brevity, niche products such as other split type DHPs, exhaust-air, etc. are not included.

From these data, information on currently available R290 DHPs and results from heat exchanger simulations and other system components, refrigerant charge for R1270, R290, R1234yf and R152a is estimated for a range of nominal heating capacities from 3 – 30 kW and with a high seasonal coefficient of performance (SCOP) of 6.5.

Considering the capacity range, flammable refrigerant charge limits specified within the revised safety standard, IEC 60335-2-40: 2022, along with cost and practical implications of applying appropriate risk mitigation concepts (RMCs) provides a framework for determining the applicability of DHPs using alternative refrigerants with GWP < 150. Differences in materials and other inputs are used to determine implications of cost, equipment size and lifetime carbon dioxide equivalent (CO_2e) emissions.

The main findings are:

- For the entire DHPs capacity range, all seasonal efficiency levels (to at least SCOP = 6.5 at 35°C water supply temperature), all the low GWP alternative refrigerants can be applied within the charge limits of the standard. With capacities above a few kW, R1270, R290 and R152a require additional RMCs, although these are also required for larger capacity systems using R32 or R1234yf.
- To achieve the same seasonal efficiency as R410A, additional heat exchanger materials are required for R152a and particularly R1234yf. Compared to R410A, quantity of materials required for R32, R1270 and R290 are similar. The cost implication is relatively small, in the order of a few Euros. For DHPs located outdoors, there is almost no cost impact for addressing safety requirements. DHPs installed indoors have a choice of different RMCs, the cost impact of which ranges from negligible to several tens of Euros. Same applies to lifetime energy costs for some RMCs and to R32 and R1234yf for larger capacity DHPs.
- Emissions associated with use of the additional materials necessary for refrigerants with GWP < 150 are negligible, especially when compared with those associated with the lifetime emissions from refrigerants with GWP > 150. Overall, significant benefits for emissions reduction are seen when switching from R410A (or from R32) to any of the GWP < 150 alternatives.
- Depending upon the choice of RMC, cost effectiveness of switching to R1270 and R290 is favourable, i.e., emissions reductions are achieved in parallel with lifetime cost reductions, particularly when the cost of refrigerant is taken into account. R152a usually mirrors R1270 and R290, but since application of R1234yf results is a cost increase, cost-effectiveness is usually around €20 – 30 per tCO_2e.
- Discussions with manufacturers and various literature sources indicate a variety of developments related to charge reduction and performance improvements, particularly with R290. Provided sufficient time is available, adoption of such improvements as well as general technology refinements, future iterations of alternative refrigerant DHPs will help further improve their cost-effective applicability.

The database, simulations and other sources from which the information was gathered is extensive. Treatment and analysis were thorough, usually being approached both practically (break-down of real equipment data) and theoretically (upward simulation of components) to confirm consistent findings. Inputs with notable uncertainties employed reasonably pessimistic assumptions so as to disadvantage GWP <150 alternatives. Accordingly, results and conclusions are considered to be robust.

Main summary

1 Introduction

- The viability of applying selected refrigerants with a GWP < 150 to domestic heat pumps (DHPs) with nominal heating capacities between 3 – 30 kW is considered. The assessment reflects the proposed revision to the European F-gas regulation, which intends to prohibit refrigerants with GWP>150 in HPs.
- DHPs currently use a variety of refrigerants, primarily R410A, but also R32, R407C, R404A, R417A, R454B, R454C, R452B, R448A, R449A, amongst others, all of which are medium or high GWP. A number of models already use R290, whilst other alternative refrigerants with GWP < 150 under consideration are R1270, R1234yf and R152a; all of which are flammable. These low GWP alternatives are assessed, relative to R410A as baseline and R32 as the leading medium GWP refrigerant.
- Unlike other types of refrigeration and air conditioning equipment, DHPs are produced in a variety of differing forms, including monoblock and split, located indoors, outdoors or both, ground-source, ground-to-liquid-source, air-source and air and water sinks, which may be for space heating and/or domestic hot water, all at differing temperatures. To simplify the assessment, generic air-to-water (ATW) and liquid-to-water (LTW) types – which may also be installed indoors or outdoors – are considered. Domestic hot water and split air-to-air type DHPs are not considered (see reversible air conditioner report) and for brevity, niche products such as other split type DHPs, exhaust-air, etc. are not included.
- Application of these alternatives is evaluated with regards to the use of flammable refrigerants within the applicable safety standard for air conditioners and heat pumps, IEC 60335-2-40: 2022 and the Ecodesign regulation for space heating (including DHPs), which specifies minimum efficiency levels.
- The criteria against which the viability is assessed are:
  - Product safety, through compliance with safety standards, primarily related to refrigerant charge limits and risk mitigation concepts (RMCs), necessary for DHPs.
  - Achieving current and anticipated future minimum efficiency levels and preferably, higher efficiency levels under energy labelling rules.
  - Cost for the consumer, through impact on material (including refrigerant) costs and whether adoption of the alternatives would result in an adverse increase in cost.
  - Environmental impact, through assessment of carbon dioxide equivalent (CO_2e) emissions, which should also account for the emissions associated with production of DHP materials, refrigerants and energy use arising from implementation of RMCs.
- The assessment is carried out through the use of a publicly-available DHP databases (Eurovent and Keymark) which were extended to include refrigerant charge, product mass and dimensions and pricing. Aspects assessed include performance analysis of the refrigerant-DHP combinations considering heat exchanger and piping design, quantification of mass and dimensions of system components, costs and associated CO_2e emissions.

2 Flammability safety requirements

- Flammability safety is addressed through adhering to industry guidance, compliance to safety standards and conformity to the applicable European directives and regulations.

- The product safety standard, IEC/EN 60335-2-40: 2022 specifies flammable refrigerant charge limits and provides a variety of additional measures, risk mitigation concepts (RMCs) to enable the safe application of flammable refrigerant in equipment such as DHPs.
- Provided the additional mitigation measures are adopted, sufficiently large refrigerant quantities are permitted to allow DHPs to provide nominal heating capacities (NHC) across the desired range using refrigerants with GWP < 150, i.e., R1270, R1234yf and R152a.

3 Heating loads

- Various studies and industry sources were used to identify appropriate values for across Europe.
- However, for the ATW and LTW type DHPs, the thermal load (in relation to size of the heated space) does not have any bearing on the selection and application of alternative refrigerants.

4 DHP performance and product data

- To provide DHP baseline data the Eurovent and Keymark databases were used, which include NHC, SCOP and heating capacities and efficiencies at different conditions and various other values. The databases were extended to include refrigerant charge quantities, mass and volume of the DHPs and retail price. Eurovent data only includes R410A, R407C and R32 models and only R290 models were extracted from the Keymark database (i.e., about 420 from the total 5300 listed models).
- This amounted to about 2150 DHPs, excluding air-to-air, exhaust air and split types. Analysis of the data indicates that across all these models, there are only about 150 – 200 different refrigerating systems. Thus, numerous models only differ by colour/ packaging, functional features and so on.
- Seasonal coefficient of performance (SCOP) values for "average climate" and water supply temperature of 35°C, range from 3.0 to 5.5 for both ATW and LTW types.
- Many models also have cooling (reversible) function, but this is not considered further.
- Broadly, DHPs total volume and mass increase as NHC does, although this is confused by the huge amount of scatter. This is largely affected by inclusion (or not) of supplementary heating (electric or gas boiler), hot water tank(s) unusual housing shape designs (for aesthetics) and so on. Thus, it is not easy to observe reliable trends to draw conclusions. However, for many models this indicates that size and weight constraints are not given high priority.
- Refrigerant charge roughly rises with higher NHC, although, for a given capacity the highest charge quantity can be up to eight times that of the lowest charge quantity, although in most cases this can be due to inclusion of direct domestic hot water heater tanks, etc. Nevertheless, this is a strong indication that little regard is given to charge minimisation and that significant reductions are feasible in most cases. For a given NHC, there is no clear variation in charge across the range of SCOP.
- Retail price is also seen to have considerable variation, by up to a factor of five at the same NHC. There is no discernible influence of SCOP on the price and there is not even a correlation between price and total material mass. It is not possible to see whether refrigerant selection has an influence on price.

5 Mandated efficiency levels

- Current minimum efficiency is SCOP = 2.95 to 3.33 (depending upon the DHP temperature level) with highest efficiency label set to SCOP >3.95 to 4.58. A large number of DHPs exceed this highest efficiency label level, including a greater proportion of R290 models.

- Proposed revised minimum efficiency is SCOP = 2.9 to 3.42 with highest energy efficiency level set to SCOP = 3.95 to 4.89.

6 Efficiency and alternative refrigerants

- As a first step, alternative refrigerants are assessed with regards to their thermodynamic cycle efficiency. This cycle efficiency represents a theoretically ideal case in absence of thermal and other energetic losses, representing the potential performance that a refrigerant could achieve in absence of practical and economic constrains.
- At selected conditions, R410A, R32 and R1234yf exhibit lowest cycle efficiencies, R290 and R1270 and have about 4 – 5% higher and R152a about 9% higher COP. R290 and R1270 have about 7 – 9% lower compression ratio – which is associated with higher compressor efficiency – than the other refrigerants. R290, R1270 and R1234yf also have notably lower compressor discharge temperatures, which are usually favoured for compressor and system longevity.
- R1270 and R290 have higher suction swept volume than R410A and R32 and R1234yf and R152a higher values still. Swept volume is an indicate of circuit volume flow rates and thus potential flow velocities and circuit pressure drops (although fluid viscosity must also be considered). Assuming the same component (piping, heat exchangers, etc.) sizes, a higher swept volume infers greater pressure drop, leading to a higher rate of efficiency degradation from the ideal thermodynamic cycle. Alternatively, larger (which may or may not be more costly) components are used to negate the higher pressure drop.
- For a specified system efficiency (for all refrigerants), tolerable condenser- and evaporator-side losses are determined for which heat exchanger selections and interconnecting pipe sizes are made. Broadly, when targeting high system efficiency according to the proposed revised Ecodesign rules, R1270, R290 and R1234yf can tolerate about 0.5 K and R152a, 1.0 K larger condenser and evaporator temperature differences compared to R410A and R32.

7 Approximation of charge amounts

- Useful for the assessment is an approximation of required refrigerant charge for DHPs using alternative refrigerants.
- Whilst theoretically, it is supposed that higher efficiency DHPs would demand greater refrigerant charge, the data for all DHPs show huge scatter, implying that (at least at present) many systems are not well charge-optimised. This is likely due to the unrestrictive charge limits in safety standards and the relatively low cost of currently-used refrigerants.
- A simple correlation for lower/lowest specific charge as a function of NHC and SCOP is developed from the database. Adjustments are included according to refrigerant density and a weighting related to the importance of charge reduction. Heat exchanger and other component working charges from simulations are used to cross-check refrigerant charge across the various alternatives.
- It is observed from recent literature that R290 heat pumps have been developed with substantially lower specific charges than what the correlation yields, suggesting that any conclusions drawn from these charge amounts err on the pessimistic.

8 DHPs charge requirements in relation to charge limits

- Charge quantities for all models from the databases and estimated values for notably higher SCOPs are below the upper charge limits (UCL) specified within IEC 60335-2-40 for all of the alternative refrigerants, indicating all options are feasible.
- However, except for smaller NHCs with R1270, R290 and R152a and medium to large capacities for R32 and R1234yf, additional RMCs need to be implemented. These RMCs either need to limit the releasable charge to quantities small enough to avoid the need for a large minimum room area (6 – 10 m^2, the size of a utility room or cellar) or employ a ventilated enclosure (VE) where leaks within the DHP housing are exhausted to the outside.

9 Component material mass and costs

- With refrigerants have differing thermophysical properties, each have distinctive potential impacts on material requirements. Based on a high efficiency 10 kW DHP, main system component material requirements are approximated using simulations and catalogue data, as appropriate, with regards to the target efficiency, corresponding temperature differences and associated pressure drops. Costs arising from research and development, modifications to production, etc. are not accounted for, partly because of the wide variability and partly due to them diminishing over time.
- For system piping, diameter is determined according to a fixed change in saturated refrigerant temperature, corresponding to 0.1 K per metre; thus, those with higher viscosity and lower saturation pressure usually necessitate a larger diameter and greater material mass, although conversely, wall thickness can be thinner for lower pressure refrigerants in turn reducing or offsetting the mass. Material requirements for R410A, R32 and R1234yf are similar, whereas R1270, R290 and R152a potentially require about 20 – 40% less copper and corresponding cost.
- Analysis of catalogue data for rotary, reciprocating and scroll compressors shows that material mass, primarily steel, is almost identical for a given NHC, irrespective of refrigerant. Similar trade-offs between volume and wall thickness observed with piping similarly apply to compressors. There is no evidence to suggest that there should be any difference in compressor cost amongst the alternative refrigerants.
- Performance of numerous evaporators and condensers designs are simulated and the most appropriate are selected based on target temperature differences and tolerable pressure drops to ensure design capacity with minimum material cost and refrigerant mass, as appropriate. For finned-tube heat exchangers (HX) and brazed plate HX, R32, R410A, R1270 and R290 require least mass, followed by R152a and R1234yf demanding most materials. The difference between R1234yf and R32 is almost 50%.
- Considering total component cost of the HX, i.e., including cost for the operating refrigerant mass, skews the ranking, but nevertheless amplifies the difference between R32 and R1234yf; doubling the differences for mass alone.
- Examination of materials and costs for implementing the RMCs are quantified, primarily active limited releasable charge (ALRC), integrated airflow (IAF) and VE. ALRC and IAF are found to invoke a notable cost, that under some circumstances can cancel out the cost benefit from lower refrigerant cost for R1270, R290 and R152a, whilst the impact for VE is much less. For DHPs installed outdoors, these costs do not apply.

- Total incremental cost, also including for shipping and storage of slightly larger physically sized DHPs, indicates R410A, R32, R1270, R290 and R152a within ±€20 to 30 of each other, depending upon the type of DHP. Once refrigerant cost is taken into account, R32, R1270 and R290 have least cost, then R410A and R152a. In contrast is R1234yf, for which the incremental cost is over €150. However, over time the incremental costs for GWP < 150 will become smaller as the "GWP premium" for medium and high GWP refrigerants escalates.
- Compared to the average retail price of a 10 kW DHP, the incremental cost of all alternatives is usually well within ±1%, except with one costly RMCs and almost any case with R1234yf.

10 Greenhouse gas emissions

- CO_2e emissions are calculated according to the differences in materials mass from system construction associated with each alternative refrigerant and the refrigerant emissions themselves during the assumed lifetime of the DHP. Conventional assumptions are used for the latter. Emissions arising from energy consumption are negated since DHPs for all alternatives are designed for the same seasonal efficiency. However, energy related emissions due to RMCs (such as fan motors and solenoid valves) are included where applicable.
- Emissions associated solely with the construction materials (including RMCs) are generally around 150 to 250 kgCO_2e, although for R1234yf exceed 350 kgCO_2e. Incremental material emissions (excluding refrigerant) associated with the choice of refrigerant are within ±20 kgCO_2e for R410A, R32, R1270, R290 and R152a, with R1270 and R290 being the least and R410A the highest, except R1234yf which is responsible for an additional 50 – 100 kgCO2e.
- Considering the effect of medium and high GWP refrigerant emissions during in-use and at end of life, the contribution of construction materials is negligible by comparison. R32 contributes about 1300 kgCO_2e and R410A, about 4100 kgCO_2e, between 7 and 20 times the production emissions. Using a 20 y GWP significantly exaggerates this, doubling the value for R410A and tripling the value for R32 (and R152a). Total emissions for R1270, R290 and R234yf are unaffected by the choice of GWP time horizon.
- Considering the incremental costs with respect to emission reductions, all the alternatives show attractive cost-effectiveness (regardless of whether R410A or R32 are used as the baseline). R1270, R290 and R152a are almost always below €0/tCO_2e (i.e., infinite). However, R1234yf has values of about €20 – 30/tCO_2e, being relatively expensive with respect to credits under the European carbon Trading Scheme.
- Except with the case on one of the RMCs, the use of R1270, R290 or R152a always lead to significant emissions reductions below R410A (and R32) along with negative incremental costs.

11 Further considerations

- Other considerations associated with the application of alternative refrigerants should also be considered in light of the introduction of those with GWP < 150.
- The extended operating envelope for R1270 and R290 compressors offer additional benefits for DHPs in terms of functioning under a wider range of conditions and also reducing reliance upon supplementary (gas or electric) heating.
- A review of recent articles covering improvement of energy efficiency and charge reduction was carried out, to provide an impression of possible near-term developments. These studies indicate significant

improvements are imminent and that the “highest efficiency for least charge” boundary is still some way off.

- Whilst not easily quantifiable, a crucial issue pertinent to the discussion is that of development duration. Although revised designs of DHPs using alternative refrigerants, including R1270 and R290, can be realised relatively quickly (a year or two), availability of compressors and manufacturer internal expertise on the application of alternative refrigerants can impose a hindrance. Since there are already some 400 – 500 R290 models on the market, it appears that there is already adequate options available. Arguably, the main hurdle for DHP manufacturers is the allocation of development resources for converting existing HFC models to low GWP, whilst trying to accommodate the high demand anticipated over the next decade or so.
- Similarly, concerns arise over the development time caused by the extensive number of test conditions necessary and the possibility for third-party verification tests.

12 Concluding remarks

- The key technical viability criteria for adopting refrigerants with GWP < 150 are easily satisfied across the cases examined. Specifically:
  - the desired efficiency levels can be achieved;
  - (flammable) refrigerants can be applied within the constraints of the applicable safety standards and regulations;
  - lifetime CO_2e emissions are sufficiently below the levels arising from the use of medium or high GWP refrigerants (considering uncertainty ranges);
  - the incremental costs do not adversely affect the product price and are within tolerable limits of additional expenditure.
- R1270 and R290 offer the greatest advantages in terms of minimal (and negative) incremental costs and emissions reduction. R152a also offers benefits over R1234yf and R32.
- When selecting any flammable refrigerant, an important criterion is considered choice of RMC(s) so as to minimise first cost and lifetime energy consumption. Some can offset cost benefits and some emissions reduction achieved through the refrigerant selection, whilst other RMCs can have imperceptible impacts on costs and emissions.
- In terms of greenhouse gases, such a switch reduces lifetime emissions regardless of which GWP < 150 alternative refrigerant is selected. Usually, emissions reductions are achieved in parallel with lifetime cost reductions. Quantifying emissions with GWP based on 20 y ITH amplifies the climate benefits by a factor of two or three.
- The database, simulations and other sources from which the information was gathered is extensive and the treatment and analysis was thorough and approached with different methodologies. Furthermore, where there is uncertainty associated with any inputs, pessimistic assumptions were used. Thus, the results and conclusions are considered to have a high level of confidence.
- If the market for alternatives with GWP < 150 is as extensive as the current one for high and medium GWP refrigerants, the accompanying increase in research and development would lead to significant technological advances, providing further improvements in cost-effectiveness.

- Although not addressed in detail, the societal implications of switching to low GWP – such as hindering learning curve, slow research and development and insufficient competent technicians – are also deemed to be largely unsubstantiated.
- In conclusion, from a technical (and non-technical) perspective there is no rational justification for retaining high or medium GWP alternatives in favour of GWP < 150.

Nomenclature

Abbreviations

ACL	allowable charge limit
ALRC	active limited releasable charge
ATEX	ATmosphères EXplosibles (directive)
ATW	air-to-water
COP	coefficient of performance
DHP	domestic heat pump
DHW	domestic hot water
ETD	effective temperature difference
ETRS	enhanced tightness refrigeration system
EU	European Union
F-gas	fluorinated gas
GWP	global warming potential
HC	hydrocarbon refrigerant
HFC	hydrofluorocarbon
HP	heat pump
HX	heat exchanger
IAF	integral airflow
IEC	International Electrotechnical Commission
ITH	integration time horizon
LFL	lower flammability limit
LTW	liquid-to-water
NHC	nominal heating capacity
PLRC	passive limited releasable charge
RMC	risk mitigation concept
SCOP	seasonal coefficient of performance
SLHX	suction-liquid heat exchanger
UCL	upper charge limit
VE	ventilated enclosure

Symbols

a	a constant
A_{rm}	room floor area [m^2]

A_o	air discharge opening area [m^2]
A_p	area of heat exchanger plate [m^2]
b	a constant
c	a constant
CC	coefficient for electricity generation efficiency [?]
C_f	floor concentration [kg m^{-3}]
F	concentration factor [-]
g	a constant
G	a “correction” intended to account for electrical power consumption [-]
h_0	installation height of lowest refrigerant-containing parts [m]
h_{rm}	room height [m]
L	additional refrigerant piping length [m]
LFL	lower flammability limit by mass [kg m^{-3}]
m	an index
M	molar mass [kmol/kg]
m_{ACL}	allowable charge limit of refrigerant [kg]
m_C	system refrigerant charge [kg]
m_{hx}	mass of heat exchanger material [kg]
$\dot{m}_{leak}$	assumed refrigerant leak rate [kg/s]
m_p	mass of heat exchanger plate [kg]
m_{RC}	releasable refrigerant charge [kg]
m_{UCL}	upper charge limit of refrigerant [kg]
n	an index
N_p	number of heat exchanger plates [-]
Q_c	nominal heating capacity [kW]
$SCOP$	seasonal coefficient of performance [-]
$T_{f,in}$	fluid (air or liquid) inlet temperature [°C]
$T_{f,out}$	fluid (air or liquid) outlet temperature [°C]
T_d	discharge temperature [°C]
$T_{d,sat}$	saturated discharge temperature [°C]
$T_{s,sat}$	saturated suction temperature [°C]
V_{rm}	room volume [m^3]

$\dot{V}_{o,min}$	minimum volume airflow rate [m^3/s]
Δp	pressure drop as refrigerant saturation temperature [K]
$\Delta T_{c,eff}$	condenser effective temperature difference [K]
$\Delta T_{e,eff}$	evaporator effective temperature difference [K]
η_c	heating efficiency [-]
μ_c	specific refrigerant charge [kg/kW]
ρ_l	density of saturated liquid refrigerant [kg/m3]
ψ	charge matching coadjuvant [-]
ϑ	retained charge coefficient [-]

1 Introduction

1.1 Background

Hydrocarbon refrigerants (HC), such as R290, have excellent performance and negligible global warming potential (GWP). Within the context of the Kigali Amendment of the Montreal Protocol and national and regional legislation on fluorinated fluids, such as the European “F-gas” regulation, there is a desire for reducing the use of hydrofluorocarbons (HFCs). Within this context, HCs are a potentially viable substitute for many applications.

Significant growth in the deployment and use of domestic heat pumps (DHPs) is expected over the next decades as a means to reduce reliance on oil and gas and carbon dioxide emissions through substituting fossil fuel heating with electricity from renewable sources. For example, the number of DHPs is expected to more than quadruple by 2030 (IEA, 2020). Most DHPs use HFCs with GWP, thus leading to significant CO2e emissions over the 20 – 30-year lifetime of the DHP systems.

Some studies as far back as the 1950s proposed to use hydrocarbon refrigerants (HCs) in heat pumps (HPs) (Haselden and Klimek, 1957); Klimek, 1959; Schnitzer, 1983). Following the advent of the Montreal Protocol, there were a large number of studies investigating and proposing the use of HCs – primarily R290 – in HPs (e.g., Bivens, 1991; Eggen, et al., 1994; Frehn, 1993a; Frehn, 1993b; Granryd et al., 1994). About the same time, several manufacturers began producing R290 models to the European market. However, whilst they were mainly using R22 compressors for R290 systems, the introduction of the European Pressure Equipment Directive (PED) in 1997 hindered this development since it meant that compressors had to be certified for the refrigerant they were being used for and compressor manufacturers did not pursue compliance. As a result, the growth in HC HPs was curtailed and HFCs became the dominant refrigerants, although as shown in Palm (2008), several manufacturers persisted with certain DHP models. Currently (2022), there are in excess of 130 models that use HCs, which, although is less than 3% of all products, represents around 20% of all models using low- or medium-GWP refrigerants (Oltersdorf et al., 2022). These are mostly air-to-water (ATW) “monoblock” and a small number of liquid-to-water (LTW) monoblock systems.

The application of DHPs is also governed by the European Ecodesign regulation on space heating[1], which specifies minimum efficiency levels, according to type and operating conditions. Relevant performance test standards (EN 14511 and EN 14825) are invoked for demonstrating performance of the DHP.

Unlike with split air conditioners, the safety standard IEC 60335-2-40 does not necessarily pose obstructive requirements, although arguably the narrow choice of safety concepts may have restricted the development of more imaginative approaches. The recently approved revision of this standard, though, does offer some benefits for easier applications of HCs in DHPs. The reasons for the small proportion of models using HCs are varied, but include avoidance of (perceived) additional costs to implement safety features, availability of suitable compressors, lack of internal expertise, hesitance to work with higher flammability refrigerants (possibly leading to litigation or reputational damage), company internal philosophies and peer pressure amongst companies and trade bodies.

[1] Commission Regulation (EU) No 811/2013 of 18 February 2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of space heaters, combination heaters, packages of space heater, temperature control and solar device and packages of combination heater, temperature control and solar device Commission Regulation (EU) No 813/2013 of 2 August 2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters

The proposed European F-gas regulation[2] includes new placing on the market prohibitions, including for stationary split air-conditioning and split heat pump equipment and "plug-in room and other self-contained air conditioning and heat pump equipment that contain fluorinated greenhouse gases with GWP of 150 or more". This prohibition is proposed to apply from 1st January 2025. Some industry lobby groups have unsurprisingly objected to such elements of the proposal, arguing that there are inadequate alternatives or preparedness for such sub-sectors.[3, 4]

There are several alternative refrigerants that may be considered to address this prohibition. Whilst there are several pure HCs suitable as refrigerants and numerous blends, those of primary interest are R290 and R1270. Also, with relatively low GWP are R152a and R1234yf, which are also considered. All of these alternatives are flammable (class A3, A2 or A2L). GWP, safety class and lower flammability limit (LFL) (ISO 817: 2014) of these refrigerants, as well as those currently widely used for DHPs (R410A and R32) are listed in Table 1. GWPs for each are listed, both for a 100 y and 20 y integration time horizon (ITH) and including those from the IPCC 4th Assessment report (AR4) as used for emissions accounting and the latest 6th Assessment report (AR6). Other high GWP (R134a, R404A, R407C and R417A, R448A, R449A) and medium GWP refrigerants (R452B, R454C and R513A) are not considered.

In addition to the above, R744 (with GWP = 1) is also used in HPs, although due to the characteristics of the necessary transcritical cycle, competitive efficiencies are generally mainly achieved when heating to a high temperature from a low initial temperature, such as is typically required for domestic hot water (DHW) and similar applications (Rony et al., 2019). As such there is a fairly extensive range of domestic, commercial and industrial HP systems dedicated to hot water production (e.g., food and beverage processes, hospitals, leisure facilities, and so on). Recently, there have been a small number of products made available that also supply space heating as well as DHW, mainly targeted for retrofitting of systems using conventional gas boilers[5]. Due to the emerging status and different drivers for adoption of the technology, R744 will not be addressed further.

Table 1: GWPs, safety classification and LFL of refrigerants under consideration

GWP	R410A	R32	R1270	R290	R1234yf	R152a
GWP (100 y) †	2088	675	[not listed]	[not listed]	0.5	124
GWP (20 y) †	4340	2330	[not listed]	[not listed]	2	591
GWP (100 y) ‡	2256	771	<0.1	<0.1	0.1	164
GWP (20 y) ‡	4715	2690	<0.1	<0.1	0.5	591
Safety class	A1	A2L	A3	A3	A2L	A2
LFL [kg/m^3]	–	0.307	0.046	0.038	0.297	0.130

† IPCC 4th Assessment report, AR4 (Chapter 2; Forster et al., 2007)
‡ IPCC 6th Assessment report, AR6 (Chapter 7; Forster et al., 2021)

The objective of the present study is to identify whether such a proposal for the F-gas regulation is viable or not. Within this context "viability" is broadly a subjective concept. On one hand, the proposal is viable

[2] https://ec.europa.eu/commission/presscorner/detail/en/IP_22_2189
[3] https://www.fluorocarbons.org/news/trade-associations-respond-to-f-gas-proposal/
[4] https://epeeglobal.org/wp-content/uploads/2022/04/Joint-industry-Press-Release-on-F-gas-Regulation-proposal-5-April-2022.pdf
[5] E.g., https://group.vattenfall.com/uk/newsroom/pressreleases/2022/vattenfall-launches-heat-pump-solution-to-replace-gas-boilers

because technically, all DHPs can still be provided with or without F-gases (with a GWP < 150), but there may be cost, environmental and convenience implications. On the other hand, the proposal may be regarded as not viable, since it could cause at least some disruption (however large or small) to the DHP sector, maybe leading to higher costs, less convenience, fewer sales and loss of profit. Therefore, some tangible criteria must be applied.

1.2 Viability criteria

Viability criteria may be based upon the financial, safety and "inconvenience" impact to stakeholders throughout the producer-to-user chain. If the viability is to be assessed, it must involve defined and quantifiable parameters. In this regard, the following criteria are considered.

Extent of required R&D for manufacturers

DHPs are continually under a revision cycle and (another) change of refrigerant would be combined with such an R&D process. Further, developments are not carried out solely related to one aspect (such as refrigerant); improvements addressing efficiency, reliability, noise, cost reduction, functionality, etc., are carried out simultaneously. It is not practical or realistic to isolate this one aspect from other R&D activities and therefore the general implications associated with integration of an alternative refrigerant with GWP < 150 cannot easily be isolated.

Satisfying minimum efficiency rules

As the working fluid within the refrigerating system, the refrigerant has an influence on the efficiency that the system achieves. Amongst numerous other products, regional legislation prescribes the minimum efficiency that DHPs must provide and in addition energy labelling identify levels of efficiency that equipment (manufacturers) should aspire to. Adoption of any alternative refrigerant must not preclude the DHP from attaining the highest level of efficiency and beyond.

Inconvenience for the supply-chain

Manifestation of inconvenience if often cited as a barrier to implementation of alternative refrigerants. However, some stakeholders have argued that the additional training, knowledge, awareness, etc. of suppliers and installers can be beneficial (e.g., Rajadhyaksha et al., 2015). For example, higher levels of competence during selection and installation, more precise matching of DHP to the application, better servicing, etc. is more favourable to the end user and improves product reputation. Conversely, there may be instances where it is not feasible for the proposed DHP product to be installed (e.g., when using a large charge of higher flammability refrigerant), thus causing consternation and inconvenience to the seller and end user.

Product safety

Product safety is one of the key reported concerns associated with the adoption of flammable and higher flammability refrigerants. Accordingly, application is to be considered with respect to the new safety standard, IEC 60335-2-40. However, it may be noted that despite more tolerant requirements than previous editions of the standard, the degree of stringency imposed by the standard is beyond what flammability risk assessments may deem necessary.

Competency of technicians

Although a portion of the DHP installer base may be less well trained and often conduct activities in a "slap-dash" manner, there is an increasing importance for technicians to carry out refrigerant handling practices in a responsible manner, even when non-flammable refrigerants are involved. Poor service practices lead to a variety of other safety hazards, poor system performance, adverse functioning and reduced efficiency and greater refrigerant-related and energy-related emissions. There is an obligation for all technicians to increase competency, irrespective of the introduction of low GWP and/or flammable refrigerants.

Cost for the consumer

A crucial factor for manufacturers and retailers is the cost of DHPs to the consumer, since it is the consumer that dictates the success of their business. Adverse changes to the first cost of products may compromise the appeal of products and, largely due to reputational effects, higher on-going service and maintenance costs can similarly be detrimental. Switching to an alternative refrigerant – whilst maintaining a given efficiency – infers possible cost implications to system components, additional safety features and subsequent lifetime costs. As such, these need to be examined.

Environmental impact

The primary purpose of switching to low GWP alternatives is to reduce environmental impact, particularly CO_2e emissions. There are numerous sources of such emissions, including those associated with production of construction materials, transportation and leakage of refrigerants over the product lifetime. Monetary value is increasingly applied to carbon emissions, or more precisely, the avoidance of emissions. Therefore, it is important to consider whether any additional emissions associated with the production offsets gains achieved by reducing those arising from lifetime leakage and how these relate to the costs of adopting alternatives with GWP < 150.

Considering the assessable aspects from this list, the criteria to be quantified and evaluated are:

- Enabling a DHP to achieve the requisite minimum efficiency, the highest energy label level and above.
- Product safety, through compliance with safety standards, primarily related to refrigerant charge limits necessary for high efficiency DHPs.
- Cost for the consumer, through impact on material costs and whether adoption of the alternatives would result in an adverse increase in cost
- Environmental impact, through assessment of carbon dioxide equivalent emissions

From a societal perspective and within the context of the drive to address climate change, the latter two criteria can sometimes be considered in tandem as a means to gauge the cost-effectiveness of a given technology.

1.3 Methodology and data-sources

The methodology adopted follows the general sequence:

- Summarise refrigerant charge limits in revised safety standard
- Identify heating thermal loads in DHP applications
- Characterise DHP performance and construction aspects using available databases
- Present performance characteristics of alternative refrigerants
- Approximate DHP charge amounts for alternative refrigerants (based on capacity and efficiency)
- Consider estimated charge amounts with respect to safety standard charge limits
- Quantify material masses, component volumes and associated costs for alternative refrigerants, including shipping and storage

- Estimate production and lifetime emissions associated with the application of alternative refrigerants
- Discuss the trade-off between cost-effectiveness, safety limitations and possibly other practical implications, such as implementation time

To provide a baseline for the assessment, against which the impact of applying the alternatives can be gauged, demands comprehensive data on existing DHP products. For this, data was extracted from the Eurovent and Keymark databases for all DHPs of less than about 25 kW nominal heating capacity (Eurovent, 2022; Keymark, 2022), thus representing models applicable to domestic application. Some key values (refrigerant charge, weight, dimensions and price) are not usually included in the databases, so additional details were obtained from product literature, when available. Where performance values stated on product sheets and manuals differed from those listed in the databases – as is frequently the case – values from the database were selected for the analysis.

To help enable adjustments and validation of the analysis, available data for a R290 models was also obtained. No DHP models for R1270, R1234yf or R152a have been found, so the assessment for these fluids is based solely on their properties and characteristics, using established methods, along with supporting information from published research. Furthermore, interviews and discussions were held with several manufacturers to better understand DHP product development and implementation.

1.4 Characterisation of DHPs

Unlike other similar products (such as split air conditioners), DHPs extend to several different forms, according to end user requirements. This makes analysis of the equipment (with regards to application of alternative refrigerants) more complicated than with similar products. Accordingly, general types and characteristics of the various DHP products need to be identified in order to carry out an assessment.

DHPs can be categorised according to the thermal source (where the heat is taken from, such as the air or the ground) and the thermal sink (where the heat is "discharged" to and for what purpose, e.g., space heating or domestic hot water or both). They may also be identified according to general system construction format (e.g., monoblock or split). These characteristics can have an impact on how the viability of alternatives is assessed.

In terms of the thermal source, DHPs will extract thermal energy from a variety of sources, including outside air, indoor air (being discharged to the outside), the ground (at shallow depths or with deep piles), surface water (lakes, rivers, etc.) and solar collectors. In the majority of cases, heat will be transferred to the refrigerant from air or through an intermediate water circuit. (A small number of systems use refrigerant to extract heat directly from the ground or water, but these are uncommon due to increased likelihood of leakage.)

Similarly, DHP systems will discharge heat to air or water or both. For space heating, DHPs usually supply heat to water sinks at two temperature levels: one at higher temperature for DHW, used for hot water taps, washing, etc. and one at moderate temperature levels intended for central heating by means of radiators, convectors and underfloor heating. In some instances, air is heated directly whilst being circulated in a room (as in a reversible split air conditioner) or ducted throughout the dwelling. Accordingly, DHPs are categorised according to the medium from which the heat is transferred to the refrigerant, i.e., liquid (water or antifreeze) or outside air, and the medium to which the heat is directly transferred to, i.e., water or indoor air. Figure 1 illustrates the main configurations of DHPs.

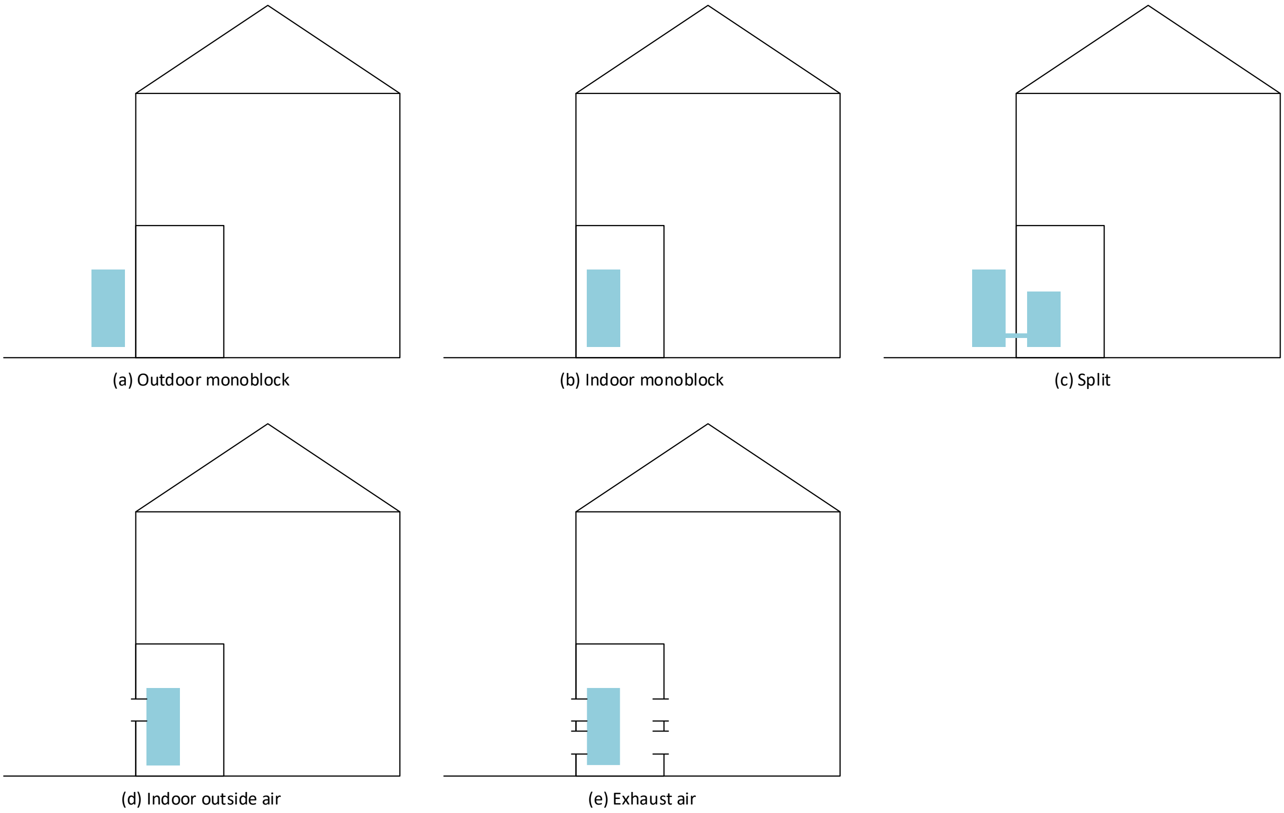


Figure 1: Basic DHP arrangements according to design and installation characteristics

These basic arrangements can be summarised as follows.

(a) Outdoor monoblock

The entire refrigerating system is located outdoors and is factory-sealed. Usually air-to-water, where the sink hot water is then piped into the building, providing DHW and space heating.

(b) Indoor monoblock

The entire refrigerating system is located indoors and is factory-sealed. Usually liquid-to-water (water- or brine-to-water), where the thermal source is the ground or surface water (lakes, rivers, etc.) and sink hot water provides DHW and space heating.

(c) Split

Part of the refrigerating system is located outdoors and the other part indoors (similar to a split air conditioner), where the compressor could be in either location. The two parts are connected with refrigerant piping. The outdoor part could be water-source – see (b) – but is usually air-source. Hot water provides DHW and space heating from the indoor part.

(d) Indoor outside air

The entire refrigerating system is located indoors, but outside air is drawn indoors through ducting, to the DHP evaporator as the thermal source. Hot water then provides DHW and space heating.

(e) Exhaust air

The refrigerating system is usually located indoors. It uses stale air being rejected from the building as the thermal source and heats fresh air drawn from the outside to be fed into the building. Sometimes there is an additional DHW function.

EHPA publish annual data on the European DHP market, with the data for 2021 being in Table 2. Increase in sales from the previous year was by 556,900, corresponding to a growth of 34%. This high rate of growth is expected to continue for at least the next decade or two.

Table 2: DHP sales by type within Europe in 2021

Type	Sales (rounded)
Air-Air	823 400
Air-water	984 500
Brine-water	128 200
Sanitary hot water	245 800
Other	1 000
Total	2 182 900

Across the range of available products there are numerous variations amongst DHP arrangements, but for brevity, the focus of this study will be limited to these as described. The basic categories are:

- Liquid-to-water (LTW), where "liquid" could be antifreeze (brine/glycol) or water
- Air-to-water (ATW), where air is outside air, directly or drawn inside via ducting

Space heating water may be supplied at temperatures usually ranging from +35°C to +55°C, depending upon whether the dwelling and heating system has been designed for DHP application (lower supply temperature) or if the dwelling is being retrofitted from an existing gas boiler (higher supply temperature). Since the majority of DHPs are supplied for the former case, this study will be primarily focussed on a +35°C supply temperature. Most applications also demand DHW, which involves heating a storage tank to at least 60°C. However, the proportion of thermal energy dedicated to this purpose is usually around 15% of total annual thermal demand and it is therefore opted to be neglected.

Furthermore, the following are also not included in the assessment:

- DHPs for multi-family occupancies
- Ait-to-air split systems (i.e., reversible split or heating-only air conditioners)

For split-type air-to-air heat pumps, which are effectively heating-only split ACs, refer to GIZ Proklima (2022); they will not be addressed herein.

For both the LTW and ATW models, conditions are:

- High efficiency, with SCOP = 6.5 at supply temperature of 35°C
- 10 kW nominal heating capacity (NHC)
- Monoblock – or effectively monoblock (direct coupled units) – meaning no additional interconnecting piping

These generic models are considered to be representative of the models currently sold within Europe.

2 Flammability safety requirements

Since all the alternatives presently under discussion are flammable, consideration is given as to the constraints to their application.

Within the EU, there is a hierarchy of rules applicable to the placement of products on the market, where (iii) takes priority over (ii) and (ii) over (i).

i) Industry safety guidance, usually providing practical information related to safe design, installation and servicing, such as the BWP guidelines[6] and the GIZ HC safety handbook[7].

ii) European safety standards which provide formal rules relating to the safe design, installation and servicing of systems, primarily EN 378 (2016) and EN 60335-2-40 (2022). These standards are harmonised to applicable health and safety requirements of relevant European directives. These standards are not necessarily mandatory, but they are broadly accepted by industry and test houses/notified bodies and provide a practical interpretation the legislative requirements. In addition, there are generic standards dealing with flammable gases, such as EN 60079-10-1 (area zoning) and EN 1127-1 (flammability risk assessment).

iii) European directives and national regulations, which lay down generic mandatory requirements that must be complied with when constructing, installing and working on systems. The directives that the above standards are harmonised to include the pressure equipment and machinery directives. The two directives specifically dealing with flammability hazards, ATEX (equipment) and ATEX (workplace) are not harmonised with the RACHP standards, however, the majority of the applicable requirements have informally been integrated into EN 378 and/or EN 60335-2-40. The generic flammable gas standards are in general harmonised with ATEX directives.

Crucially, though, none of the directives mentioned impose limits on quantities of flammable substances, irrespective of the size of the relevant spaces and therefore such constraining limits imposed by the safety standards can be bypassed, provided a suitable risk assessment is carried out and alternative mitigation measures applied, as and where necessary.

It is noted that the ATEX guidelines[8] imply that the directives do not apply to domestic equipment ("the directive shall not apply to … equipment intended for use in domestic and non-commercial environments where potentially explosive atmospheres may only rarely be created, solely as a result of the accidental leakage of fuel gas"), although this depends upon whether the refrigerant is considered to be "fuel gas" or not; since the exemption was originally formed to exclude products that fall under the Gas Appliances Directive – which DHPs do not – then this implied exemption may not apply.

2.1 Flammability risk mitigation concepts (RMCs)

Within the safety standards, EN 378 (2016) and EN 60335-2-40, there are a variety of flammability risk mitigation concepts (RMCs) that can be applied to DHPs, according to their type, installation location, refrigerant charge and so on.

[6] https://www.waermepumpe.de/fileadmin/user_upload/BWP_LF_Kaeltemittel_WEB.pdf

[7] https://mia.giz.de/cgi-bin/getfile/53616c7465645f5f7928ab98e186ae0bd4afd619dcfbece7bff1c1908c4a11f2b55aa26129092f7943fd447e15ac791325683f0f41fb2d4aa5ceebf2ad876b94acb09ae26aae04e0/giz2013-0686en-hydrocarbon-refrigerants.pdf

[8] https://ec.europa.eu/docsroom/documents/41403

Minimum separation distances

These are not prescribed within RACHP safety standards, but are required under the ATEX directives and are covered generically by hazardous atmosphere standards such as EN 60079-10-1.

For outdoor monoblock systems and the outdoor part of split systems, the primary requirement is to apply minimum separation distances from the installed unit. This is to ensure that leaked refrigerant cannot come into contact with external ignition sources or migrate into buildings. There is no formally specified separation distance, but amongst various manufacturers, distances ranging from about 0.5 m to 5 m are stated[9]. The larger the distance, the greater the inconvenience to the installer and owner.

Provided that realistic separation distances are specified, they do not invoke additional costs and are unlikely to impose inconvenience, except perhaps under some unusual circumstances. Note that such DHPs using non-flammable refrigerants similarly require minimum separation distances for the purpose of ensuring adequate airflow to achieve the requisite evaporator performance.

Charge reduction to "< 150 g" or 4×LFL

For indoor monoblock systems and the indoor part of split systems, there is no minimum room area requirements in the refrigerant charge per circuit is 4×LFL (e.g., about 4×0.038 = 0.152 kg of R290) or less. As a result, there are and have been extensive efforts to minimise the charge to meet this target. For instance, the LC150 project[10] aims to minimise the R290 charge of LTW indoor monoblock systems to below 0.15 kg, whilst maintaining high efficiency. So far, this has been achieved for systems of NHC around 5 – 10 kW, although highest COPs required about 0.20 kg of R290. Thus, it is likely that for the time-being, systems in excess of this NHC range will be subject to other RMCs, such as minimum room areas,

Minimum room area with integrated airflow (IAF) and active limited releasable charge (ALRC)

For indoor monoblock systems or indoor parts of split systems, where the refrigerant charge is between 0.15 kg and 26×LFL (approximately 1.0 kg of R290), minimum room floor area may be considered. In order to ensure the greatest flexibility of application of DHPs, it would be desirable to select the option leading to the smallest minimum room area, i.e., with integrated airflow (IAF); see below. However, with 1 kg of R290, the minimum floor area would be about 24 m^2, which is substantially larger than most utility rooms or cellars where indoor LTW DHPs are usually installed. As such, additional measures may be applied to assist, for example, an active limited releasable charge (ALRC) concept. Whilst a well-designed ALRC mechanisms could limit the releasable charge to less than 30% of the total charge, it is unlikely that monoblock systems could withhold any more than half of the total charge (unless multiple shut-off valves were applied throughout the circuit). Alternatively, if an intelligent sensing and control system was developed which could reliably identify from which side of the system a leak had occurred, it is conceivable that (in combination with a pump-down cycle) a much greater proportion of the charge could be withheld.

Ventilation

For exhaust air DHPs, "general ventilation" is employed for products already using R290; although the concept is not specifically described within the applicable safety standards. Where stale air is drawn into the DHPs as the thermal source for the evaporator, the air if further passed through the body of the DHP before being discharged to the outside. In this way, any leaked refrigerant will be diluted and similarly discharged

[9] Testing has found that with relatively large assumed leak rates under near-quiescent outdoor conditions, even at 0.5 m from the DHP, concentrations of R290 do not exceed one-fifth of the LFL. It is likely that narrower separation distances would suffice.

[10] https://www.ise.fraunhofer.de/en/research-projects/lc-150.html

outside. A leak from the condenser (bearing in-mind, the straight tubes are not considered as potential leak points, within IEC 60335-2-40) would similarly be drawn into the heated air stream and would be rapidly be diluted. Studies have demonstrated the leaks within ducts are very effectively homogenously diluted (to well below LFL) within a metre or so along the duct (Colbourne et al., 2018).

Ventilated enclosure

The ventilated enclosure concept is well established and used for indoor R290 HPs with charges of up to 130×LFL (about 5 kg). An enclosure, housing all the refrigerant-containing parts, is well sealed, except for a supply and return ventilation duct to the outside. A fan provides a specified airflow, either continuously or upon demand of leak detection. In either case, an airflow sensor is required to prove the minimum ventilation airflow rate; absence of sufficient airflow results in termination of the DHP operation. Ventilation airflow rate is proportional to the refrigerant charge, although if the system is enhanced tightness (ETRS), a smaller flow rate is usually required.

Naturally ventilated enclosure

Considering the cost and reliability implications of the ventilated enclosure concept, a variation has been studied, that relies on "natural ventilation", either stack effect, thermally-inducted convection or wind pressure effects. For DHPs installed at or above ground level, the enclosure can simply be fitted with a downward sloping conduit (for example, 25 mm diameter) to the outside and a small opening at the top of the DHP enclosure. With a leak inside the housing, the denser-than-air refrigerant will gravitate into the conduit and disperse to the outside. It is necessary that the housing is relatively tight, as the refrigerant can otherwise easily migrate to the surrounding space, forming a potentially flammable mixture. Nevertheless, if well designed, the enclosure will have exhausted sufficient refrigerant to remain below the LFL (e.g., Meljac et al., 2021). For DHPs installed below ground level, such as in a cellar, a similar concept can be applied where a pair of upwards conduits are fitted along with dissimilar outlet geometries. Any ambient air movement (wind) in any direction induces a pressure differential across the conduits, thus creating a natural ventilation current. Meljac et al. describe results of preliminary testing and subsequent testing and simulations indicate that provided the outdoor air conditions are not entirely quiescent, the refrigerant release can be exhausted within several minutes.

For either approach, the design can be assessed using the "surrounding concentration test" within the draft EN 378-1 (similar to that for commercial refrigeration appliances in IEC 60335-2-89), in order to demonstrate that a leak of refrigerant does not form a potentially flammable mixture within the installation space. The advantage of these approaches is that the components have negligible cost (compared to the conventional ventilated enclosure) and that there is no reliance upon mechanical components that may be subject to failure, such as gas sensors, flow switches and fan/motors. The two parameters that must be assured are no blocking of the conduits and the enclosures remaining relatively tight, in particular, technicians ensuring housing panels are correctly replaced after service and maintenance.

Other RMCs

Other concepts have been proposed and trialled to address the mitigation of larger releases of flammable refrigerants in smaller spaces.

For example, an approach has been described (Sonner et al., 2019; Sonner et al., 2022) that uses a quantity (less than 2 kg per kg of refrigerant) of activated carbon as part of the HP, to adsorb any release of refrigerant. The activated carbon is very cheap (i.e., charcoal) and can be formed into any shape. One possibility is to construct portions of the HP enclosure from the activated carbon, so that any leak from the refrigerating system inside the housing will be adsorbed in preference to migrating beyond the enclosure.

Ultimately the refrigerant will be desorbed from the enclosure, but at such a low rate that natural ventilation of the surrounding will easily disperse it before a flammable mixture forms.

2.2 Charge limit requirements

As with split ACs, the charge limit requirements imposed by safety standards can be a hinderance to the application of R290 or other low GWP refrigerants in certain DHP arrangements. The recently approved revision of the air conditioner and heat pump safety standard (IEC 60335-2-40, 2022) permits larger quantities of flammable refrigerants per unit of floor area of the room that the equipment is installed within. Greater quantities of refrigerant are allowed on the condition of additional mitigation measures intended to lower concentrations of refrigerant if leaked into the room. Essentially these mitigation measures include improved system tightness, provision of airflow to disperse leaks and integration of valving to limit the amount of refrigerant that could leak from the system.

In summary:

- Enhanced tightness refrigeration system (ETRS), where the assumed leak rate is much smaller than non-ETRS. In this case, allowable charge limit (ACL) is equation (1).

  $$m_{ACL} = F \times LFL \times h_0 \times A_{rm} \tag{1}$$

  where the concentration factor, F = 0.35, h_0 is the lowest installation height of the refrigerant-containing parts and A_{rm} is the floor area of the room that the equipment is installed within.

- Systems which use integral airflow (IAF), where an indoor fan operates continuously or in response to leak detection. Systems may be ETRS or non-ETRS; this only affects the required minimum airflow rate to disperse a leak. The ACL is equation (2).

  $$m_{ACL} = F \times LFL \times h_{rm} \times A_{rm} \tag{2}$$

  where the concentration factor, F = 0.50 and h_{rm} is the room height, usually taken to be 2.2 m.

- ACL calculation for the basic method (as included in previous editions of the standard) is equation (3).

  $$m_{ACL} = 2.5 \times LFL^{1.25} \times h_0 \times \sqrt{A_{rm}} \tag{3}$$

- For systems within a ventilated enclosure (VE) or located entirely outdoors, that comply with the applicable design requirements, the limit is equation (4).

  $$m_{ACL} = 130 \times LFL \tag{4}$$

  The minimum ventilation airflow for VEs is a function of the charge amount or if the system is ETRS it is fixed irrespective of the charge quantity.

- Limited releasable charge, where, if the releasable charge can be determined by test, the resulting mass can be assumed rather than the charged amount (equation 5). This can be considered to fall into two categories:

  — “Passive” limited releasable charge (PLRC), which typically accounts only for the mass retained in refrigerant oil and the system volume at atmospheric pressure, and

  — “Active” limited releasable charge (ALRC), which employs features such as safety shut-off valves to hold charge within the outdoor unit in response to leak detection.

  $$m_{RC} \cong (1 - \vartheta) \times m_C \tag{5}$$

where the retained charge coefficient, ϑ, may be around 0.8 – 0.9 for PLRC and anywhere from 0.05 to 0.75 for ALRC, based on experiments. For ALRC, the smaller the internal volume of the indoor part of the system (relative to the whole system) and the faster the response time of the leak detection system, the lower ϑ will be.

Example values are shown in Table 3. Overriding all of these ACLs is an upper charge limit (UCL) of 26×LFL, corresponding to about 1 kg of R290, 1.2 kg of R1270 and 3 kg of R152a, although for VEs the UCL is 130×LFL (i.e., the same as the ACL). There is no rational basis for these limits, except that it provides some sort of "comfort boundary" to those involved with drafting and approving the standard (Colbourne et al., 2020).

Considering the charge amounts of current indoor systems – ranging from about 0.5 kg to 4 kg (see section 4.3) – the charge according to the basic method, ETRS or even IAF are seldom useful. Taking the UCL of 26×LFL, a minimum room area of about 125 m^2 would be required for ETRS or about 25 m^2 for IAF, whereas typical utility rooms are less than around 10 m^2. With ALRC and IAF, minimum room areas could be below 10 m^2 (assuming ϑ = 0.4, although no appropriate value has been verified for such DHP systems). Within the Keymark database, only 35% of R290 systems use less than 26×LFL, with the remainder requiring larger charges and thus demanding use of a VE.

Table 3:Charge limit boundaries for various alternative refrigerants

	R410A	R32	R1270	R290	R1234yf	R152a
No min. room area	none	1.8	0.18	0.15	1.8	0.78
Indoors	none	16.0	1.20	0.99	15.4	3.38
Outside or within VE	none	79.8	5.98	4.94	77.2	16.90

2.3 Avoidance of potential sources of ignition

It is necessary to minimise the possibility of igniting leaked flammable refrigerant within the HP equipment and this is achieved through ensuring against the presence of potential sources of ignition. These may be electrical arcs from switching components, excessively hot surfaces from heaters or naked flames. There are numerous ways and means of achieving avoidance of ignition sources, such as high ventilation rates, repositioning of electrical components so flammable concentrations are avoided at such locations, using sealed enclosed to prevent ingress of flammable concentrations, limiting size of gaps within enclosures so flames cannot propagate outwards to the larger mixture and ensuring electrical current and voltage is below certain values such that sparks have insufficient energy to ignite a flammable mixture. Additional aspects are usually necessary, such as suitably robust housing materials and good quality parts, to guarantee protection against ignition for the lifetime of the equipment.

Whilst it may be intuitive to address ignition sources through selection of "Ex-type" components, this can be costly and also lead to unanticipated risks, where faulty Ex components may be replaced with non-Ex components by non-competent technicians. Thus, avoidance of ignition sources considering the principle of fail-safeness, potentially sparking components are used but suitably positioned away from where leaked refrigerant could migrate, is often preferred (and can assist with reduced costs).

2.4 Minimum ventilation/airflow rates

For IAF, a minimum airflow rate is required to disperse a leak of refrigerant. IEC 60335-5-2-40 provides formula to determine this value, depending upon whether the system is normal tightness (equation 6) or ETRS (equation 7).

$$\dot{V}_{o,min} = \frac{8\sqrt{A_o}\dot{m}_{leak}^{3/4}}{LFL^{3/4}}\left(\frac{F^{1/4}}{1-F}\right) \quad (6)$$

$$\dot{V}_{o,min} = \frac{5\sqrt{A_o}\dot{m}_{leak}^{3/4}}{h_o^{1/4}[LFL(1-F)]^{5/8}} \quad (7)$$

where A_o is the aperture area of the air discharge, $\dot{m}_{leak}$ is the assumed leak rate, $F \leq 0.50$ and h_o is the height of the air discharge. With normal tightness systems (equation 6), $\dot{m}_{leak} = m_c/240$ and with ETRS (equation 7), $\dot{m}_{leak}$ is specific to the refrigerant: for R32 and R1234yf, $\dot{m}_{leak}$ = 0.00278; for R1270, $\dot{m}_{leak}$ = 0.00135; for R290, $\dot{m}_{leak}$ = 0.00112; for R152a, $\dot{m}_{leak}$ = 0.00092 kg/s.

The minimum ventilation airflow rate for a VE is equation (8).

$$\dot{V}_{o,min} = \frac{24.5\dot{m}_{leak}}{FM} \quad (8)$$

where F = 0.25, 24.5 is a constant and M is the molar mass of the refrigerant and the assumed leak rate $\dot{m}_{leak}$, similarly depends upon the system tightness.

Example output of the minimum airflow calculation are shown in Figure 2 for IAF (where charge amounts are 2 kg for HFCs and equivalent 1 kg for HCs) and Figure 3 for VE (assuming 4 kg and 2 kg charge).

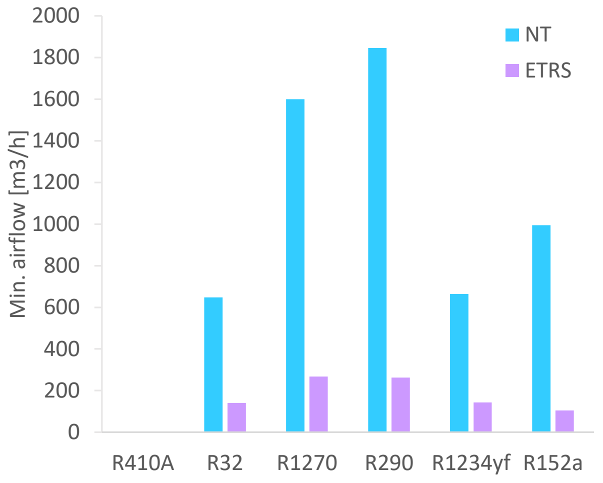


Figure 2: Minimum airflow rates for IAF, assuming normal tightness and ETRS

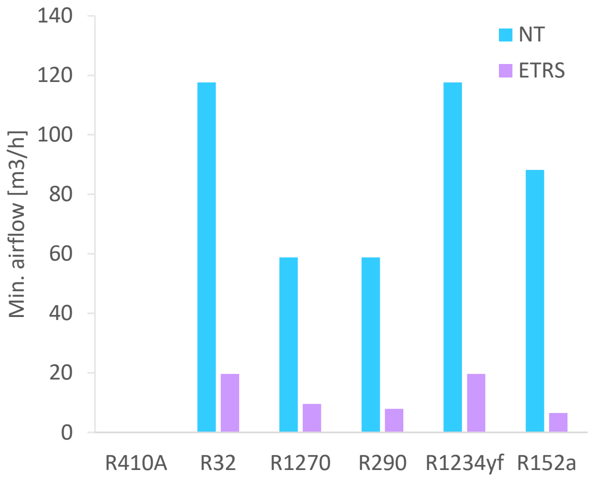


Figure 3: Minimum airflow rates for VE, assuming normal tightness and ETRS

To provide an impression of the hardware requirements (such as fan assembly volume and weight and motor power) for various airflow rates, typical values are listed in Table 4.

Table 4:Physical parameters for different sized fans

Airflow rate [m^3/h]	Fan/motor volume [m^3]	Fan/motor mass [kg]	Power [W]
10	≅ 0.00003	≅ 0.03	≅ 1
100	≅ 0.0004	≅ 0.4	≅ 10
1000	≅ 0.008	≅ 2	≅ 100

3 Heating loads

There are various studies (e.g., Persson and Werner, 2015; Kemna, 2014; Zangheri et al., 2014) reporting on average heating and cooling loads across Europe, spanning the range of climate conditions and building characteristics (age, construction, size, purpose, etc.). Across these studies, largest nominal heating demand across Europe is about 150 W/m^2, being for Prague. For more northern regions, such as Helsinki, maximum heating demand is lower (about 110 W/m^2), which is due to better thermal construction of buildings. Additional load may be considered for DHW, usually depending upon the number of occupants. Normally for DHP selections, case-by-case assessments on building heating loads and DHW requirements are carried out.

Nevertheless, unlike for split ACs (and indeed heating-only air-to-air DHPs), the thermal load associated with the application has almost no relationship to the suitability of the deployment of DHPs using alternative refrigerants.

4 DHP performance and product data

The Eurovent and Keymark databases hold extensive data for registered DHPs. They include certified performance data (thermal capacity, efficiency, etc.) for cooling and heating modes, including part-load conditions, electrical inputs and noise level data. Whilst the Keymark database includes data on refrigerant charge amounts, the Eurovent does not and neither hold data on product weight or dimensions or costs/prices. Therefore, these latter data have to be extracted from product sheets and retail internet sites[11].

Amongst the nearly 2150 models, a large number are evidently repetitions of the same refrigerating system (same compressor, HXs, housing size, etc.). Based on discrete capacity and efficiency ratings and total housing size, it is estimated that there are only about 200 different refrigerating systems. In other words, there are on average about 10 variations associated with each refrigerating system. Amongst these variations are differences in housing colour and other aesthetic features, functionality, product names (e.g., for targeting at different end use categories) and so on.

In terms of refrigerants, the Eurovent data includes only R410A, R407C and R32 models. Although the full Keymark database includes products with these and a variety of other refrigerants, only those using R290 – 420 – models were included at present. (Thus, data for the other 4,931 DHP models using refrigerants other than R290 were not part of the analysis, due to time limitations.) 203 of the R290 models were ATW and 127 were LTW, with the remainder being exhaust air, DHW-only, indoor air-to-air and other unspecified types. By comparison, for HFC products, there are 408 LTW and 1095 ATW models. As with the Keymark database, it is evident that almost all of the LTW models listed as ground-source are also presented as water-source; the difference being that they are simply tested at different conditions. Similarly, those listed as for space "heating", "cooling" and "hot water" are also listed at "heating only", "heating and cooling" and "heating and hot water".

Initially, an overview of product data is presented to help provide an impression of the range and characteristics of the products.

4.1 Efficiency

Figure 4 and Figure 5 plot NHC against SCOP for LTW DHPs, based on 35°C water supply ("W35") and 55°C supply ("W55") conditions, respectively, for average climate conditions. (The small number of data-points in Figure 5 is due to W55 data not being listed in the database.) The different markers refer to the functionality, i.e., space heating ("heat"), space heating and cooling ("heat cool"), space heating and domestic hot water ("heat hot wat") and space heating, cooling and domestic hot water ("heat cool hot wat"); noting that many are superimposed on top of each other, demonstrating they are essentially the same model.

SCOP ranges from about 3.5 to 6.5, with NHCs in excess of 40 kW. SCOP remains consistent across all NHCs. Available R290 products have efficiencies in the upper range of the various models and extending to a similar range of NHCs.

[11] Due to the absence of various values (performance, charge, unit mass, volume, price, etc.) for many models, significant information gaps exist in the presented data.

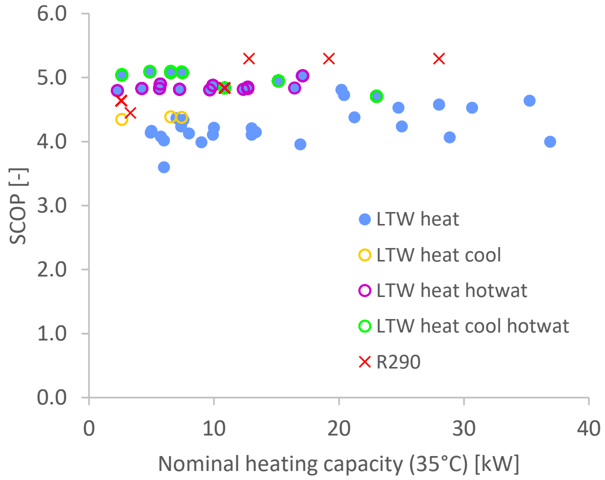


Figure 4: Seasonal efficiency and nominal heating capacity of LTW models (for supply at 35°C)

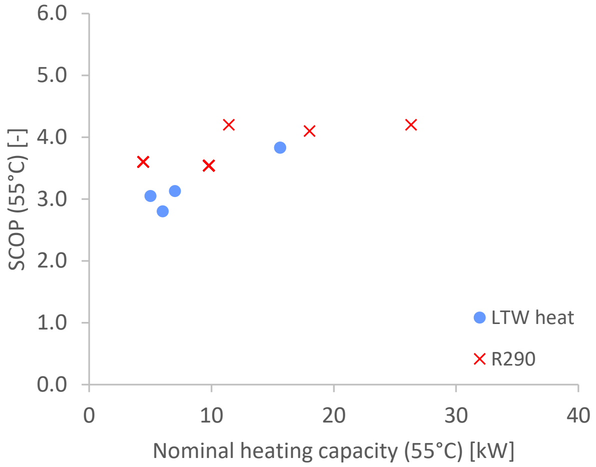


Figure 5: Seasonal efficiency and nominal heating capacity of LTW models (for supply at 55°C)

Data for air-source (ATW) models is included in Figure 6 and Figure 7 for W35 and W55 conditions, respectively. Again, fewer data-points are available for W55 conditions and the Keymark database does not differentiate between functionalities (i.e., for the R290 models). The relationship between SCOP and NHC appears independent. NHC generally ranges from about 3 kW to about 12 kW, with a few extending up to about 20 kW and SCOP is from about 3 to 5 (at W35 conditions). With both W35 and W55, a notable proportion of R290 models have higher SCOP than the R410A and R32 models.

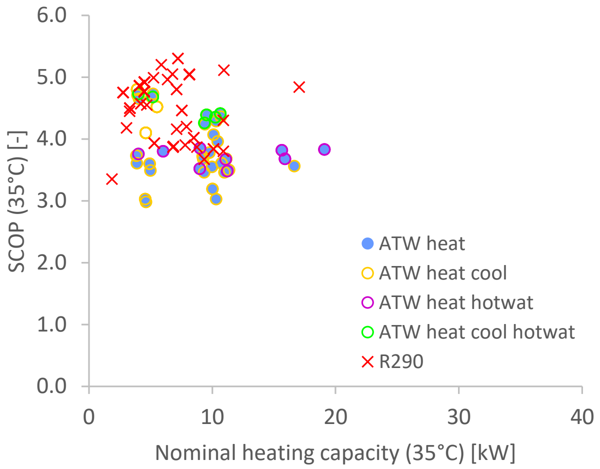


Figure 6: Seasonal efficiency and nominal heating capacity of ATW models (for supply at 35°C)

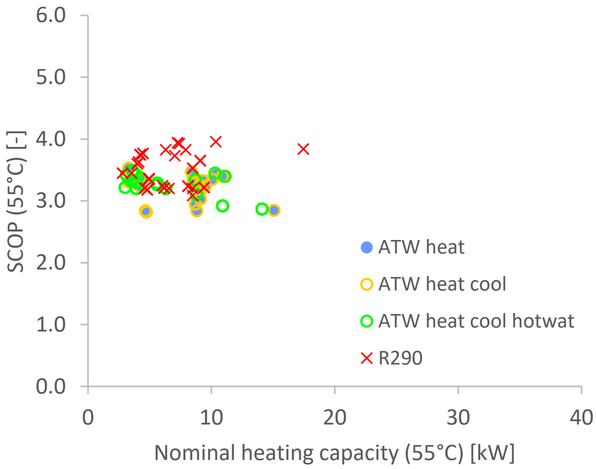


Figure 7: Seasonal efficiency and nominal heating capacity of ATW models (for supply at 55°C)

4.2 Size (volume and mass)

The total volume of the DHP units is shown in Figure 8 and Figure 9. Note that there are inconsistencies since some products come with integrated water tanks whilst others are offered with one or a selection of remote water tanks (which have been excluded); similarly supplementary electric or gas heaters may be integrated or remote (excluded).

For LTW units, volume seems to remain almost the same across all NHCs, which is likely due to the same housing being used for all models within a product range. Large shifts in volume are due to inclusion of water tanks and other ancillaries. Conversely for ATW units, larger NHC units usually correspond to greater volume, although smaller NHC models seem to be no less than 0.15 m^3. Scatter is quite wide, where for a given NHC, total volume extends across ±1 m^3, again, due to inclusion of ancillaries. Volume of R290 models is broadly consistent with the HFC models and there is no clear differentiation between the two.

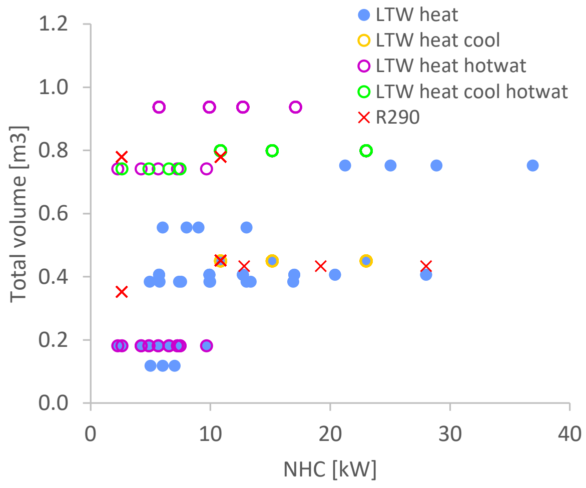


Figure 8: Total unit volume for LTW models against NHC

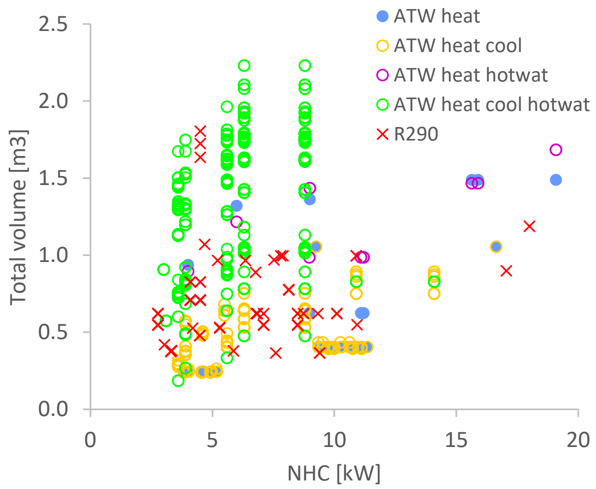


Figure 9: Total unit volume for ATW models against NHC

Similar conclusions may be drawn from the data shown in Figure 10 and Figure 11 where it is seen that generally for LTW, a greater SCOP does not lead to a larger volume, whereas a slight rise can be perceived with ATW units, but given the wide scatter it is inappropriate to state that it is due to corresponding differences in system components.

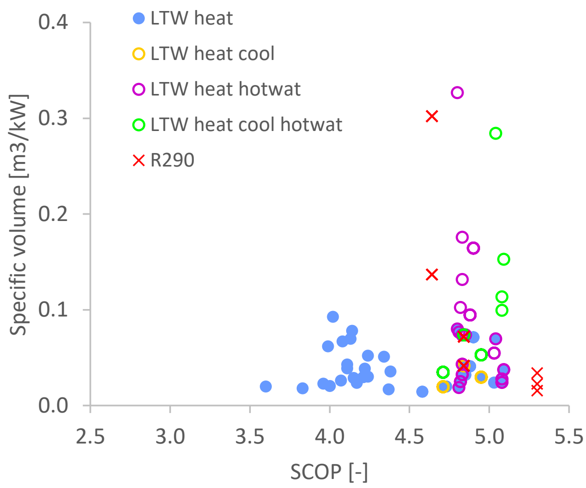


Figure 10: Specific volume for LTW models against SCOP

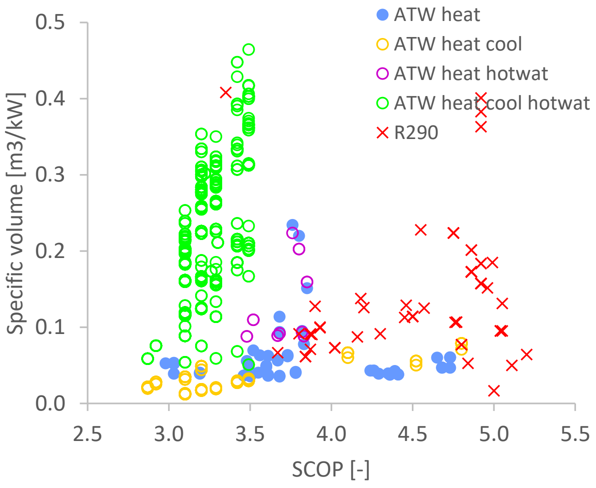


Figure 11: Specific volume for ATW models against SCOP

Data for total mass of DHP in Figure 12 and Figure 13 and specific mass in Figure 14 and Figure 15 closely match the patterns in the corresponding figures for volume and specific volume, respectively.

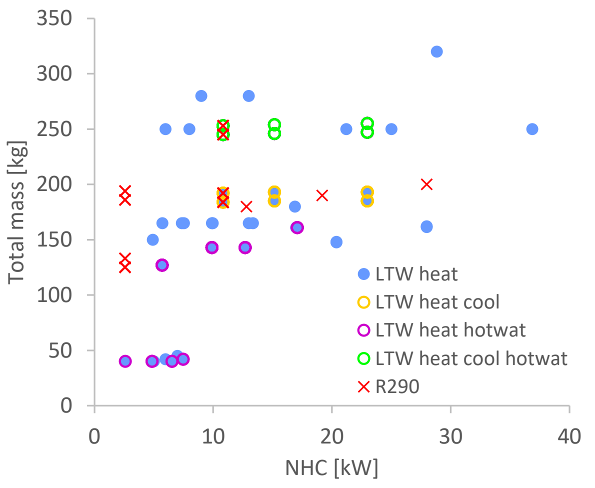


Figure 12: Total mass (includes IDU +ODU, where applicable) for LTW models against NHC

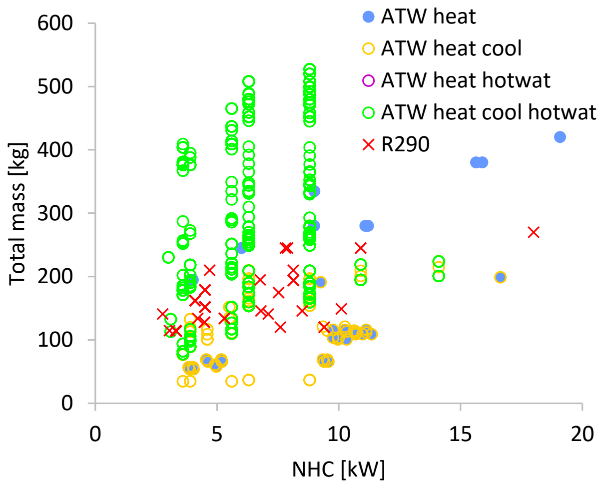


Figure 13: Total mass (includes IDU +ODU, where applicable) for ATW models against NHC

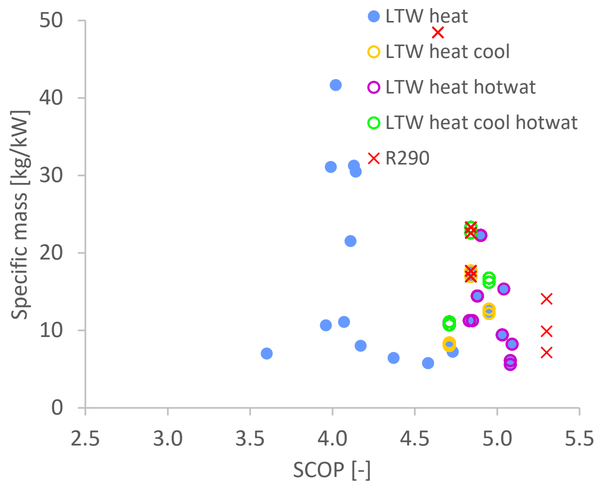


Figure 14: Total specific mass for LTW models against SCOP

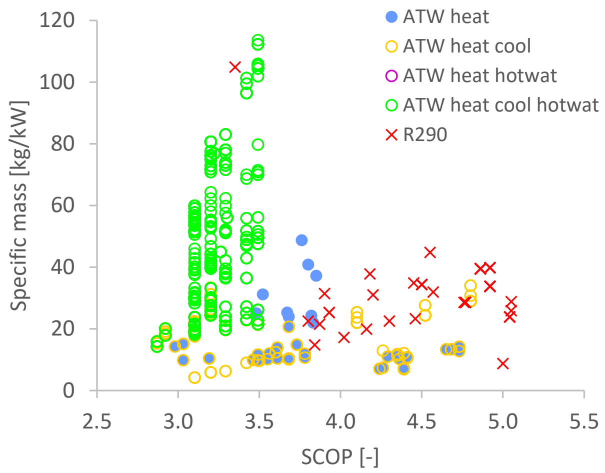


Figure 15: Total specific mass for ATW models against SCOP

Interestingly, plotting the specific mass against specific volume (Figure 16) shows that the majority of DHPs have a fairly consistent “density”.

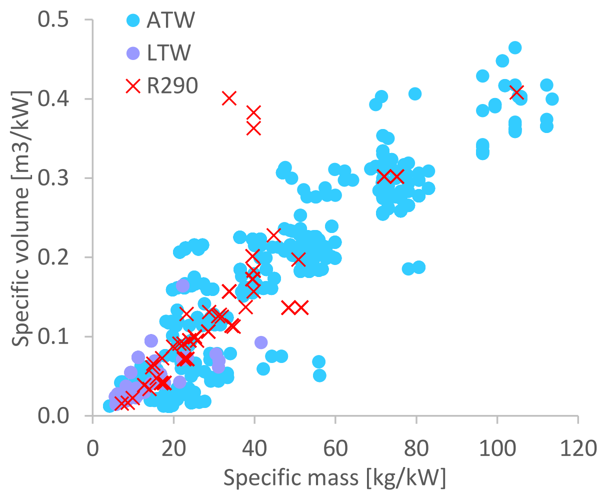


Figure 16: Total specific mass against total specific volume

4.3 Refrigerant charge

Figure 17 and Figure 18 provide refrigerant charge for the LTW and ATW DHPs, respectively. For LTW, there is a general increase in charge with higher NHC, but for any given NHC there remains a wide variation by a factor of three or four. Charges are generally higher for ATW units and for some types an increase in charge with NHC can be seen, but also (for R290 models) relatively high capacities are available with no more than 1 kg. Most LTW models using R290 are also below about 1 kg, whilst the ATW with largest R290 charge is about 4 kg.

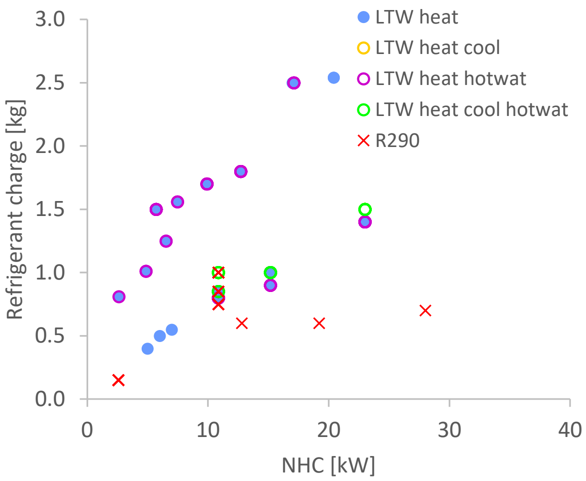


Figure 17: Refrigerant charge for LTW models against NHC

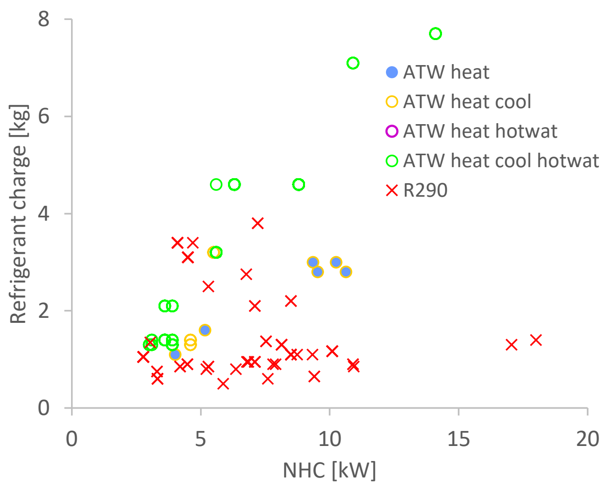
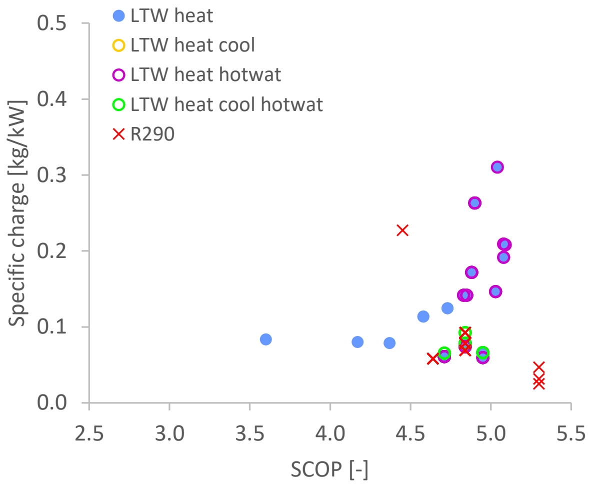


Figure 18: Refrigerant charge for ATW models against NHC

Specific charges are in Figure 19 for LTW and Figure 20 for ATW. Although some of the data-points imply that higher SCOP requires a greater charge, it can also be seen that some high SCOP models – including those with R290 – can suffice with very low charges, as low as 25 g/kW with R290. With ATW models a similar general trend can be observed, where most cases use larger specific charges as SCOP increases, yet many of those with higher efficiency only require about 50 g/kW. Evidently those products with direct DHW tanks, also seen in Figure 17 and Figure 18, use considerably more refrigerant than those with indirect DHW heating.



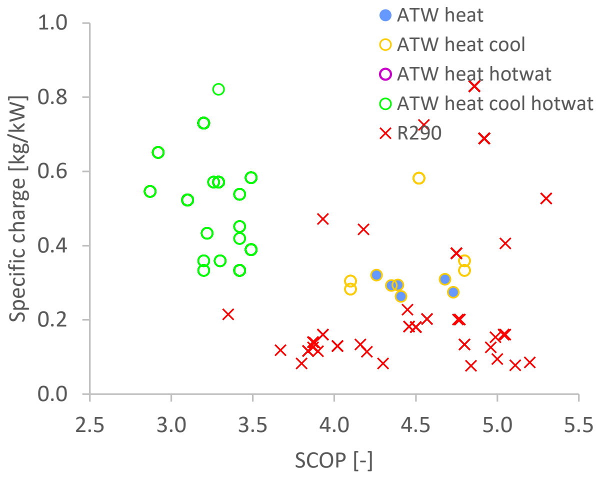
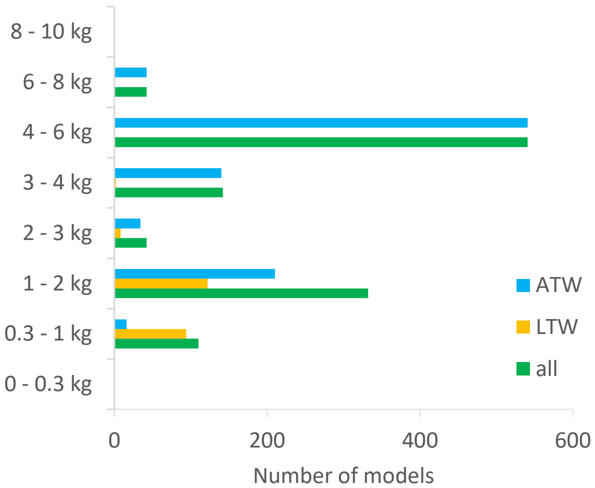


Figure 19: Specific charge for LTW products against SCOP

Figure 20: Specific charge for ATW products against SCOP

Lastly, Figure 21 and Figure 22 give the distribution of refrigerant charge across HFC and R290 models, respectively, also broken down according to DHP type. Whilst the greatest number of HFC models have charges of 4 – 6 kg, most R290 models are between 0.5 – 1 kg, which is half the equivalent HFC charge.



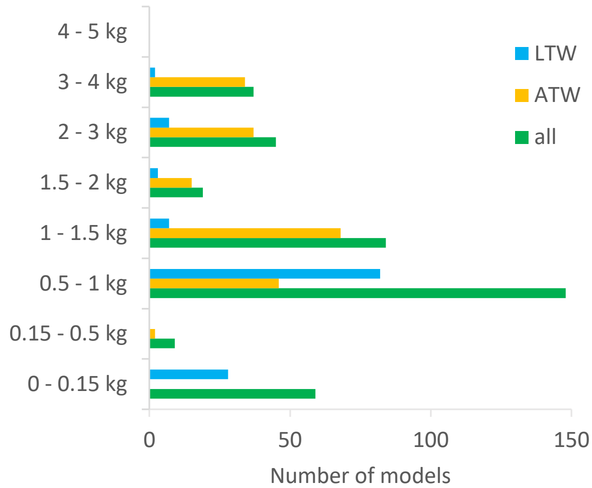


Figure 21: Distribution of refrigerant charge for HFC models

Figure 22: Distribution of refrigerant charge for R290 models

4.4 Retail price

As many retail prices were gathered for DHPs as could be readily found on the internet. Prices for some models could only be found on internet sites of certain countries; all models could not be sourced from any one country or any one retailer. Furthermore, it was possible to find prices for 1100 or 75% of the HFC models but only 75 or 20% of the R290 models and thus can be considered as providing a less representative impression of price distribution. Prices for models found only in countries outside the Euro-zone were converted to Euros using the exchange rate on 22nd – 26th August 2022. Where prices were listed as inclusive of VAT, they were reduced according to that country's VAT rate. Prices listed as exclusive of VAT were used as-is. In some cases, prices quoted without reference of VAT were assumed to be exclusive of VAT. Many internet sites listed products with "RRP" (recommended retail price) but listed them with a discount price, representing a reduction of between 10 – 50%. In some countries, legislation states that such discounts must be “genuine” (i.e., such a product must be retailed at the RRP for a certain proportion of the year), whilst in others no such rules apply. Given the uncertainty arising from these practices, a price corresponding to the arithmetic mean of the RRP and discounted price were used. Products sold in Roubles were negated due to the unrepresentative exchange rate at the time.

The price for LTW is plotted in Figure 23 against NHC and in Figure 24 for ATW models. As with other parameters, there is considerable variation. ATW products extend across a wider price range than LTW, having (for a given NHC) both lower and higher prices than LTW. For both types there is a rough increase in price with greater NHC.

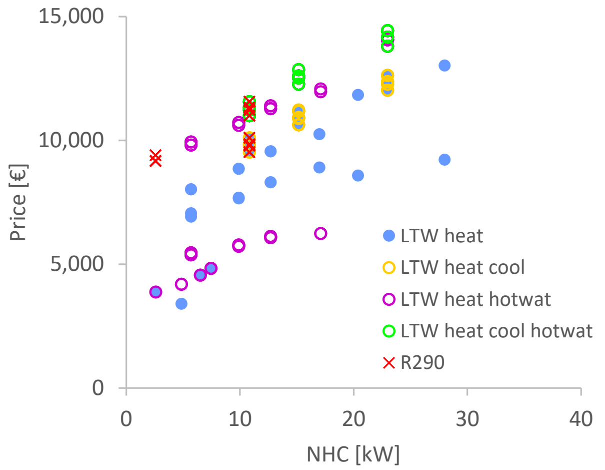


Figure 23: Normalised retail price for LTW models against NHC

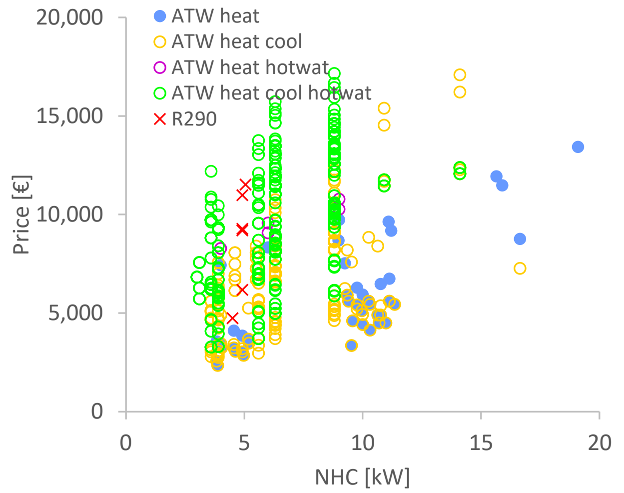


Figure 24: Normalised retail price for ATW models against NHC

Comparing specific price in Figure 25 for LTW shows a huge difference across a narrow range of SCOP, suggesting efficiency has little practical impact on price. A similar conclusion can be drawn for ATW models in Figure 26. Overall, there are too few datapoints for R290 to draw meaningful observations.

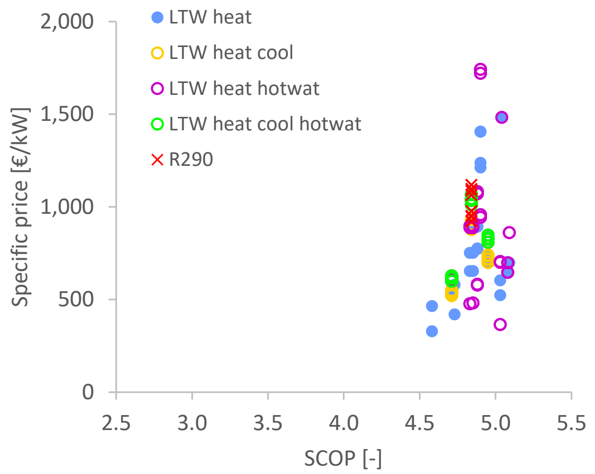


Figure 25: Specific price of LTW products against SCOP

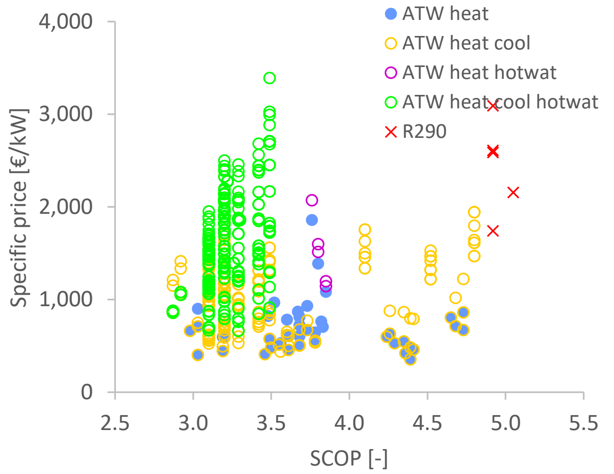


Figure 26: Specific price of ATW products against SCOP

It may be concluded that neither refrigerant nor efficiency level have any discernible influence on the retail process of DHPs. Theoretically, an increased efficiency infers more costly components which should increase price. However, these observations suggest that other factors, such as inclusion of ancillary parts (water tanks, supplementary heaters, etc.), electronics/control features and gimmicks, regional pricing philosophies, branding and producer, importer and retailer margins dwarf the contributions of material aspects. These aspects have a far more dominant effect on pricing that the impact of refrigerant characteristics.

5 Mandated efficiency levels

As with many other regions, European legislation prescribes minimum efficiency of domestic heating equipment. The Ecodesign regulation[12] sets minimum efficiency levels and labels for a variety of equipment used for the same (heating) purpose, including DHPs. Currently, minimum efficiency (expressed as "seasonal space heating energy efficiency", η_c) is 110% for DHPs in general and 125% for low temperature DHPs in particular (as from 2017). Similarly, efficiency levels for energy labelling are prescribed within the applicable regulation[13]. Table 5 lists the current minimum efficiency and efficiency label levels and the corresponding SCOP (at W35) and equivalent approximated fixed-point COP (based on +5°C source temperature).

For reference, the calculation of seasonal space heating energy efficiency from SCOP is shown in equation (9).

$$\eta_c = \frac{1}{CC} \times SCOP - G \tag{9}$$

where CC is a coefficient for electricity generation efficiency and is set at 2.5[14] and G is a "correction" intended to account for electrical power consumption associated with controls and pumps and for the equipment presently under consideration is set to 8%.

Table 5: Current efficiency levels for Ecodesign regulation

Efficiency level	Low temperature DHPs			Non-low temperature DHPs		
	η_c [%]	SCOP [-]	Equiv. COP [-]	η_c [%]	SCOP [-]	Equiv. COP [-]
Minimum	125	3.33	3.2	110	2.95	2.1
A+++	≥175	≥4.58	4.3	≥150	≥3.95	2.8
A++	150 to <175	3.95 to <4.58	3.8 – 4.3	125 to <150	3.33 to <3.95	2.3 – 2.8
A+	123 to <150	3.28 to <3.95	3.1 – 3.8	98 to <125	2.65 to <3.33	1.9 – 2.3
A	115 to <123	3.08 to <3.28	2.9 – 3.1	90 to <98	2.45 to <2.65	1.7 – 1.9
B	107 to <115	2.88 to <3.08	2.7 – 2.9	82 to <90	2.25 to <2.45	1.6 – 1.7
C	100 to <107	2.70 to <2.88	2.6 – 2.7	75 to <82	2.08 to <2.25	1.5 – 1.6
D	61 to <100	1.73 to <2.70	1.6 – 2.6	36 to <75	1.1 to <2.08	0.8 – 1.5

The Eurovent and Keymark data is plotted in Figure 27 along with boundaries for minimum efficiency and various energy efficiency label levels. It can be seen that the majority of DHP fall above the minimum efficiency levels (those that don't are sold outside the EU) and also with an energy label of A or A+ and higher. Whilst 21% of the HFC DHPs exceed the boundary for an A+++ energy label, 33% of the R290 models do.

[12] Regulation (EU) No 813/2013 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters
[13] Regulation (EU) No 811/2013 of 18 February 2013 supplementing Directive 2010/30/EU of the European Parliament and of the Council with regard to the energy labelling of space heaters, combination heaters, packages of space heater, temperature control and solar device and packages of combination heater, temperature control and solar device
[14] European Directive 2012/27/EU

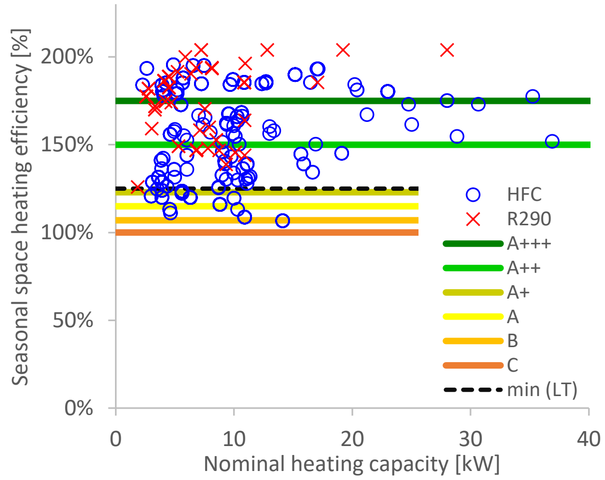


Figure 27: Current seasonal space heating efficiency levels for low temperature DHPs according to current Ecodesign regulation[15]

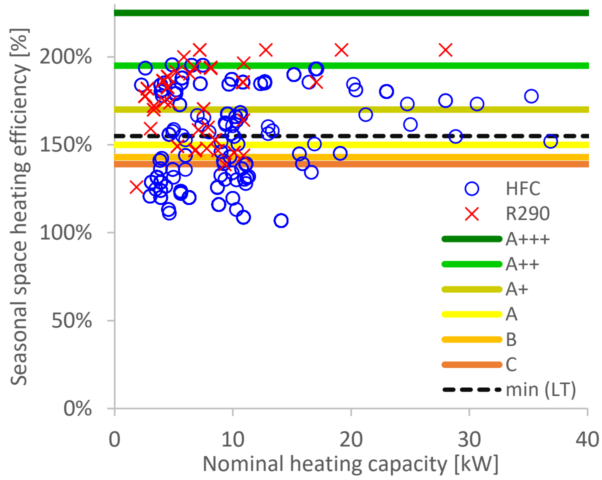


Figure 28: Proposed seasonal space heating efficiency levels for low temperature DHPs according to revised Ecodesign regulation proposals

It is anticipated that with the next revision of the EU Ecodesign regulations will revise the minimum efficiency and energy efficiency label levels upwards[16].

The proposed minimum efficiencies in a draft revised regulation[17] and energy labelling levels are listed in Table 6 (Holsteijn, 2019). In addition to the proposed new efficiency levels, the coefficient for electricity generation efficiency (CC) would be lowered to 2.1 (as a result of reduced emissions from electrical power generation). This means that η_c resulting from a SCOP will be 15% higher or conversely, the SCOP of a DHP would need to be 15% lower to achieve the same η_c (with the change in CC alone). Minimum efficiency levels would increase to 155% and 130%, but since this is less than 15% it represents a reduction in SCOP of DHPs. The levels for low temperature DHPs are superimposed on the data in Figure 28. None of the listed models reach the highest (A+++) labelling level and only 12 (0.8%) of the HFC products and 11 (2.6%) of R290 products achieve the second highest (A++) level.

Table 6: Proposed efficiency levels for revised Eco-design regulation

Efficiency level	Low temperature DHPs			Non-low temperature DHPs		
	η_c [%]	SCOP [-]	Equiv. COP [-]	η_c [%]	SCOP [-]	Equiv. COP [-]
Minimum	155	3.42	3.3	130	2.90	2.0
A+++	≥225	≥4.89	4.6	≥180	≥3.95	2.8
A++	195 to <225	4.26 to <4.89	4.0 – 4.3	150 to <180	3.95 to <4.58	2.3 – 2.8
A+	170 to <195	3.74 to <4.26	3.6 – 4.0	130 to <150	3.28 to <3.95	2.0 – 2.3
A	150 to <170	3.32 to <3.74	3.2 – 3.6	110 to <130	3.08 to <3.28	1.7 – 2.0
B	143 to <150	3.17 to <3.32	3.0 – 3.2	98 to <110	2.88 to <3.08	1.6 – 1.7
C	139 to <143	3.09 to <3.17	2.9 – 3.0	93 to <98	2.70 to <2.88	1.5 – 1.6
D	135 to <139	3.00 to <3.09	2.8 – 2.9	87 to <93	1.73 to <2.70	1.4 – 1.5

[15] Products falling below minimum efficiency levels are being sold outside Europe.

[16] Explanatory Memorandum on Revision of Commission regulations (EU) 813/2013 and (EU) 811/2013 on respectively Ecodesign and Energy Label of central hydronic space and combination heaters as well as (EU) 814/2013 and (EU) 812/2013 water heater and hot water storage tank regulations

[17] Draft Ecodesign regulation space / combination heaters COMMISSION REGULATION (EU) No of implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for space heaters and combination heaters, repealing Commission Regulation (EU) No 813/2013

6 Efficiency and alternative refrigerants

Ordinarily, refrigerating system efficiency is termed "coefficient of performance" (COP) and is the ratio of useful thermal flux (condensing capacity for a DHP) to the energy used to drive the system (compressor power). This may be extended to the "coefficient of system performance", where the energy used to drive the system also includes the ancillary electrical power for controllers, fan motors, pumps, crankcase heaters and so on, necessary for the functioning of the DHP.

Within the performance test standard, EN 14511 (2018), "COP" is used to express the system efficiency in heating mode, which includes for the compressor, fans, controls, etc. Additionally, systems must also be evaluated for part-load performance, which follows EN 14825 (2022) and includes different weightings for each condition as specified for the three European climate classes ("warm", "moderate" and "cold"). Along with part-load tests being conducted at lower outdoor temperatures, the seasonal efficiency or SCOP is determined.

Impact of refrigerant selection on SCOP cannot be easily analysed due to the complexities of isolating refrigerant properties from transient mechanisms within the various components of the refrigeration cycle. Therefore, efficiency implications are addressed for full-load/fixed-point conditions, with the assumption that the conclusions can be considered as a reasonable indicator of part-load and seasonal temperature conditions. Further, it is noted that the auxiliary sources of electrical power consumption (fan motors, controls, etc.) are unrelated to refrigerant selection – including those associated with safety features (see section 9.5) – need not be considered for the efficiency assessment.

In analysing efficiency with respect to the alternative refrigerants, the following elements are considered:

- Thermodynamic cycle efficiency
- Component pressure losses
- Heat exchanger performance (effects within the compressor and piping is neglected)

First, Table 7 lists basic cycle performance parameters, based on -5°C evaporating and +45°C condensing temperatures, assuming a source temperature of +5°C and a water supply temperature of 35°C.

All refrigerants require about 1% larger evaporating capacity than R410A to achieve the same heating capacity, except for R152a, which requires about 3% more. R290 and R1270 have higher heating cycle efficiencies (about 3 – 5%) than R410A, R32 and R1234yf, although R152a has much higher efficiencies (about 7 – 9%). Still. R290 and R1270 have lower compression ratios, which potentially lead to higher compressor efficiencies, whilst R152a has a significantly higher compression ratio. There are substantial differences in swept volume, with R290 and R1270 being about 1½ times larger than R410A and R32 and R1234yf and R152a being about 2½ times larger. There is a similarly large variation in discharge temperatures, with R32 being about 20 K higher than R410A, R1270 and R290 about 15 K lower and R1234yf about 25 K below R410A, whilst R152a is similar. Although various performance indicators should not be regarded as practical values as they neglect various system losses, they are nevertheless important as they indicate the extent of potential efficiency, such as what can be gained from system optimisation.

Table 7: Thermodynamic cycle performance of selected refrigerants

Parameter	R410A	R32	R1270	R290	R1234yf	R152a
Cooling capacity [kW]	7.61	7.66	7.71	7.72	7.66	7.81
Heating capacity [kW]	10.00	10.00	10.00	10.00	10.00	10.00
Heating COP [-]	4.18	4.28	4.36	4.38	4.27	4.56
Swept volume [m^3/h]	6.9	6.2	9.7	11.8	17.2	17.3
Compression ratio [-]	4.03	4.04	3.67	3.78	4.34	4.72
Discharge temperature [°C]	79.8	101.8	68.8	62.4	52.1	75.6

Conditions: Evaporating temperature: +5°C – 10 K = -5°C; superheat: 10 K; condensing temperature: 35°C + 10 K = 45°C; subcooling: 5 K; compressor efficiency: 80%; no circuit pressure losses; interconnecting piping is adiabatic.

In order to characterise system components (HX, piping, etc.), the corresponding evaporating and condensing temperatures should be determined. This is done to obtain an equal target system efficiency (fixed-point COP) for all refrigerants. The resultant parameters are then used subsequently for HX selections. Here, the effective temperature difference (ETD) for the evaporator and condenser are defined as equations (10) and (11), respectively.

$$\Delta T_{e,eff} = \frac{T_{f,in}+T_{f,out}}{2} - T_{s,sat} \tag{10}$$

$$\Delta T_{c,eff} = T_{d,sat} - \frac{T_{f,in}+T_{f,out}}{2} \tag{11}$$

where $T_{f,in}$ and $T_{f,out}$ are the inlet and outlet fluid temperatures (water/brine or air) of the respective HX, $T_{s,sat}$ is the saturated suction (evaporator outlet) temperature and $T_{d,sat}$ is the saturated discharge (condenser inlet) temperature. The applicable cycle points are illustrated in Figure 29.

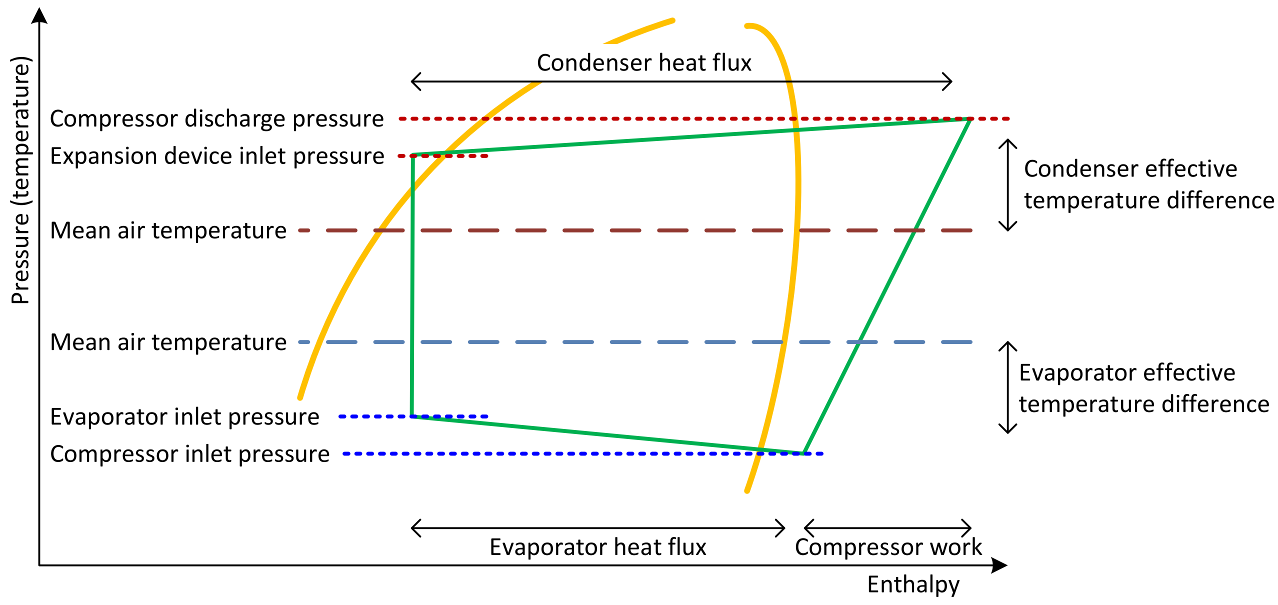


Figure 29: Cycle diagram indicating effective temperature differences

The target efficiency should correspond to a value greater than the highest proposed energy label level, so a SCOP of 6.5 has been chosen. Since calculation must be based on fixed-point values, though, a corresponding COP must be identified, which can be based on average conditions. According to the bin hours for an average climate in EN 14825: 2022, the arithmetic mean source temperature is 5.1°C; so, a nominal source temperature of 5°C is selected. From the databases, full load COPs are fairly similar to the SCOP; see for example, Figure 30 and Figure 31. Across the data for ATW DHPs (and interpolating between +2°C and +7°C source temperature), the fixed-point COP is between 0.65 to 0.85 × SCOP. For LTW DHP, fixed-point COP is between 0.8 to 1.1 × SCOP. These factors apply regardless of sink water supply temperature. Taking intermediate values, COP = 0.75 × SCOP for ATW and COP = 0.95 × SCOP for LTW. With a target SCOP of 6.5 for 35°C water supply temperature, fixed-point COPs of 4.9 for ATW and for LTW, COP of 6.2, are selected for the component calculations.

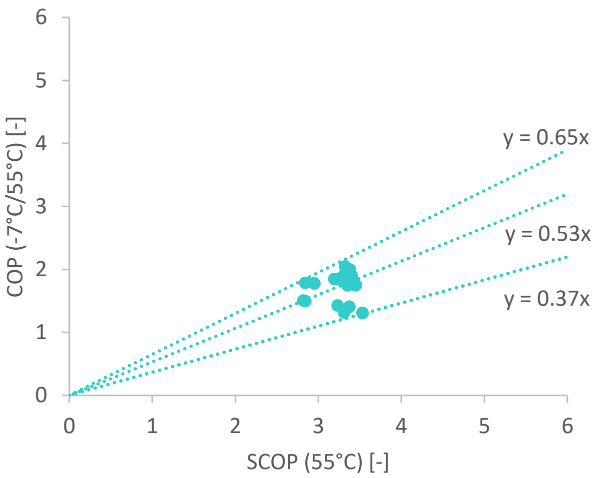


Figure 30: Comparison of SCOP and COP for ATW heating only DHPs at -7°C source and +55°C sink temperatures

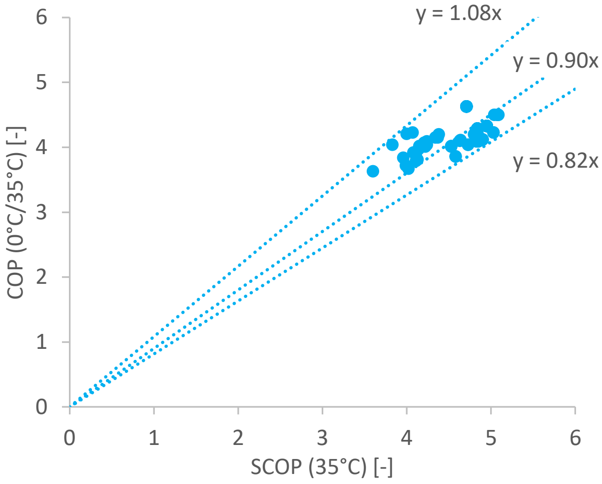


Figure 31: Comparison of SCOP and COP for LTW heating only DHPs at 0°C source and +35°C sink temperatures

With these values, corresponding ETD was determined for all refrigerants (as listed in ISO 817: 2014) and the results for the calculation are shown in Figure 32 for ATW and Figure 33 for LTW; the alternative refrigerants presently under consideration are identified. Values are plotted against saturation pressure of the refrigerant. Broadly, it can be seen that refrigerants with a lower saturation pressure can tolerate a slightly higher ETD.

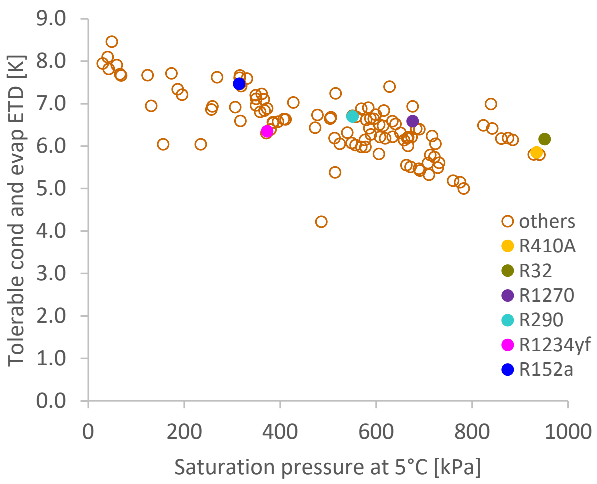


Figure 32: Effective temperature difference for refrigerants necessary for ATW target fixed-point COP = 4.9

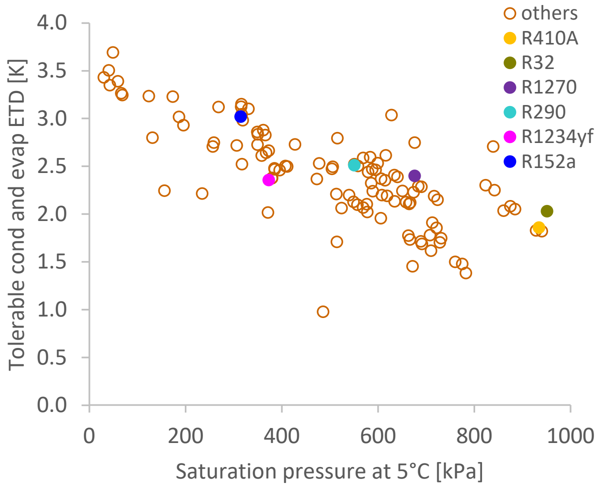


Figure 33: Effective temperature difference for refrigerants necessary for LTW target fixed-point COP = 6.2

ETD values for the selected refrigerants are also provided in Table 8. Specifically, R1270 and R290 can tolerate 10 – 15% larger ETD than R410A and R32 and R152a about 25% larger for the ATW DHP conditions. For LTW, R1270 and R290 can tolerate about 25% larger ETD and 50% larger for R152a. In all cases, R1234yf demands a slightly larger ETD than R410A and R32. Accepting a slightly higher ETD in the HX helps compensate for the comparatively higher pressure drop of these refrigerants (see section 9.4). Lower pressure refrigerants tend to be able to tolerate a lower saturation (corresponding to compressor suction) temperature because they have a higher cycle efficiency and can thus accept greater losses when reaching the target efficiency.

Table 8: Target effective temperature differences for selected refrigerants with T_s = 5°C for fixed COP

Type	Target COP	R410A	R32	R1270	R290	R1234yf	R152a
ATW	4.9	5.85	6.16	6.59	6.70	6.35	7.47
LTW	6.2	1.86	2.03	2.40	2.51	2.36	3.02

These differences are elaborated on further in Figure 34, where ETD is presented over a range of COPs (based on equal ETD assigned to evaporator and condenser). All refrigerants behave in a similar manner, where smaller ETDs correspond to higher COPs. Lower pressure refrigerants (R1234yf and R152a) tolerate larger ETDs whereas higher pressure refrigerants consistently require smaller ETDs and the ranking for each refrigerant remains the same across the range of COPs. For any given refrigerant, with a fixed thermal flux, a smaller ETD demands a larger heat exchanger. Depending upon the refrigerant, a smaller ETD may be applied (i.e., relatively larger HX) to the evaporator or condenser, but since most systems are usually reversible, an equal weighting is applied to both HX. Importantly, though, for lower target COPs the difference in tolerable ETD across the selected refrigerants is about 2 K, whereas for higher target COPs it is about 1 K. Practically, this means that for higher target efficiencies, the rate of increase in HX size is greater for higher pressure refrigerants, or, as minimum efficiencies increase, higher pressure refrigerants will demand increasingly larger HXs, relative to lower pressure refrigerants.

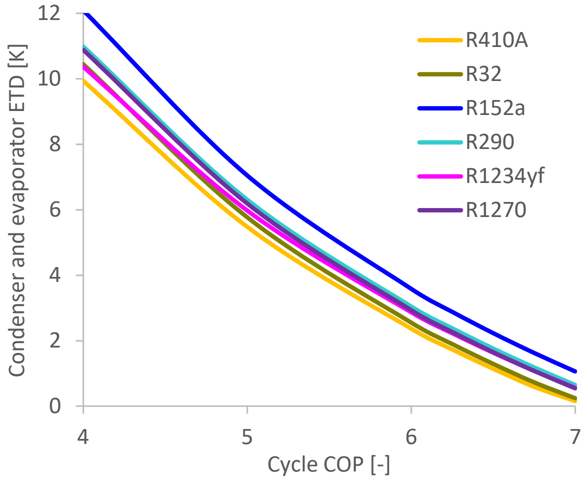


Figure 34: Effective temperature difference for selected refrigerants over a range of COPs

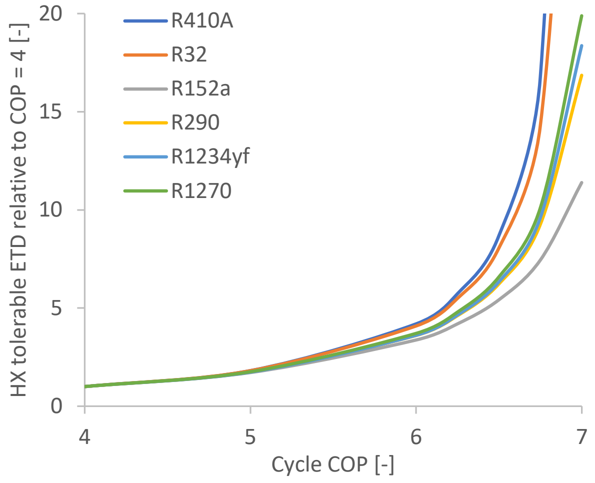


Figure 35: ETD relative to R410A over a range of COPs

The data is plotted in Figure 35, relative to R410A. As the target COP increases, the greater the variation in tolerable ETDs, reflecting the ranking of refrigerants' cycle COP. Assuming a consistent refrigerant heat transfer coefficient, the graph reflects the increasing size of a refrigerants' HX required to achieve the higher COP. A COP = 4 to 5.5 requires a tripling of the HX size regardless of the refrigerant. But notable divergence amongst refrigerants occurs with further improvements, for example, at COP = 6.5, R32 and R410A require 10 – 12 times larger HX, whereas the others would be about 7 times and R152a only 6 times. This difference amongst the refrigerants continues to escalate in favour of R1270, R290, R1234yf and especially R152a. As DHPs push for higher COPs, it will become increasingly more cost-effective for these low GWP options (in terms of HX sizing). Another interpretation of this characteristic behaviour is that under part-load conditions, those refrigerants that tolerate higher ETD will yield greater gains in COP since for a fixed HX size, smaller actual temperature differences will be achieved. In other words, R290, R1270, R1234yf and especially R152a will yield more efficiency benefits under part-load conditions.

Table 9 is an extension of Table 8 and presents values used subsequently to select appropriate HX in order to assess influence of refrigerant, component mass, volume and associated costs and emissions. Note that an additional nominal 0.5 K saturated temperature pressure drop is assigned to all refrigerants on the high side and low to account for interconnecting piping.

Table 9: ETD and saturated suction and discharge temperatures for achieving fixed-point COPs

Target COP	Parameter	R410A	R32	R1270	R290	R1234yf	R152a
(ATW) 4.9	$\Delta T_{e,eff}$ [K]	5.85	6.16	6.59	6.70	6.35	7.47
	$\frac{T_{a,in}+T_{a,out}}{2}$ [°C]	3.5	3.5	3.5	3.5	3.5	3.5
	$T_{s,sat}$ [°C]	-2.4	-2.7	-3.1	-3.2	-2.8	-4.0
	$\Delta T_{c,eff}$ [K]	5.85	6.16	6.59	6.70	6.35	7.47
	$\frac{T_{w,in}+T_{w,out}}{2}$ [°C]	32.5	32.5	32.5	32.5	32.5	32.5
	$T_{d,sat}$ [°C]	38.4	38.7	39.1	39.2	38.8	40.0
	T_d [°C]	68.0	86.7	60.4	55.2	45.7	68.1
(LTW)	$\Delta T_{e,eff}$ [K]	1.86	2.03	2.40	2.51	2.36	3.02

Target COP	Parameter	R410A	R32	R1270	R290	R1234yf	R152a
6.2	$\frac{T_{l,in}+T_{l,out}}{2}$ [°C]	3.5	3.5	3.5	3.5	3.5	3.5
	$T_{s,sat}$ [°C]	1.6	1.5	1.1	1.0	1.1	0.5
	$\Delta T_{c,eff}$ [K]	1.86	2.03	2.40	2.51	2.36	3.02
	$\frac{T_{w,in}+T_{w,out}}{2}$ [°C]	32.5	32.5	32.5	32.5	32.5	32.5
	$T_{d,sat}$ [°C]	34.4	34.5	34.9	35.0	34.9	35.5
	T_d [°C]	59.1	73.1	53.0	48.9	41.4	58.9

7 Approximation of charge amounts

It is intended to estimate the refrigerant charge of systems using the alternative refrigerants, across the range of DHP capacity sizes and efficiency levels. For this, DHP charge quantities are treated on the basis of specific charge (μ); the ratio of initial refrigerant charge to NHC. This enables the charge to be expressed more clearly in relation to the size of the system under consideration. Specific charge across the database has been plotted against efficiency and NHC in order to identify patterns.

As is observed in section 4.3, there is no strong relationship between refrigerant charge, NHC and SCOP in the data. Furthermore, unlike other RACHP equipment types (such as split air conditioners), there is not usually a direct link between required refrigerant charge and the size of the room that the system is located in (i.e., it is less sensitive from a safety perspective). As such, a precise approach for estimating DHP charge amount is not justified, so instead an approximation is simply based on fitting to the existing data. Any data-fit should nevertheless satisfy the following criteria:

- Higher efficiency requires greater specific charge
- Specific charge reduces with increasing NHC
- Reflect the density of the refrigerant and the effort to minimise charge to reflect its safety classification

Specific charge for space heating-only DHPs (i.e., no direct DHW tank) is approximated from equation (12):

$$\mu_c = (a \times \eta_c \times Q_c^{-1/3}) \times \rho_l \times \psi \qquad (12)$$

where ρ_l is the liquid density of the refrigerant at condensing temperature, η_c is the SCOP and Q_c is NHC. For LTW systems, a = 0.00005 and a = 0.00009 for ATW systems. ψ is a coadjuvant used to reflect the level of "effort" likely to be involved in charge minimisation according to the flammability class of the refrigerant; ψ = 1.2 for class A1 refrigerant, 1.1 for A2L, 1.05 for A2 and 1.0 for A3.

Equation (12) using SCOP = 4 is superimposed on Figure 17 and Figure 18 for refrigerants R410A, R32 and R290; these indicate a reasonable approximation for lower charge DHPs (without DHW tanks); see Figure 36 and Figure 37.

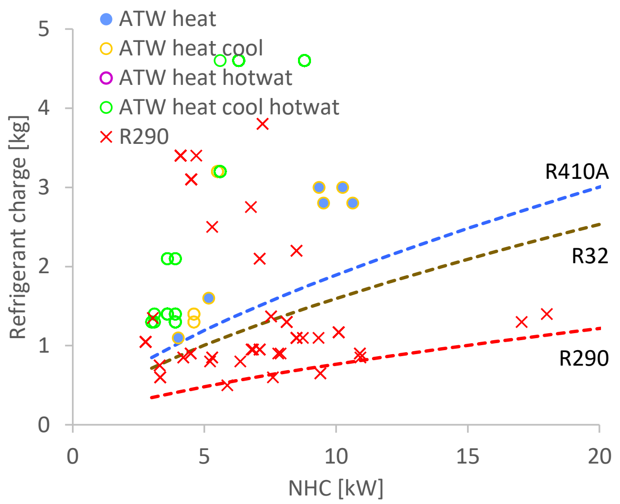


Figure 36: Charge estimation from equation (12) superimposed on Figure 17 for ATW, using typical SCOP = 4

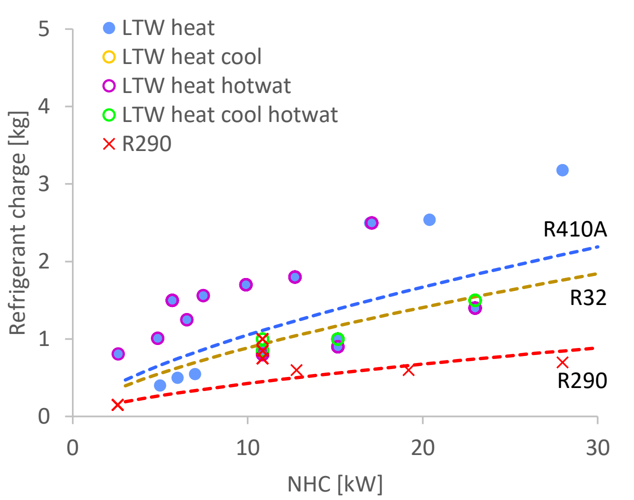


Figure 37: Charge estimation from equation (12) superimposed on Figure 18 for LTW using typical SCOP = 4

The result of equation (12) is presented in Figure 38 and Figure 39 to include all selected alternative refrigerants and with high efficiency, SCOP = 6.5.

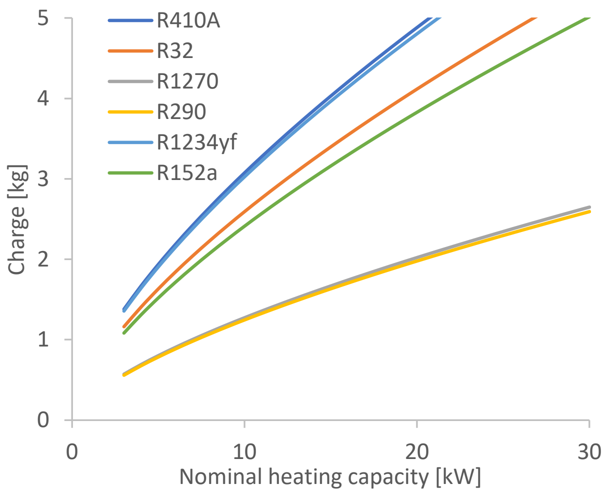


Figure 38: Charge estimation from equation (12) for ATW for selected refrigerants with SCOP = 6.5

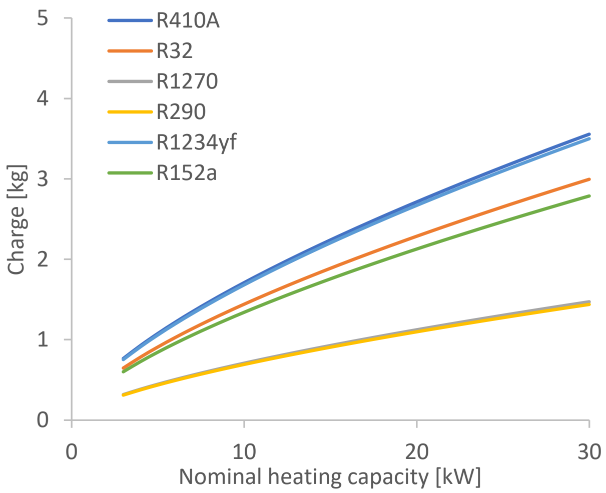


Figure 39: Charge estimation from equation (12) for LTW for selected refrigerants with SCOP = 6.5

Values for the assessment case with NHC of 10 kW are given in Table 10.

Table 10: Refrigerant charge for 10 kW ATW and LTW models with assigned COP

Type	Parameter	R410A	R32	R1270	R290	R1234yf	R152a
ATW	Specific charge [kg/kW]	0.31	0.26	0.13	0.12	0.30	0.24
	System charge [kg]	3.08	2.59	1.27	1.25	3.03	2.41
LTW	Specific charge [kg/kW]	0.17	0.14	0.07	0.07	0.17	0.13
	System charge [kg]	1.71	1.44	0.71	0.69	1.68	1.34

Lastly, as a cross-check, the output of equation (12) has been compared against the summation of the charge obtained from the HX simulations (see section 9.4), piping, valves and compressor internal volume; all based on SCOP = 6.5. The resulting charge amounts are plotted in Figure 40 and Figure 41 for ATW and LTW, respectively. For both ATW and LTW, equation (12) traces the charges fairly closely, although with the ATW, it overestimates the charge by a factor of two, for all refrigerants. This is due to the design of the components for ATW being based on fixed-point COP of 4.9 (as a proxy for SCOP of 6.5), thus using less charge. Since in both cases the charge is similar or at least proportional across the range, it provides reasonable confidence in the estimation.

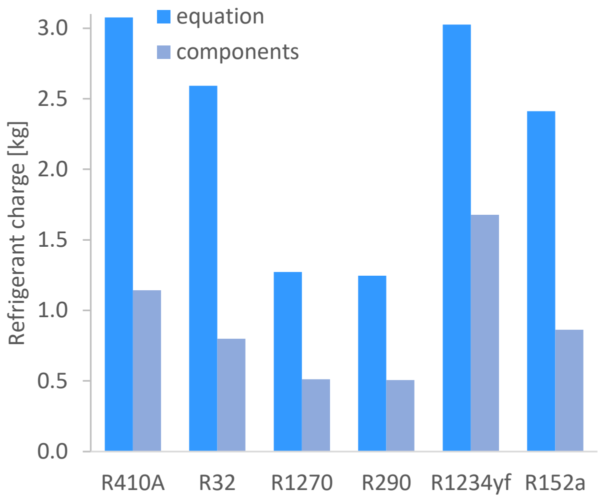


Figure 40: Comparison of equation (12) for ATW with SCOP = 6.5 and calculated component refrigerant mass

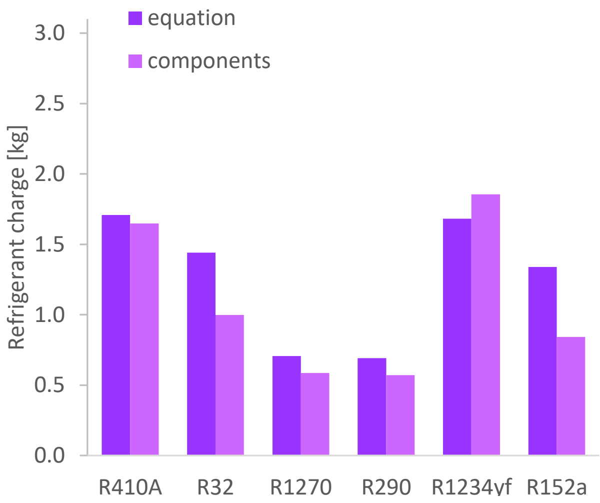


Figure 41: Comparison of equation (12) for LTW with SCOP = 6.5 and calculated component refrigerant mass

It is noted that whilst the values obtained from equation (12) and the component simulations are used for the present exercise, there are various recent studies focussing on DHPs with significantly lower R290 charge amounts. Equation (12) indicates a specific charge of about 95 g/kW for ATW and 55 g/kW for LTW, but the literature reports on values as low as 10 g/kW. Montagud et al/ (2014) developed a small ATW HP with 60 g/kW. Andersson et al. (2018) reports on a LTW DHP development with an R290 charge of 15 g/kW. Sánchez-Moreno-Giner et al. (2022) report just over 20 g/kW for LTW. Methler et al. (2022) and Dankwerth et al. (2020) as part of the LC150 project, demonstrated designs with as low as 20 g/kW at SCOP of 5. Navarro-Peris et al. (2014) developed a larger capacity HP (about 40 kW) with specific charge as low as 10 g/kW, although with a relatively low COP of around 3.8. Similarly, Palm (2014) developed a 30 kW LTW HP with 21 g/kW. As such, it should be born in mind that it can be anticipated that R290 DHPs will likely be more widely available with lower charges, for example, 200 g for a 10 kW unit; around one-third of the charge amounts used here (Table 10).

8 DHPs charge requirements in relation to charge limits

The applicability of refrigerant charge required for DHP products (section 7) can be gauged by considering them in light of the charge limits imposed by safety standards (section 2). This is carried out by comparing the charge limits and the required charge amounts, with regards to the assumed thermal load for the conditioned space (section 3).

ATW and LTW DHPs are seldom installed in the open area that they are intended to serve; instead, they are either installed outside (ATW) or within a cellar or utility room. Unlike split air conditioners and reversible air-to-air HPs, for example, the allowable refrigerant charge is unrelated to the thermal capacity/demand. This in many cases, for flammable refrigerants, poses additional hurdles. For instance, a DHP may be providing 20 kW of thermal energy to a 100 m^2 dwelling and using 1 kg of R290. Ordinarily, if in a 100 m^2 space, the 1 kg of R290 would be regarded as "safe" as a release of the entire charge would dilute to a concentration of 10% of the LFL. Conversely, if the DHP was located in a 6 m^2 utility room, a release of the entire charge would form a concentration of nearly two times the LFL, thus creating a potentially high-risk situation. Accordingly, alternative RMCs have to be implemented, as discussed in section 2. For outdoor units, the only constraint is the UCL. For indoor units, the charge (or releasable charge) needs to be limited to below 4×LFL for A2 or A3 refrigerants or 6×LFL for A2Ls if the case of no minimum area or perhaps about double this if a small minimum room area were to be specified, such as around 6 – 10 m^2. Otherwise for indoor units, a ventilated enclosure would be required, where the main constraint is the UCL.

Figure 42 and Figure 43 superimpose the various UCLs (4×LFL/6×LFL for no additional RMCs; 26×LFL/52×LFL for cases with A_{min} and 130×LFL/260×LFL for outdoor or VE) on to the estimated typical charge for ATW units.

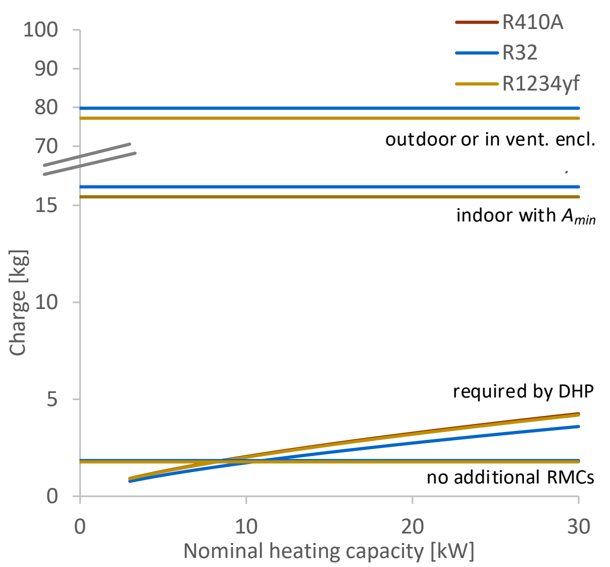


Figure 42: Charge assessment for R410A, R32 and R1234yf limits against required charge for ATW

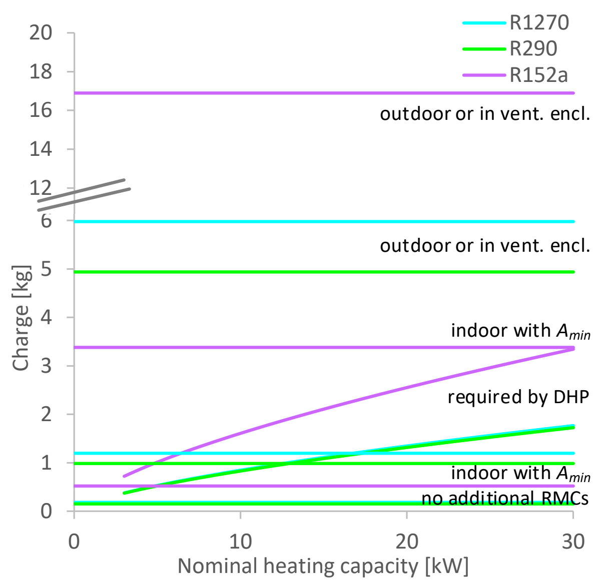


Figure 43: Charge assessment for R1270, R290 and R152a limits against required charge for ATW

Figure 44 and Figure 45 superimpose the various UCLs (4×LFL/6×LFL for no additional RMCs; 26×LFL/52×LFL for cases with A_{min} and 130×LFL/260×LFL for outdoor or VE) on to the estimated typical charge for LTW units.

Up to a NHC of about 25 kW, LTW units using R32 or R1234yf could rely on no additional RMCs. Up to the larger charge amount of about 3 kg, it could be feasible to rely on a minimum room area, which could be as small as 6 – 10 m^2, broadly consistent with utility rooms or cellars. Otherwise, if VEs were used there is effectively no refrigerant charge limit constraint.

For R1270, R290 and R152a, the situation is quite different, where only relatively small capacity DHPs would satisfy the charge criteria for no additional RMCs[18]. Whilst the DHP charge amount remains below 26×LFL, relying on A_{min} is broadly impractical for systems with charges in excess of about 0.4 kg of R1270 or R290 or around 1.3 kg of R152a, where room areas would have to exceed sizes of typical utility rooms or cellars. Therefore, to enable a greater level of freedom when applying these DHPs, other RMCs need to be relied upon, primarily the VE or a variation thereof.

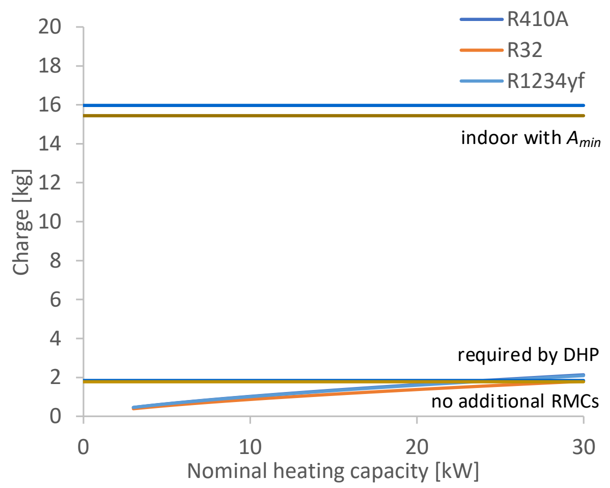


Figure 44: Charge assessment for R410A, R32 and R1234yf limits against required charge for LTW

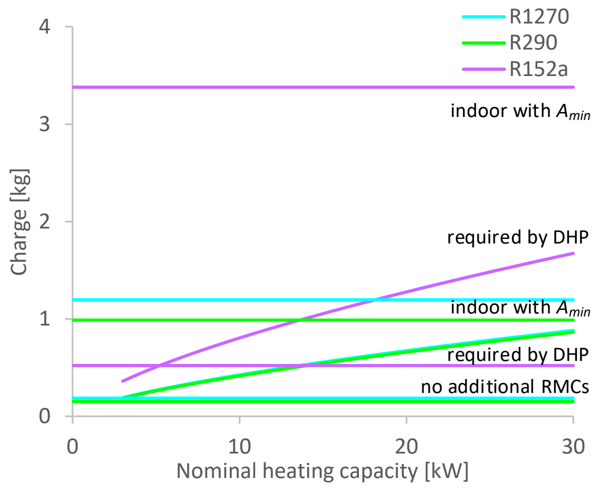
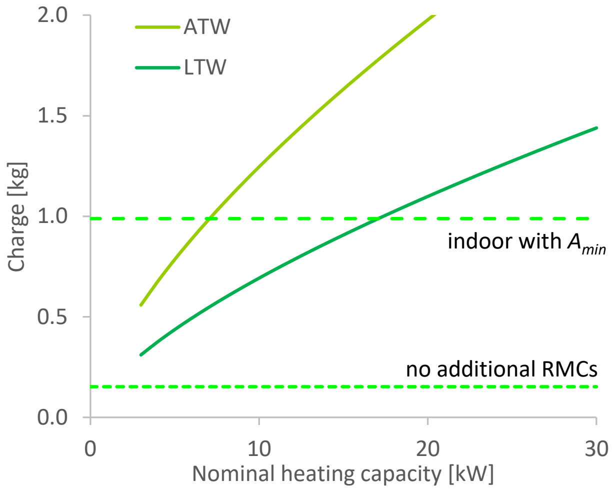


Figure 45: Charge assessment for R1270, R290 and R152a limits against required charge for LTW

In some instances, it may be feasible to implement ALRC to a DHP system, so that any release charge would lead to an A_{min} consistent with the size of cellars or utility rooms or indeed so that the releasable charge remains below 26×LFL. Whilst the conventional ALRC approach attempts to retain the charge in the outdoor unit, for systems entirely indoors, multiple shut-off valves can be used to separate the circuit into a number of discrete sections, each containing a small fraction of the charge.

Figure 47 provides the charge distribution for the R290 DHPs from the current assessment (sections 9 and 10). Assuming each component could be isolated, an ATW unit with, say 0.6 kg of R290 could have a releasable charge of less than 4×LFL. Alternatively, either ATW or LTW units with a charge of 0.9 kg could have a releasable charge limited so that A_{min} remains below about 10 m^2. However, across various studies, the charge distribution amongst DHP system components is seen to vary widely (Oltersdorf et al., 2013), so it would be feasible to optimise the charge distribution to be almost equal amongst system parts. In a well-designed system where the maximum releasable charge from any part would be 0.25 kg (from a 1 kg system) would yield A_{min} of around 6 m^2. Having multiple additional shut-off valves would have an incremental cost impact.

[18] Refer elsewhere to the current findings of the LC150 project.



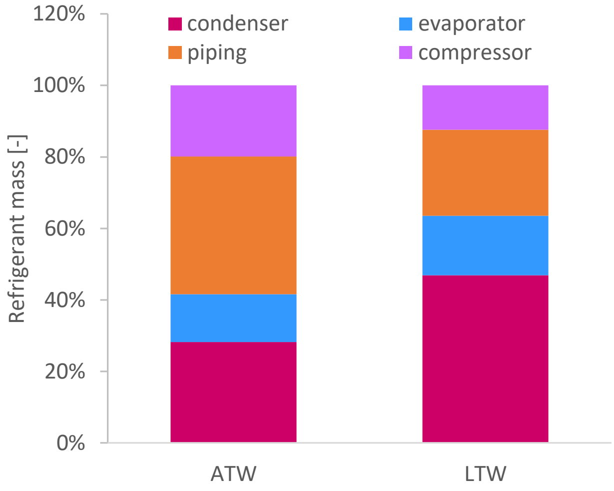


Figure 46: Charge for R290 limits against required charge for both DHPs

Figure 47: Example charge distribution for 10 kW R290 DHPs

Note that the refrigerant quantities for DHPs used here are indicative and – as seen in the data in section 4 – there is a wide variation in charge for all types of DHPs, depending upon functionality, design approach, structure, selection of components and so on. Although the charge amounts are based on very high efficiency (SCOP = 6.5), the data shows that efficiency plays a minor part in the charge for most systems and that considerable reduction in charge for high efficiency models can be achieved.

In general, DHPs with alternative refrigerants with GWP < 150 can satisfy the range of thermal capacities required for space heating and DHW of single-family dwellings (i.e., at least up to about 30 kW and probably more). Neglecting cost and environmental implications (see sections 9 and 10), R1270, R290, R1234yf and R152a can easily satisfy the application requirements within the constraints of charge limits. However, with R152a and particularly R1270 and R290, additional RMCs may need to be applied for LTW, whether it is elaborate ALRC or a VE.

9 Component material mass and costs

Quantifying the relative impact of the use of construction materials, considering cost and emissions implications, is important when selecting alternative refrigerants. Accordingly, a detailed assessment of changes in material requirements due to differences in HX, compressor, piping and other DHP components arising from a switch of refrigerants, has been carried out. This has been executed through analysis of component catalogue data, requirements of safety standards, previous implementation projects and detailed design and performance simulation of HXs.

There are further implementation costs associated with transition to alternative refrigerants.

These can include usual R&D activities (planning, study time, testing, sourcing components and materials, prototyping, etc.), changes to production line (reorganisation/sequencing, charging equipment, additional detection and ventilation equipment, level of component and tightness testing, electrical tests, etc.), possible rearrangement of storage areas and warehousing and so on. The extent of the costs associated with these aspects depend upon production output, number of models, existing production set-up, selected mitigation measures and so on. Cost implications are correspondingly diverse and it is thus impossible to draw any generalised conclusions on this aspect. But as discussed in section 1, such redesign and replacement of production equipment is already part of the usual redevelopment cycle anyway.

9.1 Piping

Refrigerant piping is part of finned-tube heat exchangers, for linking system components, including interconnecting piping between the indoor and outdoor units. In general, pipe size is selected to provide a tolerable pressure drop (in terms of equivalent change in saturation temperature). Figure 48, Figure 49 and Figure 50 shows results of calculations for piping for all refrigerants listed in ISO 817, plotted over saturation pressure corresponding to 0°C as a reference.

For each, refrigerant mass flow is calculated based on 25°C and 10 kW heating capacity and the internal diameter (Figure 48) is determined according to a pressure drop equivalent to 0.1 K/m for that refrigerant, be it liquid or vapour phase. As a general trend, lower pressure refrigerants demand a larger pipe diameter, which is primarily due to 0.1 K corresponding to a smaller frictional pressure loss than for higher pressure refrigerants. For a given saturation pressure, the scatter arises from different thermophysical properties of the refrigerants, such as density and viscosity.

With knowledge of the pipe diameter and its expected maximum operating pressure (here, taken as that corresponding to 65°C saturation temperature), the pipe wall minimum thickness can be calculated. The method prescribed in EN 14276-2 (2020) is used, assuming seamless copper tube at 100°C. For a fixed pipe diameter, the thickness increases smoothly with greater pressure. However, the scatter occurring in Figure 49 does so because of the variations in pipe diameter already seen in Figure 48.

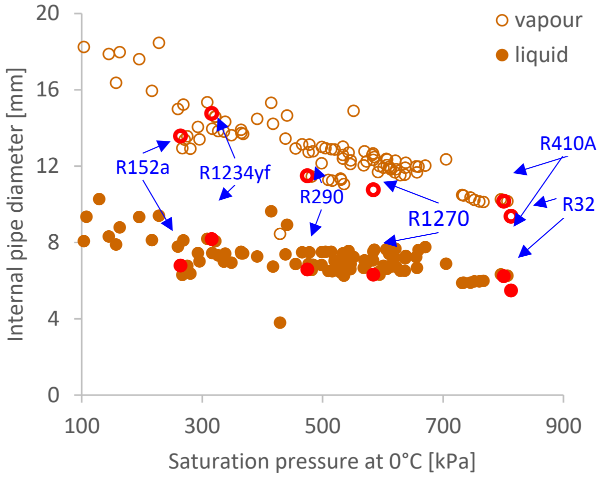


Figure 48: Internal pipe diameter for various refrigerants, assuming 25°C, 10 kW refrigerating capacity and 0.1 K/m pressure drop

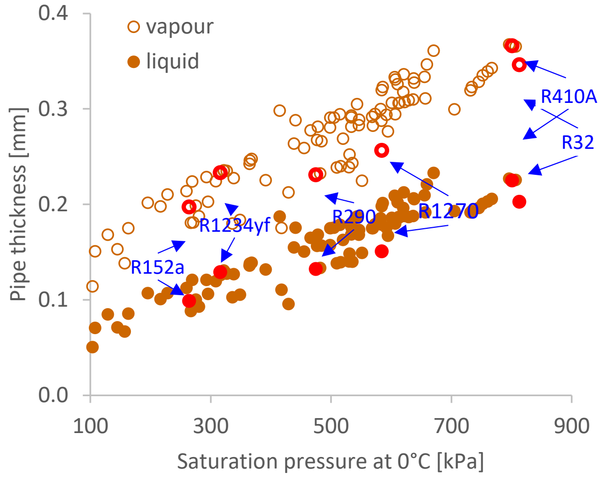


Figure 49: Pipe thickness for Figure 48, based on a maximum operating pressure corresponding to 65°C

Knowing the diameter and thickness allows the mass of copper to be calculated, as shown in Figure 50. Broadly, the higher the saturation pressure, the greater the material mass required. Although higher pressure refrigerants enjoy smaller diameters, they still require thicker walls; generally, the smaller diameter does not offset the increased mass associated with the thicker walls. Some exceptions do arise, such as with R1234yf and R410A, where the higher viscosity and smaller enthalpy difference (leading to higher mass flow rates) result in disproportionately greater pressure drops.

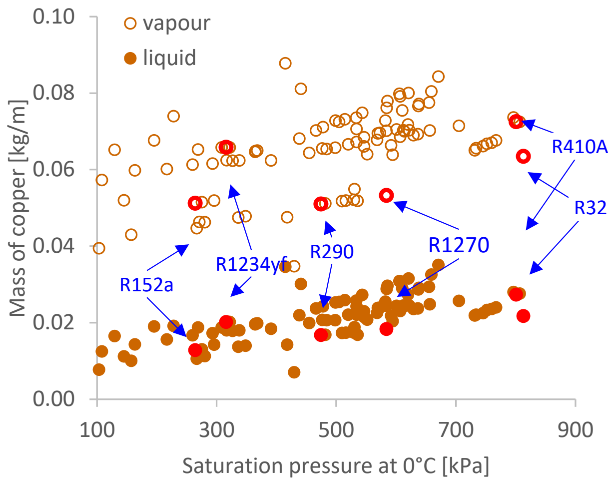


Figure 50: Comparison mass of copper per metre piping for various refrigerants

A summary of the results for the selected conditions are presented in Table 11, listing outside diameter and mass of copper. Switching from R410A to any of the alternative refrigerants always results in lower mass of copper. R32 requires 10% less, R1270 and R290, about 30 – 40% less, R1234yf, between 10 – 25% less and 30 – 50% less for R152a. All refrigerants, except R1234yf, require less mass than R32.

It is noted, however, that refrigeration pipe is usually supplied in incremental diameters and usually, two, three or four thicknesses, so optimal selection for a given refrigerant is seldom possible. Moreover, at least for lower pressure cases, overly thin pipes are avoided since further protection is usually required against external impacts, damage, etc.

Table 11: Tube diameter and mass per metre

Refrigerant		R410A	R32	R1270	R290	R1234yf	R152a
Vapour	Outside diameter [mm]	10.9	10.1	11.3	12.0	15.2	14.0
	Mass [kg/m]	0.073	0.063	0.053	0.051	0.066	0.051
	Cost [€/m]	0.58	0.50	0.42	0.41	0.53	0.41
Liquid	Outside diameter [mm]	6.7	5.9	6.6	6.8	8.4	7.0
	Mass [kg/m]	0.027	0.022	0.018	0.017	0.02	0.013
	Cost [€/m]	0.22	0.18	0.14	0.14	0.16	0.10

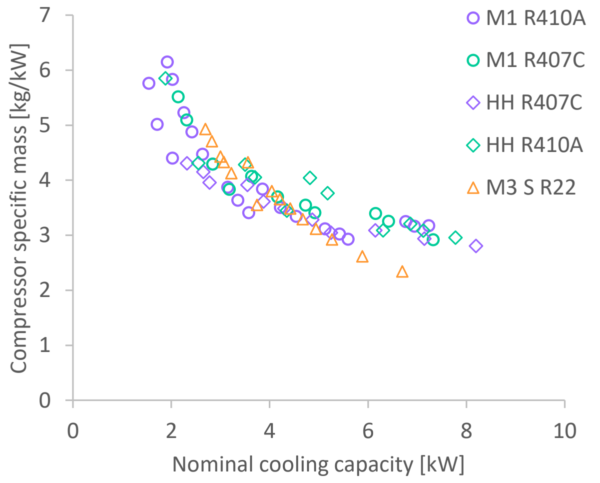
9.2 Compressors

As seen in Table 7, the alternative refrigerants have different operating pressures and swept volumes, which implies differences in the compressor design and construction. Consideration may be given to identify whether variations in material requirements and associated costs also differ. An exercise is carried out to identify the possible changes in compressor mass and volume associated with alternative refrigerants.

DHPs generally use hermetic rotary compressors or hermetic scroll compressors and sometimes hermetic or semi-hermetic reciprocating types. Catalogue data from different manufacturers has been collated for the same class of rotary and hermetic reciprocating compressors and scroll and for different refrigerants. Analysis of the data reveals implications associated with adopting other alternative refrigerants. Figure 51 compares the specific mass (compressor mass divided by nominal cooling capacity at standard conditions). When comparing compressors from the same manufacturer for high pressure (R410A) and medium pressure (R22 or R407C) refrigerants, no difference is observed. In some cases, high pressure refrigerants are associated with greater mass whereas in other cases, greater mass is for the medium pressure refrigerants. Specifically, despite R407C having over 1.5 times the swept volume of R410A, the required mass is usually within ±3%.

It is therefore reasonable to regard the swept volume (compressor displacement) as having no discernible impact upon the mass of the compressor. Indeed, the same trade-off between size and wall thickness, as discussed for pipe mass – similarly applies to R290, R1270, R1234yf and R152a. Further, motor materials (copper, aluminium, rare earth metals, etc.) would not change for the same shaft power.

The same conclusions may be drawn from the data for compressor specific volume (shell volume divided by nominal cooling capacity); Figure 52. Again, there is no clear difference between the high and medium pressure refrigerant. Based on this data it is concluded that the same would be applicable to the other alternative refrigerants under consideration.



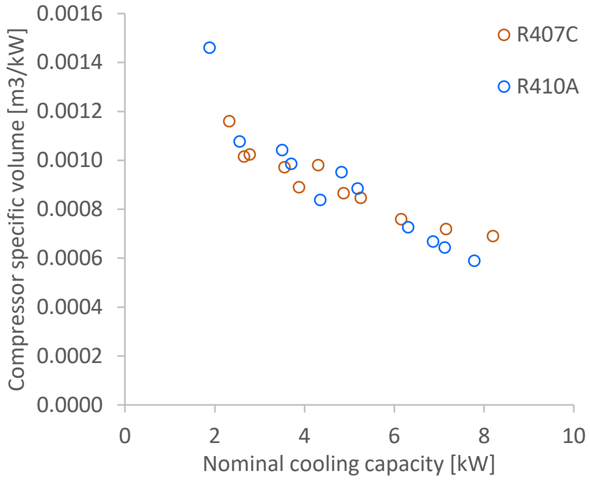
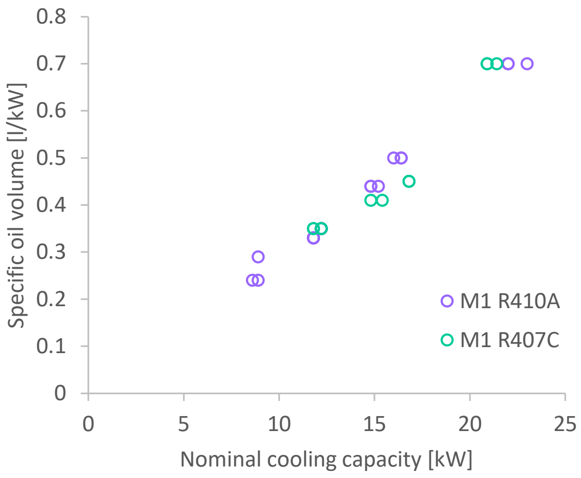


Figure 51: Comparison of rotary compressor specific mass for R410A, R22 and R407C

Figure 52: Comparison of rotary compressor specific volume for R410A, R22 and R407C

Figure 53 shows the variation of compressor oil volume for high and medium pressure refrigerants. Again, there is no distinguishable difference between the two refrigerants. Furthermore, due to the desire to minimise refrigerant charge for class A3 refrigerants, additional efforts are made in the development of R290 compressors to reduce oil charge (e.g., Gao et al., 2012; Gao et al., 2014; Zhang et al., 2014; Wu and Chen, 2015). The same is likely to apply to R152a.



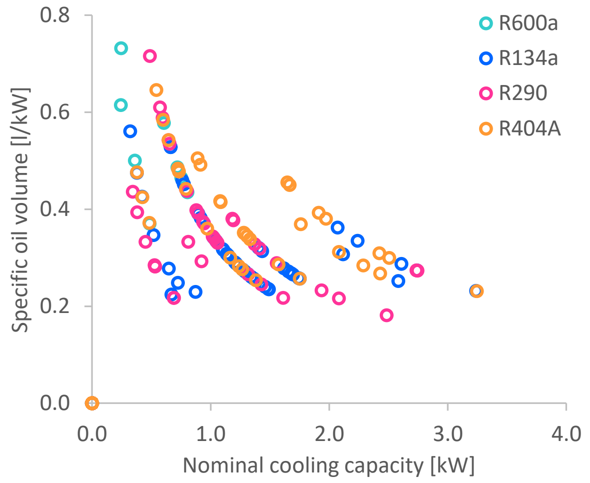


Figure 53: Comparison of rotary compressor specific oil volume for R410A, R22 and R407C

Figure 54: Comparison of hermetic reciprocating compressor specific oil volume for various refrigerants

In carrying out the same assessment on hermetic reciprocating compressors for refrigerants R404A, R290, R134a and R600a (Figure 54, Figure 55 and Figure 56), there is also no consistent differentiation between specific mass and specific volume across these refrigerants. This further provides confidence in the conclusion that the refrigerant characteristics do not impose a cost or dimensional impacts on the compressor characteristics. The observable differences arise from different compressor frames, which are seen to be used for all refrigerants.

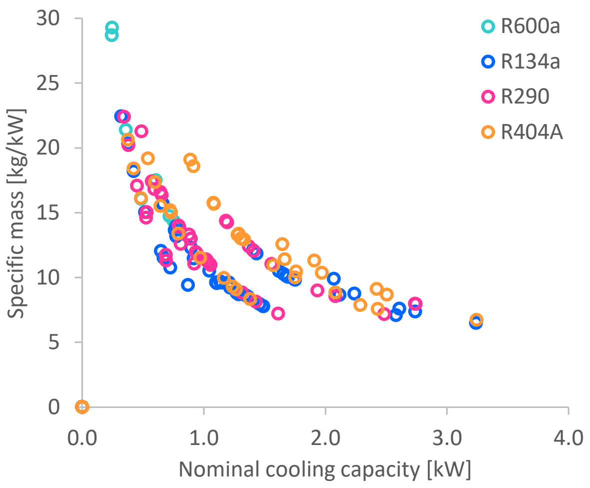


Figure 55: Comparison of hermetic reciprocating compressor specific mass for various refrigerants

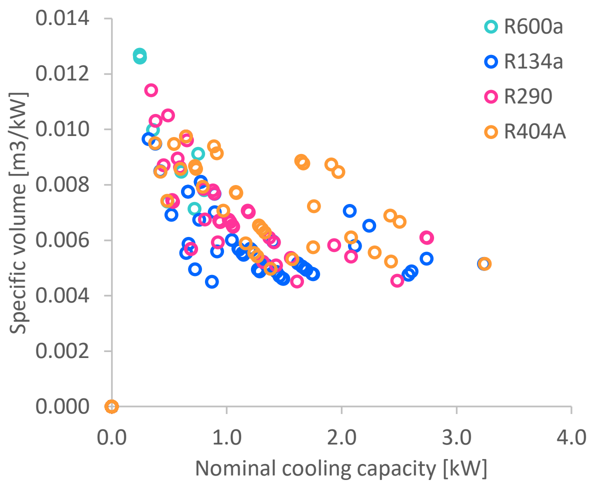


Figure 56: Comparison of hermetic reciprocating compressor specific volume for various refrigerants

In conclusion, there is no evidence to suggest that a shift to an alternative refrigerant (with lower pressure) would lead to a substantive change in compressor mass or volume or quantity of oil and as such, no impact on cost. There would be no change to the size of the unit or expenditure on materials. Furthermore, considering that usually, DHPs housings generally have significant internal free space and that housing may be used for several different NHC models, it is unlikely that any minor differences in compressor volume would pose an impact.

Compressor development is a continuous process, involving numerous iterations of refinements for improving efficiency, reliability and cost reduction. Whichever refrigerants are used on a larger-scale in DHPs, compressors will benefit from this continuous development.

9.3 Expansion devices, valves and other line components

Expansion devices may include capillary tubes, short-tube restrictors, thermostatic and electronic expansion valves (EEV). Most higher efficiency models use EEVs.

Refrigerant type has a negligible influence in expansion device cost and size. Arguably the connections may differ slightly (see section 9.1 on piping) but the vast majority of the EEV materials are related to its function and are unrelated to the refrigerant pressure or such properties.

The same considerations and conclusions apply to other valves and line components, such as solenoid valves, reversing valves and hand valves.

9.4 Heat exchangers

The construction of the primary HX – evaporator and condenser – can be notably influenced by refrigerant selection and the desired contribution to system efficiency. HXs are designed or selected to help achieve a desired system efficiency. Different construction characteristics affect their performance and these in turn often infer cost and resource implications, in terms of mass of materials (copper, aluminium), features (rifled tubes and textured fins) and manufacturing processes.

The majority of ATW DHPs employ finned-tube type evaporators, whilst LTW almost entirely use brazed plate HX (PHX). Similarly, LTW DHP use brazed PHX for both evaporators and condensers.

Within this narrow scope, there still exists a variety of different HX designs; tube shapes and sizes, tube alignments, circuitries, air-side enhancements, internal enhancements with different refrigerants and so on, the optimal of which can differ for each specific refrigerant. There are numerous strategies applied for examining the design, rationale and methodology of complex HX circuitry (e.g., Joppolo et al., 2015; Cotton et al., 2018; Wu et al., 2012; Li et al., 2018; Li et al., 2021; Mancini et al., 2019; Satalagaon et al., 2019; Blanco Castro et al., 2005). However, for brevity, within the scope of the current work HX analysis is conducted on the basis of relatively basic designs. It is considered that the conclusions arising from this would be representative of the optimised and refined designs expected from DHPs manufacturers. For this exercise, HX performance is assessed using IMST-ART simulation software[19], where various finned-tube and plate HXs are simulated, based on standard fixed-point rating conditions.

HX design and selection is not only based on thermophysical properties but other parameters that relate to the application and safety considerations. In terms of application and safety considerations, with a non-flammable refrigerant the objective may solely be lowest HX mass (once requisite capacity and efficiency are satisfied). R410A has the luxury to choose the HX providing smallest mass and size (without exceeding the target pressure drop). Conversely, with an A3 refrigerant, a key objective may be smallest refrigerant charge (whilst maintaining a pressure drop that will not lead to degradation of efficiency below the target level), which could be to the detriment of material mass and/or physical volume. (Although there is an approximate correlation between HX mass and refrigerant charge). With lower flammability yet more costly refrigerants, such as R1234yf, there remains a motivation to minimise charge for economic reasons. Due to its inherent high pressure drop, though larger HX must usually be selected if the efficiency target is to be met and accordingly refrigerant and material mass are unavoidably high.

Ultimately, the HX analysis is carried out on the basis of the system balance-points detailed in Table 9, to account for the cycle efficiencies of the alternative refrigerants. Whilst a NHC of 10 kW has been used to represent the larger-end capacity of the high population products, the findings are deemed to be extrapolatable to other NHCs. The outputs from selecting the most appropriate HX design include mass of copper, aluminium and operating refrigerant charge.

9.4.1 Finned tube evaporator (ATW)

Evaporator designs consistent typical ATW type DHPs were applied, with a target evaporating capacity of 8.37 kW, to yield a 10 kW condensing capacity. The baseline characteristics and subsequent design variations are listed in Table 12. Output data is presented graphically in Annex A for each of the alternative refrigerants.

Table 12: Finned tube evaporator configurations

Parameter	Base	Variations
Tube configuration	U, staggered	
No tubes [-]	3 × 50	3 × 50, 51 … 70
No circuits [-]	25	26, 27, … 35
Width [m]	0.7	
Nominal outside diameter [mm]	5.0	5.5, 6.0, 6.5, 7.0
Tube thickness [mm]	0.5	
Airflow rate [m^3/h]	7900	

Assuming the manufacturer intends to retain the exiting frame plate size, the evaporator width is retained throughout. Only tube diameter, number of tubes and circuitry is varied. The HX calculation was

[19] http://www.imst-art.com/?page_id=67

successively iterated with increasing refrigerant mass flow until the design evaporator capacity was achieved. Objective parameters are the saturated outlet temperature of the refrigerant from the evaporator ($T_{e,sat}$) and thus the corresponding pressure drop (expressed as the change in saturation temperature). Other than the direct effect of the refrigerants thermophysical properties on heat transfer coefficients and pressure drop, HX length, tube diameter and number of circuits affect the ETD. The suitable HX design was selected to provide a close match to the target ETD and assigned tolerable pressure drop, as listed in Table 9. A tolerance of ±0.1 K is afforded to the selection. Where there were multiple designs that provided the target ETD and pressure drop, that with the lowest material mass was selected.

Table 13: Values corresponding to evaporator balance points

	R410A	R32	R1270	R290	R1234yf	R152a
Tube diameter	5.5	5	5.75	6	7.5	7.5
No of tubes	50	52	50	50	62	52
No of circuits	25	27	25	25	31	27
Pressure drop [K]	0.4	0.4	0.4	0.4	0.4	0.4
ETD [K]	5.9	6.2	6.6	6.7	6.4	7.5
Mass of steel [kg]	0	0	0	0	0	0
Mass of aluminium [kg]	8.0	8.7	8.0	8.0	10.0	8.7
Mass of copper [kg]	7.7	7.5	8.1	8.5	13.4	11.6
Mass of metal [kg]	15.7	16.2	16.1	16.5	23.3	20.3
Refrigerant mass [kg]	0.11	0.07	0.05	0.05	0.17	0.09
Coil volume [m3]	0.059	0.063	0.059	0.059	0.073	0.063
Component cost [€]	81.7	81.6	84.7	87.8	131.8	114.8

A graph comparing the mass of materials for the evaporator is shown in Figure 57. R410A, R32, R1270 and R290 all use a similar quantity of metal. However, since R1270 and R290 require slightly larger diameter tubes, the mass of copper is greater, thereby increasing the overall cost, albeit by about 5%. R1234yf and R152a both require considerably more aluminium and copper and as a result the total mass is about 40% greater and cost is 50% more than R410A.

9.4.2 Brazed plate evaporator (LTW)

Evaporator designs consistent typical LTW DHPs were applied, with a target thermal capacity of 8.58 kW. The baseline characteristics and subsequent design variations are listed in Table 12. Output data is presented graphically in Annex A for each of the alternative refrigerants.

Table 14: Brazed plate evaporator configurations

Parameter	Base	Variations
Channel type	M	
No plates [-]	16	18, 20, … 100
Length [m]	0.4	0.4, 0.45, … 0.75
Width [m]	0.12	
Water flow rate [kg/h]	2470	

Assuming the manufacturer intends to retain the exiting frame plate size, the evaporator width is retained throughout and only number of plates and length is varied.

Objective parameters are the saturated outlet temperature of the refrigerant from the evaporator ($T_{e,sat}$) and thus the corresponding pressure drop (expressed as the change in saturation temperature). Other than the direct effect of the refrigerants thermophysical properties on heat transfer coefficients and pressure drop, HX length, tube diameter and number of circuits affect the ETD. The suitable HX design was selected to provide a close match to the target ETD and assigned tolerable pressure drop, as listed in Table 9. A tolerance of ±0.1 K is afforded to the selection. Where there were multiple designs that provided the target ETD and pressure drop, that with the lowest material mass was selected.

The mass of PHX varies amongst manufacturers due to different construction characteristics. A general approximation based on data from various manufacturers can be made from equation (13).

$$m_{hx} = c \times A_p + N_p \times m_p \tag{13}$$

where A_p is the area of the plate, N_p is the number of plates and m_p is the mass of a single plate. Presently, the constant c = 30 has been used, although it is observed that this varies widely amongst manufacturers (in some cases exceeding c = 120).

Table 15: Values corresponding to evaporator balance points

	R410A	R32	R1270	R290	R1234yf	R152a
No. of plates	24	25	30	36	80	56
Length [m]	0.6	0.65	0.48	0.44	0.47	0.42
Width [m]	0.12	0.12	0.12	0.12	0.12	0.12
Pressure drop [K]	0.1	0.1	0.1	0.1	0.1	0.1
ETD [K]	1.9	2.0	2.4	2.5	2.4	3.0
Mass of stainless steel [kg]	6.9	7.7	6.5	6.8	14.1	9.3
Mass of aluminium [kg]	0	0	0	0	0	0
Mass of copper [kg]	0	0	0	0	0	0
Mass of metal [kg]	6.9	7.7	6.5	6.8	14.1	9.3
Refrigerant mass [kg]	0.36	0.22	0.10	0.10	0.43	0.17
Coil volume [m3]	0.0081	0.0092	0.0081	0.0089	0.0212	0.0133
Component cost [€]	34.5	38.5	32.4	34.0	70.4	46.3

Results of the mass of metal is included in Figure 58. Again, values for R410A, R32, R1270 and R290 are similar and that for R152a and particularly R1234yf is significantly greater. Whilst the physical size of R410A and R32 PHX are smaller than for R1270 and R290, the mass of material is similar due to the construction of PHX (see section 9.4.4) and costs reflect this compared to R410A. R152a requires about 50% more mass of R1234yf about double the mass and associated cost.

9.4.3 Brazed plate condenser (ATW and LTW)

Condenser designs consistent typical DHPs were applied, with a target thermal capacity of 10.0 kW. The baseline characteristics and subsequent design variations are listed in Table 16. Output data is presented graphically in Annex A for each of the alternative refrigerants.

Table 16: Brazed plate condenser configurations

Parameter	Base	Variations
Channel type	M	
No plates [-]	16	16, 18 … 42
Length [m]	0.4	0.35, 0.40, … 0.90
Width [m]	0.12	
Water flow rate [m^3/h]	1.75	

Assuming the manufacturer intends to retain the exiting frame plate size, the condenser width is retained throughout. Only tube diameter, number of tubes and circuitry is varied.

Objective parameters are the saturated inlet temperature of the refrigerant to the condenser ($T_{c,sat}$) and thus the corresponding pressure drop (expressed as the change in saturation temperature). Other than the direct effect of the refrigerants thermophysical properties on heat transfer coefficients and pressure drop, HX length, tube diameter and number of circuits affect the ETD. The suitable HX design was selected to provide a close match to the target ETD and assigned tolerable pressure drop, as listed in Table 9. A tolerance of ±0.1 K is afforded to the selection. Where there were multiple designs that provided the target ETD and pressure drop, that with the lowest material mass was selected.

Table 17: Values corresponding to ATW condenser balance points

	R410A	R32	R1270	R290	R1234yf	R152a
No. of plates	10	9	12	12	22	16
Length [m]	0.67	0.68	0.52	0.52	0.44	0.40
Width [m]	0.12	0.12	0.12	0.12	0.12	0.12
Pressure drop [K]	0.1	0.1	0.1	0.1	0.1	0.1
ETD [K]	5.9	6.2	6.6	6.7	6.4	7.5
Mass of stainless steel [kg]	4.64	4.47	3.91	3.94	4.75	3.55
Mass of aluminium [kg]	0.0	0.0	0.0	0.0	0.0	0.0
Mass of copper [kg]	0.0	0.0	0.0	0.0	0.0	0.0
Mass of metal [kg]	4.64	4.47	3.91	3.94	4.75	3.55
Refrigerant mass [kg]	0.24	0.17	0.10	0.10	0.43	0.17
Coil volume [m3]	0.0038	0.0035	0.0035	0.0035	0.0054	0.0036
Component cost [€]	23.2	22.4	19.5	19.7	23.8	17.7

Table 18: Values corresponding to LTW condenser balance points

	R410A	R32	R1270	R290	R1234yf	R152a
No. of plates	24	18	24	24	28	24
Length [m]	0.76	0.73	0.60	0.61	0.69	0.51
Width [m]	0.12	0.12	0.12	0.12	0.12	0.12
Pressure drop [K]	0.1	0.1	0.1	0.1	0.1	0.1
ETD [K]	1.9	2.0	2.4	2.5	2.4	3.0
Mass of stainless steel [kg]	8.8	7.0	6.9	7.0	8.9	5.9
Mass of aluminium [kg]	0.0	0.0	0.0	0.0	0.0	0.0
Mass of copper [kg]	0.0	0.0	0.0	0.0	0.0	0.0
Mass of metal [kg]	8.78	7.00	6.93	6.99	8.87	5.87
Refrigerant mass [kg]	0.80	0.43	0.28	0.27	0.84	0.34
Coil volume [m3]	0.0103	0.0075	0.0082	0.0082	0.0109	0.0069
Component cost [€]	43.9	35.0	34.7	34.9	44.3	29.4

Results of the mass of metal is included in Figure 58. For both ATW and LTW condensers, values for R410A, R32, R1270, R290 and R152a are similar but R1234yf is slightly greater. Whilst the physical size of R410A and R32 PHX are smaller than for R1270, R290 and R152a, the mass of material is similar due to the construction of PHX (see section 9.4.4) and costs reflect this. R1234yf has greater mass and higher costs than the other refrigerants.

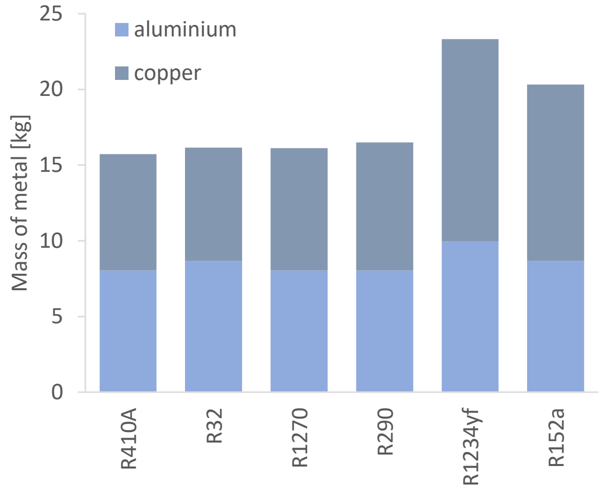


Figure 57: Material mass (copper and aluminium) for the finned tube ATW evaporator

Figure 58: Material mass (primarily stainless steel) for the various PHX

9.4.4 Remark on material mass of plate HX

Observation of resultant material (steel) mass of PHX across the various alternative refrigerants may appear counter-intuitive. In particular, values for R1270, R290 and R152a (especially for condensers) are lower than values for R410A and R32, despite the required surface area for the PHX being larger. An explanation is provided here.

Simulation results (Annex A) show that some refrigerants "prefer" longer flow paths (greater PHX height) whilst other "prefer" shorter flow paths. However, generally refrigerants characteristic of the latter case requires a larger transfer area and thus more plates. Despite this, the PHX mass is often less.

This is attributed to the heavy end-plates of PHXs. A single end-plate may have the mass equivalent to five to 10 internal plates (depending upon the manufacturer). So, if the PHX requires a greater height – but fewer plates – the overall impact can be a greater total PHX mass.

An example is shown below; see Table 19, Figure 59, Figure 60, Figure 61 and Figure 62. For the DHP condenser at ATW conditions, the mass of the R32 PHX would be 4.5 kg, which is greater than the PHX for R152a, at 3.6 kg, even though R152a requires twice as many plates and 35% more surface area. This is because the ends are a lot heavier than the individual plates; from equation (13), the end plates for R32 would be 2.4 kg, compared to 1.4 kg for R152a. Even if selecting twice the number of plates for R32 (i.e., 18) so that the height reduces to 0.50 m – now with a pressure drop of 0.01 K – the total mass is still greater (4.8 kg) because there are so many more plates. Reduction in mass (due to lower area) arising from the smaller pressure drop would be less than 1%.

Table 19: Comparison of PHX condenser mass for selected cases

Refrigerant	R32	R32	R152a
Condenser capacity [kW]	10	10	10
$T_{d,sat}$ [°C]	38.7	38.7	40.0
T_d [°C]	86.7	86.7	68.1
$T_{f,in}$ [°C]	30	30	30
Refrigerant pressure drop [K]	0.1	0.01	0.1
Number of plates	8	18	16
Plate height [m]	0.65	0.50	0.45

Refrigerant	R32	R32	R152a
Plate width [m]	0.12	0.12	0.12
Total transfer area [m2]	0.63	1.2	0.85
PHX mass [kg]	4.47	4.77	3.55
PHX end-plate mass [kg]	2.42	1.80	1.44

Thus, the unexpected result in PHX mass is a consequence of refrigerant thermophysical properties and practical construction characteristics of typical PHX.

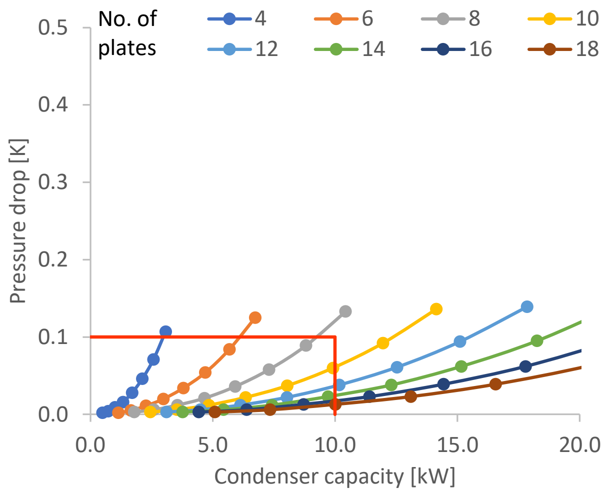


Figure 59: Balance point for R32 PHX condenser *for ATW DHP*; plate height = 0.4, 0.45, 0.5, … 0.7 m (as data-points)

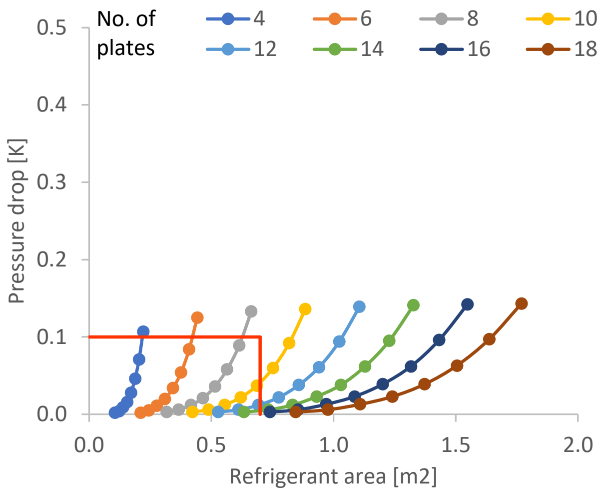


Figure 60: Surface area of R32 PHX condenser for *ATW DHP* at balance point; plate height = 0.4, 0.45, 0.5, … 0.7 m (as data-points)

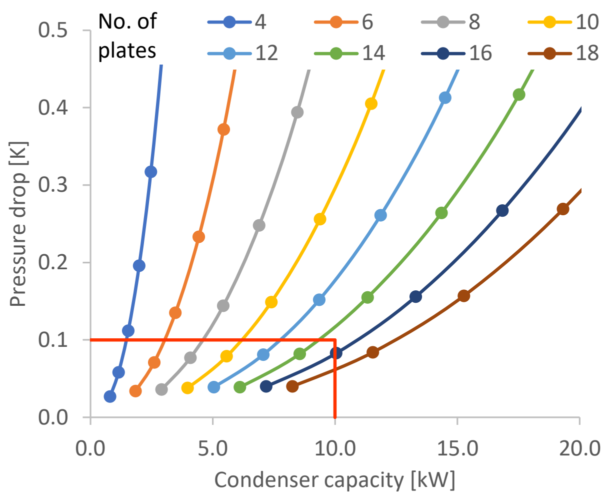


Figure 61: Balance point for R152a PHX condenser *for ATW DHP*; plate height = 0.4, 0.45, 0.5, … 0.7 m (as data-points)

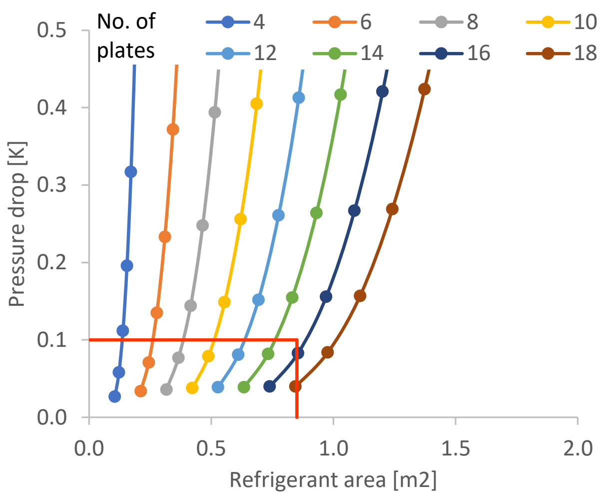


Figure 62: Surface area of R152a PHX condenser for *ATW DHP* at balance point; plate height = 0.4, 0.45, 0.5, … 0.7 m (as data-points)

9.5 Costs of safety features

When shifting from an A1 or A2L refrigerant to another refrigerant that is flammable, further constraints on charge limits may be invoked and additional costs may be incurred in order to apply further RMCs. Costs associated with compliance to the requirements to address flammability within IEC 60335-2-40 are provided and a summary of those costs is listed in Table 20.

Table 20: Summary of cost implications associated with individual safety measures

Safety measure	Cost implication	Cost for 10 kW DHP using A3 [€]
Refrigerant quantity limits	None	€0
Charge minimisation	Negative – usually leads to lower refrigerant and material cost	€0
Limited releasable charge	Negligible up to cost of sensors and valve(s)	€0 (high efficiency, reversible) to €20 (heating only)
Elaborate limited releasable charge	Additional solenoid valves	€30 (two additional)
Integral airflow	Additional fan/motor	€5 – 50
Ventilated enclosure	Including fans, ducting and sensors	€40 – 55
Naturally ventilated enclosure	Conduit and switch	€15
Leak detection	Sensors, controller	€3+
No ignition sources	Variable, but usually negligible for DHPs	€0
Enhanced tightness	Negligible, if production numbers are > 10,000's	€5
Marking and instructions	Negligible	€0

It is assumed that all systems will be ETRS, if not to reflect best practice, it always leads to a reduction cost when implementing other RMCs.

As seen in section 2, greatest increases in ACL are achieved through use of IAF and ALRC. Since most DHPs are assumed to be reversible (either for defrost purposes or extended functionality), they employ a reversing valve, which can also be used for ALRC to prevent backflow of refrigerant to the indoor unit. Thus, only a shut-off valve is required in the liquid line. For higher efficiency models, this can be satisfied with an EEV, otherwise a solenoid valve would be needed; at a cost of €15 – 30, depending upon the tube diameter. For both ALRC and IAF, leak detection is necessary. Costs associated with the controller are negligible, but sensors (ultrasonic or gas) cost anything from €1 upwards; employing three ultrasonic sensors for the leak detection system could cost €3 or with gas sensors the cost can extend to €10s (e.g., metal oxide or catalytic) or €100s (e.g., infra-red), at present. Thus, to switch to an A2 or A3 refrigerant, the costs incurred for safety features would unlikely exceed €3 or €30 when ALRC is applied (for heating only, lower efficiency models), depending upon the mitigation strategy and DHPS capacity.

Costs for individual and combined RMCs for the ATW and LTW DHPs are listed in Table 21. Overall costs are provided, either based on minimum or maximum cost depending upon the combination. For ATW, the cheapest option is locating the entire unit outdoors; this is not considered to be an option for LTW. For the overall cost assessment (section 9.8), the maximum overall cost from Table 21 is chosen.

Table 21: Estimation of overall cost implications associated with safety measures

Type	Safety measure	R410A	R32	R1270	R290	R1234yf	R152a
ATW	ETRS [€]	5	5	5	5	5	5
	Outdoor [€]	0	0	0	0	0	0
	Indoor with ALRC and IAF [€]	0	0	50	50	0	20
	Indoor with VE [€]	0	0	40	40	0	40
	Indoor with naturally VE [€]	0	0	15	15	0	15

Type	Safety measure	R410A	R32	R1270	R290	R1234yf	R152a
	Overall highest [€]	5	5	55	55	5	45
	Overall lowest [€]	5	5	5	5	5	5
LTW	ETRS [€]	5	5	5	5	5	5
	Outdoor [€]	n/a	n/a	n/a	n/a	n/a	n/a
	Indoor with ALRC and IAF [€]	0	0	50	50	0	20
	Indoor with VE [€]	0	0	40	40	0	40
	Indoor with naturally VE [€]	0	0	15	15	0	15
	Overall highest [€]	5	5	55	55	5	45
	Overall lowest [€]	5	5	20	20	5	20

9.5.1 Refrigerant quantity limits

For flammable refrigerants, the possibility of a leak being ignited by an uncontrolled potential source of ignition within the wider area needs to be minimised. This is achieved by limiting the quantity of refrigerant that can leak out such that potentially flammable mixture will not form beyond the RACHP equipment itself. Thus, refrigerant quantities may be limited according to an “allowable charge limit” (ACL), that is based on the size of the space that the system is located within, installation characteristics of the system and auxiliary equipment and the flammability and/or toxicity characteristics of the refrigerant. Further, there may be an “upper charge limit” (UCL) which is a capped quantity that overrules the ACL.

Materials: Essentially, implementation of the quantity limit does not necessarily demand additional hardware. However, where a manufacturer or installer wishes to use a certain refrigerant in a situation with restrictive quantity limits further design tasks may be required such as charge minimisation (see below), limited releasable charge techniques (see below) and/or multiple refrigerant circuits.

Costs: Whilst there are no direct costs associated with refrigerant quantity limits, practically, any costs involved are associated with the R&D activities to minimise charge (see below) or applying limited releasable charge techniques (see below), as and when required. Splitting the system into two or more refrigeration system circuits (to halve the charge) can result in a significant cost increase, up to 1.5 times the cost of the original system.

9.5.2 Refrigerant charge minimisation

In order to comply with refrigerant quantity limits, or indeed to lower the risk associated with a given system, charge minimisation may be exercised. Historically, little consideration was given to the reduction of refrigerant charge and often oversized refrigerant reservoirs were applied to prolong the duration that a leaky system would operate for without the need of a service or “topping-up” of the refrigerant. Nowadays, systems may be designed with numerous features, such as avoidance of liquid receivers (or if necessary to handle alternate operating modes, reduced to a necessary volume), smaller heat exchanger tubes (and associated total internal volume), smaller diameter interconnecting tubing and compressors with reduced internal volumes and lubricant charge. It should be noted, though, that within the constraints of current technologies, there is likely a limit of how far charge reduction can go, whilst maintaining a certain efficiency level of a system.

Materials: Generally, charge reduction corresponds to a less system materials, whilst individual component’s function essentially remains unchanged. In some cases, construction materials may change, for example, when switching from a finned (aluminium) tube (copper) to micro-channel heat exchanger (all aluminium).

Costs: Charge minimisation typically has a negative cost impact (cost saving) through reduced refrigerant costs, both at manufacture and in-use, but also reduced construction material costs by means of smaller or lighter components. Additional costs are largely associated with R&D resources involved with sourcing and trialling alternate components.

9.5.3 Limited releasable charge

Ordinarily, it is assumed that the leaked amount of refrigerant equals the charged amount. However, there are passive and active ways to reduce the released refrigerant, whereby overall risk can be reduced and refrigerant quantity limits can be more easily satisfied, as described in section 2.

Materials: For PLRC, no additional hardware is required, only to quantify the releasable charge. This may be done by calculation or test or a combination of both. For ALRC, additional components may be needed. For a system with a reciprocating compressor and liquid line solenoid valve, or a rotary compressor with a reversing valve and electronic expansion valve, or similar arrangements, no additional hardware is necessary. Otherwise, additional solenoid valve(s) may have to be fitted.

Costs: For PLRC, the costs are solely associated with the resources necessary to quantify the releasable charge. The same applies to ALRC and possibly an additional solenoid valve(s), the cost of which is dictated by the tube diameter and may range from a few Euros upwards.

9.5.4 Ventilation

For certain types of DHPs, such as exhaust air DHPs, the inlet and outlet ventilation is used, which disperses refrigerant in the event of a leak.

Materials: Since this ventilation is employed anyway as part of the functioning of the DHP, no additional equipment is necessary.

Costs: Negligible.

9.5.5 Ventilated enclosure

A naturally ventilated enclosure is considered, as described in section 2.1.

Materials: Ventilated enclosures involve considerable additions for compliance with the safety standard. These include:

- Flow and return ducting
- Fan
- Airflow confirmation (e.g., flow switch)
- Leak detection (in absence of continuous airflow)
- Controller/integration with existing controller

Costs: According to Figure 3, the minimum airflow required would be about 50 m^3/h for normal tightness and 10 m^3/h for ETRS. This corresponds to duct diameters around 60 mm and 25 mm, respectively. Typical prices for flexible conduit are used. Axial fans providing these airflow rates are relatively low cost, although an additional allowance is included for Ex-certification. Usually for such fans (e.g., EC types) no special features are required for compliance with the applicable Ex-type standards (EN 60079-series), only the approvals process. Electronic mass airflow sensors are widely produced at fairly low cost. For leak detection, gas sensors are considered. Alternatively continuous fan operation can be used which could be more cost-effective, especially for ETRS (approximately €2 per year, as opposed to about €10 per year for normal tightness systems). Costs associated with integration of electronics and ensuring tightness of the enclosure are negligible. The only additional costs that may be deemed relevant are for the additional installation time, involving drilling holes in the wall and laying the ducting, likely less than one hour. Example costs are summarised in Table 22.

Table 22: Costs for ventilated enclosure assuming 2 kg of R290 and assumed 2 × 3 m duct lengths

	Normal tightness	ETRS
Fan	€ 15	€ 5
Ducting/conduit	€ 30	€ 20
Airflow sensor	€ 10	€ 10
Gas sensor	€ 5	€ 5
Electronics	€ 0	€ 0
Enclosure tightness	€ 0	€ 0
Total	€ 60	€ 40

9.5.6 Naturally ventilated enclosure

A naturally ventilated enclosure is considered, as described in section 2.1.

Materials: A naturally-ventilated enclosure relies on one or two lengths of conduit and ensuring the enclosure if sufficiently tight. It may also be appropriate to employ a magnetic reed switch to ensure that the enclosure access panel has been correctly closed to maintain sufficient tightness.

Costs: The ducting/conduit could be as indicated in in Table 22, although for DHPs installed at ground level only a single length would be necessary and the use of smaller diameter plastic conduit would be viable. Improved tightness of the enclosure would have negligible cost, although a magnetic reed switch is in the order of €5. It is estimated the that the total incremental cost would be about €15.

9.5.7 Integral airflow

Typically for flammable refrigerants, IAF can be used to mix a leak with the surrounding air to ensure that the concentration remains below the lower flammability limit (LFL). It is not intended to exhaust the mixture from the space, only dilute it, as described in section 2.

Materials: Airflow could usually be achieved with a fan that is part of the system (such as with an indoor ATW model), provided the airflow rate and discharge velocity into the space were high enough to disperse a leak. Otherwise, for other indoor ATW or LTW models, an additional dedicated fan would be necessary, the size of which would depend upon whether the system was ETRS or not.

Costs: Where an additional auxiliary fan is required, the cost is a function of the airflow rate (see Figure 2), which is dependent upon the assumed leak rate, according to the level of tightness of the system. Estimation of cost of fan/motor assembly in Euros is about $2.2 \times \dot{V}^{0.45}$, where $\dot{V}$ is the volume airflow rate in m^3/h. For example, 10 m^3/h about €5, 100 m^3/h about €15 and 1000 m^3/h about €50.

9.5.8 Warning alarms

In the event that an RMC is activated, operation of the DHP will usually be terminated and a signal will be indicated.

Materials: No additional materials are involved since the electronic and indicator will already be part of the DHP.

Costs: Negligible incremental costs.

9.5.9 Leak detection

A leak detection system is often required as part of the arrangement involving VE, ALRC and IAF, as detailed in section 2. The type of leak detection may employ gas detection, ultrasonic sensors, system operating parameters (pressure, temperature, etc.) or other means such as liquid level sensing, liquid flow fractions, etc. The choice of technology depends upon the construction and function of the system, the type of RMC employed and the desired effectiveness.

Materials: A leak detection system involves one or more sensors and a signal processor and controller. There are a wide variety of different sensors and often more than one may be needed.

Costs: Costs can vary extensively. Gas sensors can be from catalytic or metal oxide type, ranging from a few Euros upwards, to infra-red or laser gas detection, which can cost in excess of €1000; generally, reliability and precision is reflected in the cost. However, if the use of gas detection proliferates throughout the RACHP sector, there is no reason why more reliable and precise technologies cannot drastically reduce in cost. Ultrasonic sensors are significantly more reliable and are extremely low cost (<€1), although several sensors may be needed, depending upon the equipment they are applied to. Specifically for DHPs, this technology may be less suitable due to the high portion of the system being insulated, which inhibits transmission of ultrasound. Sensor for measuring system parameters include pressure transducers and thermocouples. Whilst the latter have a negligible cost (and may be used anyway) pressure transducer can be in the order of tens of Euros upwards, again depending upon the refrigerant and component size. Signal processing and control units usually have negligible cost and can nevertheless be integrated into existing electronics of the DHP.

9.5.10 No ignition sources

A detailed in section 2.3, ignition sources must be eliminated and the implications of this are largely dependent upon the type and construction of the DHP. However, provided that the main electrical components are housed in a compartment sectioned off from the refrigerating circuit, there is usually minimal effort required.

Materials: Given the wide variety of options for avoiding ignition sources there are a similarly extensive number of material implications. Usually, there will be additional plastic enclosures or dividing plates, depending upon the type and size of equipment and refrigerants involved. Most component manufacturers now offer products (compressors, valves, fans, etc.) that are approved for use with flammable refrigerants, against the applicable safety standards.

Costs: Similarly, additional costs can range from negligible to thousands of Euros (e.g., for large industrial installations), although in most cases, economically effective approaches have been found. With large-scale introduction, there should be cost parity with equipment not intended for use with flammables.

9.5.11 Increased system tightness

By introducing appropriate design and construction measures to the refrigerating system, the likelihood of leakage and the possibility of larger leak holes can be reduced. Various safety standards and guidelines include such measures and are referred to as “enhanced tightness refrigeration systems” (ETRS); see section 2. Requirements may include better system components and fittings, avoidance of constructional circumstances likely to lead to leakage, more rigorous testing and leak checking and quality control programmes.

Materials: System components and fittings that have been tested and approved to the applicable standards.

Costs: Whilst some tested and approved components and fitting will be higher cost than basic ones, as more and more manufacturers adopt the regime, additional costs should become negligible. More rigorous leak testing requires additional equipment and procedures for production lines and quality control programmes demand additional resources of workers. However, provided the output of a production facility is large enough, additional costs become negligible (per unit produced). Furthermore, regardless of whether the refrigerant is conventional or a more hazardous alternative refrigerant, reduced leakage has significant advantages in terms of better system reliability, lower lifetime service costs and consequent reputational benefits.

Based on the requirements of IEC 60335-2-40 clause 22.125, Table 23 summarises the cost implications for DHPs. Assuming a reasonable annual production of the facility, say 10,000 units per year, the incremental cost for complying with ETRS would be no more than €5 per unit.

Table 23: Cost implication associated with elements of ETRS

Requirement	Cost implication
a) compressors, pressure relief devices and pressure vessels of the refrigerating system shall be located in locations other than the occupied space,	None (condensing unit must be outside)
b) refrigerant distribution assemblies shall meet all applicable requirements of this standard,	None (not applicable to single splits)
c) refrigerating systems shall use only permanent joints indoors except for site-made joints directly connecting the indoor unit to the refrigerant piping, or factory-made mechanical joints in compliance with ISO 14903,	None (all indoor joints must be brazed)
d) refrigerant containing parts in indoor units shall be protected from damage in the event of catastrophic failure of moving parts, e.g., fans, belts,	None (demonstrated that IDU fan cannot damage HX or piping)
e) refrigerant containing pipes in the occupied space in question are installed in such a way that they are protected against accidental damage	Negligible (low level piping must be sheathed)
f) the refrigerating system of each indoor unit shall be tightness tested at the factory with detection equipment with a capability of 3 grams per year of refrigerant or less under a pressure of at least 0,25 times the maximum allowable pressure, no leak shall be detected.	Negligible (leak test production equipment – usual cost approx. €250,000, but assuming production of 100,000 units over 5 years, cost would be about €0.5 per unit)
g) vibrations exceeding 0,30 G RMS, when measured with a low pass filter at 200 Hz, are not allowed in the refrigerant containing parts in the occupied space under normal operation.	Negligible (type test for each model)
h) indoor heat exchangers shall be protected from damage in the event of freezing, as demonstrated by a test.	Negligible (type test for each model)
i) the maximum speed of the indoor fan, in normal operation, shall be less than 90 % of the maximum allowable fan speed as specified by the manufacturer of the fan wheel. If the manufacturer does not specify a maximum allowable fan speed, then the fan wheel shall be tested.	Negligible (type test for each model)

9.5.12 Instructions/marking

Information about safe practices needs to be relayed to workers (such as technicians) and to users. Additional information may relate to the hazardous characteristics of the refrigerant. This additional information usually involves instructions in manuals and marking or signage on the equipment. Examples include details about safe handling of the refrigerant during servicing of the equipment and “flammable refrigerant” warning triangles.

Materials: Additional pages in a manuals and adhesive stickers.

Costs: Material costs are negligible, the main cost implication being that associated with sourcing and drafting the relevant information.

9.5.13 Energy use of RMCs

For several RMCs there is an energy penalty associated with their operation. These must be considered in terms of cost to the user and emissions associated with electricity production.

Amongst the RMCs, the elements that draw electrical energy are fan motors and normally closed solenoid valves that are being maintained in an open state whilst the refrigerating system is operating. Ordinarily, the function of shut-off valves would be performed by a reversing valve and/or EEV, but assuming that an elaborate ALRC is employed (section 2), it is assumed that one (for R152a) or two (for R1270 and R290) additional solenoid valves are employed. Solenoids are assumed to be energised during system operation and a typical annual operating time fraction of 0.35 is used. Depending upon the function of the RMC, fan motors may operate upon demand of the leak detection (taken as one hour per year) or continuously, being a time fraction of 1. Class A1 refrigerants do not require these RMCs (i.e., all time fractions are 0) and according to the estimated charge amounts, A2L would only require RMCs for the LTW DHP, but as a pessimistic illustration, they are assumed not to be required for the present exercise. Table 24 lists the corresponding time fractions according to the RMC and its function.

Table 24: Active operation time fraction for fan motors and solenoid valves for different RMCs

RMC	function	Item	R410A	R32	R1270	R290	R1234yf	R152a
Outdoor	n/a	Fan	0	0	0	0	0	0
		Solenoid	0	0	0	0	0	0
ALRC and IAF	upon sensor	Fan	0	0	0.0001	0.0001	0	0.0001
		Solenoid	0	0	0.35	0.35	0	0.35
	continuous	Fan	0	0	1	1	0	1
		Solenoid	0	0	0.35	0.35	0	0.35
VE	upon sensor	Fan	0	0	0.0001	0.0001	0	0.0001
		Solenoid	0	0	0	0	0	0
	continuous	Fan	0	0	1	1	0	1
		Solenoid	0	0	0	0	0	0
Naturally VE	n/a	Fan	0	0	0	0	0	0
		Solenoid	0	0	0	0	0	0

Considering the airflow rates and associated fan power calculated in section 2.4, as well as the active power of the normally closed solenoid, taken to be 8 W, the lifetime energy consumption is calculated; Table 25. In all cases, it is assumed that the system is ETRS, since this always leads to substantial savings. Electronic controller energy is assumed not to be affected since it is providing functionality of other features anyway.

Table 25: Lifetime energy consumption for RMCs

RMC	function	R410A	R32	R1270	R290	R1234yf	R152a
Outdoor [kWh]	n/a	0	0	0	0	0	0
ALRC and IAF [kWh]	upon sensor	0	0	368	368	0	368
	continuous	0	0	3888	3816	0	1747
VE [kWh]	upon sensor	0	0	0.01	0.01	0	0.01
	continuous	0	0	125	104	0	85
Naturally VE [kWh]	n/a	0	0	0	0	0	0

Since in Table 25 it is evident that continuous fan operation consumes not insignificant energy, application of the RMC upon demand of a sensor (leak detection) is considered subsequently. It is expected that implementation of sensors for the purpose of leak detection to be more economical than having continuous fan operation (i.e., costing several hundred Euros over the lifetime).

Emissions associated with the production of the RMC equipment should also be included. For fan/motor assemblies, Medas et al. (2016) offer values in the range of 2 – 10 $kgCO_2/kg$, so a value of 6 $kgCO_2/kg$ will be used. The mass of the assembly [kg] can then be approximated from $\cong 0.001 \times \dot{V} + 0.05$, where $\dot{V}$ is the volume flow rate [m^3/h]. Ducting and conduit are based on the material mass for the required length and additional solenoid valves are treated as with standard system valves. Small sensors are neglected.

9.6 Materials and refrigerant

Material costs are based on standard values per kg, as listed in metal exchange internet sites (August 2022); Table 26. Product material costs are these multiplied by the corresponding material masses. Note that material comprising equipment for safety measures are not included, since they are accounted for separately.

Table 26: Price of materials

Material	Price [€/kg]
Steel	1.5
Stainless steel	3.3
Aluminium sheet	2.5
Copper tube	8.0

Refrigerant prices vary widely, depending upon region, quantity purchased, supplier and other factors. Prices listed in Table 27 are considered to be representative for Europe in May 2022. The majority of DHPs for the European market are manufactured within Europe and any products containing F-gases are subject to quotas so associated costs are considered to be lumped in with the product cost. Refrigerant cost is separated into two categories: “bulk” according to manufacturer purchases (i.e., assigned to the first fill) and “service” as for technicians providing top-up or repair (i.e., applied to the refrigerant used for servicing during the remainder of the DHPS lifetime). Service refrigerant cost also applies for filling additional interconnecting piping. Recovered refrigerant at end of life is not considered to have a value. Pricing data is taken from Öko-Recherche (2022) and commercial sources where data was absent.

Table 27: Price of refrigerant

	R410A	R32	R1270	R290	R1234yf	R152a
Bulk [€/kg]	14	10	2	2	45	4
Service [€/kg]	22	18	8	8	66	10

Figure 63 and Figure 64 show the summed system material mass and component volume from the various components and refrigerant (first fill). Overall, there is no significant difference amongst the alternatives with wither ATW or LTW DHPs, except with R1234yf, which involve both greater mass and larger volume, being about 10% and 35%, respectively.

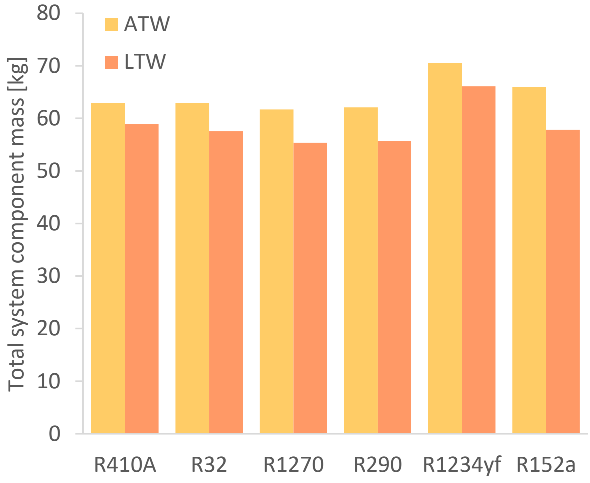


Figure 63: System material mass

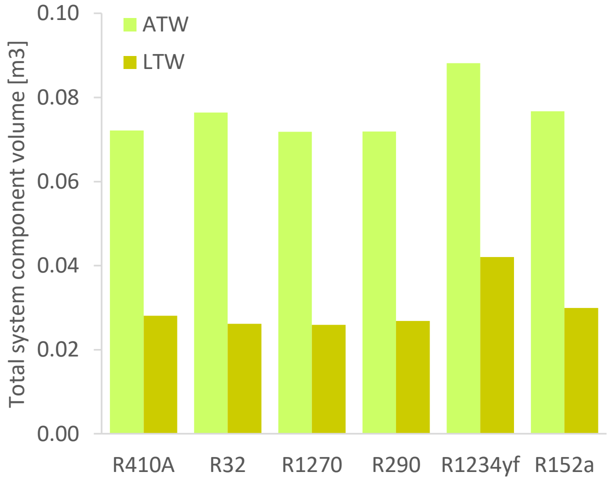


Figure 64: System component volume

For reference, the average volume of an LTW DHP is $\cong 0.2 \times Q_e^{0.4}$ and $\cong 0.55 \times Q_e^{0.3}$ for ATW. Average mass of an LTW is $\cong 40 \times Q_e^{0.6}$ (approximately 160 kg for 10 kW) and $\cong 100 \times Q_e^{0.4}$ for ATW (approximately 250 kW for 10 kW). Typically, refrigeration system components account for about a quarter to a third of the DHP mass.

9.7 Transportation and storage

Additional transportation costs are based on any greater mass of system components arising from use of an alternative refrigerant. For example, where a larger evaporator is required, the additional haulage cost (assuming an average distance by road of 500 km across Europe is around €0.04 per kg) is applied to the additional evaporator mass. Similarly, for storage, a cost corresponding to €18 per m^3 per year has been applied (assuming the DHPs are in storage for one year). The storage cost is highly variable depending upon location and size of storage facility, but since it is a relatively minor contribution variations can be deemed negligible.

9.8 Overall costs

Mass of materials from sections 9.1 to 9.7 have been summed and included in Table 28. Material mass includes first charge of refrigerant for each system and the primary metals associated with piping, compressor, HXs, valves, etc. Materials associated with plastics, insulation, electronics and so on have been neglected since there is unlikely to be any significant variation arising from refrigerant choice.

For comparison, the baseline cost is according to $\cong 3100 \times Q_e^{0.5}$ for LTW and $\cong 2600 \times Q_e^{0.5}$ for ATW models. As seen in section 4.4, the retail price covers a huge range and is largely unrelated to efficiency, so the baseline cost is simply based on the median.

A summary of the material masses and costs for the ATW are listed in Table 28, including for the highest cost RMCs for the A2 and A3 refrigerants.

Table 28: Summary of material mass and associated costs for ATW DHP

Item		R410A	R32	R1270	R290	R1234yf	R152a
Material mass	Refrigerant – first fill [kg]	1.4	1.2	0.6	0.6	1.3	1.1
	Mass of steel [kg]	37.0	37.0	37.0	37.0	37.0	37.0
	Stainless steel [kg]	4.6	4.5	3.9	3.9	4.8	3.5

Item		R410A	R32	R1270	R290	R1234yf	R152a
	Mass of aluminium [kg]	8.0	8.7	8.0	8.0	10.0	8.7
	Mass of copper [kg]	11.8	11.6	12.2	12.6	17.5	15.7
	Total (all) [kg]	62.9	62.9	61.7	62.1	70.5	66.0
Material costs	Refrigerant – first fill [€]	19	12	1	1	61	4
	Steel [€]	56	56	56	56	56	56
	Stainless steel [€]	23	22	20	20	24	18
	Aluminium sheet [€]	20	22	20	20	25	22
	Copper [€]	95	93	97	100	140	126
	Safety features* [€]	5	5	55	55	5	45
RMC energy use* [€]		0	0	0.002	0.002	0	0.001
Additional shipping cost [€]		0.00	0.00	-0.05	-0.03	0.31	0.13
Additional storage cost [€/y]		0.00	0.08	-0.01	0.00	0.29	0.08
Total cost (excl. refrigerant) [€]		198	197	247	251	250	266
Total cost (incl. refrigerant) [€]		240	224	252	255	377	278
DHP baseline price [€]		8222	8222	8222	8222	8222	8222
Incremental cost (excl. reft) [€]		0	-1	14	17	51	42
Total DHP (excl. reft) [€]		8222	8206	8234	8237	8358	8260
Relative		0.00%	-0.19%	0.14%	0.18%	1.66%	0.46%
Incremental cost (incl. reft) [€]		0	-16	-23	-20	137	13
Total DHP (incl. reft) [€]		8222	8221	8271	8274	8273	8289
Relative		0.00%	-0.01%	0.60%	0.64%	0.62%	0.82%

* Includes ETRS for all and VE for R1270, R290 and R152a only. Results for other RMCs included in subsequent graphs.

A summary of the material masses and costs for the LTW are listed in Table 29, including for the highest cost RMCs for the A2 and A3 refrigerants.

Table 29: Summary of material mass and associated costs for LTW DHP

Item		R410A	R32	R1270	R290	R1234yf	R152a
Material mass	Refrigerant – first fill [kg]	2.1	1.7	0.8	0.8	2.0	1.6
	Mass of steel [kg]	37	37	37	37	37	37
	Stainless steel [kg]	15.7	14.7	13.4	13.8	23.0	15.1
	Mass of aluminium [kg]	0.0	0.0	0.0	0.0	0.0	0.0
	Mass of copper [kg]	4.1	4.1	4.1	4.1	4.1	4.1
	Total (all) [kg]	58.9	57.5	55.4	55.7	66.1	57.8

Item		R410A	R32	R1270	R290	R1234yf	R152a
Material costs	Refrigerant – first fill [k€]	29	17	2	2	91	6
	Steel [€]	55.5	55.5	55.5	55.5	55.5	55.5
	Stainless steel [€]	78.4	73.5	67.0	69.0	114.8	75.7
	Aluminium sheet [€]	0.0	0.0	0.0	0.0	0.0	0.0
	Copper [€]	33.1	32.9	32.8	32.7	32.9	32.7
	RMCs* [€]	5	5	55	55	5	45
RMC energy use* [€]		0	0	0.002	0.002	0	0.001
Additional shipping cost [€]		0.00	-0.05	-0.14	-0.13	0.29	-0.04
Additional storage cost [€/y]		0.00	-0.03	-0.04	-0.02	0.25	0.03
Total cost (excl. refrigerant) [€]		172	167	210	212	209	209
Total cost (incl. refrigerant) [€]		235	207	217	219	399	227
DHP baseline price [€]		9803	9803	9803	9803	9803	9803
Incremental cost (excl. reft) [€]		0	-5	3	5	37	12
Total DHP (excl. reft) [€]		9803	9798	9806	9808	9840	9815
Relative		0.00%	-0.05%	0.03%	0.05%	0.37%	0.12%
Incremental cost (incl. reft) [€]		0	-27	-53	-51	165	-32
Total DHP (incl. reft) [€]		9803	9776	9750	9752	9968	9771
Relative		0.00%	-0.28%	-0.54%	-0.52%	1.68%	-0.33%

* Includes ETRS for all and VE for R1270, R290 and R152a only. Results for other RMCs included in subsequent graphs.

Overall, it may also be observed that the cost of materials corresponds to less than 10% of the DHP retail price. The remaining 90 – 95% may be accounted for by research and development, manufacturing whole unit and components, storage and distribution, business costs such as sales and marketing, warrantees and of course company profit. As such, a change of, say, 10% in cost of component material mass corresponds to less than 1% difference to the overall price.

Incremental cost associated with the alternatives is shown relative to the cost for R410A. From Figure 65, it is seen that only R32 has a negative incremental cost when excluding cost of refrigerant, but when accounting for the refrigerant, R1270, R290 and R152a all have a negative incremental cost, too. Conversely, R1234yf shows a significant increase, both without and particularly when including its high refrigerant cost. Whilst the material masses for R1270 and R290 are similar to that of R32 and R410A, the assumed high costs for implementing RMCs is detrimental, although evidently offset by the relatively low cost of refrigerant. Lifetime “top-up” costs associated with refrigerant leakage are about equal to the first fill cost, so would provide further reduction of incremental cost. Considering the anticipated rise in price of F-gases over the next decade, it is likely that these negative incremental costs for some alternatives would be enhanced over time.

Figure 66 shows the results for the LTW DHP. In this case, the incremental costs of R32, R1270 and R290 are similar, showing cost parity with R410A when excluding refrigerant and about the same level of negative cost increment when including the cost of refrigerant. Cost increments associated with R152a and R1234yf are similar to the ATW case.

In Figure 65 and Figure 66, the cost increments as a percentage of the DHP retail price are also included, showing that except for R1234yf, they are always within 0.5% of the product price.

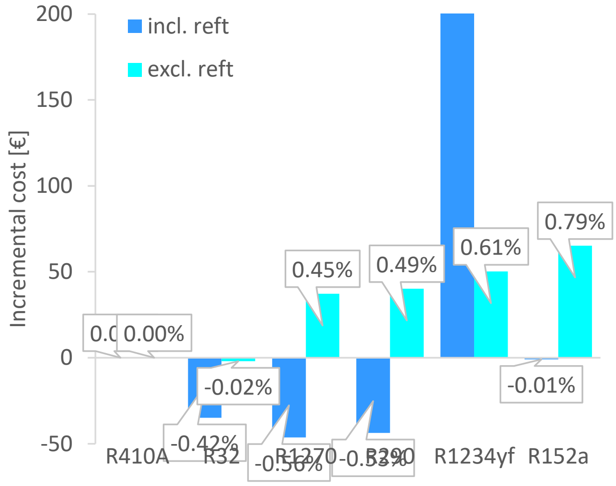


Figure 65: Incremental material costs relative to R410A for ATW DHP; indoor with VE

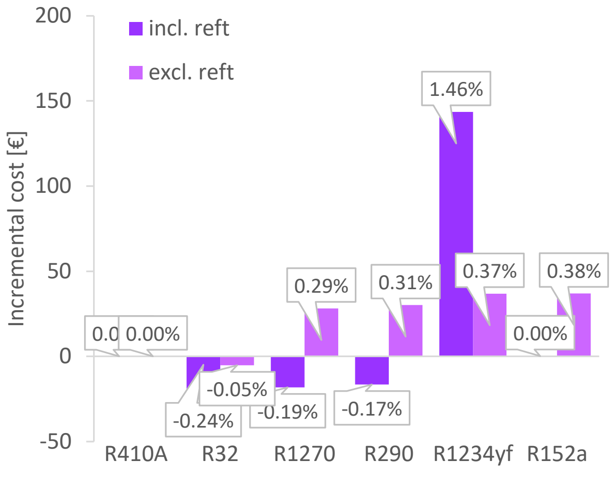


Figure 66: Incremental material costs relative to R410A for LTW DHP; indoor with VE

Figure 67 and Figure 68 present incremental costs for the same cases, but assuming a naturally ventilated VE. Due to the reduced costs for the RMC, the impact is lower and in the case of R1270 and R290 in ATW DHP, they start to show cost parity with R410A and R32 even before the contribution of refrigerant is included. The effect is less beneficial for the LTW DHP.

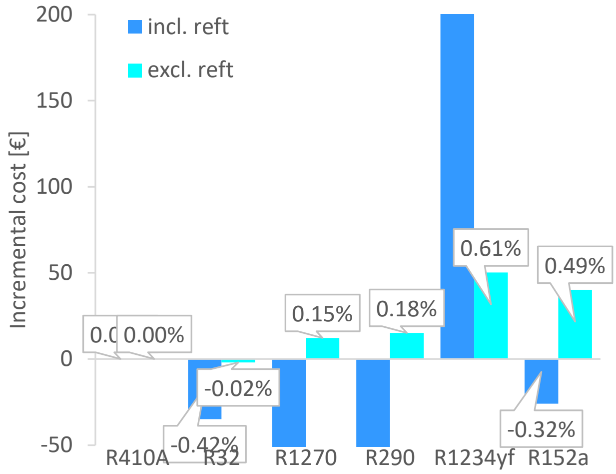


Figure 67: Incremental material costs relative to R410A for ATW DHP; indoor with naturally VE

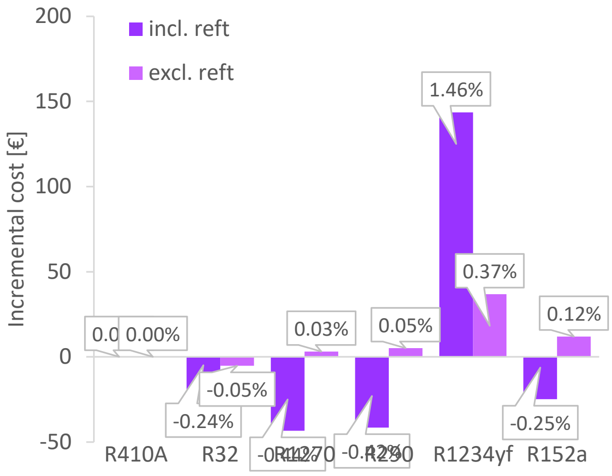
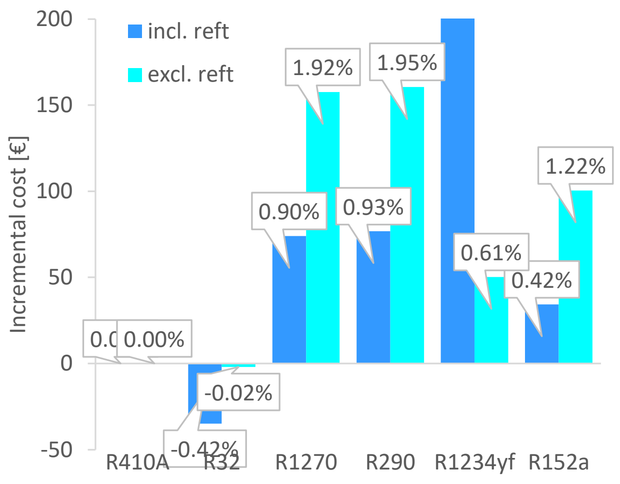


Figure 68: Incremental material costs relative to R410A for LTW DHP; indoor with naturally VE

Instead of a VE, Figure 69 and Figure 70 include costs for the ALRC and IAF type RMCs. Due to the use of additional valves and sensors, the incremental costs are similar to that of a conventional VE for both ATW and LTW.



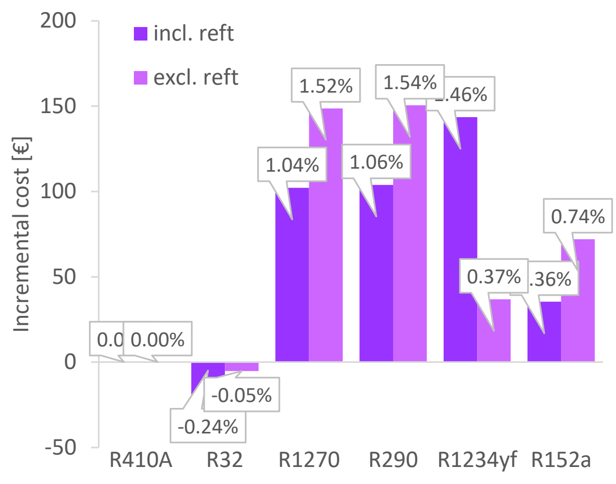
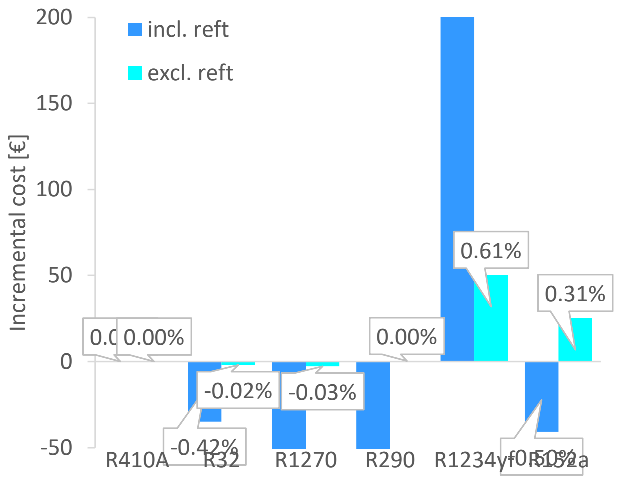


Figure 69: Incremental material costs relative to R410A for ATW DHP; indoor with ALRC and IAF

Figure 70: Incremental material costs relative to R410A for LTW DHP; indoor with ALRC and IAF

Lastly, results are shown for the ATW and LTW, both being assumed to be located outdoors (although it is recognised that LTW DHPs are seldom installed outdoors, if at all). Since the only additional cost is for implementing ETRS, Figure 71 and Figure 72 have notably lower incremental costs for R1270, R290 and R152a, in most cases being negative or in parity with R410A. Only R1234yf remains with a significantly higher cost.



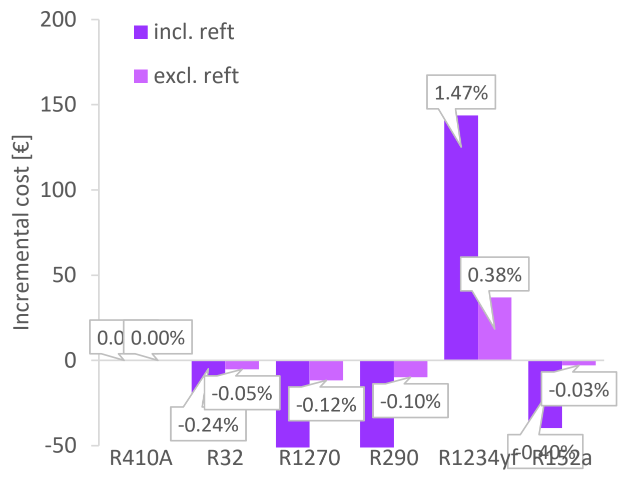


Figure 71: Incremental material costs relative to R410A for ATW DHP; outdoor

Figure 72: Incremental material costs relative to R410A for LTW DHP; outdoor

These incremental costs may also be considered relative to manufacturing net profit margin for DHPs. Whilst such data is not freely available, general information reports that for large domestic appliances, net profit margins are generally around the range of 5 – 10% of the retail price. Thus, for the example case under consideration, this could be in the order of €400 – 800 and thus the adoption of R1234yf would be financially questionable.

10 Greenhouse gas emissions

Greenhouse gas emissions are quantified for the DHPs using the alternative refrigerants. This involves estimating CO_2-equivalent (CO_2e) emissions for the production of used materials (metals and refrigerants; emission factors in Table 30) and the direct contribution of leaked refrigerant. Emissions associated with the production of other materials (plastics, electronics, etc.) are not included as it is assumed they remain identical across all the refrigerants. Similarly, emissions associated with the generation of electricity used for operating the DHPs is also negated since the designs are intended to achieve the same seasonal efficiency (i.e., SCOP = 6.5); as discussed in section 6 and section 9. Additional emissions associated with the RMCs are also included as appropriate. Calculations are carried out for GWPs (AR6) using both 20-year and 100-year time horizon based on IPCC AR6 values, in order to reflect current views on the matter.

Emission factors for production of materials are as reported in common literature sources. The literature reports a wide range of values for production of refrigerants, influenced by factors such as production route, age of plant and of course, source of funding for the study. Representative values are taken from Johnson (2011). Material production emissions are again based on the product of emission factors and mass of materials. Lifetime emissions consider only annual leakage and end-of-life releases of refrigerant, according to IPCC guidelines and application reports (i.e., SKM Enviros, 2012).

Table 30: Production emission factors for materials and refrigerants

	R410A	R32	R1270	R290	R1234yf	R152a
Refrigerant [$kgCO_2/kg$]	15	10	1.5	1.5	40	5
Steel [$kgCO_2/kg$]	2.2					
Stainless steel [$kgCO_2/kg$]	3.3					
Aluminium sheet [$kgCO_2/kg$]	7.1					
Copper tube [$kgCO_2/kg$]	3.4					

Average emissions factor for the EU, projected over the lifetime of the DHP is estimated to be 0.113 $kgCO_2/kWh$[20].

Table 31: Summary of emissions for ATW DHP (neglecting refrigeration system energy)

	Item	R410A	R32	R1270	R290	R1234yf	R152a
Material production emissions	Refrigerant production [$kgCO_2$]	46.1	25.9	1.9	1.9	121.1	12.1
	Steel [$kgCO_2$]	81.4	81.4	81.4	81.4	81.4	81.4
	Stainless steel [$kgCO_2$]	15.3	14.8	12.9	13.0	15.7	11.7
	Aluminium [$kgCO_2$]	57.1	61.6	57.1	57.1	70.8	61.6
	Copper [$kgCO_2$]	42.6	41.4	43.0	44.1	61.2	54.7
	RMC equipment* [$kgCO_2$]	0.0	0.4	3.9	3.9	0.4	3.9
	Total production emissions [$kgCO_2$]	242.5	225.5	200.2	201.4	350.5	225.4
	Incremental emissions [$kgCO_2$]	0.0	-17.0	-42.3	-41.1	108.1	-17.1
Lifetime emissions	RMC energy use* [$kgCO_2$]	0	0	0.001	0.001	0	0.001
	Annual leakage [%/y]	5%	5%	5%	5%	5%	5%
	EOL emissions [%]	80%	80%	80%	80%	80%	80%
	Lifetime [y]	15	15	15	15	15	15
	Lifetime emissions (100 y) [kgCO2]	10755	3099	0.2	0.2	0.5	613

[20] https://www.eea.europa.eu/data-and-maps/daviz/co2-emission-intensity-10/download.csv

	Item	R410A	R32	R1270	R290	R1234yf	R152a
	Total emissions (100 y) [kgCO2]	10997	3324	200	202	351	839
	Lifetime emissions (20 y) [kgCO2]	22477	10811	0.2	0.2	2	2209
	Total emissions (20 y) [kgCO2]	22720	11036	200	202	353	2435

* Includes for VE for R1270, R290 and R152a only

Table 32: Summary of emissions for LTW DHP (neglecting refrigeration system energy)

	Item	R410A	R32	R1270	R290	R1234yf	R152a
Material production emissions	Refrigerant production [kgCO_2]	25.6	14.4	1.1	1.0	67.3	6.7
	Steel [kgCO_2]	81.4	81.4	81.4	81.4	81.4	81.4
	Stainless steel [kgCO_2]	51.8	48.5	44.3	45.5	75.8	50.0
	Aluminium [kgCO_2]	0.0	0.0	0.0	0.0	0.0	0.0
	Copper [kgCO_2]	13.9	13.9	13.8	13.8	13.9	13.8
	RMC equipment [kgCO_2]	0.0	0.0	3.6	3.6	0.0	3.6
	Total production emissions [kgCO_2]	172.7	158.2	144.1	145.3	238.3	155.4
	Incremental emissions [kgCO_2]	0.0	-14.6	-28.6	-27.4	65.6	-17.3
Lifetime emissions	RMC energy use* [kgCO_2]	0	0	0.001	0.001	0	0.001
	Annual leakage [%/y]	5%	5%	5%	5%	5%	5%
	EOL emissions [%]	80%	80%	80%	80%	80%	80%
	Lifetime [y]	15	15	15	15	15	15
	Lifetime emissions (100 y) [kgCO2]	5975	1721	0	0	0	341
	Total emissions (100 y) [kgCO2]	6148	1880	144	145	239	496
	Lifetime emissions (20 y) [kgCO2]	12487	6006	0.1	0.1	1	1227
	Total emissions (20 y) [kgCO2]	12660	6164	144	145	240	1383

* Includes for VE for R1270, R290 and R152a only

A breakdown of the emissions associated with the production of the system components is given in Figure 73 and Figure 74 for the ATW and LTW, respectively. It can be seen that the emissions are dominated by the main construction materials and that those for the RMC equipment are negligible.

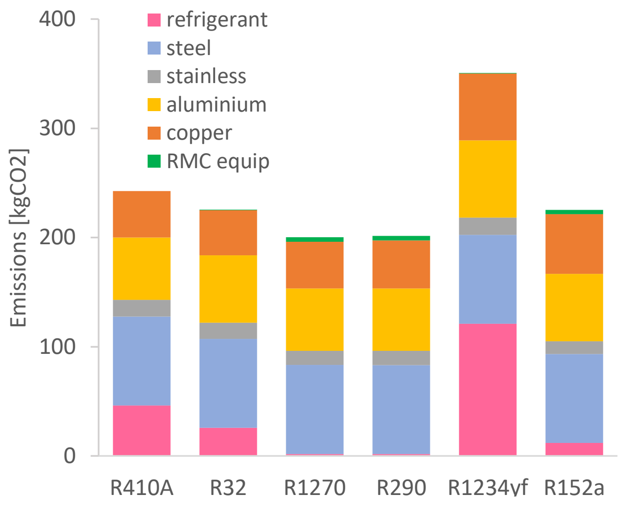


Figure 73: Breakdown of production emissions for ATW

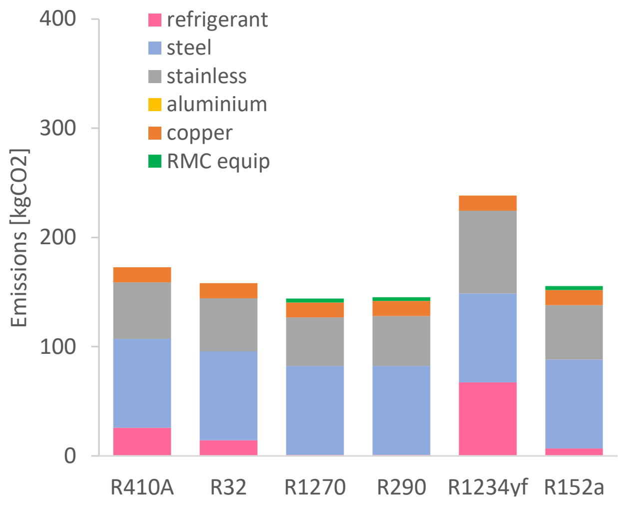


Figure 74: Breakdown of production emissions for LTW

A summary of the overall results for the ATW DHP emissions are shown in Figure 75 and Figure 76 using 100-year and 20-year ITH. Similarly, Figure 77 and Figure 78 present emissions for the LTW DHP.

For R410A and R32 the refrigerant-related emissions dominate and even for R152a, they are more than double the production-related emissions. When using the 100 y ITH for GWP, emissions for R32 are about one-third of R410A. However, on a 20 y basis, the emissions associated with R32 represent half of those of R410A. The same amplification in effects is seen with R152a. For all cases, R1270 and R290 show the lowest emissions and in-use emissions (including those for operating the RMC) is about equal to the production emissions. Whilst Figure 75 to Figure 78 are based on the VE RMC, results for emissions when selecting other RMCs – indoor with naturally VE, indoor with ALRC and IAF and outdoors – cannot be differentiated from these.

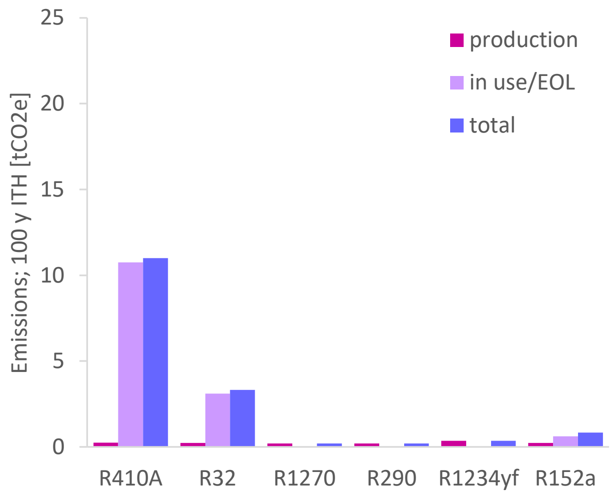


Figure 75: Production, lifetime and total emissions for ATW, using 100-year ITH GWPs; indoor with VE

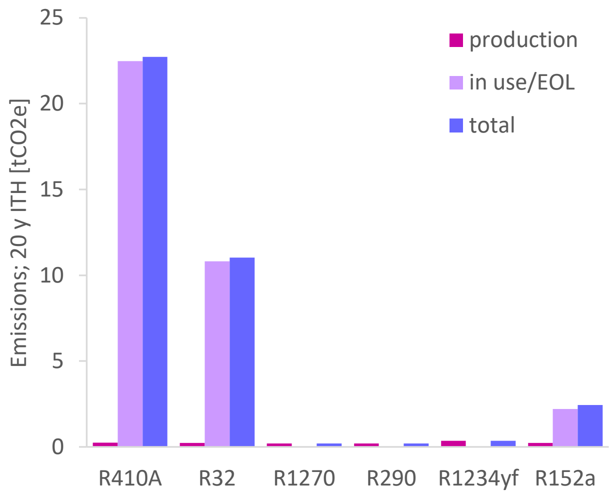


Figure 76: Production, lifetime and total emissions for ATW, using 20-year ITH GWPs; indoor with VE

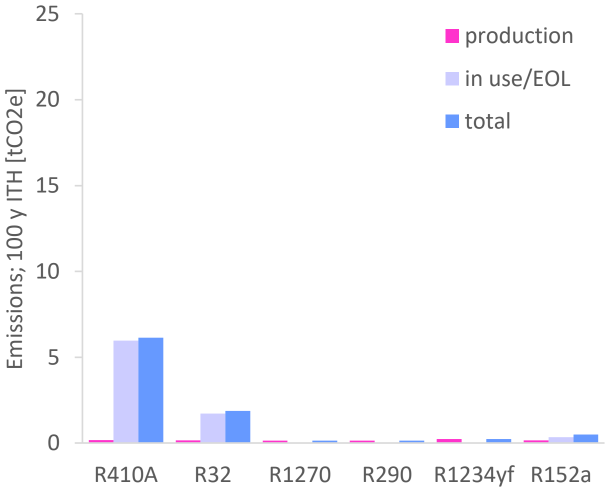


Figure 77: Production, lifetime and total emissions for LTW, using 100-year ITH GWPs; indoor with VE

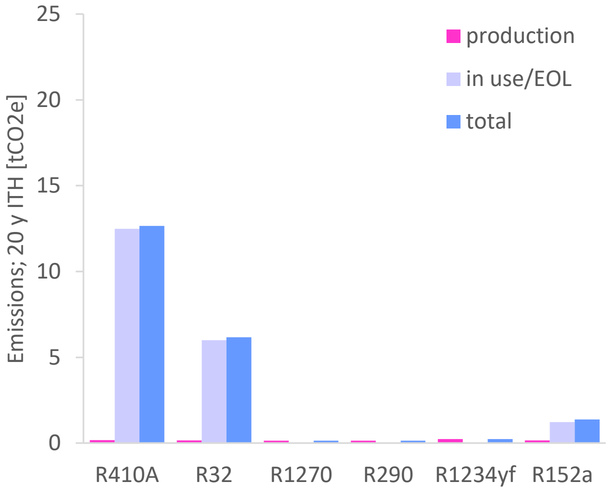


Figure 78: Production, lifetime and total emissions for LTW, using 20-year ITH GWPs; indoor with VE

Combining with the results for the costs from section 9.8, cost-effectiveness values are given in Table 33 for ATW and LTW in Table 34, relative to the R410A baseline. Results are presented for four different RMCs (as above). Where a particular option leads to a negative incremental cost (i.e., cost saving) along with an emissions reduction, the cost-effectiveness is not applicable (i.e., and infinite benefit, "inf")[21]. This is always the case for R32, R1270 and R290; the exception is for the RMC "indoor with ALRC and IAF", due to the cost associated with two ancillary solenoid valves and the energy consumption involved with keeping them open.

Table 33: Cost-effectiveness for ATW, based on 100 y ITH GWPs

RMC	Parameter	R410A	R32	R1270	R290	R1234yf	R152a
Indoor with naturally VE	Emissions reduction [tCO_2]	0.00	7.67	10.80	10.80	10.65	10.16
	Incremental costs [€]	0.0	-34.9	-86.4	-83.6	242.6	-40.7
	Cost-effectiveness [€/$kgCO_2$]		inf	inf	inf	22.8	inf
Indoor with VE	Emissions reduction [tCO_2]	0.00	7.67	10.71	10.71	10.65	10.12
	Incremental costs [€]	0.0	-34.9	74.0	76.8	242.5	34.3
	Cost-effectiveness [€/$kgCO_2$]		inf	6.9	7.2	22.8	3.4
Indoor with ALRC and IAF	Emissions reduction [tCO_2]	0.00	7.67	10.80	10.80	10.65	10.16
	Incremental costs [€]	0.0	-34.9	-46.4	-43.7	242.5	-0.9
	Cost-effectiveness [€/$kgCO_2$]		inf	inf	inf	22.8	inf
Outdoor	Emissions reduction [tCO_2]	0.00	7.67	10.80	10.80	10.65	10.16
	Incremental costs [€]	0.0	-34.9	-71.4	-68.7	242.5	-25.9
	Cost-effectiveness [€/$kgCO_2$]		inf	inf	inf	22.8	inf

Table 34: Cost-effectiveness for LTW, based on 100 y ITH GWPs

RMC	Parameter	R410A	R32	R1270	R290	R1234yf	R152a
Indoor with naturally VE	Emissions reduction [tCO_2]	0.00	4.27	6.01	6.01	5.91	5.66
	Incremental costs [€]	0.0	-23.4	-58.2	-56.4	143.7	-39.6
	Cost-effectiveness [€/$kgCO_2$]		inf	inf	inf	24.3	inf
	Emissions reduction [tCO_2]	0.00	4.27	6.01	6.01	5.91	5.66

[21] When the cost is negative, the resulting cost-effectiveness is meaningless; both a larger or a smaller negative quotient could be favourable, whereas with a positive cost, a smaller quotient is always more favourable.

Indoor with VE	Incremental costs [€]	0.0	-23.4	-58.2	-56.4	143.7	-39.6
	Cost-effectiveness [€/kgCO_2]		inf	inf	inf	24.3	inf
Indoor with ALRC and IAF	Emissions reduction [tCO_2]	0.00	4.27	6.01	6.01	5.91	5.66
	Incremental costs [€]	0.0	-23.4	-58.2	-56.4	143.7	-39.6
	Cost-effectiveness [€/kgCO_2]		inf	inf	inf	24.3	inf
Outdoor	Emissions reduction [tCO_2]	0.00	4.27	6.01	6.01	5.91	5.66
	Incremental costs [€]	0.0	-23.4	-58.2	-56.4	143.7	-39.6
	Cost-effectiveness [€/kgCO_2]		inf	inf	inf	24.3	inf

Switching from R410A to R32 always results in emissions reductions and a cost reduction. But switching from R410A to any of the low GWP alternatives (R1270, R290, R1234yf or R152a) always results in further emissions reductions; about 50% more than when using R32 (or double, when using 20 y ITH GWP values). The incremental cost reduction when switching to R1270, R290 or R152a is greater than when choosing R32, except with the RMC option “indoor with ALRC and IAF”; in this case, the additional costs for components and operation of the solenoids offsets any potential cost reduction from materials and refrigerant. However, if refrigerant charge was sufficiently minimised (although still above 4×LFL), the supplementary solenoid valves could be dispensed with and this RMC option could offer a similar benefit to the other RMCs. When using R1234yf, the incremental cost of is always quite high, even though the additional costs for RMCs is negligible. This is caused by the high cost of refrigerant and greater HX material, required to overcome the high pressure drop. Nevertheless, cost-effectiveness remains below the €30 – 90 per tCO_2 value under the European Carbon Trading Scheme, suggesting that it could be acceptable in absence of any other alternatives.

11 Further considerations

Some additional aspects are considered with respect to the application of alternatives, primarily, operating envelopes, refrigeration system advances for DHPs and development schedules.

11.1 Operating envelope

The compressor operating envelope is an important factor for DHP development.

Operating envelope affected by parameters such as discharge temperature, pressure, compression ratio, oil solubility, etc. Across most of the determining factors, HCs have more desirable characteristics. Overall, R1270 and particularly R290 operating envelopes have wider (evaporating temperatures) and taller (condensing temperatures) ranges than R410A, R32 and R152a. R1234yf also possesses some such benefits (Xu et al., 2018).

This means that DHPs using R1270 and R290 can operate across a wider range of temperature conditions, for example, they can provide DHW whilst outdoor temperatures are low. As a consequence, it lessens the need for supplementary electric or gas heating at those conditions. For several types of DHPs, this is regarded as a major advantage.

11.2 Advances in efficiency improvement and charge reduction

A review of the recent literature identifies various studies with consistent objectives, where approaches for reducing charge and increasing thermal capacity and/or efficiency are reported. For example:

- Andersson et al. (2018) – LTW DHP that uses only less than 150 g of R290 to achieve 10 kW NHC.
- Berman et al. (2020) – 15% higher efficiency with SLHX without additional charge.
- Bo and Shen (2020) – 17% higher efficiency with the same charge and nominal capacity through HX and compressor optimisation.
- Chen et al. (2018) – 30% increase in heating capacity and 20% higher efficiency with R290 using vapour injection.
- Fujino et al. (2014) – a novel MCHX evaporator design that resolves the condensate water problem.
- Huang et al. (2018) – computationally and experimentally analysed MCHX evaporator designs for HPs.
- Methler et al. (2022) and Dankwerth et al. (2020) as part of the LC150 project, demonstrated designs with as low as 20 g/kW at SCOP of 5.
- Palm (2014) developed a 30 kW LTW HP with 21 g/kW.
- Panda et al. (2019) – novel MCHX evaporator headers to resolve maldistribution problem for reversible systems.
- Radermacher et al. (2017) – novel HX designs, which showed the leading options could improve performance by more than 15%, lessen material mass by 20% and reduce overall system refrigerant charge by 30 – 40%.
- Ren et al. (2014) – reduce charge by >5% and increase capacity and efficiency by 5% by addition of a special liquid-suction heat exchanger (SLHX).
- Ribeiro and Barbosa (2019) – 10% of R290 charge reduction in ACs by using a novel HX design.
- Sánchez-Moreno-Giner et al. (2022) report just over 20 g/kW for LTW.

- Satoshi et al. (2012) – special microchannel HX design used effectively in reversible systems (evaporator and condenser), which reduces material mass by a third and quarters the refrigerant quantity, compared to a finned-tube HX.
- Tancabel et al. (2020) – 30% in R290 charge through an alternative HX design.
- Wang et al. (2019) – air conditioner with R290 using various techniques leading to improvement of up to 45% in efficiency.
- Zhou and Gan (2019) – reduce R290 charge by xx% through use of a novel HX design.

Ongoing developments associated with charge reduction and efficiency improvements will lead towards smaller DHP specific charge amounts in parallel with higher efficiencies thereby enabling greater flexibility for design and installation of systems in the near future.

11.3 Development schedules

An important consideration in this discussion are development schedules for products using alternative refrigerants. Revised designs of DHPs for alternative refrigerant can be realised relatively quickly (a year or two) but compressor design and optimisation and refinement iterations can take longer, as well as scaling-up extended production. However, given the large number of models already using some GWP < 150 refrigerants – primarily R290 – it is evident that there is already a relatively healthy number of compressors available. Many European DHP manufacturers are relatively small, yet tend to have a large number of different models relative to the size of the enterprise; this corresponds to fewer staff to revise designs for new refrigerants. In particular, the number of staff with intimate knowledge of flammability/risk and applicable standards may be few. In parallel, current rapid growth in the DHP sector means that staff are being prioritised for satisfying demand, rather than investment in research and development. On the other hand, many manufacturers do already produce models with flammable refrigerants so for those which anticipated such a transition, in-house expertise should already be at some level. Arguably, it could require more than two or three years for most producers to achieve steady output of revised models using refrigerants with GWP < 150. One notable implication, though, is the number of tests necessary for generating rating data and moreover, that such tests may need to be repeated by a third party, according to proposals for the revision of the Ecodesign regulations. Such an extensive test regime could lead to further delays in R&D time.

12 Concluding remarks

Detailed assessment of alternative refrigerants for DHPs is potentially complex, due to the variety of configurations, installation locations, functions and operating conditions. As such, the number of combinations has to be rationalised to help simplify the process. Cases analysed are outdoor ATW and indoor LTW models, providing 10 kW of nominal heating capacity. From a technical perspective, the main considerations associated with the viability of universally applying refrigerants with GWP < 150 to DHPs are whether:

- the desired efficiency levels can be achieved;
- (flammable) refrigerants can be applied within the constraints of the applicable safety standards and regulations;
- lifetime CO_2e emissions are sufficiently below the levels arising from the use of medium or high GWP refrigerants (considering uncertainty ranges);
- the incremental costs do not adversely affect the product price and are within tolerable limits of additional expenditure.

From the analysis, it is evident that these four technical criteria can be achieved with at least two (R1270 and R290), possibly three (and R152a) of the refrigerants with GWP < 150. It is evident that all common DHPs could be substituted with such alternatives, whilst gaining cost and emissions reductions and raising efficiency as required.

The databases, simulations and other sources from which the information was gathered is extensive, and the treatment and analysis was thorough and approached with different methodologies. Furthermore, where there is uncertainty associated with any inputs, pessimistic assumptions were used. Thus, the results and conclusions are considered to have a high level of confidence.

There are no outright obstructions to the adoption of alternatives with GWP < 150 due to refrigerant charge limitations. Where charge amounts (particularly of A3 refrigerants) are large, additional RMCs need to be applied. Selected RMCs should be done with due consideration of first cost, lifetime energy use and associated emissions. (For example, continuously operating fans intended to dilute a mixture in the event of a possible leak, is usually undesirable due to the potentially large energy consumption which could outweigh GWP-based emissions reductions.) Evidently, more versatile requirements within safety standards would be of benefit so that more innovative RMCs could be developed and applied.

Choice of integration time horizon (ITH) for refrigerant GWP has a major effect on the quantification of emissions and it is not proportional. Although selecting R1270, R290, R1234yf or R152a over R32 results in a one-third reduction in emissions compared to R410A when using 100 y ITH GWPs, this is nearly doubled when calculating with a 20 y ITH. Given the narrowing forecast for achieving emissions reductions, this is an increasingly important result.

For A2 (R152a) and A3 (R1270 and R290) refrigerants, certain RMC options can lead to higher incremental costs, either due to hardware/equipment requirements or lifetime energy consumption (from active solenoids or continually operating fans). However, use of leak detection can negate these costs. Even where the application of RMCs leads to higher component costs, it is usually a fraction of 1% of the DHP retail price. Whilst the RMC costs for R1234yf are relatively minor, there are potentially detrimental cost implications, relative to R410A, driven by HX material and refrigerant costs.

Nonetheless, the overall cost benefit of using any low GWP alternatives is likely to escalate over time, as the restriction and consequent costs of using medium and high GWP refrigerants within the European market (i.e., due to the quota system) heighten. Recent literature reporting on research and development activity associated with use of alternative refrigerants in DHPs indicate not only significant interest, but also beneficial advances with the technology. Thus, as the market for alternatives with GWP < 150 expands, there will undoubtedly be an accompanying increase in research and development for their application in DHPs, leading to significant technological advances, providing further improvements in cost-effectiveness.

There are other societal implications associated with adoption of the proposed alternatives, that should also be given consideration.

For those manufacturers with limited experience with (A2 and A3) flammable refrigerants, there is a learning curve to be followed. But flammability safety is only one of many other aspects (other safety, efficiency, reliability, noise, cost reduction, functionality, etc.) that manufacturers have to handle, so it should not be put forwards as a unique type of barrier. Similarly, the inconvenience of flammability issues to the supply chain is unlikely to be a tangible issue; after all, heat pumps are replacing gas burning equipment.

Lack of technician competence is often cited as an argument against the adoption of flammable refrigerants. But since the majority of DHPs are monoblock types they do not require refrigerant handling at installation. Refrigerant handling is only required during repair to the refrigerant circuit, corresponding to about one-hundredth of the installed base of DHPs. Taking the current annual output of 2 million DHPs per year, that is 20,000 “refrigerant handlings” per year or 30 per day; equally distributed, that is about one or two additional competent flammable refrigerant technicians per country per year. Obviously, this is a gross simplification, but at least it provides a perspective on the situation.

More broadly, a service sector tends to respond to a demand. As more and more flammable refrigerant competent technicians are needed, untrained technicians subject themselves to the desired training to satisfy that demand in a progressive manner. In reality, there will not, suddenly one day, be tens of millions of new DHPs leaking flammable refrigerants, demanding competent technicians to repair leaks or change compressors.

In conclusion, from a technical (and non-technical) perspective there is no rational justification for retaining high or medium GWP alternatives in favour of GWP < 150.

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Annex A Heat exchanger characteristic data

ATW finned-tube evaporator

ATW finned-tube evaporator characteristic data are presented in Figure 79 to Figure 102. All characteristic data is based on successive iterations to converge at coil evaporating capacity (8.37 kW) required for the nominal heating capacity (10.0 kW). The evaporator selection is based on:

- Closest selection to pressure drop of 0.4 K and least material (metal) cost.

Selections and associated parameters are listed in Table 13.

Pressure drop

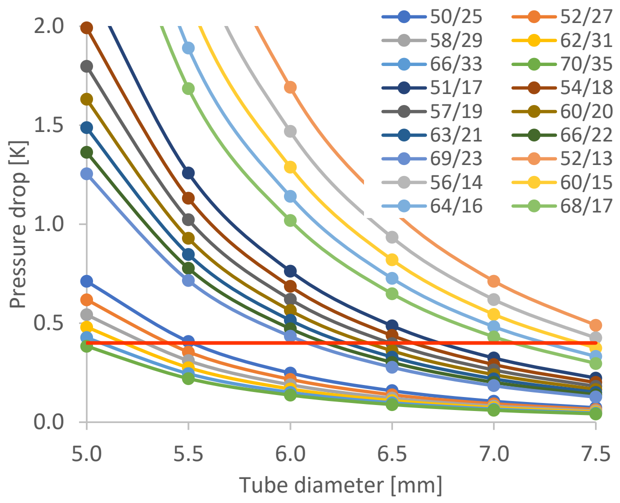

Figure 79: Selection balance point for R410A evaporator

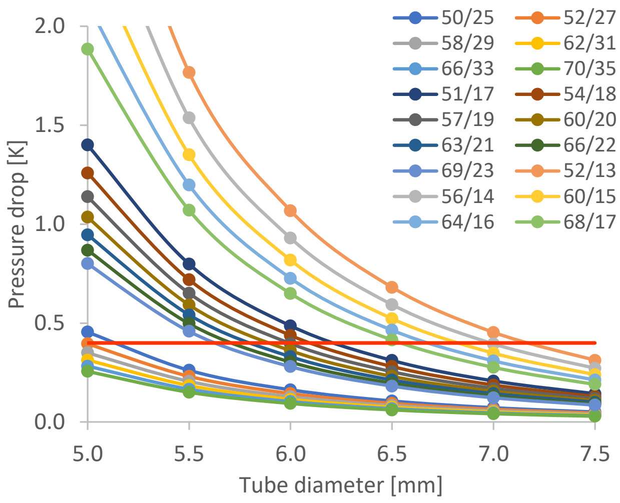

Figure 80: Selection balance point for R32 evaporator

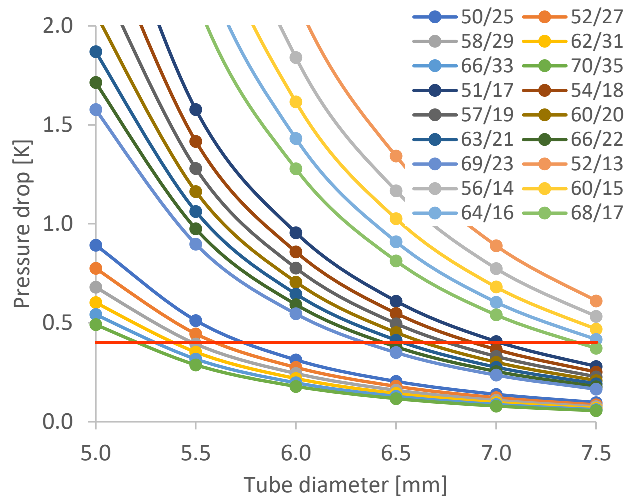

Figure 81: Selection balance point for R1270 evaporator

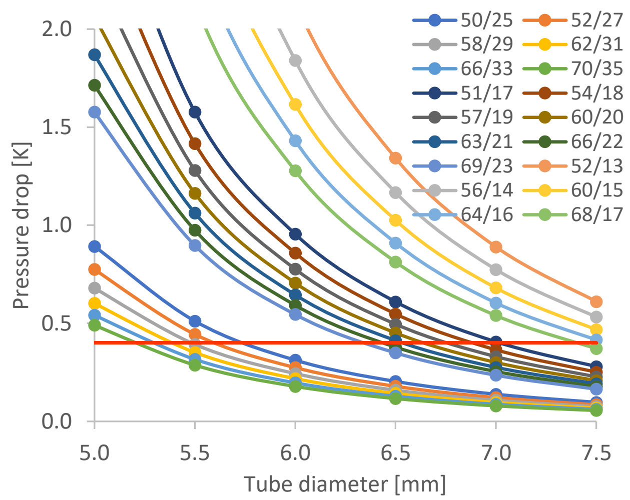

Figure 82: Selection balance point for R290 evaporator

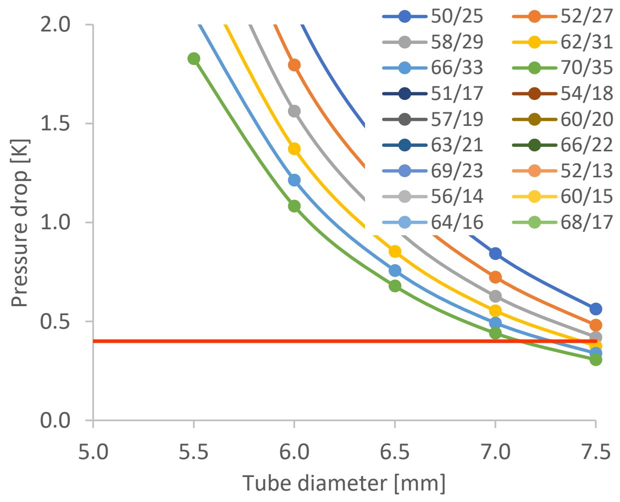

Figure 83: Selection balance point for R1234yf evaporator

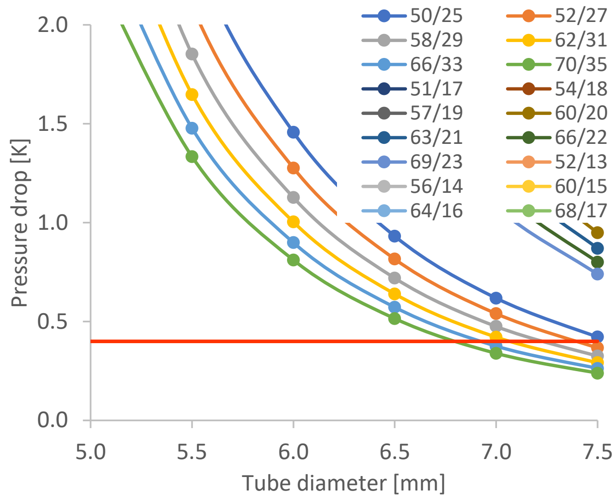

Figure 84: Selection balance point for R152a evaporator

Coil volume (same for all refrigerants)

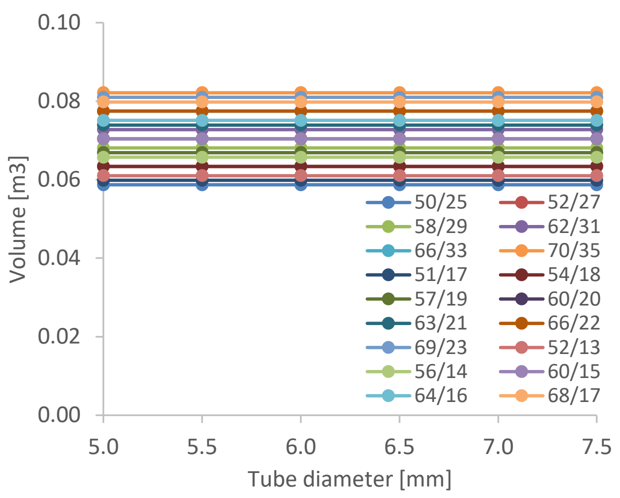

Figure 85: Coil volume for R410A

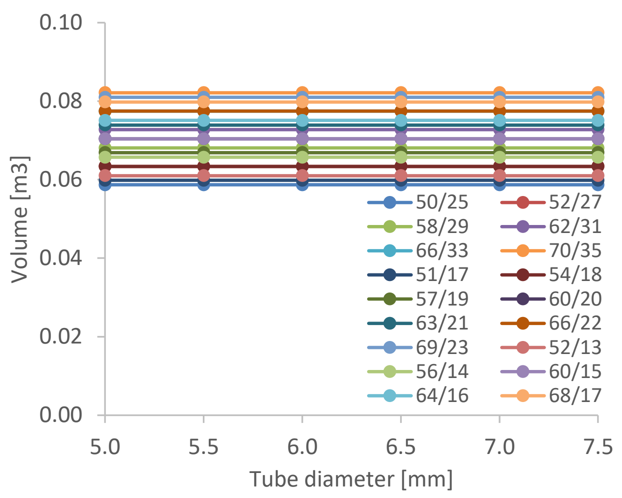

Figure 86: Coil volume for R32

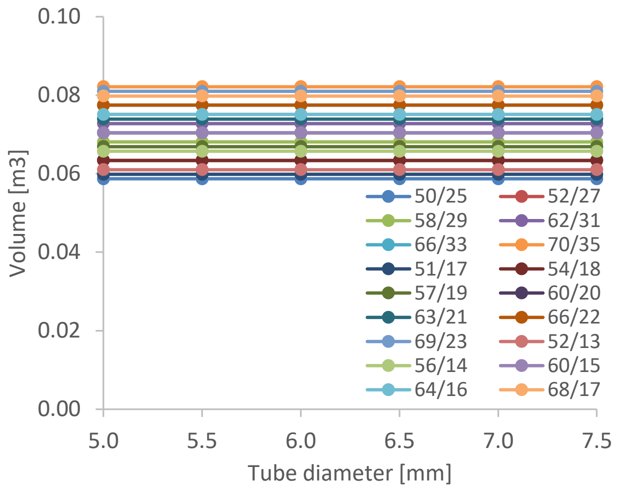

Figure 87: Coil volume for R1270

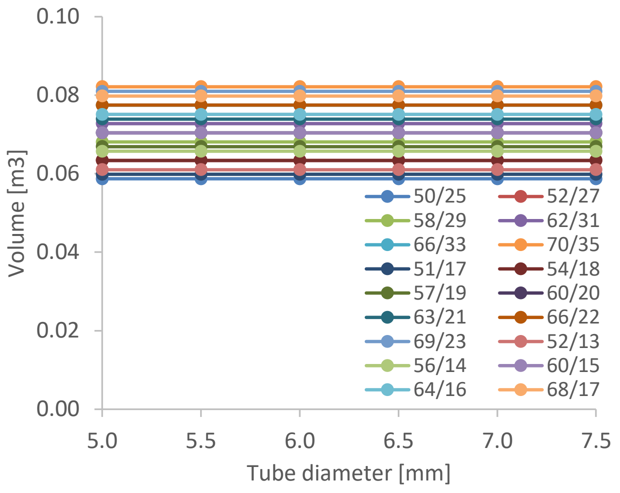

Figure 88: Coil volume for R290

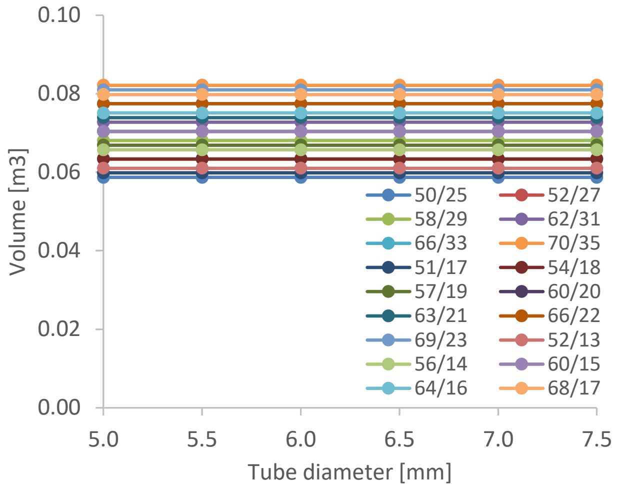

Figure 89: Coil volume for R1234yf

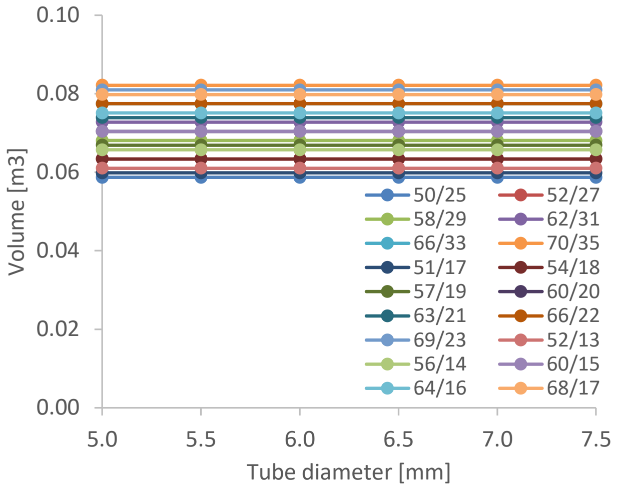

Figure 90: Coil volume for R152a

Refrigerant mass in operation

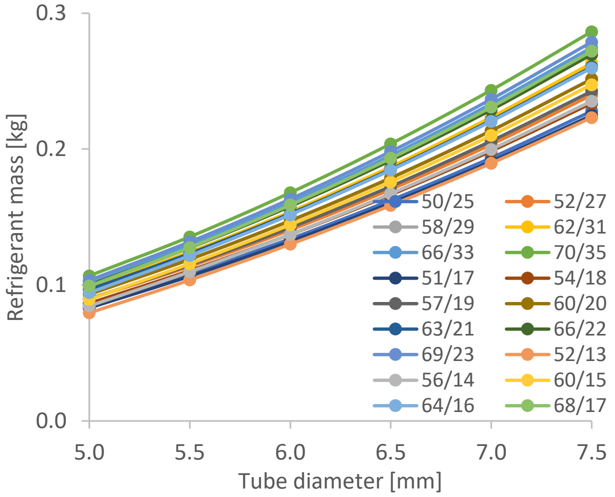

Figure 91: Mass of R410A in operating evaporator

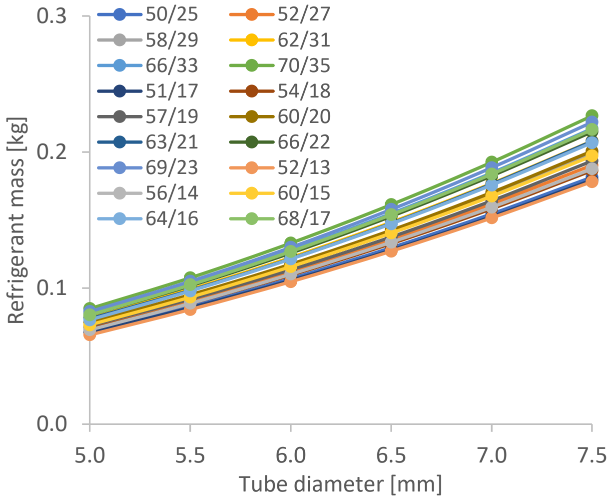

Figure 92: Mass of R32 in operating evaporator

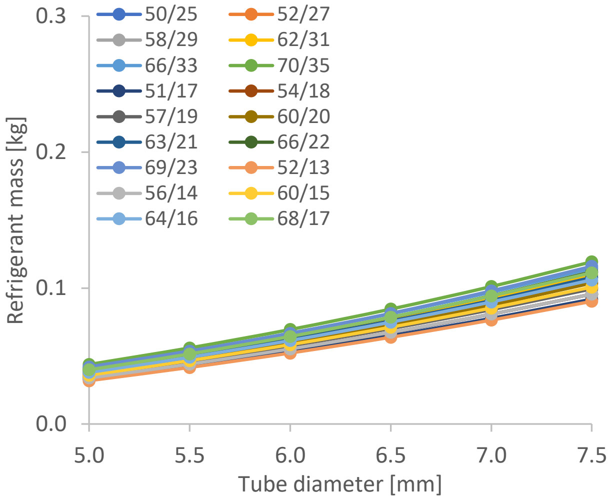

Figure 93: Mass of R1270 in operating evaporator

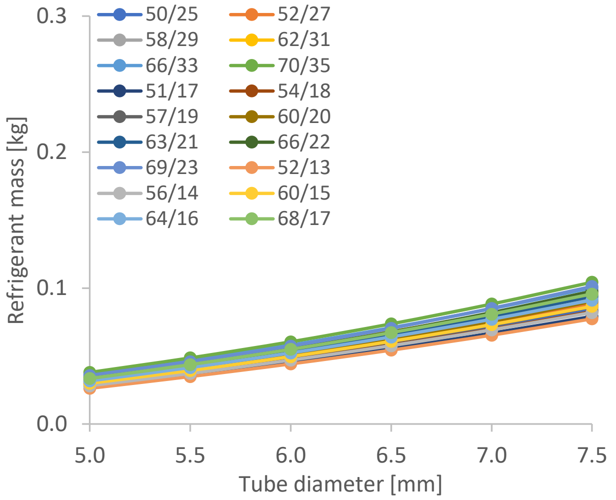

Figure 94: Mass of R290 in operating evaporator

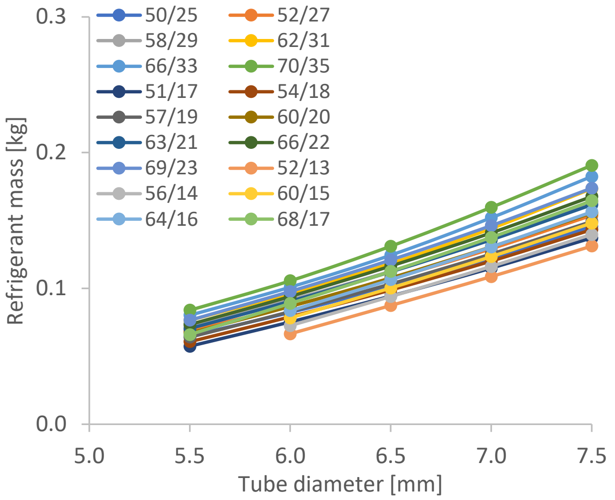

Figure 95: Mass of R1234yf in operating evaporator

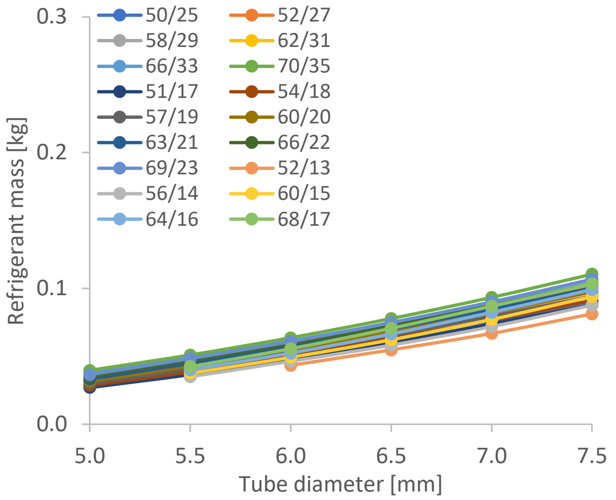

Figure 96: Mass of R152a in operating evaporator

Component material cost (including copper tubing and aluminium fins)

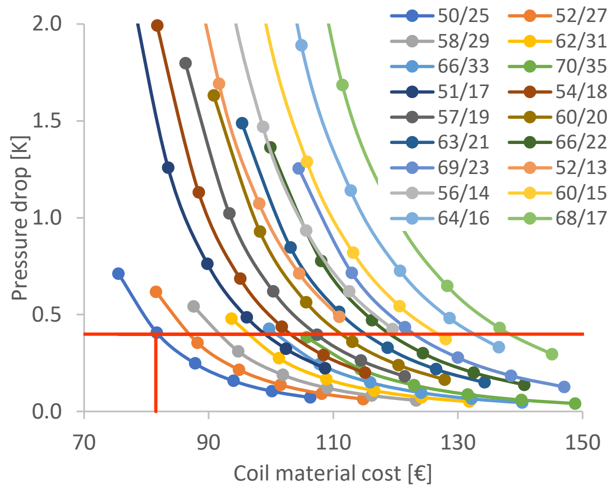

Figure 97: Sum of material costs for R410A

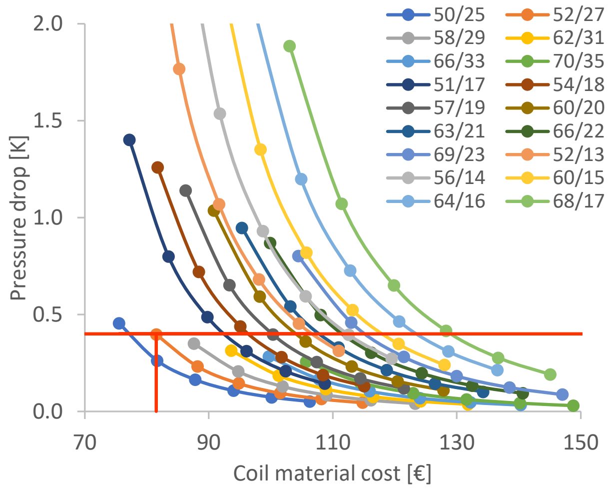

Figure 98: Sum of material costs for R32

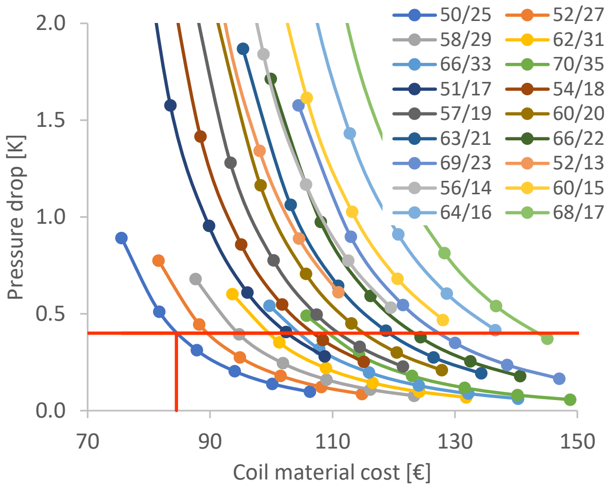

Figure 99: Sum of material costs for R1270

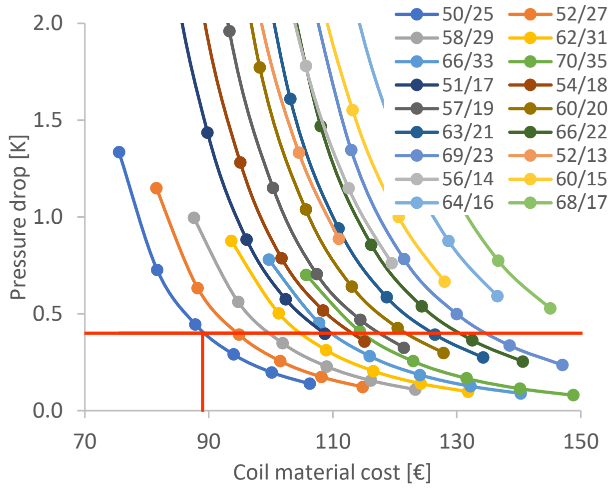

Figure 100: Sum of material costs for R290

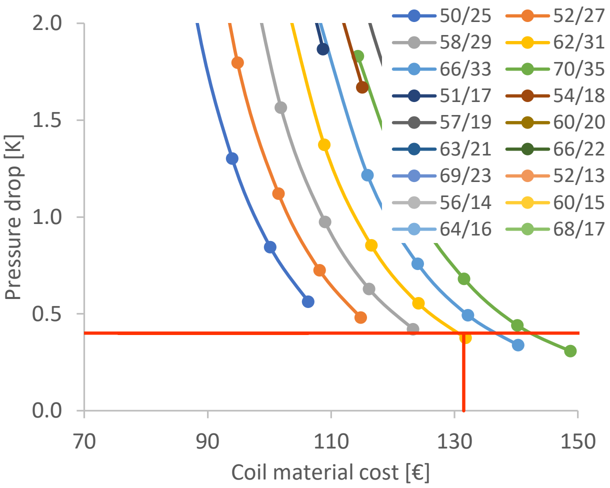

Figure 101: Sum of material costs for R1234yf

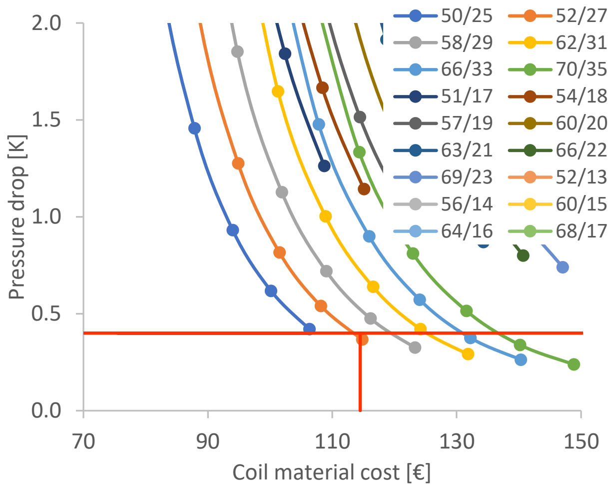

Figure 102: Sum of material costs for R152a

ATW plate HX condenser

ATW plate HX condenser characteristic data are presented in Figure 79 to Figure 102. The condenser selection is based on the following:

- All selections at nominal heating capacity.
- Closest selection to pressure drop of 0.1 K.

Selections and associated parameters are listed in Table 13.

Balance point

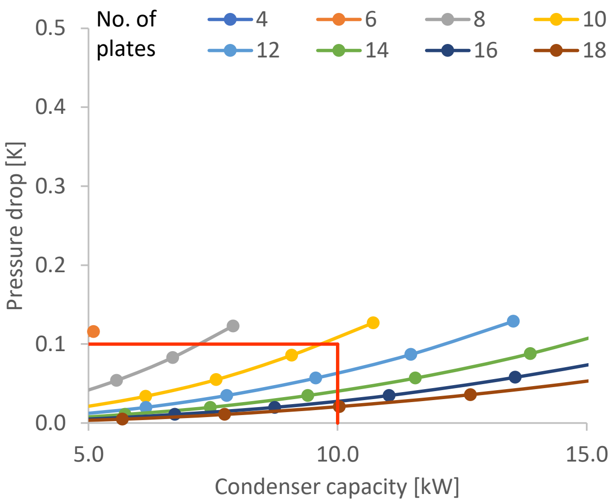

Figure 103: Selection balance point for R410A condenser

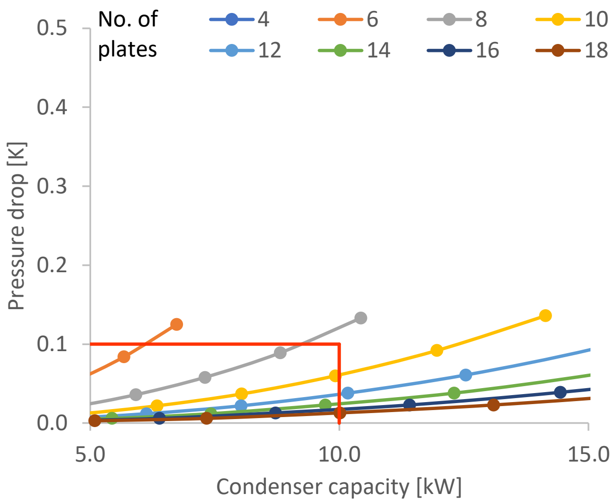

Figure 104: Selection balance point for R32 condenser

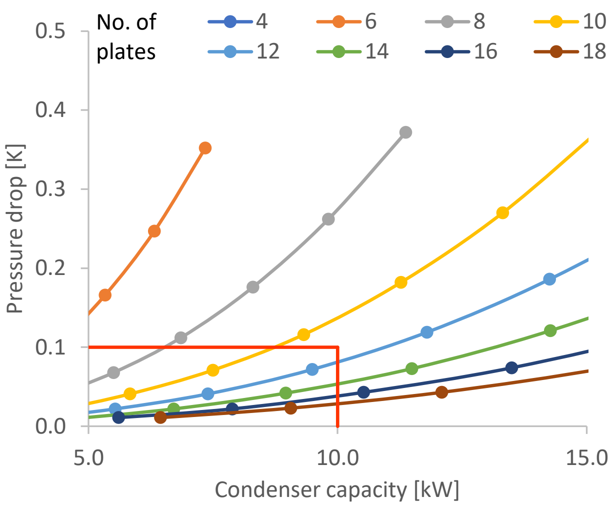

Figure 105: Selection balance point for R1270 condenser

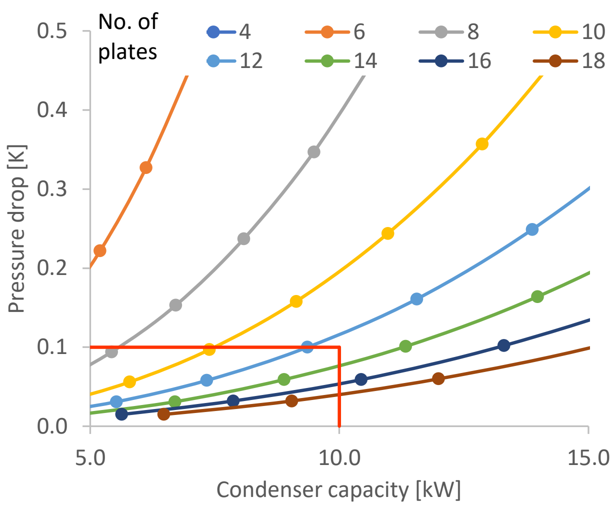

Figure 106: Selection balance point for R290 condenser

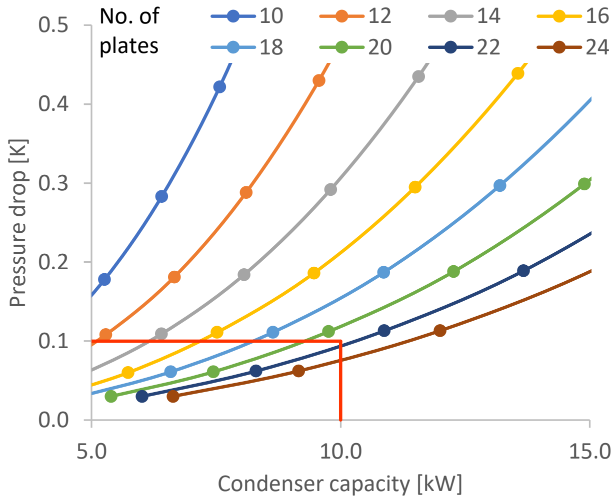

Figure 107: Selection balance point for R1234yf condenser

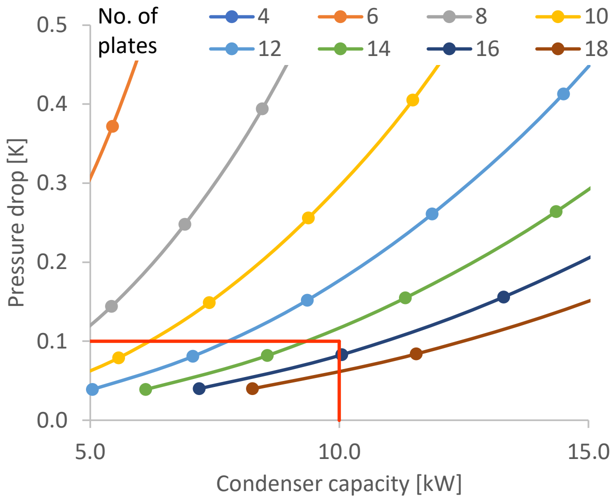

Figure 108: Selection balance point for R152a condenser

PHX volume

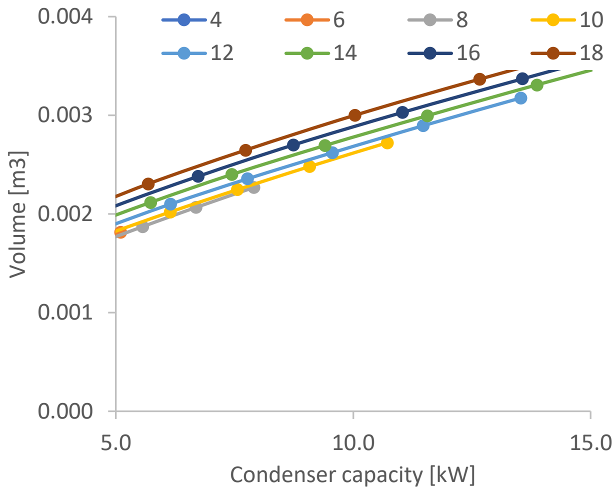

Figure 109: Coil volume for R410A

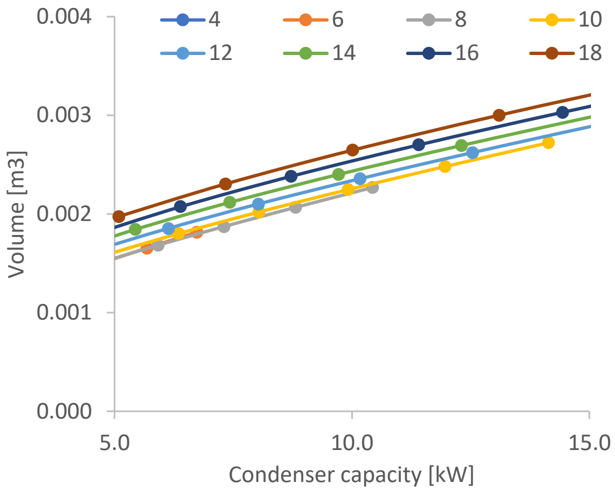

Figure 110: Coil volume for R32

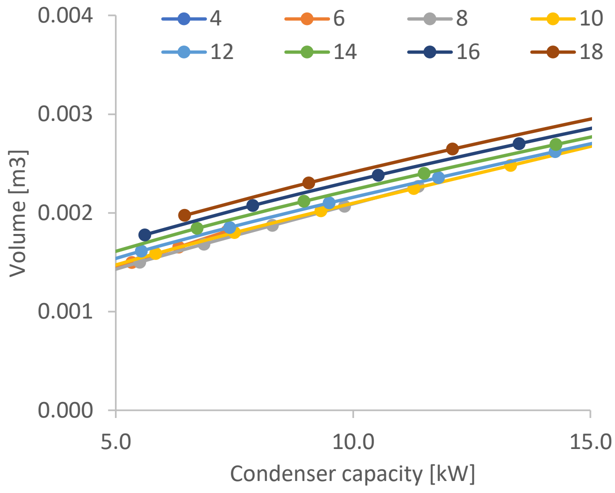

Figure 111: Coil volume for R1270

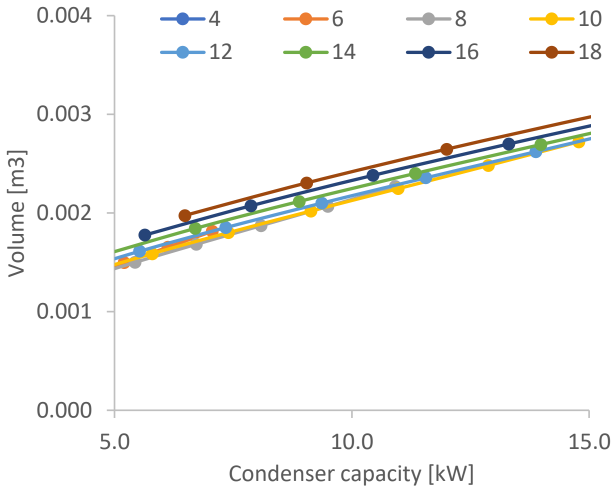

Figure 112: Coil volume for R290

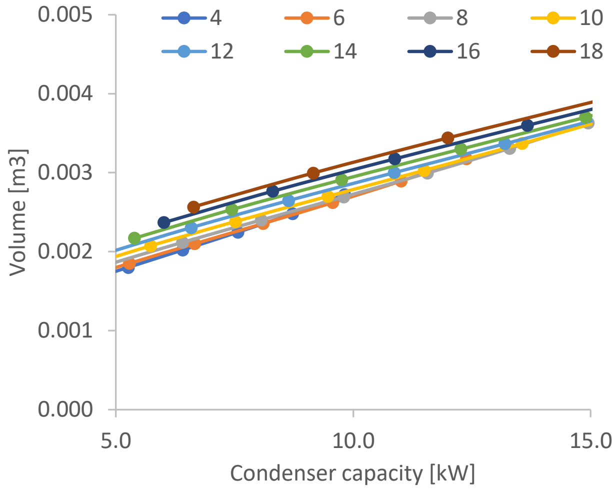

Figure 113: Coil volume for R1234yf

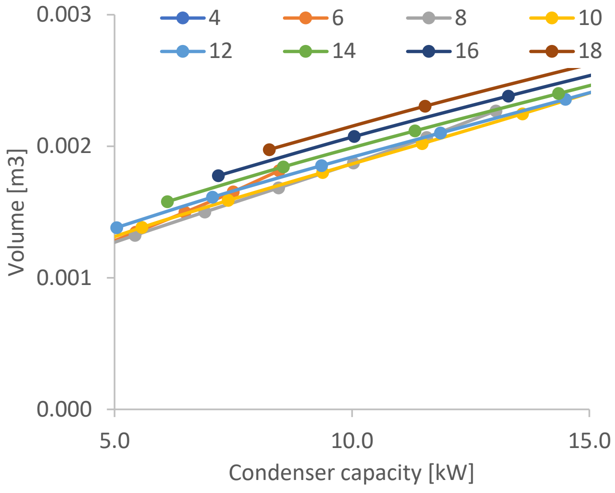

Figure 114: Coil volume for R152a

Refrigerant mass in operation

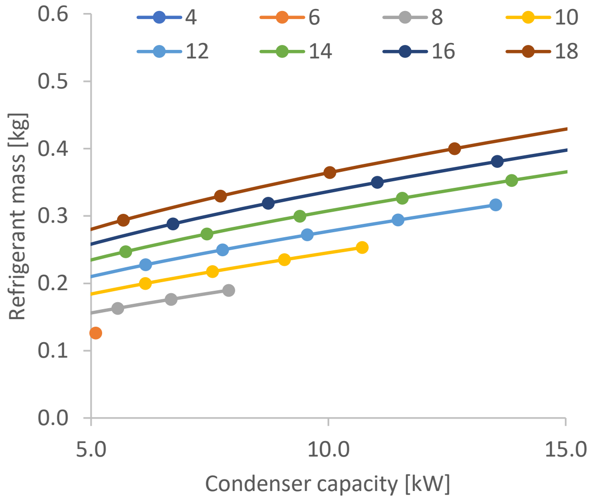

Figure 115: Mass of R410A in operating evaporator

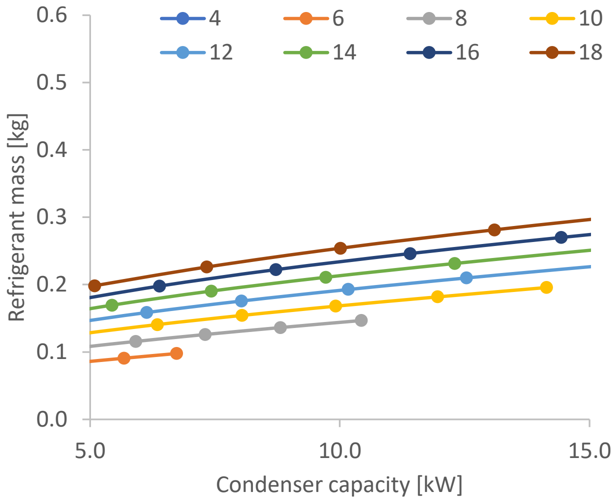

Figure 116: Mass of R32 in operating evaporator

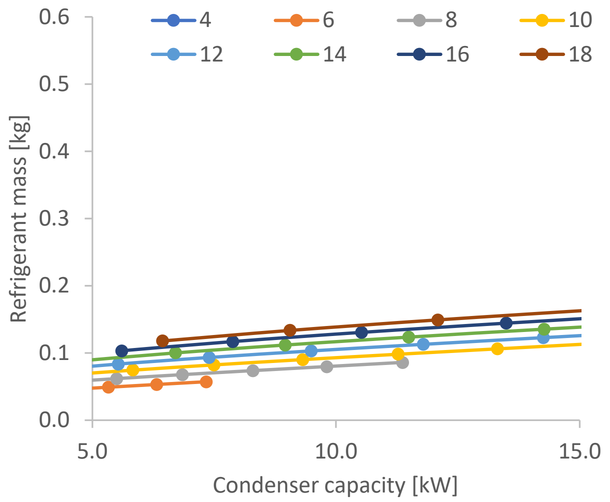

Figure 117: Mass of R1270 in opera ting evaporator

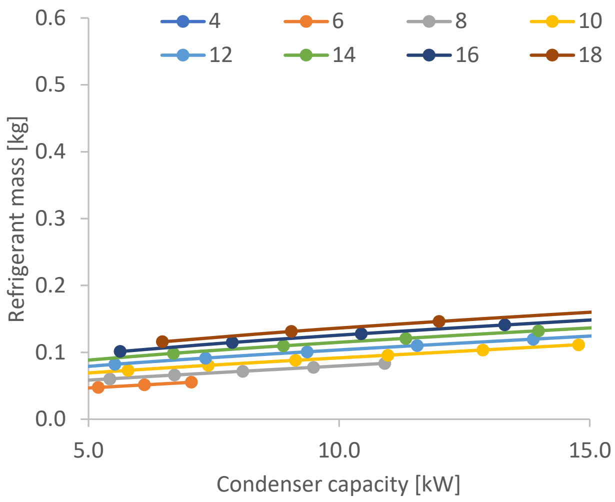

Figure 118: Mass of R290 in operating evaporator

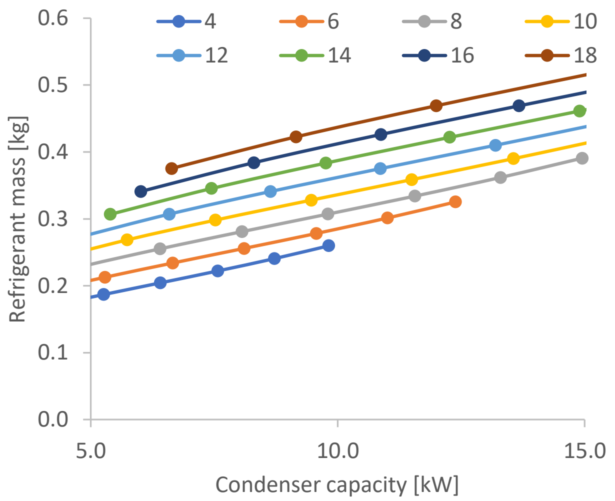

Figure 119: Mass of R1234yf in operating evaporator

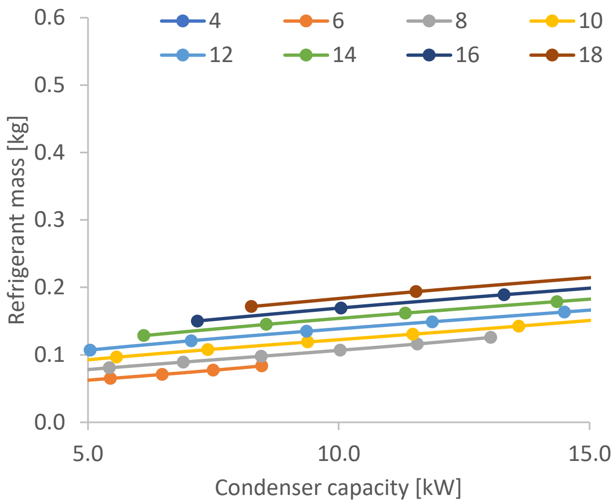

Figure 120: Mass of R152a in operating evaporator

Component material cost (including copper tubes, aluminium fins, refrigerant)

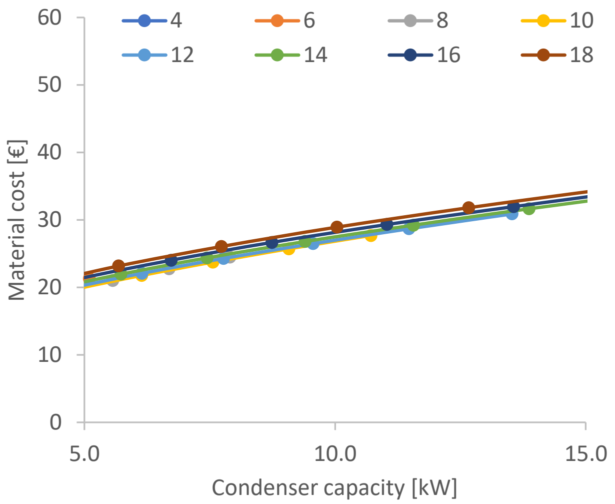

Figure 121: Sum of material costs for R410A

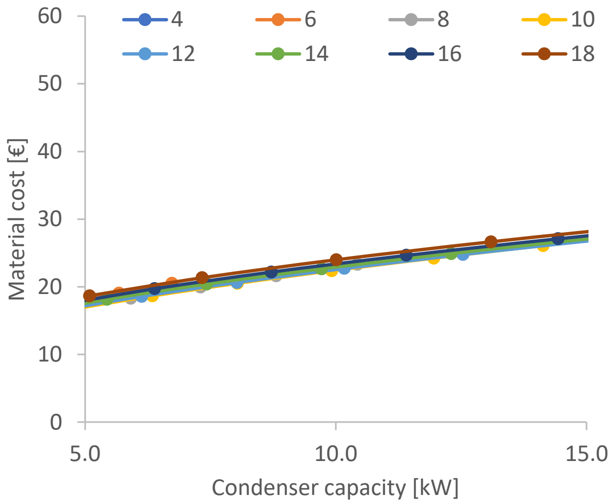

Figure 122: Sum of material costs for R32

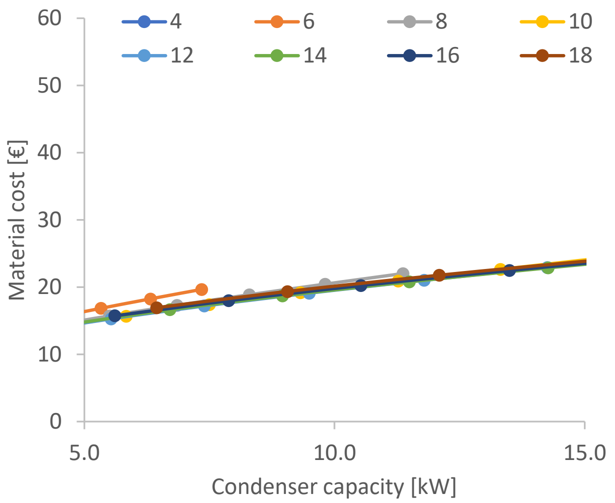

Figure 123: Sum of material costs for R1270

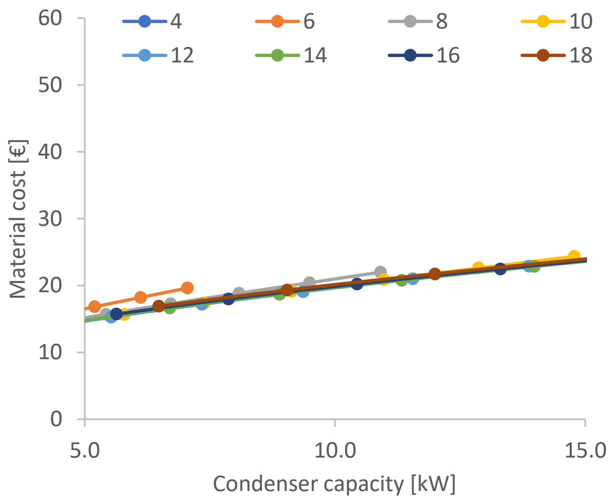

Figure 124: Sum of material costs for R290

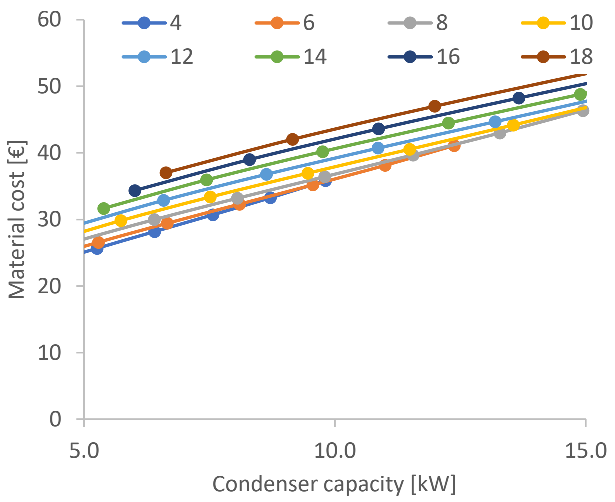

Figure 125: Sum of material costs for R1234yf

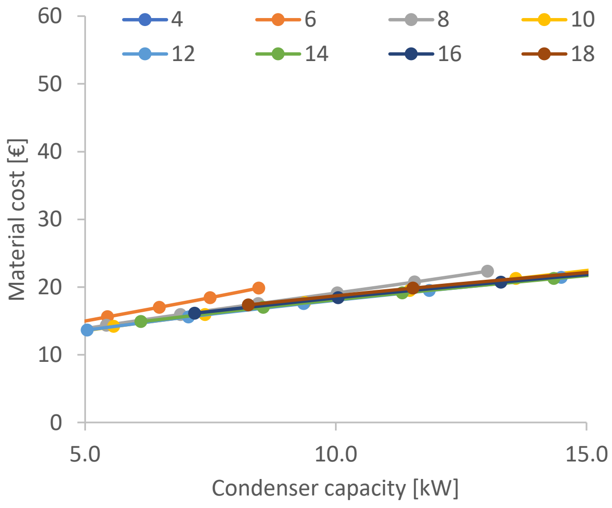

Figure 126: Sum of material costs for R152a

LTW plate HX evaporator

Evaporator characteristic data are presented in Figure 127 to Figure 150. The evaporator selection is based on the following:

- All selections at evaporating capacity required for nominal heating capacity.
- Closest selection to pressure drop of 0.1 K.

Selections and associated parameters are listed in Table 17.

Balance point

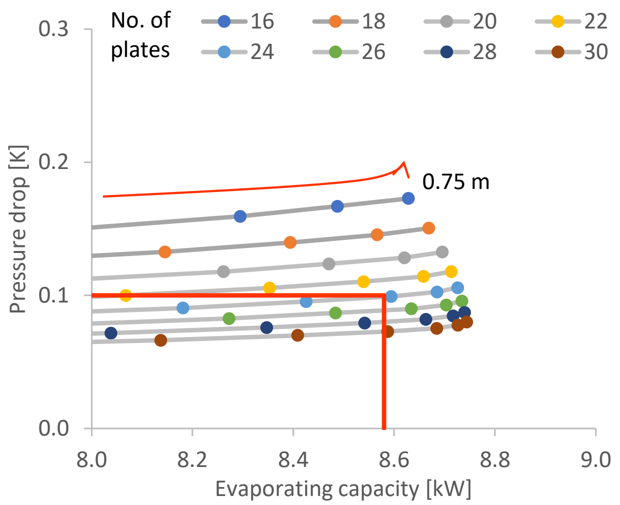

Figure 127: Selection balance point for R410A evaporator

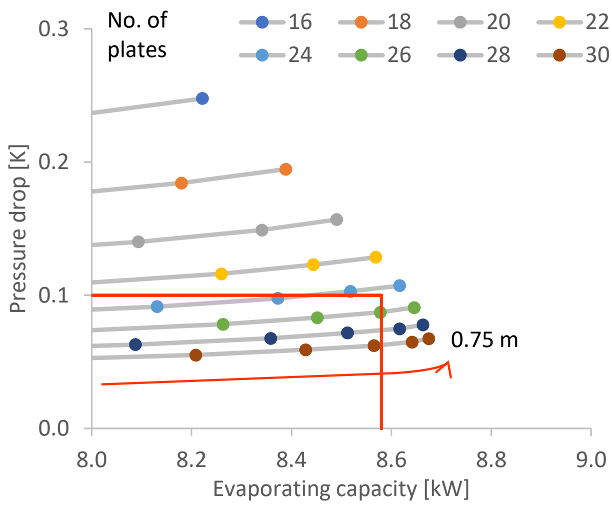

Figure 128: Selection balance point for R32 evaporator

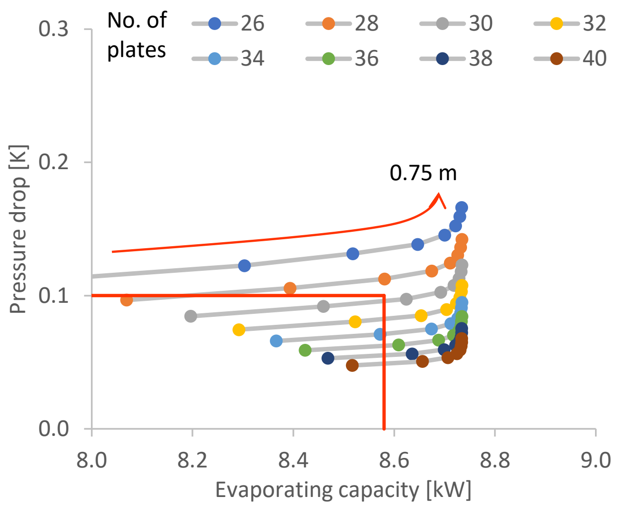

Figure 129: Selection balance point for R1270 evaporator

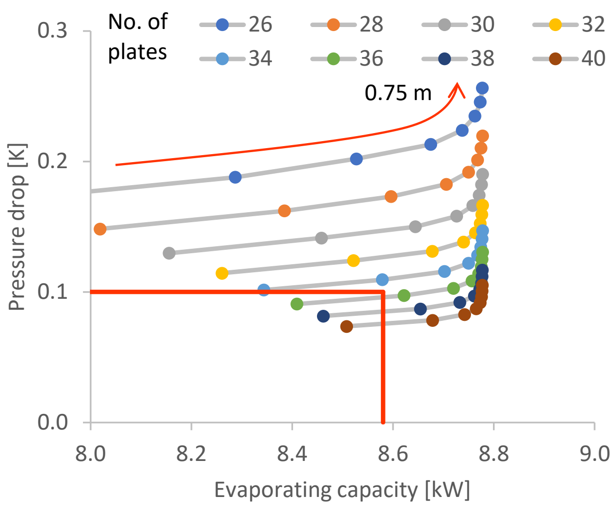

Figure 130: Selection balance point for R290 evaporator

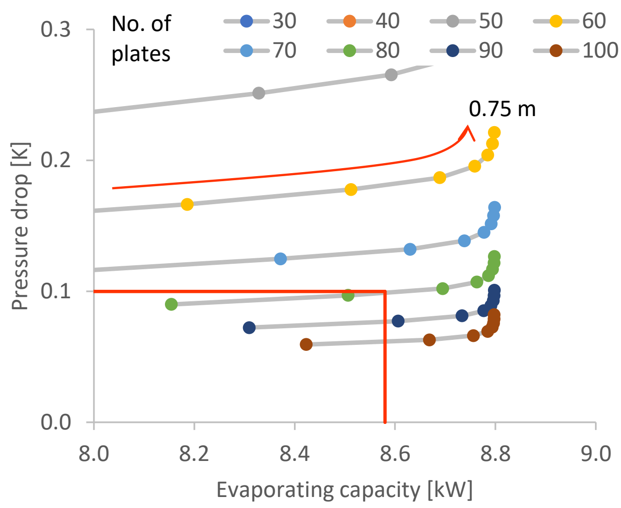

Figure 131: Selection balance point for R1234yf evaporator

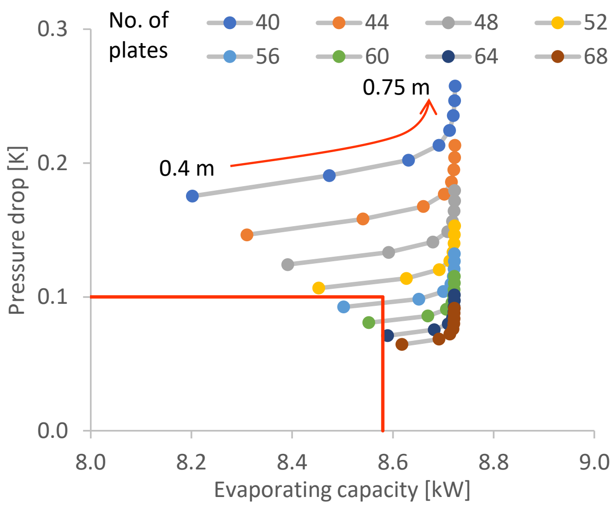

Figure 132: Selection balance point for R152a evaporator

PHX volume

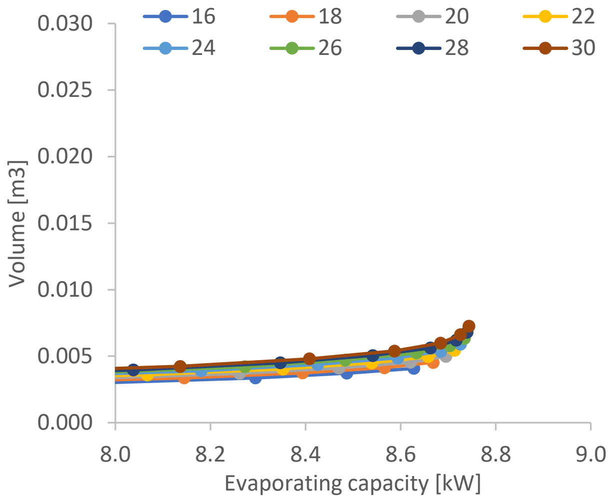

Figure 133: Coil volume for R410A

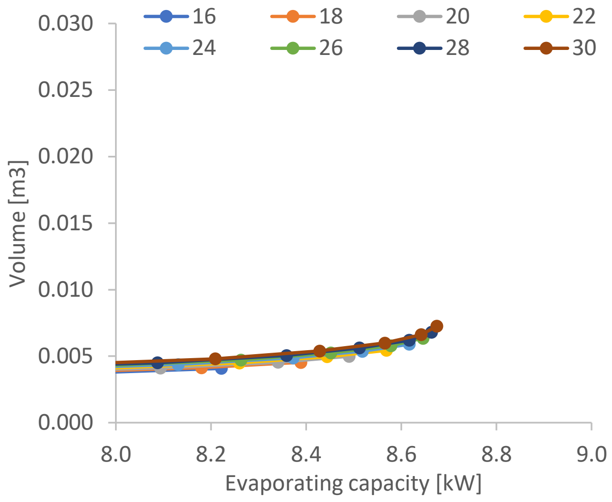

Figure 134: Coil volume for R32

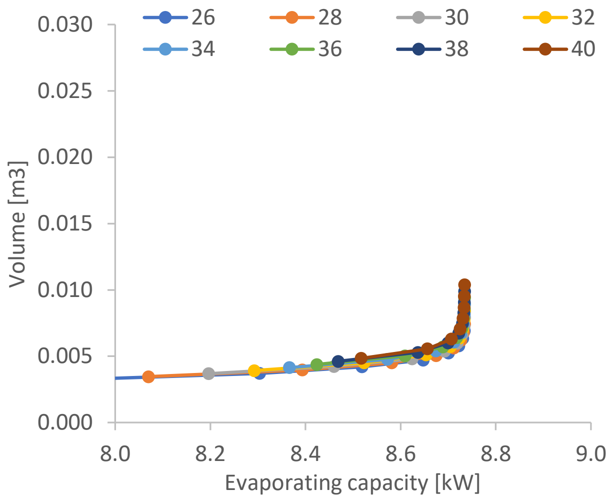

Figure 135: Coil volume for R1270

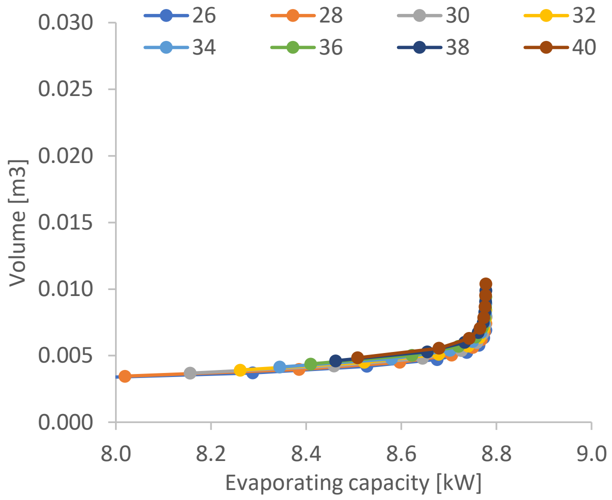

Figure 136: Coil volume for R290

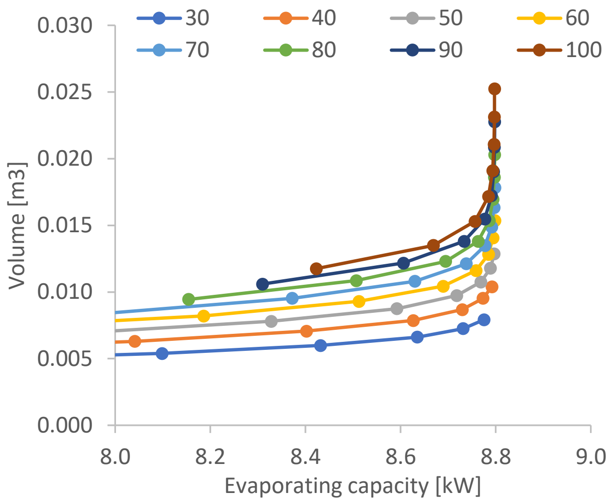

Figure 137: Coil volume for R1234yf

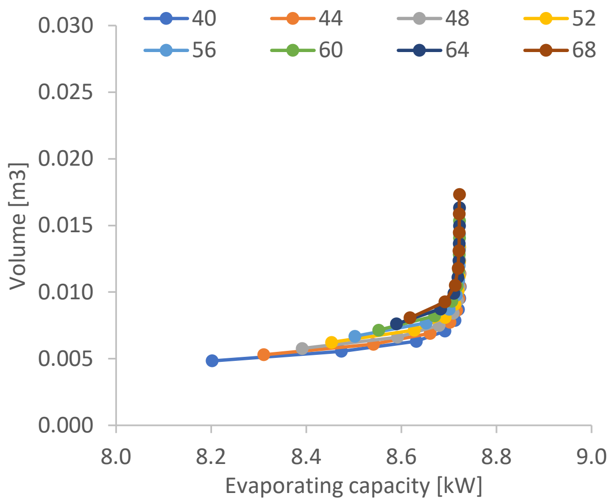

Figure 138: Coil volume for R152a

Refrigerant mass in operation

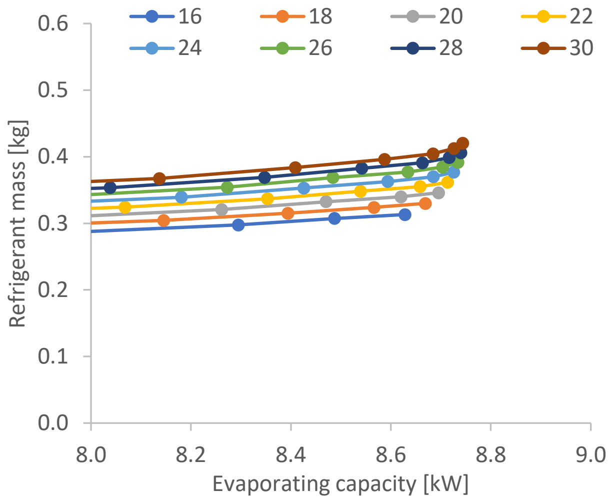

Figure 139: Mass of R410A in operating condenser

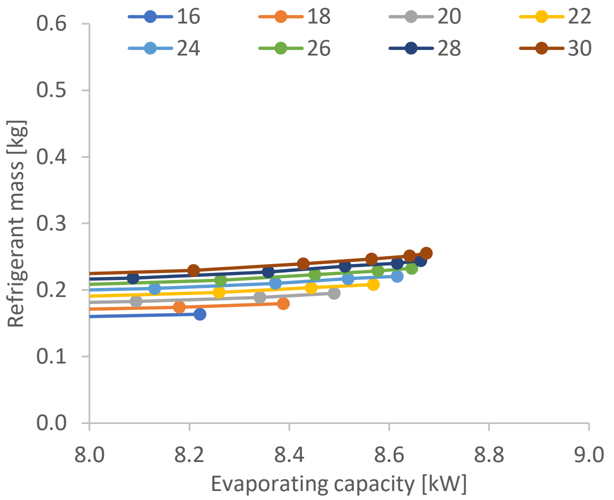

Figure 140: Mass of R32 in operating condenser

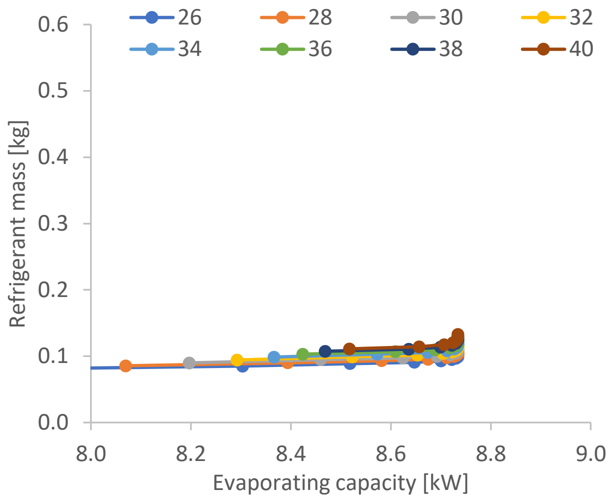

Figure 141: Mass of R1270 in operating condenser

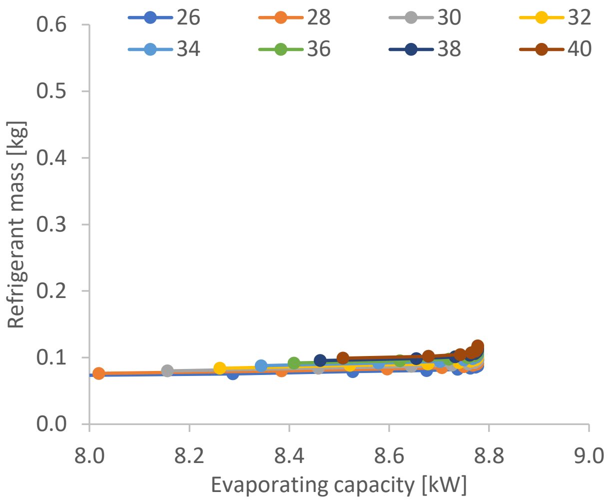

Figure 142: Mass of R290 in operating condenser

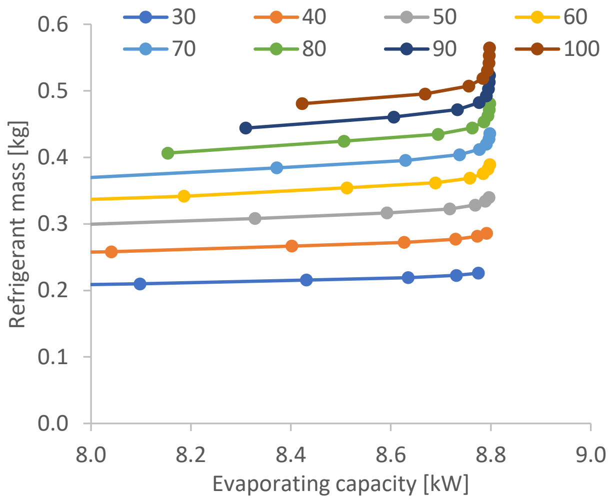

Figure 143: Mass of R1234yf in operating condenser

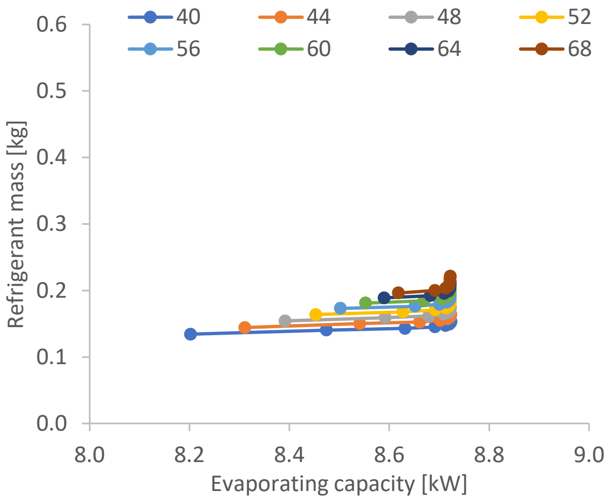

Figure 144: Mass of R152a in operating condenser

Component material cost (including copper tubes, aluminium fins and refrigerant)

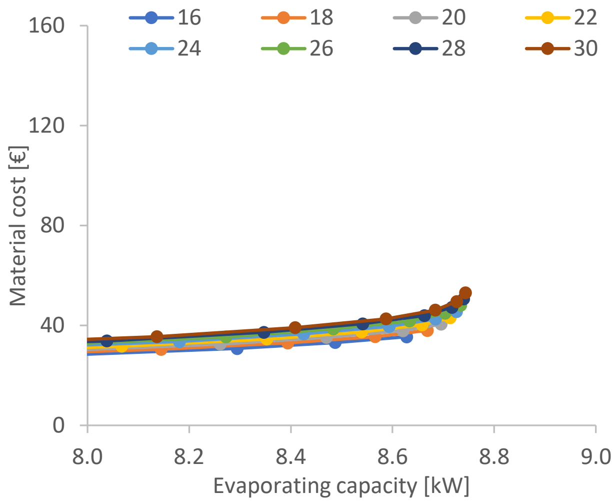

Figure 145: Sum of material costs for R410A

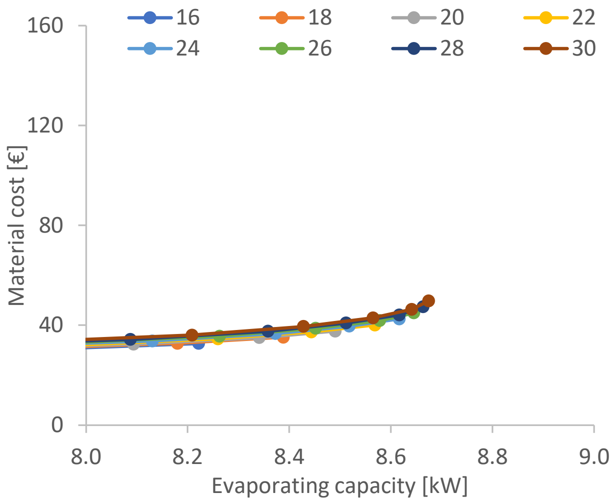

Figure 146: Sum of material costs for R32

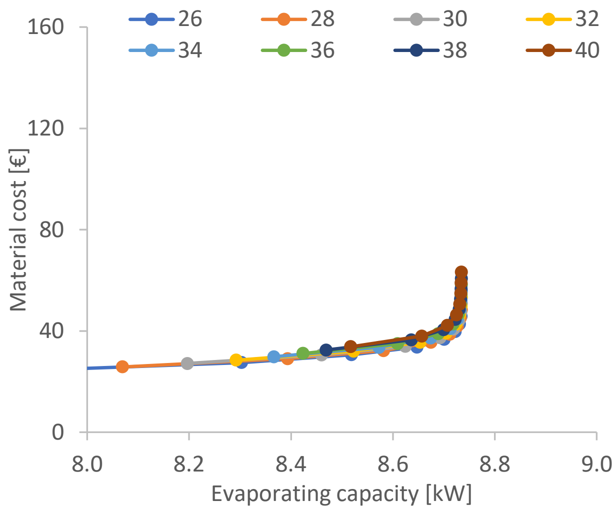

Figure 147: Sum of material costs for R1270

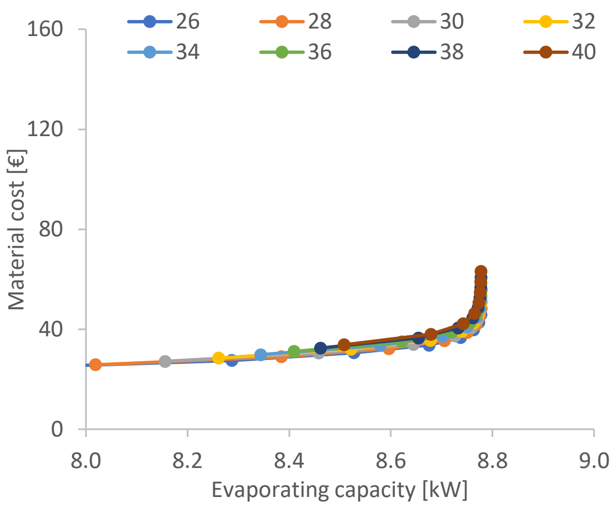

Figure 148: Sum of material costs for R290

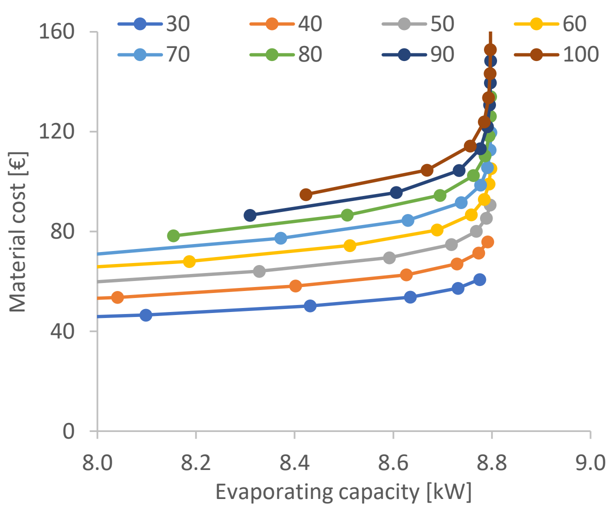

Figure 149: Sum of material costs for R1234yf

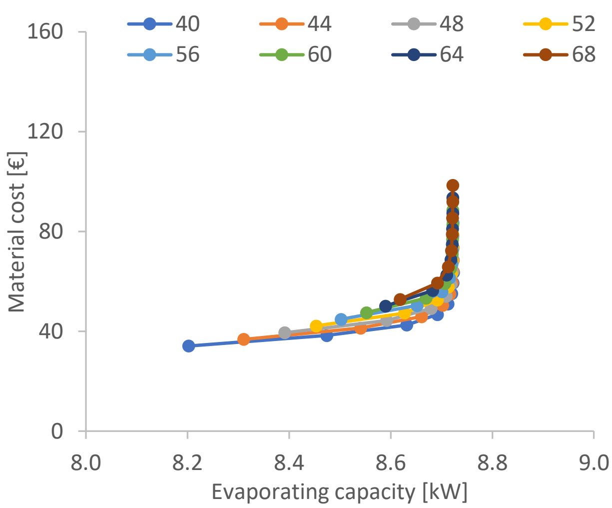

Figure 150: Sum of material costs for R152a

LTW plate HX condenser

Condenser characteristic data are presented in Figure 127 to Figure 150. The condenser selection is based on the following:

- All selections at nominal heating capacity.
- Closest selection to pressure drop of 0.05 K.

Selections and associated parameters are listed in Table 17.

Balance point

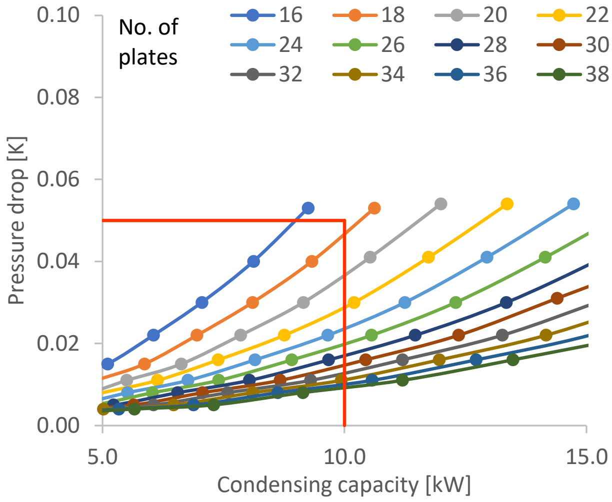

Figure 151: Selection balance point for R410A condenser

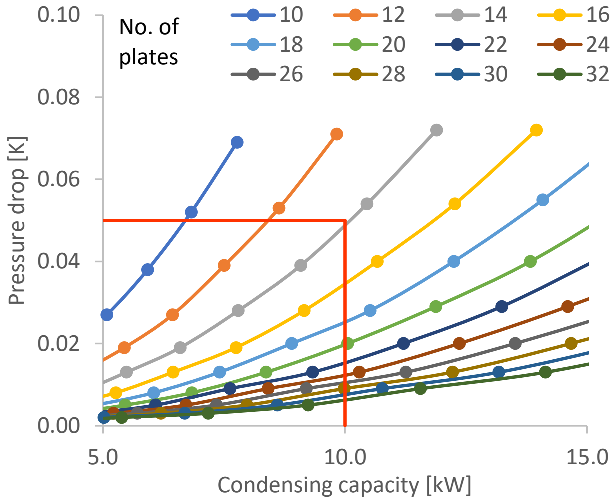

Figure 152: Selection balance point for R32 condenser

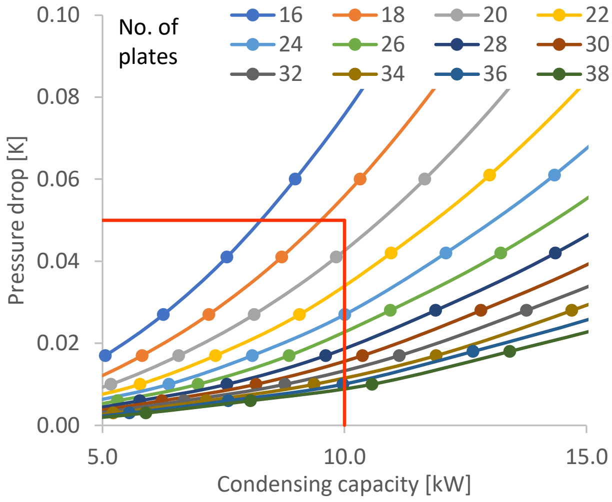

Figure 153: Selection balance point for R1270 condenser

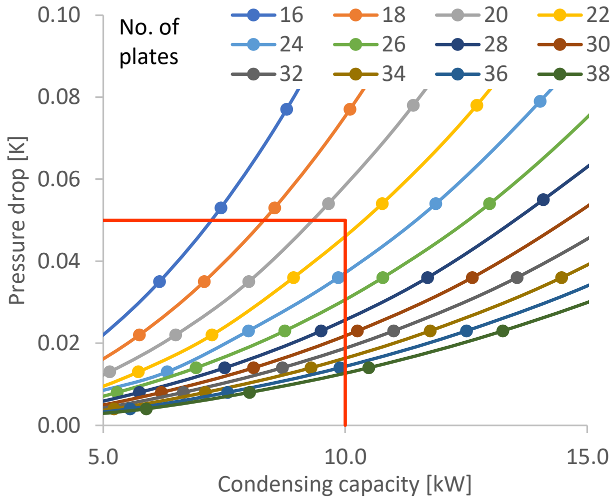

Figure 154: Selection balance point for R290 condenser

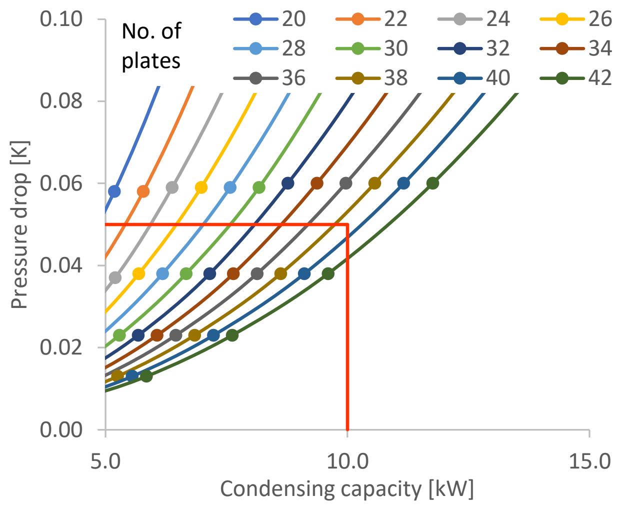

Figure 155: Selection balance point for R1234yf condenser

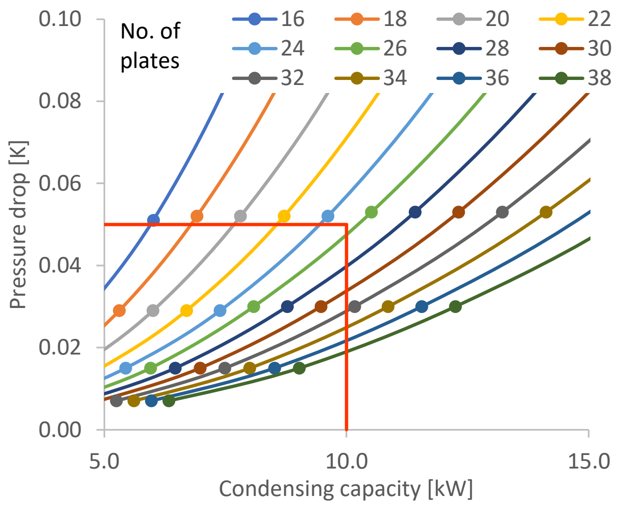

Figure 156: Selection balance point for R152a condenser

PHX volume

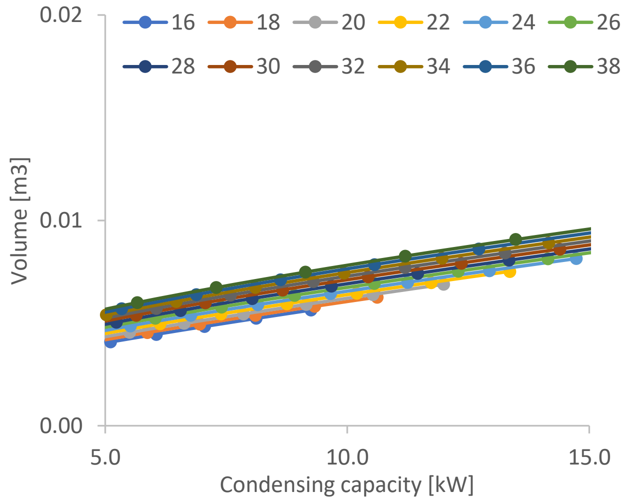

Figure 157: Coil volume for R410A

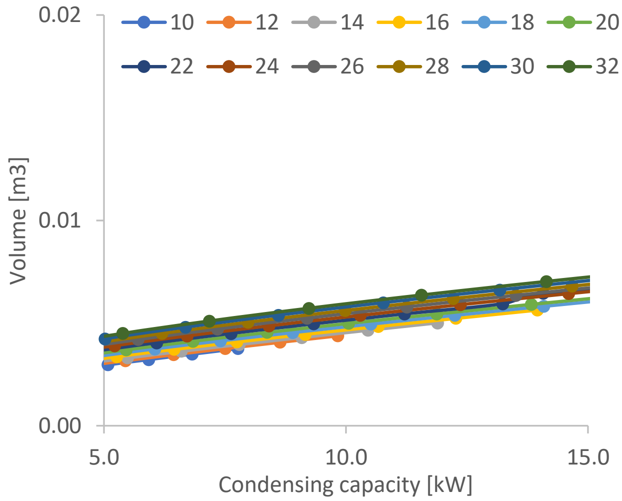

Figure 158: Coil volume for R32

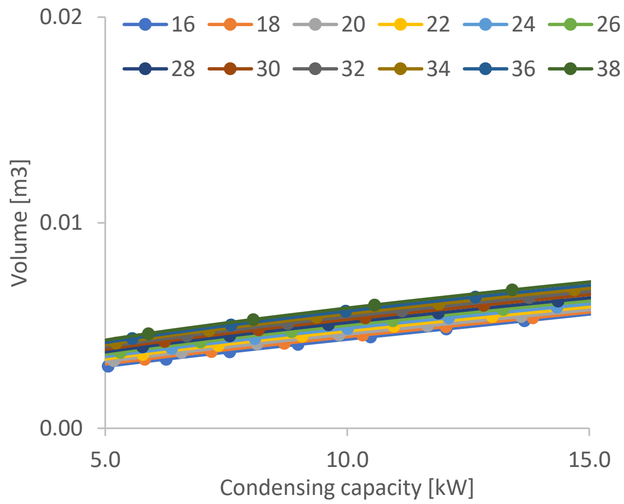

Figure 159: Coil volume for R1270

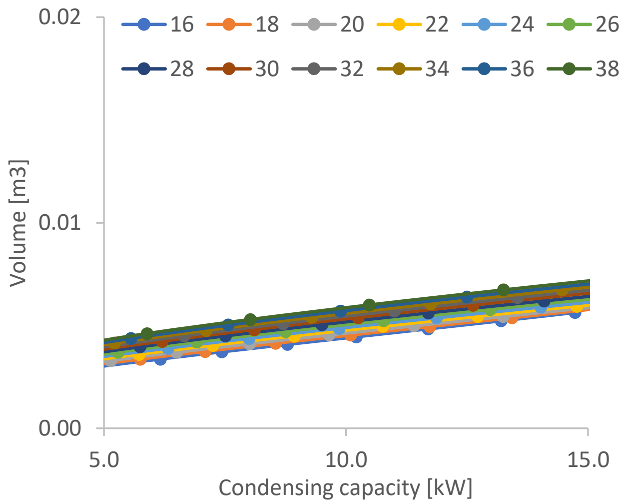

Figure 160: Coil volume for R290

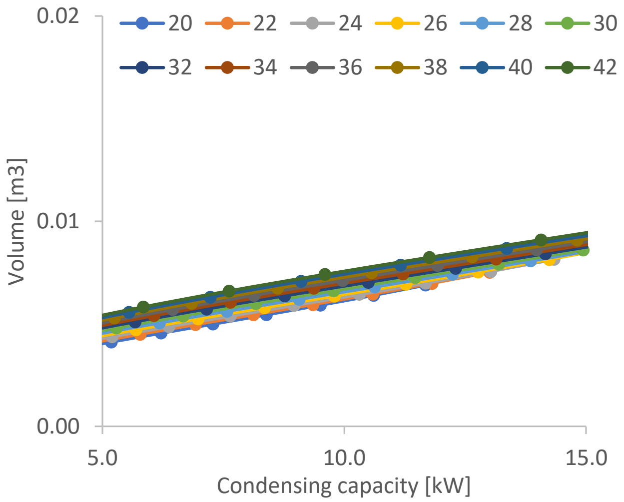

Figure 161: Coil volume for R1234yf

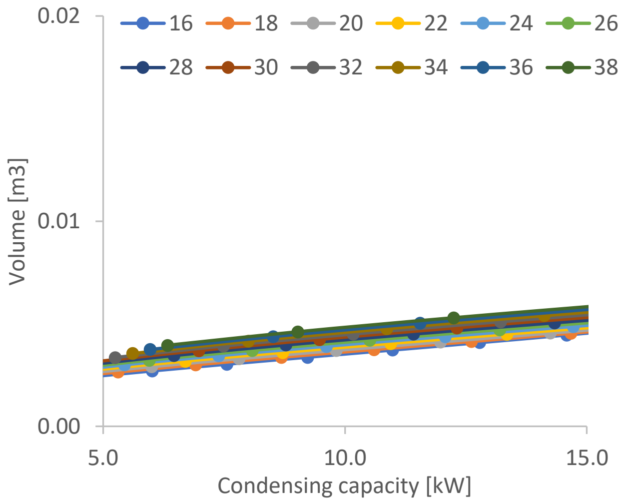

Figure 162: Coil volume for R152a

Refrigerant mass in operation

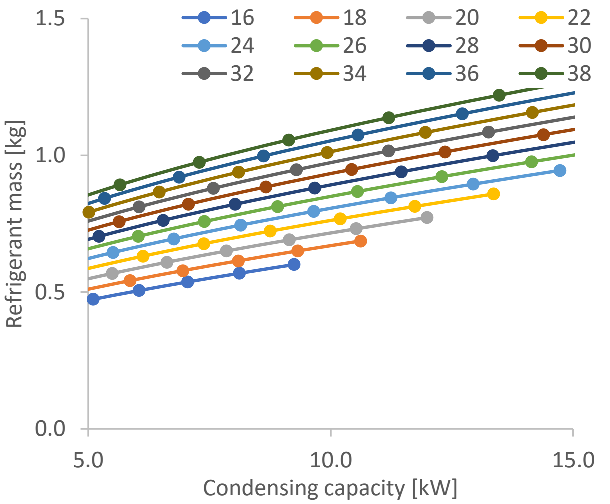

Figure 163: Mass of R410A in operating condenser

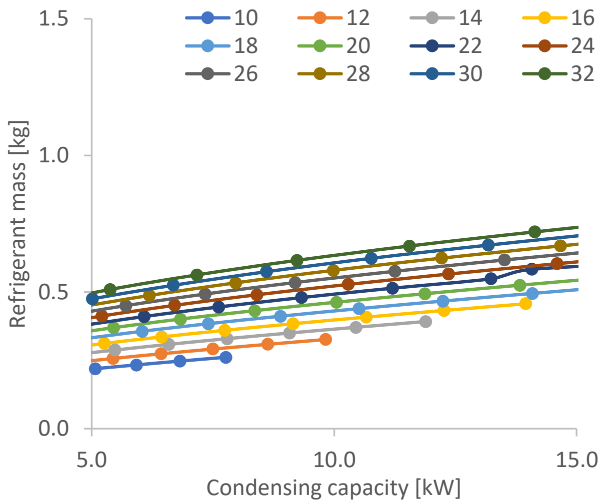

Figure 164: Mass of R32 in operating condenser

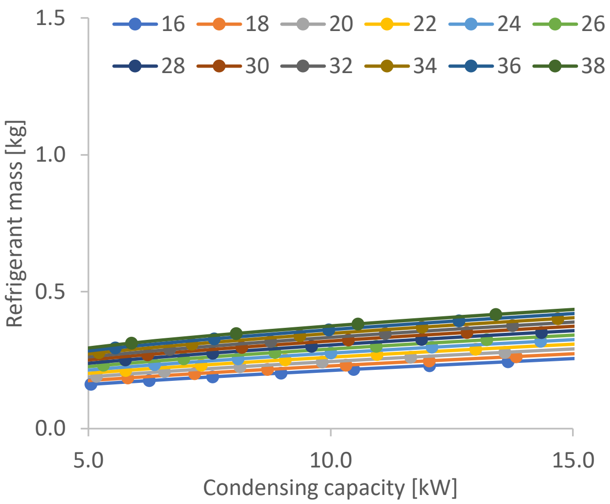

Figure 165: Mass of R1270 in operating condenser

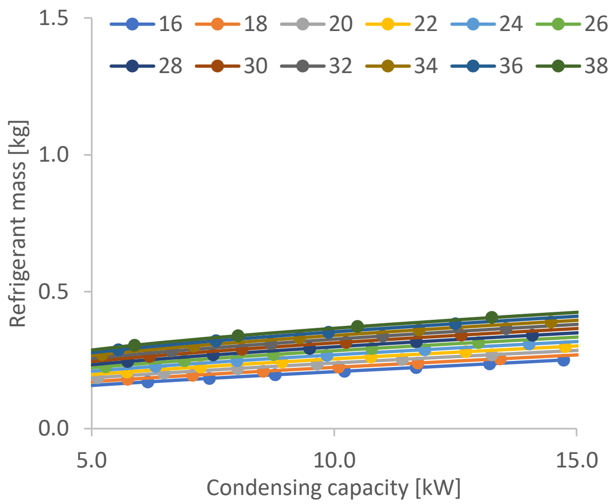

Figure 166: Mass of R290 in operating condenser

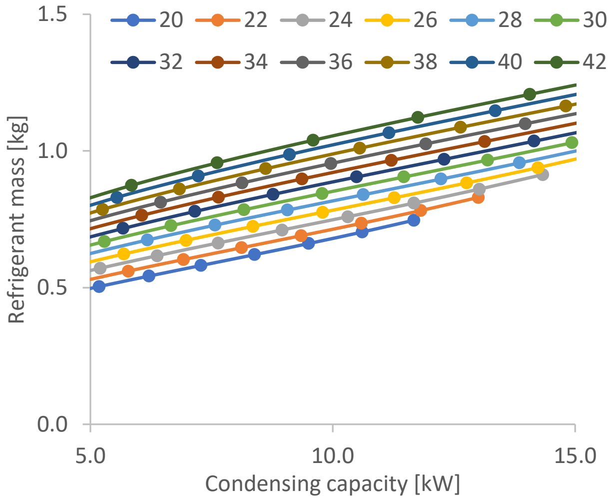

Figure 167: Mass of R1234yf in operating condenser

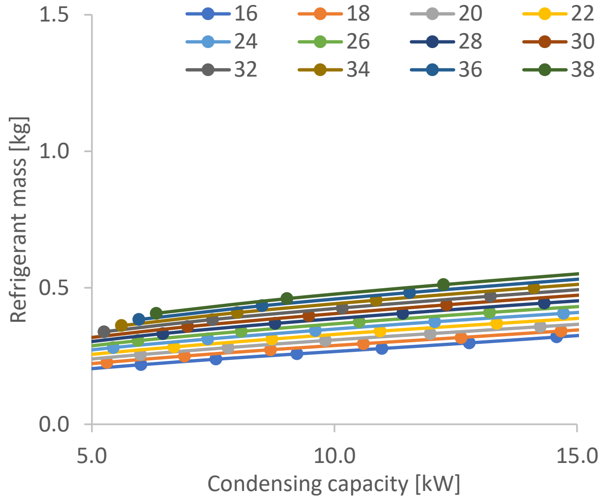

Figure 168: Mass of R152a in operating condenser

Component material cost (including copper tubes, aluminium fins, refrigerant)

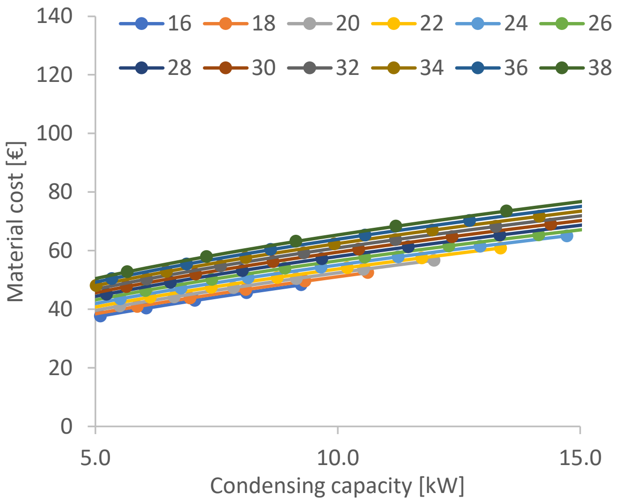

Figure 169: Sum of material costs for R410A

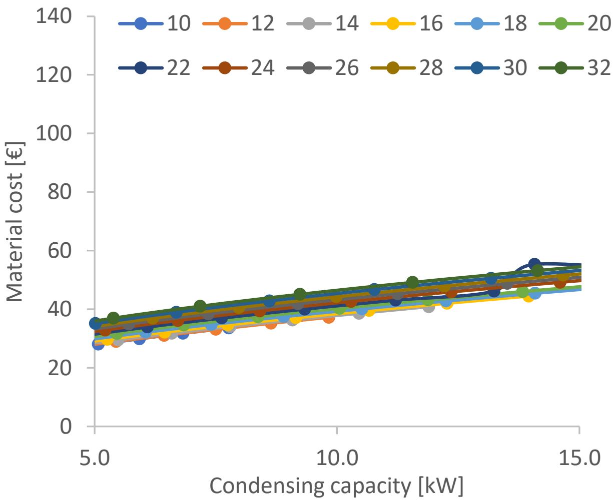

Figure 170: Sum of material costs for R32

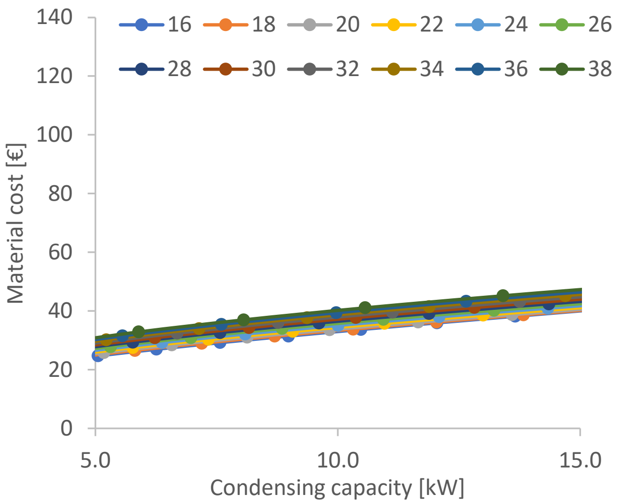

Figure 171: Sum of material costs for R1270

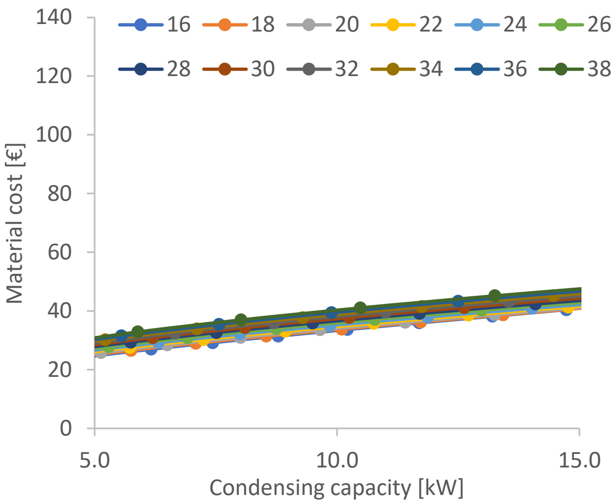

Figure 172: Sum of material costs for R290

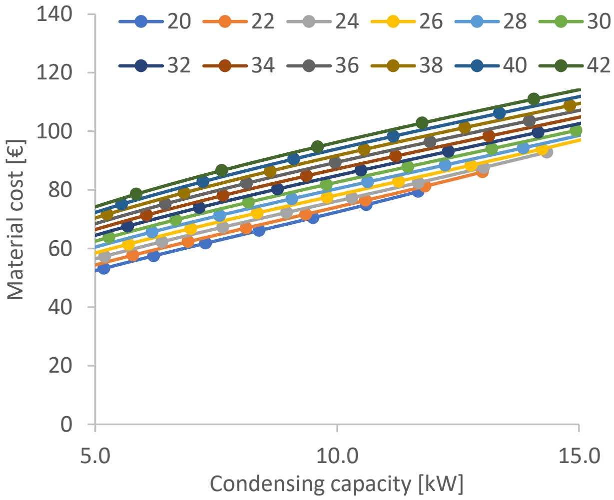

Figure 173: Sum of material costs for R1234yf

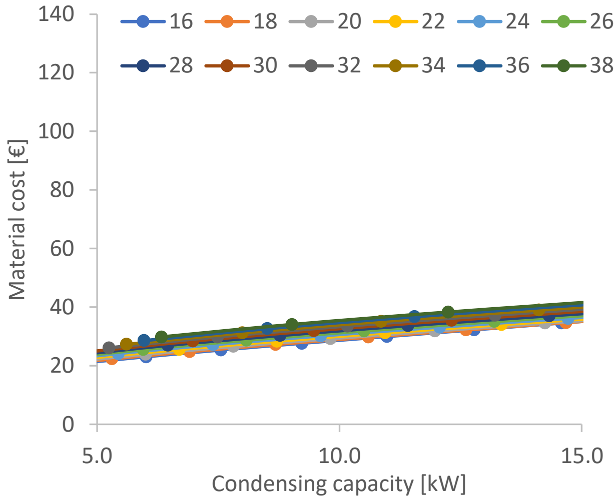

Figure 174: Sum of material costs for R152a