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A review of compressors for high temperature heat pumps

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A B S T R A C T

The development of high temperature heat pumps for waste heat recovery benefits industrial processes by meeting demand, increasing energy efficiency, and reducing emissions. The aim of such systems is to upgrade waste heat streams, typically around 50 °C to 100 °C, to higher temperatures ranging from 100 °C to around 200 °C. A primary challenge in getting the required high temperatures is the compression system. There is a plethora of recently published research papers including reviews that address high temperature heat pumps. However, there has been no comprehensive review addressing compression systems, despite their major influence in the successful development of high temperature heat pumps; this paper provides a comprehensive review of such compressors. Firstly, an overview of heat pump systems is provided, which covers cycle arrangements and working fluid selection. This is followed by a review of the different compressor technologies used, and the development of relevant modelling and design tools. Finally, suggestions for future directions in research for high temperature heat pump compressors are provided. It was found that screw compressors have been the obvious choice for heat pumps due to the experience gained from the refrigeration industry. However, the temperatures they can handle are constrained by the maximum possible limitation to avoid oil degradation. For higher temperatures, better efficiency, and larger capacity, it seems that the alternative is turbo-compressors. Nevertheless, there is a lack of experience in this area and more research and development efforts are required to enable these machines to achieve their potential in high temperature heat pumps.

Nomenclature

Symbol	Meaning
CFC	Chlorofluorocarbon
COP	Coefficient of performance
EXP	Expander
FT	Flash tank
GWP	Global warming potential
HC	Hydrocarbon
HCFC	Hydrochlorofluorocarbon
HCFO	Hydrochlorofluoroolefin
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
HP	Heat pump
HTHP	High temperature heat pump
IC	Intercooler
IHP	Industrial heat pump
IHX	Internal heat exchanger
IWH	Industrial waste heat
LCCP	Life-cycle climate performance
ODP	Ozone depletion potential
ORC	Organic Rankine cycle
PDM	Positive displacement machine
RVC	Rotary vane compressor
RPC	Reciprocating piston compressor
SC	Sub-cooler
SEP	Separator
SG	Safety group
SS	Single stage
T_c	Critical temperature
TEWI	Total equivalent warming impact
THC	Thermal heating capacity
T_{lift}	Temperature lift
T_{sink}	Sink temperature
T_{source}	Source temperature
VCC	Vapour compression cycle
WF	Working fluid
WHR	Waste heat recovery

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1. Introduction and motivation

Worldwide, governments have committed to reduce CO_2 emissions with ambitious goals of net zero by 2050 in order to combat climate change and environmental pollution, and to ensure energy security for their countries [1]. As part of the technical solutions presented for such aspirations, which include diversifying the energy mix, large-scale energy storage technologies, and energy saving, innovations for improved energy efficiency come as a necessity [2]. Industry is the highest consumer (54 %) of the total delivered energy worldwide [3], based on a report from the US Energy Information Administration of all end-use sectors globally. Even with direct electrification of industry through the substitution of combustion mechanisms, and with indirect electrification via hydrogen from electrolysis [4], industrial processes reject a substantial amount of waste heat, with an average of 10 % of the energy consumed at temperatures ranging between 50 °C and 1,000 °C [5]. While energy efficiency measures aim to reduce the amount of heat loss in this sector, based on the fundamentals of thermodynamics only a small improvement in this aspect can be achieved, and thus the next best solution would be to utilise the rejected heat in other processes [6]. According to Papapetroua et al. [7], the waste heat potential which can be recovered from industrial sectors in the EU amounts to around 300 TWh/yr which, depending on the industry, translates to around 20 % – 51 % of the total process heat [8]. Waste heat recovery (WHR) strategies have been identified to enhance the energy efficiency of industrial processes either through the conversion or the upgrade of thermal energy that would have otherwise been wasted. A summary of WHR technologies is represented in Fig. 1 where each group is suitable for specified temperature ranges [9,10]. Several mechanisms are appropriate for making use of medium to high temperature waste heat, either through upgrade to temperatures high enough for combustion pre-heating or through conversion to electrical energy. However, the same cannot be claimed for well-established technologies utilising low to medium grade heat at not quite as high temperature levels, which are required for certain industries in which economisers, and heat pumps can come to play.

State-of-the-art technologies make use of medium to high temperature heat; examples include organic Rankine cycles (ORCs) in the range of 100 °C to 400 °C [11], and supercritical CO_2 power cycles for heat recovery from temperatures between 250 °C and 600 °C [12]. Nonetheless, even with around 40 % of the rejected industrial waste heat (IWH) being at temperatures below 100 °C [13], technologies for exploiting the lower grade range have not yet been fully adopted. A limitation for the commercial feasibility of low temperature waste heat power cycles is the poor heat to power conversion efficiencies at such temperatures [14]. Therefore, the inclination has been to make use of technologies that recycle and utilise IWH for process optimisation, such as pre-heating process fluids for combustion air [15].

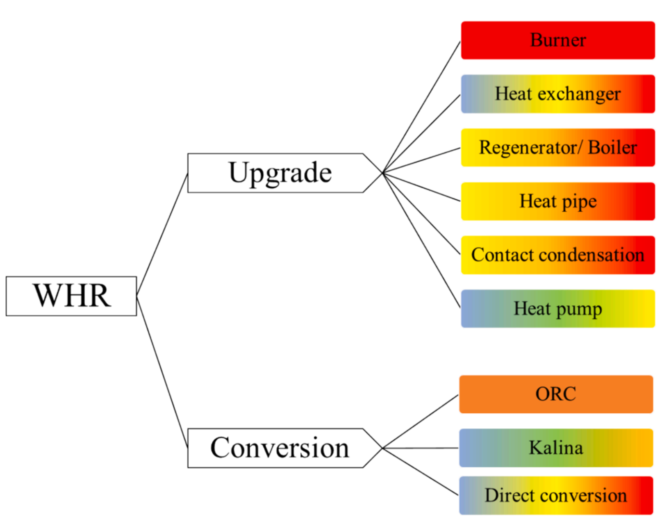

Fig. 1. Waste heat recovery technologies [9]. Colour gradient describes waste heat temperature range where blue represents low temperature and red represents high temperature. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Heat pumps (HPs) have been deemed viable environmentally friendly solutions for upgrading waste thermal energy to be utilised for electrified heating in multiple applications, and thus improving energy efficiency and reducing greenhouse gas emissions [16]. Several countries have implemented policies identifying heat pumps to be used for energy efficiency and savings. Numerous studies exist on low temperature heat pumps that are mostly used for domestic purposes [17,18,19,20], however these are of limited advantage to industrial processes requiring a purpose for their waste heat, typically bigger capacities, and a large demand for high temperature heat. In recent years, attention has been shifted towards the development of high temperature heat pumps (HTHPs) to augment IWH and reduce the use of fossil fuels in industrial heating processes to attain carbon neutrality. HTHPs, which are also referred to as industrial heat pumps (IHPs), typically elevate heat sources with temperatures around 50 °C – 100 °C, to heat sink temperatures (T_{sink}) above 100 °C [21]. Their applications can replace gas-fired boilers in supplying process heat and generating steam which are required in industrial processes.

Nellissen and Wolf [22] claim that out of the 2,000 TWh of industrial heat demand, 174 TWh can be supplied by industrial heat pumps. It is estimated that the primary energy consumption of HTHPs can be less than a third of that of electric boilers [23]. Global research interest has been sparked by the operating conditions and temperature lifts of such systems which necessitate technological advances. The high economic costs, acting as the main barrier to the deployment of HTHPs, require the development of highly efficient, reliable, and cost-effective components, and specific system-level designs that can accommodate high temperature heat sinks [24]. Arpagaus et al. [25] summarised a few notable HTHP industrial products on the market, yet in comparison to other mature WHR technologies, industrial heat pump development is still limited. Intensive research efforts are thus required to explore the fundamental challenges related to such technologies, and to investigate innovative approaches to system and component design. Enhanced research centred on developing the leading compressor component has been identified as an enabler for the uptake of HTHPs for boiler substitution [26].

The rationale and uniqueness of the current review can be summarised as follows: interest in high temperature heat pumps has been rapidly accelerating over the last few years; several review papers have been published on heat pumps since 2010, and more specifically on high temperature heat pumps since 2016; the main focus of each review paper found in literature has been summarised in Table 1. As indicated, most of the published reviews focus on the HTHP system as a whole, looking at market applications, cycle configurations, refrigerant selection, and barriers for integration. Given that the compressor is a major component required for enabling high temperature lifts, to the authors' knowledge there have not been any reviews focusing on detailed studies and advances in compressors for high temperature heat pumps. While several papers summarise and distinguish between the types of compressors used in experimental test rigs [27] or commercial products [25], they do not detail the design or operational requirements, nor their research progression and future development directions or needs. Even when two reviews (ticked in bold) have been identified to cover compressors in heat pumps, those are not necessarily covered for high temperature systems. Therefore, this work distinguishes itself from previous studies by providing a critical review of the current state-of-the art of compressors for HTHP applications, with a focus on technical and

Table 1
Focus of heat pump review papers.

Reference	Year	Industrial scale	High temperature	Refrigerant selection	Performance enhancement	Component modifications	Market	Cycles	Applications	Control
Chua et al. [31]	2010	✓			✓				✓	
Austin and Sumathy [32]	2011	✓				✓				
Mohanraj et al. [33]	2011			✓						
Sarkar [34]	2012					✓		✓		
Ma et al. [35]	2013	✓				✓		✓		
Sarbu [36]	2014	✓		✓						
Gużda and Szmolke [37]	2015					✓				
Arpagaus et al. [38]	2016				✓			✓		
Zhang et al. [39]	2016	✓							✓	
Bamigbetan et al. [40]	2017	✓	✓	✓						
Fischer and Madani [41]	2017								✓	✓
Huang et al. [42]	2017	✓								
Schiffmann et al. [43]	2017					✓				
Arpagaus et al. [25]	2018	✓	✓				✓		✓	
Cao et al. [44]	2019					✓		✓		✓
Goyal et al. [45]	2019	✓								✓
Lecompte et al. [46]	2019	✓						✓		
Schlosser et al. [47]	2019	✓			✓				✓	
Menon et al. [48]	2020	✓							✓	
Schlosser et al. [49]	2020	✓					✓		✓	
Gaur et al. [19]	2021						✓			
Jesper et al. [50]	2021	✓			✓		✓			
Wu et al. [51]	2021	✓	✓	✓						
Adamson et al. [52]	2022	✓	✓		✓			✓		
Barco-Burgos et al. [53]	2022	✓	✓							
Jiang et al. [27]	2022	✓	✓				✓	✓		
Klute et al. [54]	2024	✓	✓		✓			✓		

operational issues, and on the technological advancements that could potentially overcome the challenges, while taking into account the aforementioned system considerations of cycle configurations and working fluid selection. Whereas several products on the market claim commercial or demonstration availability for HTHP applications [28], many suppliers have provided information which is not backed up by research literature. Therefore, the emphasis will be on the research status of compression machines that accommodate temperatures above 100 °C and high temperature lifts for industrial applications. For commercial heat pumps, according to Hassan et al. [29], the current temperature limit is around 120 °C – 130 °C. Nevertheless, the authors point to the existence of heat pumps that allow higher temperatures of up to 165 °C. Promising research in this area was presented by researchers from the Norwegian SINTEF Energy Research [30]. They have developed a water-based heat pump that reaches temperatures of up to 180 °C.

To put the topic of HTHP compressors in context, this paper begins by providing an overview of high temperature heat pump systems including a brief explanation of suitable refrigerants and feasible cycle modifications, as more detailed studies are found in literature. It is then followed by an in-depth review of the various compressor types (scroll, piston, screw, and turbo) which have been investigated by researchers designing HTHPs. Given that high-temperature compressors are not exclusive to heat pump systems, an insight is provided on some expertise that could be shared from other compressors used for high temperature applications. Finally, the paper concludes with a summary of the challenges facing the development of compressors for HTHPs and provides an outlook for future trends.

2. Heat pump systems

Industrial heat pumps are broadly classified in three major categories: vapour compression systems, absorption systems, and chemical systems. The absorption cycles encompass two types: (1) absorption heat pumps where high temperature heat is supplied in the generator to upgrade low grade waste heat to a medium temperature level which has a higher power than both inputs, using two cycles with different working media (usually LiBr/ H_2O) [39,55]. Absorption heat pumps are reported to have a coefficient of performance (COP) in the range of 1.3–1.9 [56]. (2) Absorption heat transformers where medium temperature heat is upgraded to a higher temperature from the absorber and partially discharged at low temperature. Such systems have a thermal COP of around 0.5 [39]; COPs less than 1 are expected for heat pumps that do not rely on electricity consumption [57]. Chemical heat pumps make use of the reversible reactions of chemical substances to absorb low temperature heat through an endothermic reaction and upgrade stored thermal energy to be released via an exothermic reaction [58]; their typical COPs are also below 1 [59]. Hybrid systems of compression-absorption or solar assisted configurations are also suitable for industrial heat recovery applications. However, most industrial high temperature heat pumps fall under the vapour compression systems due to better performance, and thus absorption and chemical heat pumps, which do not employ compressors, are out of the scope of this paper [31].

Typically, and in simple terms, HTHPs follow the operation of closed vapour compression cycle (VCC) heat pumps, schematically illustrated as a standard cycle in Fig. 2. The VCC works on a thermodynamic heating cycle using a refrigerant as a working fluid (WF) circulating around an evaporator — where heat is absorbed ($\dot{Q}_L$) — a compressor — which raises the pressure and the temperature of the WF — a condenser — where heat is released — and an expansion valve [60].

Industrial heat pumps have several applications such as space heating, hot water, drying, steam generation, and providing process heat requirements. Techniques to improve the energy efficiency of heat pumps involve careful selection of refrigerants, improved cycle configurations and architectures, and enhanced compressor designs, all of which are determined by the heat source temperature (T_{source}) and the desired temperature lift (T_{lift}) [31]. The coefficient of performance is a thermodynamic performance measure of the HP and is defined as the ratio of heat transferred to the sink ($\dot{Q}_H$) to the input compression work ($\dot{W}$) as per Eq. (1). The volumetric heating capacity (VHC) in Eq. (2), being the ratio of the heat rejected by the compressor to the volumetric flow rate ($\dot{V}$), is another parameter used to indicate the relationship of the heat delivered to compressor size; a higher VHC implies a smaller compressor size. Greater input power and heat loss is associated with larger compressors. VHCs of typical heat pumps range between 500 kJ/m^3 – 1000 kJ/m^3 on the lower end, to up to 5000 kJ/m^3 – 6000 kJ/m^3 [61]. In this section, the possible improvements through working fluid choice, and component modifications for updated cycle configurations are discussed in light of their influence on compression systems.

$$COP = \frac{\dot{Q}_H}{\dot{W}} \tag{1}$$

$$VHC = \frac{\dot{Q}_H}{\dot{V}} \tag{2}$$

2.1. Refrigerant selection

The choice of working fluid, which directly impacts compressor design, must compromise between maximum system performance and minimum volumetric flow rate [62]. There is an extensive collection of working fluids where more than fifty compounds have been utilised over the decades in refrigeration cycles [63,64]. The selection of a suitable substance is determined by cycle conditions, as well as requirements which include cycle thermal efficiency, thermal stability, chemical compatibility, environmental compliance, safety, availability, and price [25]. In fact, when choosing a WF, a trade-off is exercised between the environmental suitability of the WF, and the consequences of a potentially lower COP which in turn affects the overall system emissions. The evolution of synthetic refrigerants has enabled the creation of chemically stable fluids that abide by the environmental considerations relating to ozone destruction and global warming potential (GWP). Second generation substances of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) have a relatively high ozone depletion potential (ODP) and are prohibited or are being phased out respectively, and thus are not discussed in this review, whereas hydrofluorocarbons (HFCs) have a high GWP. A few studies on HFC refrigerants for heat pumps employed for temperatures higher than domestic needs have been conducted primarily using R245fa, R245ca, or refrigerant mixtures [65,66,67]. An issue associated with HFCs is their immiscibility with mineral oils and the provision of synthetic lubricants for compressors that might demand regular servicing due to their moisture-absorbing nature [33].

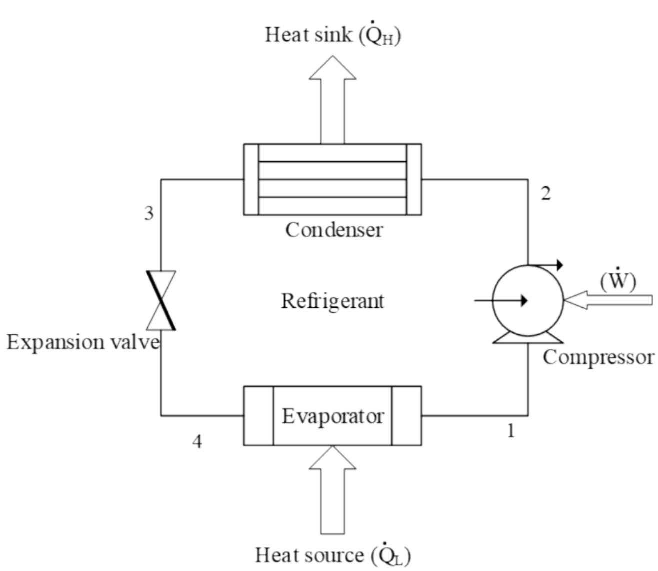

Fig. 2. Vapor compression heat pump cycle.

Characteristically, refrigerants having a zero ODP, and a GWP lower than 10 are recommended [33]. More recently, measures of the environmental impact associated with the whole life cycle of refrigerants, such as the total equivalent warming impact (TEWI) and the life-cycle climate performance (LCCP) indices, have also been used to assess the ecological suitability of working fluids [36]. Lower discharge temperatures are sought to extend the compressor life, which is correlated with refrigerants having a low specific heat ratio. Refrigerants with low boiling points tend to have smaller sized compressors due to low volumetric flow rates [40]. WFs with low saturation pressure at the heat source temperature are generally avoided to evade unreasonably low evaporation pressures. The maximum HP cycle temperature, for subcritical systems, is limited by the critical temperature (T_c) of the working fluid, while the pressure level dictates the material properties of the HP system [25]. The intrinsic critical property of the fluid can be addressed by substituting WFs with other more suitable refrigerants. Systems operating with fluids that have lower critical temperatures run at higher pressures and thus with more compact compressors, but with decreased COP and more demanding constraints on the compressor material. Fluids with high critical temperatures not only allow the cycle to operate at higher temperatures (for subcritical operation), but also seem to have better COP. Temperature-entropy diagrams of refrigerants with a high overhang might lead to the formation of liquid droplets within the compressor, increasing the risk of erosion; thus, care should be taken when selecting such refrigerants which should be balanced by additional cycle components to ensure sufficient superheat.

Hydrofluoroolefins (HFOs) have suitable environmental compatibility criteria with T_c's having a threshold of about 170 °C which needs to be taken into account for subcritical HTHPs that require higher temperature sinks [68]. The same can be inferred for hydrochlorofluoroolefins (HCFOs) which exhibit sufficient critical conditions with good safety standards and are almost ozone benign [68]. However, some noteworthy components under this refrigerant type are still not widely commercial yet, for example R1234ze(Z). Hydrocarbons (HCs) are cheap environmental alternatives, whereby heavy HCs have high critical temperatures and good miscibility with mineral oil and synthetic lubricants, but their high flammability, placing them in the A3 safety group (SG), deters their use from large systems (limited to 500 g) for safety reasons [69]. Besides that, hydrocarbon compressors for high temperature applications are not widely commercialised.

Accessible, ecological, and cheap natural refrigerants, as exploited in the earliest refrigeration systems, seem to be ideal candidates for use in HTHPs. Whereas carbon dioxide (R744) is mostly used in transcritical cycles due to its low critical temperature and high volumetric capacity, water (R718) is employed for very high temperature applications because of its appealing T_c and its favourable efficiencies at very high temperatures [40]. However, the low vapour density of R718 poses high compression ratios requiring large or high-speed compressor components; nonetheless, the choice of steam turbo-compressors has been suggested to alleviate such issues [70]. R718 is also associated with large heat exchangers to accommodate the phase change process due to its high specific volume at elevated temperatures. Additionally, the corrosive properties of water inhibit cheaper carbon steel equipment and would oblige the use of stainless steel instead. The compressor discharge temperatures with R718 are also high and thus water injection is used to regulate it. Water is not only used as a refrigerant in closed VCCs, but also commonly used in open mechanical vapour heat pump cycles for industrial applications. Ammonia (R717) attracts interest because of its large volumetric heating capacity which is particularly useful for HTHPs with a high demand [51]. Yet, several challenges are associated with NH_3 including its hazardous properties, material limitations (inappropriateness of less expensive copper and brass), and high compressor discharge temperatures and pressures [71]. Advances in compressor design and technology have permitted pressures reaching 60 bar – 76 bar for ammonia at elevated condensation temperatures reaching 97.5 °C [26,40]. Centrifugal compressors are not applicable for systems with R717 due to its low molecular mass [36].

An emerging field of study is focused on the development of new low-GWP refrigerants, and the creation of targeted refrigerant mixtures which make use of particular thermodynamic properties to adjust certain characteristics for a widened and flexible offering of choice, while maintaining environmental neutrality. Zeotropic (or non-azeotropic) mixtures — ones which do not behave as a single refrigerant when undergoing a phase change, instead experience non-isothermal characteristics through a temperature glide of 4 °C – 7 °C [33] — are the preferred mixture type for efficiency enhancement and capacity modulation, but at the risk of volatility leakages which change the mixture blend and impact the system performance [72]. For heat pump cycles that have a substantial temperature glide, the Lorenz cycle, which is a modified vapour compression cycle using a zeotropic mixture as the refrigerant, experiences better performance in comparison to the Carnot cycle [52]. Nevertheless, a specific selection of a suitable mixture, and the optimisation of the temperature glide matching with the heat source and sink temperature profiles are necessary to reap the benefits of such systems and decrease exergy destruction [72]. Unconventional bespoke mixtures are being customised to suit HTHP applications for improved COP and exergy/ second-law efficiency [73]. Recently, Abedini et al. [74] performed an optimisation analysis of different binary mixtures with the objective of maximising the COP of a HTHP. It seems that mixtures can provide higher COPs than pure fluids because of lower pressure ratios and compressor discharge temperatures; however, the optimal mixtures are those with HCs which bring forth the issues of flammability.

Zamfirescu and Dincer [75] suggested the use of a category of working fluids known as Bethe-Zel'dovich-Thompson (BZT) fluids, utilized in ORCs, which exhibit non-classical gas dynamic characteristics. Apparently, the classes of siloxanes and perfluorocarbons have advantageous qualities for HTHPs including non-toxicity, lubrication, thermochemical stability, low flammability, and suitable thermo-physical properties. A detailed showcase of the properties of seventeen BZT fluids can be found in their publication, where the refrigerant molecular mass, and critical pressures and temperatures range between 297 g/mol – 774 g/mol, 7.2 bar – 17.9 bar, and 291.9 °C – 428.1 °C, respectively, as obtained from specific equations of state. This recommendation is made based on a HP operating with a turbo-compressor component so as to minimise the risk of shock occurrences in the rotor, even when marginally supersonic flows are present for high pressure ratios. However, currently there seems to be no further exploration of such working fluids for HTHP applications.

Several researchers conducted work on evaluating the suitability of refrigerants for HTHPs, particularly for replacing older generations of working fluids by low GWP ones like HFOs, HCFOs, or with natural/ HC refrigerants. Bergamini et al. [26] suggest that for subcritical cycles using natural refrigerants, R717 performs well for lower heat sink and source temperatures (60 °C and 110 °C respectively), whereas R718 yields better performance metrics at increased temperatures; both refrigerants were able to achieve maximum COPs around 2 in single and double stage systems. Wu et al. [76] compared the performance of a selection of natural, HC, HFO, and HFC working fluids in an idealised simple heat pump system. Their results concur with the earlier reference that R718 has outstanding performance particularly at high condensation temperatures (i.e., high T_{sink}), but at the cost of increased pressure ratios, higher volumetric flow rates, and extreme compressor superheat and discharge temperatures which require addressing [76]. In a similar study, Frate et al. [77] assessed the COP and volumetric heating capacity of a simple sub-critical HTHP cycle arrangement up to 150 °C with conservative assumptions, using an array of twenty-seven working fluids with T_c's > 125 °C, autoignition temperatures < 250 °C, and appropriate chemical stability. R718, ethanol, and methanol were discarded for simple cycle architectures after the results indicated a large number of compression stages (up to 7) required for high temperature lifts because of the large enthalpy change. The HCFO R1233zd(E) provided the optimal trade-off between COP (around 2.2 for T_{lift} = 100 °C), an efficiency performance measure, and VHC, an economic indicator [77]. Two HFOs and two HCFOs were numerically investigated by Arpagaus et al. [68] to substitute two HFCs (R365mfc and R245fa) in an idealised HP cycle with T_{sink} reaching 150 °C. Based on their results, an experimental test rig using commercially available R1233zd(E) was deployed for further analysis. Similar work was conducted by Mikielewicz and Wajs [78] whereby the best COP (2.9 – 3.6) for a single stage cycle was achieved through ethanol, R601, and R1234ze(Z), in descending order, when condensation temperatures reached 130 °C. A cascade cycle with R601 and R1234ze(Z) in the upper and lower cycles respectively achieved a COP of 3.2 with a second law efficiency equal to 63.7 %. A simple VCC was employed by Yan et al. [79] in preliminary modelling of a variety of working fluids and their mixture combinations using only the critical condition properties of refrigerants; this tool could be useful for exploring new refrigerants with limited open-source data. Koundinya and Seshadri [80] carried out a multi-criteria energy, exergy, environmental, and economic evaluation of a single-stage standard VCC using fourteen refrigerants. The results obtained for COP, heating capacity, exergy efficiency, TEWI, and plant cost were assessed for rising condensation temperatures; in the highest range of 93 °C – 95 °C, the most suitable working fluid was found to be R1234ze(E), albeit not reaching temperatures over 100 °C as considered in this paper as HTHP. Also taking economic factors of low GWP refrigerants into account, Kosmadakis et al. [81] examined single and two-stage cycles for heat sink temperatures up to 150 °C. R1234ze(Z) seems to have the best performance (COP ≈ 3.0) and lowest specific equipment cost (≈ 150 €/kW – 300 €/kW) for high temperature lifts in both cycle configurations, while R1336mzz(Z) experiences the lowest COP and heating capacity.

When taking into account all factors of environmental considerations, safety, oil compatibility, leak detection, and properties affecting performance, it is usually difficult to identify one WF, from the most common refrigerants found in Table 2, that satisfies all requirements for a specific application. Therefore, design compromises need to be made on a case-by-case basis, considering the system as a whole.

2.2. Cycle configurations

Industrial heat pump systems can take on various configurations and additional components to enhance performance depending on the application and temperature requirements. An important consideration in heat pump design is cycle architecture which influences the design and use of the compressor components [61].

Simple, standard heat pump cycles with single-stage (SS) compression, such as the cycle shown in Fig. 2, can achieve a temperature lift between 30 °C and 80 °C where experimental values of COPs range between 1.7 and 3.1 for T_{sink} > 100 °C [84], with higher COPs shown to be achievable in some simulation exercises [81]. Such systems are limited by the pressure ratio of a single compression stage. For low

Table 2
Properties of refrigerants mentioned in this paper [82].

Type	Refrigerant	T_c [°C]	P_c [bar]	ODP [-]	GWP [-]	SG	BP [°C]	M [g/mol]
HFC	R134a	101.1	40.6	0	1300	A1	−26.1	102.0
	R245fa	154.0	36.5	0	858	B1	14.9	134
	R152a	113.3	45.2	0	138	A2	−24.0	66.1
HFO	R1234ze(E)	109.4	36.4	0	<1	A2L	−19.0	114.0
	R1234ze(Z)	150.1	35.3	0	1	A2	9.8	114.0
	R1336mzz(Z)	171.3	29.0	0	2	A1	33.4	164.1
HCFO	R1233zd(E)	166.5	36.2	0.00034	1	A1	18	130.5
HC	R600a Isobutane	134.7	36.3	0	3	A3	−11.8	58.1
	R600 Butane	152.0	38.0	0	20	A3	−0.5	58.1
	R601 Pentane	196.6	33.7	0	0	A3	36.1	72.2
	R290 Propane	96.7	42.5	0	3	A3	−42.1	44.1
Natural	R717 Ammonia	132.3	113.3	0	0	B2L	–33.3	17.0
	R718 Water	373.9	220.6	0	0	A1	100	18.0
	R744 Carbon dioxide	31.0	73.8	0	1	A1	−78.5	44.0
Mixture	BY-5 [83]	155	43.7	0	800	–	15.6	134.0

temperature lifts of around 20 °C, a simple HP cycle with an external sub-cooler (SC) can help improve COP by reducing flash gas throttle and reducing the total refrigerant flow rate [85]. Liquid-gas separators (SEPs) or injectors are also used to enhance the operation of standard cycles through the avoidance of liquid entering the compressor from incomplete evaporation, controlling the compressor superheat (by up to 10 K), as well as cooling the compressors [27,52]. Flash tank (FT) separators in particular are used in HTHPs with water as a refrigerant, and for steam production, where liquid water in the FT absorbs some superheat, before compression, and is sent back for evaporation [86,87]. In such cases, COPs, from experimental studies, have been shown to reach values of 6.1 for a 30 °C lift from an evaporation temperature of 85 °C [87], while simulation results produced COPs as high as 11.0 [88]. Some researchers however have shown that standard cycles are able to achieve higher COPs by using advanced working fluids. For example, Zhang et al. [83] investigated the performance of a standard heat pump cycle using a non-conventional binary near-azeotropic working fluid (BY-5); the system achieves COPs higher than 3.0 for evaporation and condensation temperatures of 70 – 80 °C and 110 – 130 °C respectively.

A standalone standard cycle underachieves compared to more sophisticated arrangements for very high condensation temperatures and large temperature lifts. Multistage, and cascaded cycles outperform simple configurations, with average COPs between 2.1 and 5 [89,90,91], particularly for high temperature lifts, but at the expense of additional complexity which translates to extra costs [92]. Multistage cycles, employing multiple compression stages, have a lower pressure ratio and higher compression efficiency per stage in comparison to single-stage compression [93]. The pressure ratio across each compression stage can be different but is usually aimed at being maintained identical in order to obtain a better COP [94]. Interstage cooling, or intercooling, is also used to enhance compression performance by cooling the WF for reduced compression work [95]. Adding an economiser is another technique of improving performance. In a two-stage cycle with an economiser, the vapour fraction of the refrigerant from the economiser is compressed by the second-stage compressor, meaning it does not pass through the evaporator and the first-stage compressor which lowers the total compression work of the system [39]. Cascade heat pumps operate as separate cycles that are coupled through a heat exchanger; they too have lower pressure ratios and work per compressor. Unlike multistage cycles, cascade systems can make use of different refrigerants for each cycle, depending on operating conditions, to maximise performance [96], with COPs in excess of 2.0 [52]. Yang et al. [97] studied a HTHP system composed of a combination of a transcritical R744 bottoming cycle and a subcritical R152a topping cycle for a maximum R152a condensation temperature of 102 °C. By comparing their results to a standard heat pump with the same operating conditions, they demonstrated that a 55 % larger COP was achieved via the combined system, and that the exergetic efficiency of their proposed HTHP is more than double that of the standard cycle. They also found that the power consumption of the compound system is around 8 % less than that of a simple heat pump cycle [97]. A transcritical cycle also has an advantage of rejecting heat in a gas cooler at pressures higher than the refrigerant critical pressure rather than in a condenser where the temperature glide is lower [32].

Bergamini et al. [26] compared the performance of different configurations, single-stage, double-stage with intercooler (IC), and cascade cycle HTHPs with four WFs. Their results show that in comparison to the SS cycle, the double-stage cycle has improved COP with R717 but no benefit for other WFs. The double-stage cycle exhibits higher exergetic efficiencies (up to 70 % for heat sink temperatures of 180 °C) for all four studied refrigerants, whereas the two-fluid cascade cycle configuration widens the operating range with minimal effect on COP [26].

Internal heat exchangers (IHXs) are auxiliary components used in either single-stage or multistage/ cascade cycles to transfer heat from the refrigerant after the condensation (in the case of a subcritical cycle) or in gas cooling (for transcritical cycles) to the refrigerant before evaporation which yields higher compressor suction and discharge temperatures accordingly and can also be used to superheat the refrigerant before compression [74]. The possible advantage of adjusting a simple cycle with the addition of a regenerative heat exchanger was suggested by Frate et al. [77] for enabling the use of the working fluids that were abandoned as discussed in Section 2.1. Mota-Babiloni et al. [98] presented a study on the optimisation of a cascade system using IHX to improve the performance of a HTHP for temperature lifts of 85 °C – 115 °C with several refrigerant arrangements in each cycle where a maximum COP of ≈ 3.2 was achieved; the IHX design and performance parameters were varied. A comparable COP (≈ 3.7) was obtained experimentally for condensation and evaporation temperatures of 140 °C and 90 °C respectively by Reißner [61] using a simple HTHP cycle with an IHX and a novel working fluid (LG6). A shell and tube IHX as a recuperator was embedded in the transcritical R744 HTHP by White et al. [99] to extend CO_2 cooling beyond the gas cooler and to sufficiently superheat the refrigerant before compression. However, their results led to the conclusion that the removal of the IHX, and the substitution with a larger gas cooler might assist in increasing the volumetric heat capacity by 20 %.

Following the trend of larger temperature lifts, irreversibility and losses linked to the throttling expansion valve become higher and of more significant impact on the HTHP performance; therefore, it is possible to replace the valve with components that could recoup some of the expansion work. Two-phase ejectors are one example of a cost-effective and straightforward solution for work recovery, loss mitigation, compressor work reduction through increased suction pressure and temperature, and evaporator enhancement from liquid feeding and flash gas bypass [34,100]. Bai et al. [101] showed that an ejector-enhanced dual-pressure HTHP cycle outperforms both standard and parallel-compression cycles in terms of lower exergy destruction (15.8 % and 8.9 % respectively), higher COP (29.5 % and 12.6 % respectively), and greater VHC for temperature lifts between 45 °C and 75 °C. Ejection systems have been commonly used in transcritical HP cycles, particularly with R744 which has a large temperature glide, to recover energy at the gas cooler outlet, but mostly for not high enough temperatures [51,102,103]. Dai et al. [104] recently applied the ejector model to varying configurations of a transcritical CO_2 system for hot water generation at a gas cooling temperature in the 120 °C range, obtaining promising performance results. On the other hand, expanders (EXPs), either positive-displacement or turbo-expanders [105], similar to those employed in ORCs [46], have slowly been making their way within HPs to recover expansion work. However, this is not conventional because of the two-phase non-ideal expansion that occurs within the refrigerant's saturation dome after condensation; possible advances in research in this field can be shared from ORC based learnings [105]. Back in 1997, Tamura et al. [106] studied the use of a screw-expander for power recovery in an ammonia/ water cascade HTHP but without further follow up investigations.

An innovative configuration was proposed by Mateu-Royo et al. [107] which consists of a single reversible system that can operate in either HTHP mode or ORC mode to maximise waste heat recovery potential for both heat and electricity generation. Their cycle runs on a simple VCC with a scroll compressor and an IHX when in HTHP setting; when operating as an ORC, the IHX acts as a regenerator, the expansion valve is bypassed and a pump is replaced, and the scroll compressor acts as an expander. From an evaporation temperature of 85 °C, the HTHP achieved a COP of 2.2 for a condensation temperature of 140 °C, and the ORC obtained a net electrical efficiency of 8.75 % for its condensation temperature of 40 °C [107]. The most thorough research work on different cycle configurations and supplementary components for HTHP performance enhancement presented in a single publication is one conducted by the repeatedly cited active research group ISTENER in Spain [65] who also accounted for the effect of several refrigerants. As anticipated, a greater temperature lift entails a more complex two-stage cascade cycle arrangement, whereas a higher temperature heat source could do with a single-stage cycle plus some additional components such as an economiser and parallel compression. This study is a good summation of the intricacies associated with selecting a unique system; the cycle optimisation is a process that depends on the available technologies and the application/ resources. It constitutes a trade-off and prioritisation between performance, emissions, and cost, all of which are determined and designed on a case-by-case basis to ensure satisfactory delivery of all requirements, instead of a one solution fits all approach.

3. Heat pump compressors

Having covered the working fluid and cycle aspects related to high temperature heat pumps, attention is turned to the focal point of this review paper which is the compressor. When designing a heat pump system, an integrated approach incorporating all elements, including compressor design, cycle optimisation, and refrigerant selection is necessary to constrain the solution space to feasible regions where compressor designs are applicable, and to reach a trade-off between component efficiency and overall system performance.

HTHPs demanding high temperature lifts subsequently necessitate high compressor pressure ratios which complicate the component design. The discharge pressure limit of a compressor also limits the heat sink temperatures, while the suction side pressure limit poses heat source temperature restrictions and compression ratio constraints. Compressor selection and design are co-dependent on the selected WF, cycle configuration, operating parameters, and temperature rise. Other technical considerations such as materials, lubricants, size/ heating capacity, and ancillary components all influence the selection of the compressor type and its design. The compressors discussed in this paper are generally categorised as per the classification chart in Fig. 3.

The schematic representation in Fig. 4 identifies the scope where the available compressor options are expected to be most suitable for heat pumps according to heating load requirements [37]. The literature shows that even when the two most common compressor types used in HTHPs are reciprocating piston and scroll compressors [109], ultimately the choice falls between screw and turbo-compressors for the higher end of heating capacity scales.

According to the survey by Arpagaus et al. [25], there are no IHPs employing rotary vane compressors (RVCs). RVCs are widely used in small capacity domestic refrigeration applications due to their simplicity, compactness, and reliability [110]. Their performance is limited by sealing issues, and clearance gaps which imply restricted pressure ratios and efficiency losses [111]. Given that the pertinence of RVCs is for small scale systems, their discussion is omitted from this paper because of their unsuitability for IHPs that usually require higher thermal heating capacity (THC) ranges.

3.1. *Positive displacement compressors*

Positive displacement compressors (PDCs) increase the pressure of a fluid by containing the gas within the compressor, reducing its volume, and then discharging it via an exit valve; they offer flexibility in delivered pressure range and have high efficiencies [112]. Reciprocating piston, scroll, and screw compressors are a classification of PDCs.

Positive displacement compressors have been the primary choice for high temperature heat pumps using HFCs, HFOs, HCs, and NH_3 (ammonia/ R717), however the technological barrier of the operating envelope of the compressors limit the attainable temperatures and pressures. Oil lubricated PDCs are chosen to complement the selected WF properties and compressor manufacturers guarantee stable operation for specific lubricant/ refrigerant combinations. The compatibility of the lubricant is based on its viscosity and thermal stability which is particularly important at high temperatures where coking is an issue. There has been extensive research for improving the performance of polyelester oils at higher temperatures than the originally suggested boundary of 180 °C [25 113]. Table 3 summarises indications of the generally available reciprocating piston compressor technology limits for selected natural refrigerants; compressors for HFC and HC WFs have lower upper limits but wider operation capacities in comparison to ammonia and carbon dioxide compressors [82]. This WF dependency is based on the operating conditions associated with R744 and R717 particularly in relation to high discharge temperatures. It is worth noting that the cited source [82] is from 2015 and since then, some progress is expected to have been achieved on that front.

3.1.1. *Scroll compressors*

Scroll compressors are typically used in heating, ventilation, and air conditioning units. The working fluid enters in the periphery of a moving scroll which orbits around a stationary scroll and moves the fluid to the central region where it is discharged [114]; a schematic representation of the geometry of the scroll set is shown in Fig. 5 [115]. Vapour injection scroll compressors are regularly used in two-stage refrigeration cycles which reduce compression work per stage, allow an increased operating envelope, and warrant lower compressor

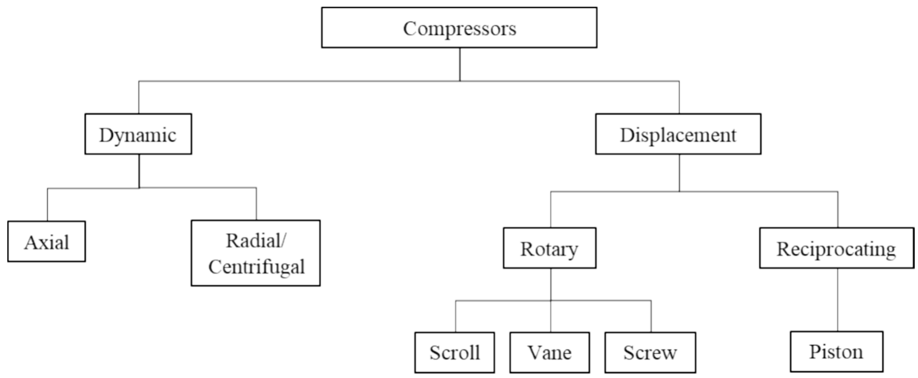

Fig. 3. General compressor classification considered in this paper [108].

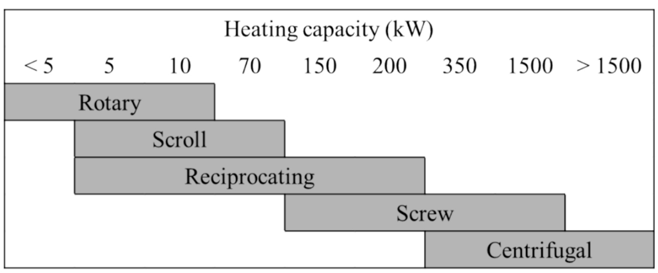

Fig. 4. Compressor options for heat pumps [37].

Table 3
Operating limits for available piston compressors [82].

WF	Pressure limit [bar]	Flow rate [m^3/h]
HFC/ HC	28	5 – 280
Low pressure R717	28	5 – 180
High pressure R717	50 76 [26]	90 – 200
Transcritical R744	140	6 – 25

discharge temperatures [116]. In comparison to reciprocating compressors, scroll compressors have a higher volumetric efficiency at single speeds [117]. They also offer the benefit of being oil free, minimising the risk of oil degradation at high temperatures for HTHPs [118].

Table 4 gives an overview of the studied research works using scroll compressors for HTHP applications. For all findings discussed hereafter, when both simulation and experimental results are present in one study, experimental results are reported in the overview table. Bobelin et al. [85] used two parallel hermetic scroll compressors with a maximum allowable suction (evaporation) temperature of 60 °C to ensure sufficient cooling of the motor. They concluded that for temperature lifts over 60 °C, the COP of their HTHP system decreases due to the high pressure ratios at which scroll compressors exhibit poor efficiencies. Scroll compressors have been investigated by researchers in Tianjin University who proposed new binary near azeotropic mixtures (BY-4 and BY-5) [83,84]. Two 23.6 kW scroll compressors working in parallel, with a maximum allowable discharge temperature of 130 °C, were employed by Xiaohui et al. [84]. Their tests achieved a highest discharge pressure of 17.3 bar with a pressure ratio of 5.03 for a maximum discharge temperature of about 112 °C. A hermetic scroll compressor with ≈ 5.9 kW power and a theoretical exhaust of ≈ 0.37 m^3/h was designed by Zhang et al. [83] for the purpose of a small-scale HTHP compressor with high efficiency and reliability, simplicity, and low noise levels. During all operation tests, the pressure ratio reached no more than around half (4.5) of the constrained ratio of 8, for the highest condensation pressure of 27.1 bar with the maximum discharge temperature reaching around 132 °C. Huan et al. [119] tested a simple HTHP cycle using a constant speed scroll compressor with R245fa under condensation temperature conditions between 70 °C and 120 °C to heat up pressurised water. The qualitative trends of the experimental results agreed with their theoretical calculations. High compressor suction superheating was required for high temperature conditions. A 5.18 experimental value of COP was obtained for an evaporation temperature of 71 °C which enabled outlet water temperatures of 114 °C [119]. Koundinya et al. [80] validated their thermodynamic model of a HTHP system with a scroll compressor. While their work was mostly in relation

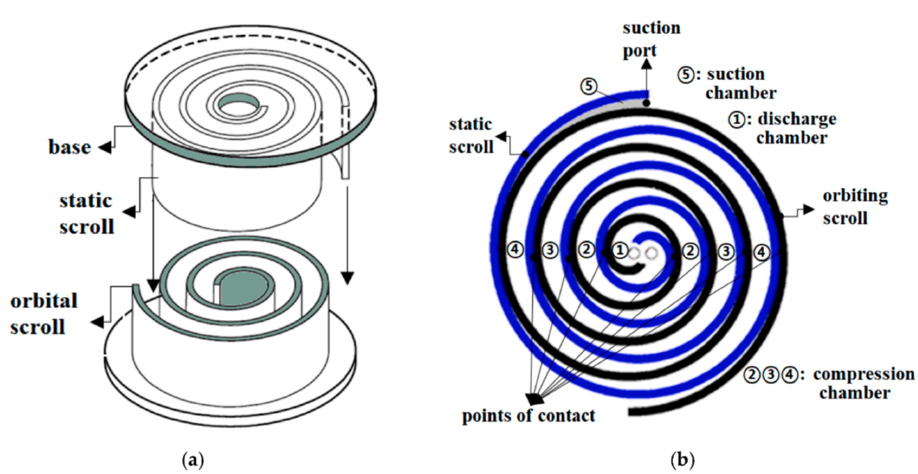

Fig. 5. Schematic figure of scroll compressor [115].

Table 4
Overview of HTHPs with scroll compressors.

Ref.	Year	Cycle	Refrigerant	T_{source}/ T_{evap} [°C]	T_{sink}/ T_{cond} [°C]	THC [kW]	COP [-]
[85]	2012	Standard	ECO3	−/35–60	−80–138	50–200	1.0–5.0
[84]	2014	Standard	BY-4	−/50–70	−/80–110	44–141	1.3–5.8
[83]	2017	Standard with separator	BY-5	−/70–80	−/110–130	15.5–19.5	2.5–3.1
[119]	2017	Standard	R245fa	−/30–90	−/70–120	6–12	4.1–7.2
[80]	2022	Standard	various	−/10	−/45–95	2–28	2.2–6.0
[120]	2019	Standard with IHX	R245fa	60–80/-	90–140/-	10.9–17.5	2.2–3.4
[121]	2019	Standard with separator	NBY-1	60–80/-	−/85–135	14–17.5	2.4–3.9
[122]	2021	Standard with separator	R245fa binary mixtures	45–65/-	80–100/-	≈12	4.0–5.7

− is used to indicate values that are not reported.
/ is used to indicate or (example, either source temperature or evporation temperature).

to the selection of the most suitable refrigerant based on a techno-enviro-economic analysis, some information regarding the compression can be withdrawn. The assumed isentropic efficiency of the compressor with a displacement of 17.1 m^3/h was 70 % and the cut-off pressure was placed at 30 bar. R290 was suggested to prolong compressor life alongside oil durability in comparison to other refrigerants. Most of the studied refrigerants had higher compression power requirements due to their higher pressures in comparison to R134a [80]. Mateu-Royo et al. [120] based their tests on an open scroll compressor with a 7.5 kW nominal power and a displacement of 121.1 cm^3/rev (2900 rpm); they internally modified the compressor to ensure its capability in running at high temperatures with proper lubrication. The compressor power consumption increased from around 3.2 kW to around 7.9 kW when the heat sink temperature varied between 90 °C and 140 °C (with little effect of temperature lift) due to an increase in mass flow rate. Lower evaporation temperatures reduce volumetric efficiency and suction density which directly influence the mass flow rate [120]. The critical refrigerant discharge temperature reached up to 160 °C (for T_{sink} = 140 °C and T_{lift} = 60 °C) which is near the maximum specifications for WF thermal stability. As expected, the compressor component had the highest fraction of the total exergy destruction with the most irreversibility experienced in it. Even though the team tested R245fa, they conducted a simulation study which showed that the low-GWP refrigerant, such as HFO R1336mzz(Z), would require a larger compressor to accommodate lower suction densities and offer comparable heating capacities [120]. A scroll compressor, using a novel refrigerant (NBY-1) with suitable HTHP properties, was experimentally tested by Deng et al. [121] in a standard VCC with a gas–liquid separator. Maximum condensation pressures in the region of 2.5 MPa were attained for a condensation temperature of 135 °C, where the compressor input power was close to 6.2 kW; with lower pressures and compression works achieved at lower sink temperatures. A 4.04 kW rated power scroll compressor with a displacement of 8.24 m^3/h and a nominal capacity of 12.11 kW was tested in a standard HTHP at Tianjin University [122] using binary mixtures of R245fa. Compressor discharge temperatures reaching around 112 °C, from suction temperatures of around 60 °C, were obtained for mixtures containing 70 % R245fa. Consequently, mixtures with high levels of R245fa produced the highest discharge pressures in the range of 1.7 MPa up to 2.2 MPa from suction pressures of 0.6 MPa — 0.8 MPa. High discharge temperatures are reported to "carbonise the lubrication oil and damage the insulation layer of the compressor", whereas high discharge pressures might harm condenser pressure limitations and increase compressor input power [122].

This type of compressor is the least common in the literature for high-temperature applications. Li et al. [123] developed a thermodynamic model for a high temperature water source cascade heat pump with a scroll compressor and semi-hermetic piston compressor. The proposed model assumes a steady state and steady flow process for all system components including the compressors. In addition, zero pressure drop and heat loss were assumed. This simplification is not appropriate for compressors as in HTHPs as they are the components with the highest irreversible losses as reported in [40].

3.1.2. Reciprocating piston compressors

Reciprocating piston compressors, illustrated in Fig. 6, which use the motion of a piston, powered by a crankshaft, within a cylinder to pressurise the WF, are the most commonly used compressors in HTHPs and HPs in general for their simplicity, low cost, reliability, and popular use in many fields of application [124].

Table 5 summarises the key parameters in the listed research studies on HTHPs that incorporated piston compressors in their work. As noted, the maximum attainable heat sink temperature reached around 180 °C and heating capacities did not exceed 400 kW. Typical temperature lifts for such systems are approximately 50 °C on average with a limited number reaching up to 100 °C. Sarbu [36] hinted that reciprocating compressors are advisable for refrigerants with a high pressure, low vapour volume, and low molecular mass. This is reflected with the most common choice of hydrocarbon working fluids (particularly R600 with a molecular mass of 58.1 g/mol) in the reviewed HTHPs with piston compressors. An early prototype investigation of a moderately high temperature heat pump (up to 90 °C) considered a reciprocating compressor to deliver high-pressure CO_2 at discharge pressures of 100 bar – 120 bar for a low-pressure ratio R744 system [99]. The compressor running at full speed (1212 rpm) with no oil addition to the working fluid achieved isentropic efficiencies of 70 % for all test cases but with a substantial decline in volumetric flow rate and large heat loss at higher compression ratios as anticipated. The influence of working fluid selection on the pressure ratio and power input of a hermetic reciprocating compressor that was formerly constructed for the high-GWP HFC R134a was investigated by Pan et al. [126]. For a temperature lift of 45 °C, a

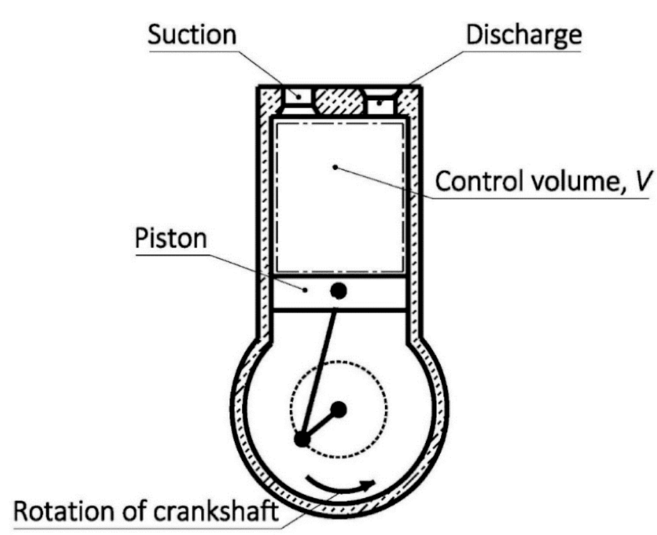

Fig. 6. Illustration diagram of reciprocating piston compressor [125].

Table 5
Overview of HTHPs with reciprocating piston compressors.

Ref.	Year	Cycle	Refrigerant	T_{source}/ T_{evap} [°C]	T_{sink}/ T_{cond} [°C]	THC [kW]	COP [-]
[126]	2011	SS	R600– R600a	−/32–44	−/60–90	2.5–5.5	3.3–5.8
			R245fa	−/40–55	−/85–100		3.3–3.8
			R600/R245fa	−/40–55	−/70–100		3.3–5.4
[61]	2015	SS + IHX	LG6	−/40–90	−/70–140	4.4–6.5	3.3–4.0
[127]	2018	SS + SEP	Several	60–100/-	100–140/-	20	1.8–3.4
[68]	2018	SS + IHX	R1233zd(E)	60–80/-	70–150/-	2.3–10.5	1.1–3.6
[128]	2019	MS	R744	–	130–145/-	90	–
[129]	2019	SS	R601	−/83–88	−/130	55–83.4	–
[130]	2019	SS	R1234ze(E)	−/82	−/150	≈11	3.3–3.7
[131,132]	2019	Cascade	R290/R600	26–63/ 46–57	115–125/ 114–118	20	2.1–3.1
[133,134]	2020/ 2022	SS + IHX	R600	60/-	110–160/-	11.4–30.7	3.1–4.5
[135]	2020	SS + IHX	R1336mzz(Z)	88/-	120–160/-	196–282	2.7–4.7
[136]	2020	SS + IHX	various	85/-	100–188/-	<200	1.9–4.3
[137]	2020	MS + IHX + SEP	R601	80–85/-	130–150/-	98–166	2.1–2.7
[138]	2020	SS + IHX	R1224yd(Z) R1233zd(E)	40–80/-	80–150/-	3–10	1.3–4.6
[139]	2021	3 HPs	R704	80/-	183	190–410	≈1.3
[140]	2021	SS + SC	R1233zd(E)	70–105/-	≈130	12.1–42.3	4–7

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lower pressure ratio of 3.06 was obtained for the R600/R245fa mixture in comparison to that of the HFC R245fa refrigerant alone (3.21). A higher compressor power was required for the R600/R245fa mixtures compared to the R245fa, where the difference is more pronounced at higher temperature lifts and condensation temperatures (up to around 24 %) [126]. Reißner [61] used a 5.5 kW two-cylinder piston compressor with a displacement volume of 0.23 l running at speeds between 900 rpm and 1800 rpm which imposed a limit to the maximum evaporation and condensation temperatures (90 °C/ 140 °C) for the developed refrigerant (LG6) which has an A1 SG and a GWP < 1. Heat losses in the compressor were expected as insulation would not be suitable in order to ensure cooling of the compressor. The heating capacity of the HTHP at a condensation temperature of 120 °C increased by around 50 % when the compressor speed was raised from 900 rpm to 1400 rpm, but the COP decreased from 4.0 to about 3.4 due to increased pressure losses [61]. The author concluded that because of the high target of the heating capacity (>500 kW) of their work, a turbo-compressor would be more suitable than a piston compressor which was used mainly for demonstration purposes. In the theoretical study by Bamigbetan et al. [127], a black box approach for modelling a piston compressor was considered by assigning volumetric and isentropic efficiencies of 70 % as well as suction and discharge temperature limits of 80 °C and 140 °C respectively, as per a butane prototype specification. Besides the dependency of the compressor side temperatures on the heat supply and delivery temperatures (operating conditions), the compressor discharge and suction pressures, its pressure ratio, and its size in terms of volumetric flow rate were also analysed as these parameters ultimately affect the compressor design and selection. Through an analysis of different working fluids, R600 and R1233zd(E) showed promise in both performance metrics and in their suitability of use in existing compressor technologies [127]. Arpagaus et al. [68] used a variable-speed semi-hermetic compressor, running between 30 Hz – 50 Hz (870 rpm – 1750 rpm) to drive their experimental HTHP system. Polyolester oil was used for its lubrication. Their results indicated that both $\dot{W}$ and THC are directly proportional to compressor speed, and that lower COPs are achieved with reduced rotational speeds. Pressure ratios between 2.9 and 5.4 were reported for temperature lifts ranging from 31 °C to 71 °C, which are typical for positive displacement compressors. A transcritical CO_2 HTHP with a serial stage pressurisation system composed of two semi-hermetic piston compressors (25 Hz – 70 Hz) was tested by Bellemo et al. [128] but with little description of parametric effects on performance. The same limited detail was provided by Marina et al. [129] for their 193 m^3/h rated capacity open type piston compressor at 1450 rpm. Similarly, a semi-hermetic compressor with 7.7 m^3/h displacement, maximum discharge pressure of 13 MPa, and a maximum motor input of 11 kW, initially designed for R744, was used by Abi Chahla et al. [130] for their transcritical HFO HTHP test rig. Bamigbetan et al. [131,132] evaluated the performance of two prototype four-cylinder semi-hermetic piston compressors in a cascade cycle. The compressors were built to withstand a maximum discharge temperature of 160 °C, with the high temperature compressor withstanding suction temperatures up to 80 °C for cooling purposes. Minimal oil and compressor degradations were observed at oil temperatures below 100 °C. The rated maximum operating pressure of 31 bar, and the tolerable pressure ratios of 3 – 6 were not exceeded [131,132]. Thermographs of temperature distribution within the compressor visualise that the maximum temperature is located at the compressor discharge with little heat transfer beyond that ensuring that both heat losses are minimal, and that other essential parts are safe. Verdnik and Rieberer [133,134] studied the effect of several operating parameters on the performance of a HTHP with R600 using a reciprocating compressor of 34.7 m^3/h capacity at 1450 rpm. Their numerical work was validated with experimental results which showed similar trends to previously discussed publications. The optimum maximum pressure level was around 18 bar after which an increase led to increased compression work and pressure ratios which decreases the compressor volumetric efficiency. Maximum COP values were achieved at the nominal compressor speed with reductions up to 4 % when the compressor frequency was at 75 Hz. The authors suggested a control strategy to maintain the optimum pressure for enhanced operation of the system [133,134]. Several studies were presented in the 13th IEA Heat Pump Conference [135,136,137,138,139,140] about emerging test facilities using piston compressors with preliminary results summarised in Table 5.

In terms of modelling for piston compressors, Hassan et al. [29] used experimental data to determine (Table 6) the empirical parameters in the form of correlations (using the regression tool of Microsoft Excel) that together with data for the other components of the HTHP were given as input to IMST-ART. This software is appropriate for the modelling of any vapour-compression refrigeration system regardless of the type of refrigerant and secondary fluids [141]. The developed model was used for initial sizing and selection of the main components of the HTHP. In an earlier study, Hassan et al. [142] analysed the operation of a selected piston compressor (model HBC-511 VHE) and designated additional parameters to describe the compressor model used in IMST-

Table 6
Overview of HTHPs with screw compressors.

Ref.	Year	Cycle	Refrigerant	T_{source}/ T_{evap} [°C]	T_{sink}/ T_{cond} [°C]	HC [kW]	COP [-]
[106]	1997	Cascade + EXP	R717/R718	40/-	180/-	–	2.6–3.0
[66]	2013	SS	R245fa/R134a	−10–40/-	65–90/-	70–228	1.9–3.8
[157]	2014	SS + FT	R718	86–94/-	–/118–122	300–335	4.5–5.5
[158]	2019	SS + FT	Various	–/50–70	–/105–130	–	2.8–4.1
[86]	2019	SS + FT	R718	–/83–87	–/120–128	190–230	5.0–3.8
[87]	2020	SS + FT	R718	/75–85	–/111–150	285–118	6.1–1.9
[76]	2020	SS + FT	Various	–/40–90	–/80–140	<300	2.5–4.7
[81]	2020	SS/ SS + IHX/ MS + FT	Various	40–100/-	100–140/-	180–800	1.5–5.5
[159]	2021	SS cascade	BY3B/BY6	55–85/-	140/-	468–542	2.3–3.1

– is used to indicate values that are not reported.
/ is used to indicate or (example, either source temperature or evporation temperature).

ART: compressor speed [rpm], compressor displacement [m^3/s], oil circulation rate [%], oil volume [m^3], external inverter efficiency [%], and heat losses [%]. The remaining two parameters of overall and volumetric compressor efficiencies were entered as a polynomial based on experimental data. The refrigerant mass flow rate and the compressor inlet electrical power were described as per values from the experiment [29].

As mentioned earlier, the type of working fluid determines the operating parameters of the compressor. In order to analyse the type of working fluid performance for HTHPs with a piston compressor, Bamigbetan et al. [127] assumed a constant isentropic and volumetric performance of the compressor using Dynamic Modelling Laboratory (Dymola) 2017 [143] to model the heat pump cycle and TIL model library [144] for modelling thermal systems. In their subsequent publication [132], the authors analysed experimentally the performance of a modified compressor prototype for a hydrocarbon heat pump with butane as the working fluid. They showed that a proportional increase in mass flow is not obtained at higher compressor speeds thus higher speeds are not recommended for compressor design. In addition, the thermograph images taken allowed the temperature distribution in the analysed prototype to be compared with the compressor used for low-temperature applications. They showed that the high-temperature area on the compressor discharge is limited to the discharge manifold only, thereby protecting other parts of the compressor, for example the electric motor.

Arnold and Stewart [145] point out that for piston compressors the flow rate is not directly equal to the displacement of the piston as there is an associated volumetric efficiency, the ratio of actual volumetric flow at inlet temperature and pressure conditions to the displacement of the piston. Hassan et al. [142], presented a thermodynamic model where the power consumption for each stage is determined in the same way assuming the mechanical efficiency (η_m) is 93 % and indicated efficiency (η_i) is 85 %. For their reciprocating compressor, Mateu-Royo et al. [146,147] demonstrated that the power consumption of the compressor is proportional to the mass flow rate and the slope of the isentropic line of each refrigerant based on results from the Engineering Equation Solver (EES) software which allows multiple differential equations to be solved numerically [148]. In addition, the authors specified that Pierre's correlations can be used to identify the volumetric and isentropic efficiencies of reciprocating compressors. The above approach for reciprocating compressors with a multi-parameter optimisation algorithm was also presented by Mota-Babiloni et al. [98] for a two-stage HTHP system with internal heat exchangers. The authors employed EES software that incorporates Powell's conjugate direction method to find the local minimum of a function of several variables, and REFPROP [149] for calculating the transport and thermodynamic properties of fluids and their mixtures.

For transcritical heat pumps, energy consumption and heating capacity in the supercritical area is influenced by compressor discharge pressure [150]. Therefore, researchers seeking to improve system efficiency in their work use thermodynamic models to optimise compressor discharge pressure [151,152]. Li et al. [150] reviewed and presented correlation methods of optimal discharge pressure; the thermodynamic models were adjusted per the correlation equation for the optimum discharge pressure of the semi-hermetic reciprocating compressor, based on experimental research for an air source transcritical CO_2 HP. It was found that the water temperature at the inlet and the CO_2 temperature at the outlet of the gas cooler have a strong influence. In the next step a control strategy was determined that took into account the correction of the degree of superheat.

3.1.3. Screw compressors

Screw compressors are commonly used in air compression and refrigeration cycles for their wide operating capacity and their associated COP improvements with respect to piston compressors [153]; noting that the comparison is only valid for part of the overlapping operating range, screw compressors offer an advantage of higher heat outputs. They have been used in heat pumps and water distribution systems for residential purposes [154]. These are the most common type of compressors used in industrial applications such as high-pressure air supply and refrigeration systems. Their simple construction, reliability, and compactness compared to other positive displacement compressors are among the main factors for their widespread use. Twin-screw compressors are made of two meshing rotors with helical grooves enclosed in a casing (as shown in Fig. 7). The rotors and casing are separated by small clearances to prevent rubbing against each other. The rotors are driven by a motor and mesh like gears, and as they rotate, the space between the two rotors and the casing is reduced progressively, thus reducing the volume, and increasing the pressure of the trapped gas between them. The gas enters the compressor in one inlet port at a position where the volume is large. The exit port is placed at a location where the trapped volume is reduced to its minimum, thus achieving the required pressure rise [155].

Screw compressors can handle a variety of working fluids, which may be gases, dry vapour, or multi-phase mixtures with phase changes taking place within the compressor. They typically involve oil flooding, or other fluids injected during the compression or expansion process. Oil acts primarily as a coolant, enabling high pressure to rise in a single compressor stage. In addition, oil acts as a sealant reducing leakage of compressed gas from high pressure to neighbouring low pressure cavities and hence improving the volumetric efficiently of the compressor. Furthermore, the oil acts as a lubricant, preventing rubbing of rotors against each other and against the casing. Oil is usually separated from the compressed gas and is re-injected into the compressor.

In some industrial applications such as food and beverage industries, it is not desirable to have traces of oil remaining in the compressed air used in the process. In this case, oil free screw compressors are potential alternatives. These are, however, more complicated as they require a timing gear to drive the second rotor, which is driven by the main rotor in oil fed compressors. The absence of oil limits the functionality of oil

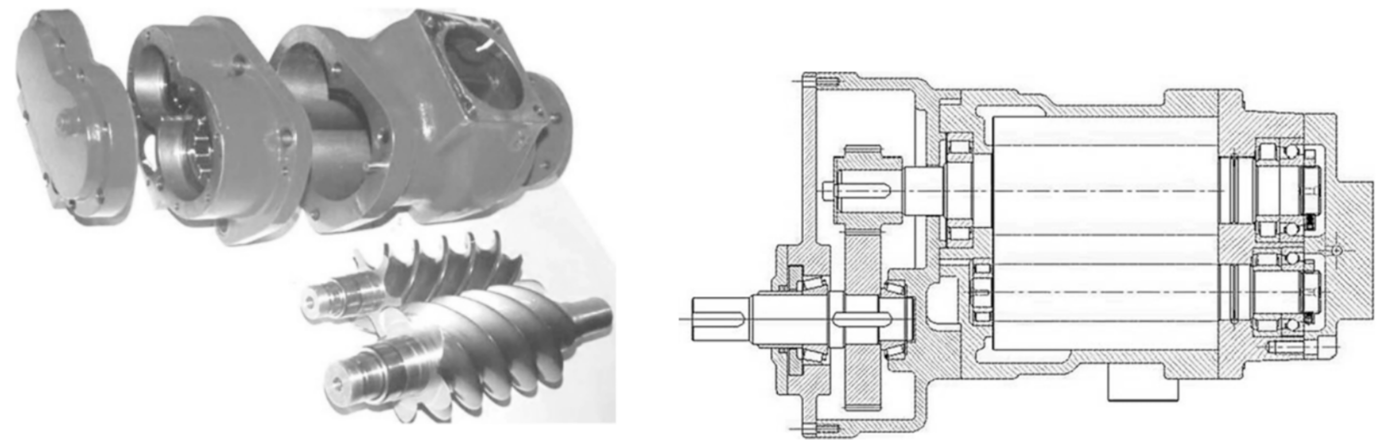

Fig. 7. Views of screw compressor rotors and casing [155]. Reproduced with permission.

free compressors as this instils a constraint on the maximum pressure rise. Also, operating life of such compressors is significantly reduced due to the resulting thermal stresses and potential rubbing of components. A comprehensive review of the state of the art in research advances and applications of screw compressors was conducted by Wang et al [156]. In this paper, the main focus is on screw compressors research for high temperature heat pump applications.

One of the early investigations of the usage of a screw compressor for HTHPs was done by Tamura et al. [106] in 1997 who developed a computational model of a HTHP system. Their work was primarily influenced by the application of an expander to their HP and did not give much detail on the compressor characteristics. Fast forward to 2013 when Kobe Steel [66] developed a heat pump, with a twin-screw compressor, which supplies hot water at temperatures up to 90 °C which is just under the 100 °C threshold of the high temperature consideration of this paper. Their modified semi-hermetic two-stage compressor required motor cooling which was supplied through the spraying of the flashed refrigerant. The two-stage system was selected to accommodate the encountered high pressure ratios (>5), and to expand the operating envelope of the compressor beyond which a single stage can uphold [66]. Chamoun et al. [157] developed a twin-screw compressor running nominally at 4700 rpm with a displacement of 6.6 l/rev and an input power of 90 kW for their HTHP system. The calculated volumetric and adiabatic efficiencies of the compressor are 82 % and 57 % respectively. A thermodynamic analysis was done by Lu et al. [158] for a HTHP with a compressor that is assumed to have an average isentropic efficiency of 72 %. For the four WFs that they simulated, the pressure ratio was always almost around 5 with the condensation pressure always being below 2.3 MPa. However, this paper [158] seems ambiguous for mentioning the use of both screw and scroll compressors but given that it is a thermodynamic analysis with no specifications of the compressor design then the discussion of the type is irrelevant, and the review is merely for heat pump performance purposes. Wu et al. [86] simulated a model of a HTHP and experimentally validated it with tests from a system using a specifically design twin-screw compressor. The compression ratio and the compressor power varied between 3 and 4.7 and 46 kW – 57 kW respectively when the condensation temperature increased from 120° to 128 °C. The changes in heating capacity however were less pronounced when the condensation temperature increased but were more affected by the evaporation temperatures [86]. Wu et al. expanded on their experimental work a year later [76,87] where they explored the performance of the oil-free twin-screw compressor with a 30.8 m^3/min displacement. Volumetric efficiencies of their compressor were in the range of 42 % to 72 % while adiabatic efficiencies varied between 39 % and 59 %; where the higher values were obtained for evaporation temperatures of 85 °C and condensation temperatures less than 120 °C which yielded a pressure ratio between 3 and 4. Compression ratios reached values as high as 9.3 when a temperature lift of 65 °C was imposed [87]. An almost linear positive gradient was established between the compressor power consumption (between around 38 kW and 83 kW) and the condensation temperature, with little effect of evaporation temperature. They also observed that the compressor discharge mass flow rate decreases with higher condensation temperatures due to a lower compression volumetric efficiency and the constant water vapour refrigerant density [87]. Kosmadakis et el. [81] used a 470 m^3/h at 2900 rpm semi-hermitic screw compressor model for their techno-economic analysis of different cycle configurations using various refrigerants. They account for volumetric losses within the compressor noting an actual flow rate of a minimum of 400 m^3/h for high pressure ratios which yields lower isentropic efficiencies, and a maximum of 460 m^3/h in other instances. From their results and noting that available screw compressors have a maximum displacement of around 1,500 m^3/h, the authors [81] state that for very large heating requirements (over 800 kW), several large compressors are required to achieve upscaling. An open — separated motor and screw to avoid damage because of high temperature — oil free 8 m^3/min at 3000 rpm twin screw compressor with rated power of 150 kW was adopted by Wu et al. [159] for their cascade heat pump system. A maximum design temperature of 200 °C mandates specific geometric constraints, such as tooth ratio, length to diameter ratio, and meshing clearance, to ensure safety of the system at high temperatures, while moving parts of the compressor (bearings and gears) were oil lubricated. Their results of the multi-objective optimisation to balance out heating capacity and annual net profit, show that for a heating capacity of around 500 kW and a T_{lift} = 74 °C, a compression ratio of 2.4 and 2.2 was achieved for the low temperature and the high temperature compressor stages respectively [159].

As pointed out in this review, most of the screw compressors were investigated with water as the working fluid. From Wu et al.'s [76] refrigerant comparison, R718 shows promising advantages for high condensation temperatures due to its low saturation pressure. However, ways to manage the compressor discharge superheat and temperatures are required for safe operation of such systems using water.

Chamoun et al. [160] presented a mathematical model of a twin-screw compressor that uses water as the working fluid. The modelling method was based on three processes occurring simultaneously for one revolution of the male rotor (2π): suction, compression, and discharge. A set of differential equations was used to describe the conservation of energy and mass which were solved in Dymola 6.1 (Dynamic Modelling Laboratory [161]) based on the open Modelica modelling language. Ahrens et al. [162] also used Dymola 2020/Modelica, however, to solve the quasi-one-dimensional numerical model of an oil-free liquid-injected twin screw compressor, specifying eight control volumes in the model. They point out that it is difficult to determine the exact leakage flows that occur and the efficiencies that can be achieved. Wu et al. [86] assumed a multi-stage compression model, with the same compression ratio for each step, for a twin-screw water compressor in a HTHP. They provided separate formulas for mass flow rate of compressed water vapour, and of injected water for each of the stages under consideration, as well as for the mass flow rate of water vapour intake and of water vapour discharge by the compressor. Ahrens et al. [162] stated that the performance of screw compressors depends on internal leakage, it is therefore crucial to determine the value of the leakage flow rate in the model by linking adjacent control volumes.

Bergamini et al. [26] indicated that the compressor has a limited impact on exergetic efficiency trends based on the heat pump models implemented in EES. However, in their analysis compressor heat losses have been ignored and isentropic efficiency and motor electric efficiency were assumed constant at 85 % and 90 % respectively. In turn, Uusitalo et al. [163] concluded that the compression power consumption at high temperature lifts lowers the COP of the heat pump, based on the conducted HP thermodynamic analysis, compressor design analysis, and experimental results for the cascade heat pump, which is why cascade systems are often used.

Though they present numerous advantages to high temperature heat pumps, the volumetric flow rate of such screw machines is constrained by manufacturing limitations on rotor size and tip speed restrictions [164]. The pressure difference across the machine affects the bearing loads, meaning that a maximum allowance on the pressure ratio, which in turn limits the temperature difference, is required to minimise thermal distortion of the compressor rotors and casing. HTHPs operating with high pressure and temperature differentials can cause severe deformation that increase leakage losses and adversely impact performance [165]. Consequently, precise clearances need to be maintained to restrict internal leakages and component wear. Furthermore, the challenging high temperature environment created within IHPs places further limitations on oil-refrigerant compatibility and restricts sufficient bearing lubrication and cooling [166].

3.2. Turbo-Compressors

Turbo-compressors are well established in many industries at various scales ranging from a few kW power input to over hundred MW. At the upper end, mostly multistage axial compressors are used, mainly in gas turbines for aerospace propulsion and power generation, as well as for compression in gas pipelines. Axial compressors can achieve high isentropic efficiencies exceeding 90 % and can handle large mass flow rates at a much smaller frontal projection than centrifugal compressors. At the lower end, mostly centrifugal compressors are used due to their design simplicity, higher pressure ratio in a single stage, and wider operating envelope among other advantages which compensate for their lower isentropic efficiency compared to axial flow compressors. They however can achieve higher efficiencies than positive displacement compressors. Centrifugal compressors (schematic diagram shown in Fig. 8) have been popular in automotive turbochargers, micro gas turbines and plenty of process industry applications. As with other compression systems, there is an overlap in practically achievable capacity where the choice may fall between the two types depending on the specific need, economic and supply chain factors.

Turbo-compressors have been in heat pumps mainly for higher outputs exceeding few hundred kW, but mostly closer to higher than one MW. The predominant factor for their selection seems to be in cases where screw compressors cannot achieve the required duty due to size limitations imposed by manufacturing constraints on these machines. Where there is overlap, the choice may fall to turbo-compressors mainly where their higher efficiency may be a more important for the application than operational range.

There is a plethora of research papers which include discussions or reviews of advances in turbo-compressors, their analysis, design, and optimisation methods, primarily for ideal gas working fluids. However,

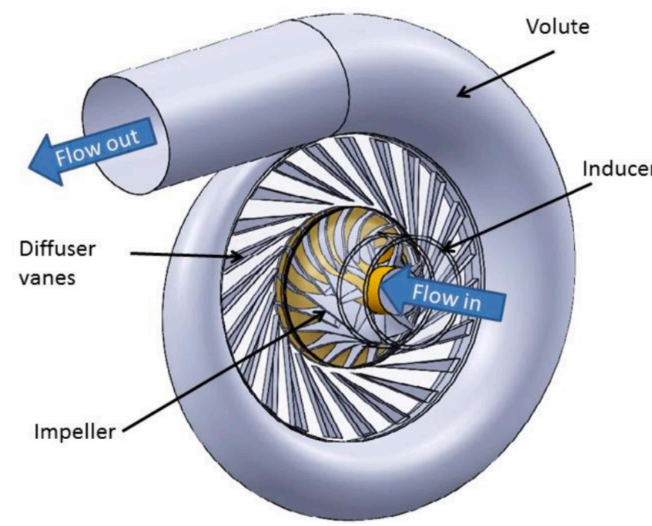

Fig. 8. Typical structure of a centrifugal compressor [167].

there has been more interest in recent years in conducting research for compressors handling non-ideal working fluids such as refrigerants, primarily water, as well as CO_2 as an organic fluid for use in heat pumps, supercritical power cycles, and organic Rankine cycles; for example, Jaatinen-Värri et al. [168] presented a study on the design of centrifugal compressors for HTHPs. This paper will mainly focus on research of compressors intended for high temperature heat pumps. Given that most of the research in this area is either for low temperature systems (<100 °C heat sink temperatures as in the case of [169,170,171]) or on preliminary numerical design and analysis, with limited experimental testing, a summary table is not presented for turbo-compressors and a discussion is given instead.

Zamfirescu and Dincer [75] considered non-conventional four mechanical compression heat pump configurations with a chemical reactor for studying the performance of seventeen BZT working fluids which are applicable to temperature lifts of around 50 °C. They suggest the use of turbomachinery for both the compressor and expander and imply that the use of BZT fluids would enable a turbo-compressor from operating shock-free even when flows reach Mach numbers slightly higher than 1, making such heat pumps suitable for high temperature applications, particularly when isentropic efficiencies are relatively high. Such studies are quite limited and might be useful for further exploration for HTHP applications. A techno-economic study was performed by van de Bor and Infante Ferreira [172] whereby the performance and costs of vapour compression cycles, with either a turbo or a screw compressor (or blower), were compared based on size and application. Their calculations infer that the cost of centrifugal turbo-compressors is less sensitive to shaft power in contrast to screw machines, while that the latter are more economical for applications where thermal duty is less than 5 MW. It is worthy to note that their approach does not consider the actual feasibility of systems which, as per state-of-the-art, are not at a maturity level reaching such large sizes. One of the early developments of a HTHP turbo-compressor was by Madsboell et al. [70] who based their design on an existing high-speed gear for automotive supercharger applications (with rotational speeds of 100,000 to 300,000 rpm) rather than the use of more expensive direct drive. Their titanium impeller steam centrifugal compressor, with a 300 kW – 500 kW capacity, was designed to withstand source temperatures of around 60 °C – 85 °C and temperature lifts between 20 °C and 30 °C. The suction volumetric flow rate of 0.3 m^3/s for the compressor running at 95,000 rpm at high pressure ratios, dictates a tip speed of 450 m/s – 550 m/s to achieve the desired sink temperatures. In timber drying applications, for a single-stage standard cycle, they reported COPs reaching values between 4 and 7, where the higher values are for cases of superheated steam [70]. Similarly, three-stage water centrifugal compression was used by Tolstorebrov et al. [173] to allow higher flow rates (0.5 kg/s – 0.7 kg/s per stage) for a steam drying process and regeneration which provides hot tap water around 400 kW. An open system with atmospheric superheat suction pressure has been shown to perform better (COP = 3.84) than a closed system with an evaporation temperature of 90 °C at 0.7 bar (COP = 3.31). A two-stage centrifugal compressor running nominally at 1630 m^3/h and 11,000 rpm using water too, albeit not for HTHP applications, was used for supplying a heating capacity in the range of 1,500 kW – 1,750 kW with temperature lift up to 30 °C giving an average COP of 5.5 [174], showing the feasibility of turbo-compressors not only for high temperature applications. Another compression cascade heat pump system using R718 was assessed for condensation temperatures between 40 °C and 160 °C from evaporation temperatures of 30 °C – 100 °C considering turbo-compressor isentropic efficiencies of 70 % and 80 % [175]; the authors reported COPs of around 7.4 – 8.4 for such configurations. Average pressure ratios ranged from 1.41 to around 5.68 for temperature lifts limited to 40 °C, however, larger differentials (T_{lift} = 50 °C – 60 °C) led to compression ratios up to 16.53 which might not be feasible given that the maximum pressure ratio per stage is around 3.8 – 4.8 to limit the turbo-compressor's tip speed to 600 m/s. Bantle and their team at SINTEF presented their work on a of a water-based two-stage high temperature heat pump over the years starting with performance evaluation of a compressor prototype [176,177,178] and moving on to experimental testing [179,180]. A radial turbo-compressor for mechanical vapour recompression heat pump system for steam drying was designed at 0.22 m^3/s flow rate and a pressure ratio of 2.7, with the impeller running at 90,000 rpm, to achieve up to 300 kW_{th} capacity. Initial evaluations gave COP values as high as 11.5 but with a limited temperature lift of 25 °C [176]. Initial testing was conducted for two identical turbo-compressors were designed for a two-stage mechanical vapour recompression HTHP delivering 2.8 bar steam at 131 °C (from atmospheric conditions) [177]. The pressure ratios of each stage was 1.69 with a mass flow rate of 400 kg/h – 600 kg/h and the obtained COP was 7.8. More elaborate experimental procedures were conducted at a later date [179] for the compressors running at 81,000 rpm and 72,000 rpm for the first and second stage respectively, delivering 500 kg/h of steam at 3 bar and 133 °C for a 300 kW_{th} system with a COP of 5.9. A total pressure ratio of 3.0 was obtained across the two-stages with a compression efficiency of 74 %. Most recently, the group reported an experimental COP of 4.54 for a two-stage 500 kW system operating at high temperature lift (46 °C) and pressure increments (4.2 bar steam from atmospheric conditions). Their compressors were able to operate at pressure ratios and isentropic efficiencies of 2.4 – 2.0 at 90 % speed, and 67 % – 77 % respectively; the team generated compressor maps for different operating conditions.

Madsboell et al. [70] used computational fluid dynamics to design a centrifugal water vapour (steam) compressor geometry and 1D correlations to analyse various parameters relating to the operation of the unit. Through the model used, they demonstrated that the trade-off between high efficiency and high pressure is determined by the blade exit angle. Steady-state, one-dimensional numerical modelling (mean-line model) of a centrifugal compressor was discussed also by Meroni et al. [143]. For this purpose, MATLAB [181] was used where the focus was on modelling the compressor rotor and the vane and vaneless diffusers. Flow and thermodynamic conditions were obtained by solving the energy balance, mass continuity, and loss equations simultaneously. Determining the choke point in the compressor stage was essential in formulating the mean-line model, therefore the proposed method identified the choke point by assessing the potential for choking conditions at the rotor inlet, rotor outlet, or vane diffuser. To avoid two-phase conditions at the compressor outlet, a degree of superheat was imposed, thus fixing the inlet temperature. Meroni et. al [143] presented a meanline method for design and off-design performance analysis of centrifugal compressors for heat pump applications where the model can result complete compressor maps.

Centrifugal compressors are a more suitable choice for systems with high volumetric flow rate demands and heating capacities over 500 kW_{th} [62]. In comparison with screw compressors, the centrifugal compressors have the benefits of higher efficiencies, lower investment costs, and more compact sizes which make them appealing for HTHPs [173]. Nonetheless, with the limited HTHPs deploying turbo-compressors on the market, it is evident that there exists a research gap for this technology in IHPs.

A guideline for compressor selection is given through the research carried out by Frate et al. [77], who investigated the feasibility of different compressor technologies, single-stage (SS) and multi-stage (MS), depending on the size of the system (evaporator thermal input: 500 kW_{th} and 2000 kW_{th}) and the type of working fluid for the vapour compression heat pump with the heat sink temperature up to 150 °C. The results presented in Fig. 9 indicate that the best solution for the case under consideration is single stage and multistage centrifugal compressor as they are adapted to the assumptions mentioned previously. However, if higher discharge temperatures are required then it is possible to use reciprocating compressors (200 °C) or oil-free liquid-injected twin screw compressor (221 °C). It can be concluded that various compressor types are possible for high-temperature heat pumps. The choice of compressor type depends on the type of working fluid and the HTHP application.

4. Conclusions and future trends

The main objective of this paper was to provide a comprehensive literature review of research articles on compressors used in high temperature heat pumps. To put this into context, additional material was provided on thermodynamic cycles, working fluid selection and compressor modelling approaches.

It was found that positive displacement compressors, primarily piston and screw compressors are the most prevalent in heat pump applications, with piston compressors covering the lower end of the spectrum. At the higher end, turbo-compressors are used mostly where the maximum size of available screw compressors cannot achieve the required capacity.

There are several factors that led to the widespread use of positive displacement compressors in heat pumps in general, which is not different from the case in high temperature heat pumps developed so far. The most important of these factors, is probably their wide operating range while maintaining efficient operation. This is unlike turbo-compressors which generally have higher design point efficiency than positive displacement compressors, but they have a much narrower operating envelope, and their performance deteriorates significantly away from design point, unless variable geometry features are used, which can improve part load performance and widen the operating envelope at the expense of more complexity. The second factor is economic, where positive displacement compressors have relatively lower capital costs which are driven by the volume production due to synergy with the well-established refrigeration industry, the design simplicity, and ease of manufacturability. Another important parameter is the much lower rotational speed required for positive displacement compressors compared to turbo-compressors providing the same capacity which puts less constrains on the design and hence cost of bearings, electrical drive and power electronics used for control.

With recent interest in achieving higher temperature lifts and higher sink temperatures in industrial heat pumps required for upgrading waste heat and reducing fossil fuels use in process industry, it is more likely that interest in turb-compressors, at capacities much lower than they have been used, will increase, in particular, for sink temperatures around or exceeding 150 °C. In these cases, oil fed positive displacement compressors suffer from the challenge of the limitation to the maximum temperature achievable with commercially available oils. This is not the case in turbo-compressors where oil is mainly required for the bearings where cooling can be easily achieved. Additionally, there has been

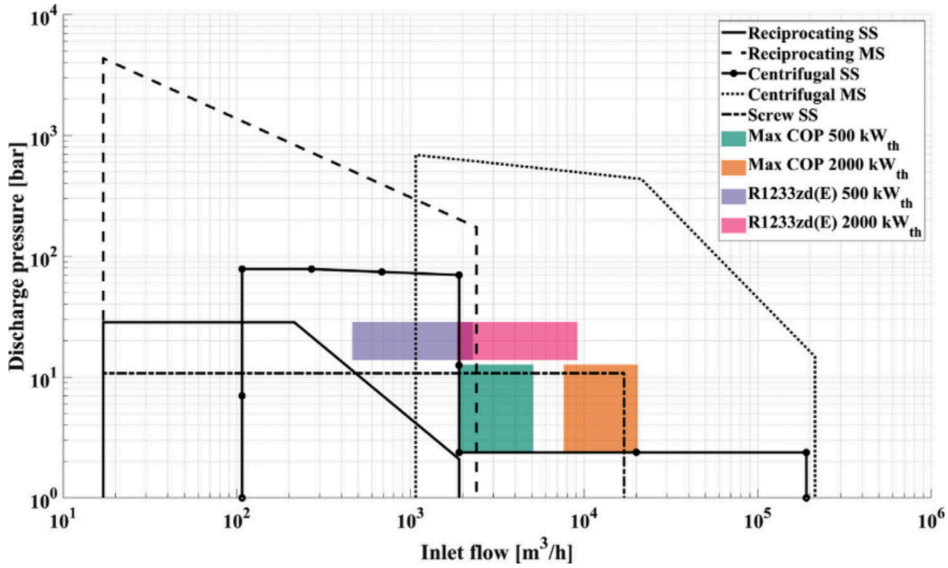

Fig. 9. The relationship between discharge pressure and inlet flow for different compressor technologies [77]. Reproduced with permission.

significant progress in recent years in oil free bearings. It is plausible to assume that oil free screw compressors may offer the potential to plug this gap at the achievable capacities, however, these machines suffer from challenges of the lower achievable pressure ratio than their oil injected counterparts and the deterioration of performance and reliability due to thermal expansion and increase in clearance gaps. Hence, turbo-compressors may be the obvious choice for the high temperature applications, particularly as higher temperatures are sought. Research and development, however, is still required to enable their commercial exploitation. Areas to address include design innovation to extend the operating envelop while maintaining high efficiency, shaft bearing arrangements and tribological aspects to ensure vibration free and reliable operation and other aspects that may lead to robustness, reliability and cost reduction.

CRediT authorship contribution statement

Tala El Samad: Writing – original draft, Data curation, Conceptualization. **Alina Żabnieńska-Góra:** Writing – original draft, Formal analysis. **Hussam Jouhara:** Writing – review & editing. **Abdulnaser I. Sayma:** Writing – review & editing, Supervision, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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