PAPER ID: 0606
DOI: 10.18462/iir.compr.2024.0606

DECARBONIZING MID-SIZE INDUSTRIAL APPLICATION WITH LARGE SCROLL COMPRESSORS: VISION AND CHALLENGES

Ivan Rangelov(a), Jonathan Vlastuin(b)

(a) Danfoss
Kolding,6000, Denmark, ivan.rangelov@danfoss.com
(b) Danfoss
Trevoux Cedex, 01600, France, j.vlastuin@danfoss.com

ABSTRACT

There are many new developments in compressors for heat pumps (HP) as well as envelope extension investigations are undergoing. However, not without technical challenges. Each technology has its advantages and drawbacks, but there are some common challenges related to compressors for heat pump applications due to the higher temperatures and operational pattern compared to refrigeration: materials and lubrication related.

There are many technologies in the market. But selecting the best one means full application mapping of the industrial processes – cooling and heating, review available technologies/refrigerants/sizes, study best practices: safety, COP, installation cost, service, future proof, etc. and select the best solution for each application.

This paper will suggests and compares a combination of layouts with scroll compressors for both cooling and heating below 100 °C in a mid-size Food & Beverage (F&B) plant < 1 MW_{th} and analyze COPs and give a pointer of indication for installation costs based on installed thermal capacities. Moreover, it will focus on scroll development for high temperature applications and showcase some solutions with new development for large scroll compressors for HP applications for process heat, and then compare with the more conventionally used semi-hermetic piston compressors.

Keywords: Industrial heat pump, Decarbonization, process heating and cooling, scroll compressors

1. INTRODUCTION

There are three megatrends, challenging Industry’s future energy supply: urbanization, climate change and energy crisis. In Europe there already are several working mechanisms, such as the European Green deal (EU-Council, 2023). Climate awareness has picked momentum also in business. The war in Ukraine from 2022 has put gas boilers under pressure. (Pachai, Hafner, & Arpagaus, 2023).

However, it is hard to select or even map the optimal IHP technologies for a given industrial application due to:

1. Refrigerants variety
2. System layouts
3. Components selection
4. Control algorithm
5. System integration
6. Technical limitations for components
    a. Lubrication

b. Materials

”Heat pumps are all about COP”. The authors disagree with that postulate. It is important to evaluate the total cost of ownership (TCO) – not just COP, but a complete feasibility study. Comparing all possible scenarios over installation life-time and select the optimum (Reinholdt, 2023). Nevertheless, (Lund, Skovrup, & Holst, 2019) claims TCO evaluation is very difficult. Efficiency is important but is not everything, therefore at least a first cost comparison can be made – based on f.ex. swept volumes required per unit of heat supply.

Target of this paper is to illustrate example layout for a mid-size Food & Beverage plant with process cooling and heating demands below 100°C and evaluate solutions with large size scroll compressors. It will further elaborate and address some of the technical challenges for the compressor technologies in this application.

2. MAIN SECTION

2.1. Potential: why heat pumps in industrial processes?

Targeting decarbonization and efficiency improvements, over the past decade there has been a significant development of heat pump technologies. Today heat pumps’ reach supply temperatures up to 100 °C is well proven, implemented in the industry and commercially available. (IEA HPT, 2023)

Several studies show the potential for implementation of heat pumps for process heat. In Europe (EU 28), the addressable heat demand for industrial processes < 100 °C is approximately 11% from the total annual process heat (Boer, et al., 2020).

Table 1 shows temperature split of the annual heat consumption in Europe. It is fair to add that for the addressable heat demand for heating and hot water – the two latter comprise of 17%, largest share of which will also be < 100 °C. In total at least a third of heat (process and space heating) will be addressable with heat pump technologies, already proven and mature technologies today.

Table 1 Summary process heat demands split by temperature and segments EU-28 (Arpagaus, 2024)

	Heat Consumption (TWh/a)	EU-28	
	Space heating	297	16%
	Hot water	25	1%
Process heat	PH <60 °C	55	3%
Process heat	PH 60 to 80 °C	53	3%
Process heat	PH 80 to 100 °C	**89**	**5%**
Process heat	PH 100 to 150 °C	**192**	**11%**
Process heat	PH 150 to 200 °C	**80**	**4%**
Process heat	PH 200 to 500 °C	151	8%
Process heat	PH 500 to 1'000 °C	376	21%
Process heat	PH >1'000 °C	504	28%
	Total Heat Consumption (TWh/year)	1'821	100%
	Total Process Heat <60 °C to >1'000 °C (TWh/year)	1'499	
	Total Process Heat 90 °C to 160 °C (TWh/year)	**237**	**16%**

In industry, such as Food & Beverage, there is most often need for both cooling and process heating. Traditionally the former is done by an industrial refrigeration system (IRF), and the latter using a fossil fuel boiler, such as natural gas. Even though very often in these plants there is energy recovery with use of the process heat, this setup is inefficient as cooling and heating systems are disconnected. In fact, in many of these industries gas boilers can be partly or completely replaced by already available and proven heat pump technologies.

Figure 1 below shows a generic process flow diagram in a Brewery:

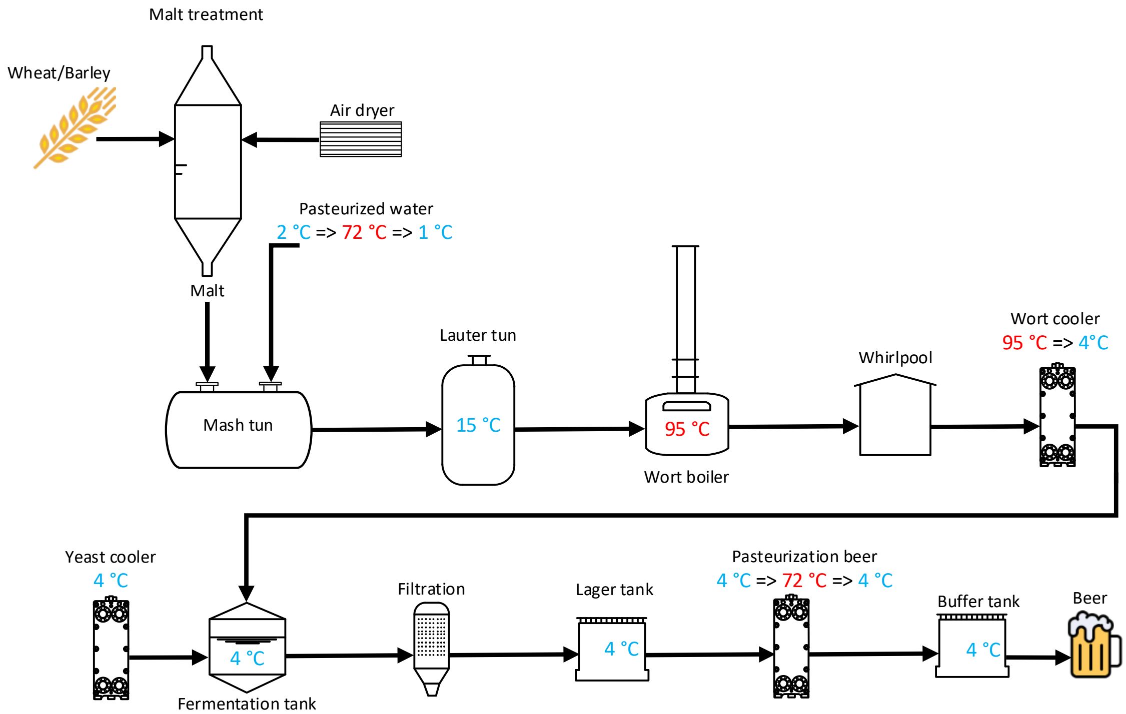

Figure 1: Simplified process diagram in Brewery

The generic process diagram of Brewery shown in Figure 1 is illustrating the sequence of cooling and heating processes during beer brewing. Even though details are not shown, it is clear that the product (water, wort, yeast and beer) is multiple times cooled down and heated up. For cooling very often is used chilled propylene glycol in temperature range -1 to -6 °C, however in large systems also direct refrigerant may be applied. For the heating needs hot water is required, and for the highest temperatures – wort heated > 95-98 °C, it is used low pressure steam (2-3 bar). Many of the processes in breweries are batch processes. To store the cold and heat carriers, there are usually large storage tanks of cold glycol and cold as well as hot water, which are also serving as buffers between the process side and the cold and heat generation plants.

Key take aways from Figure 1:

1. The product is cooled down and heated up multiple times. The waste heat from the refrigeration cycle can be a perfect source of heat for installation of a heat pump for the industrial processes. It is a sustainable way to reuse the waste heat as a source, rather than rejecting it into the environment and then burn fossil fuel into a boiler.

2. Most of the heating needs in a Brewery can be achieved with heating carrier below 100 °C, meaning already well proven heat pump technologies. (Note: still a small part of steam is required to completely disconnect from the boiler, which requires sink > 100 °C. Even though the technical readiness level of steam, generating heat pump is very high, they are not in focus of this paper).

3. The process is simplified, in real life also load profile as well as co-currence of processes must be accounted for successful integration of heat pumps in industrial processes. Cooling load typically varies, and heating is not always present, therefore:

    a. Cooling and heating must be decoupled from each other, also control for load balancing implemented

    b. Multiple sources must be used for the heat pump successful integration

To summarize, the potential is there: various processes in F&B industry, which just like in breweries, require both cooling and process heating. These are obvious places to start the decarbonization journey of the industry.

2.2. Cooling & Process heating in Brewery: which technologies and layouts?

There are multiple refrigeration technologies which can be used in F&B applications. The selection depends on multiple factors: needed cooling and heating capacity, COP, TCO, legislation, service, components availability, skilled personnel, experience, etc. (Rangelov & Lund, 2024)

Large scale breweries (MWs-size) would typically use an NH_3 IRF system for cooling, and then an add-on heat pump can be added "on top", lifting the waste heat from around 30 °C to 80-90 °C supply water with very high COPs, often in the range of 5. However, looking into small to medium size (< 1 MW_{th}) then more technologies become relevant and can compete.

Target of this study is to suggest a system layout for cooling and process heating for micro/medium size breweries – range of thermal capacity Q_{th} < 1 MW. In these capacities also a variety of environmentally friendly refrigerants – both naturals (e.g. Hydrocarbons) as well as low-GWP (HFOs) can be relevant. Furthermore, this paper strives to compare solutions for these applications with new development for large scroll compressors, targeting HP applications for process heat < 100 °C at first stage (later on 120 °C), with the more conventional used semi-hermetic piston type compressors.

A simplified layout example of such a system for cooling and process heat is shown in Figure 2 below:

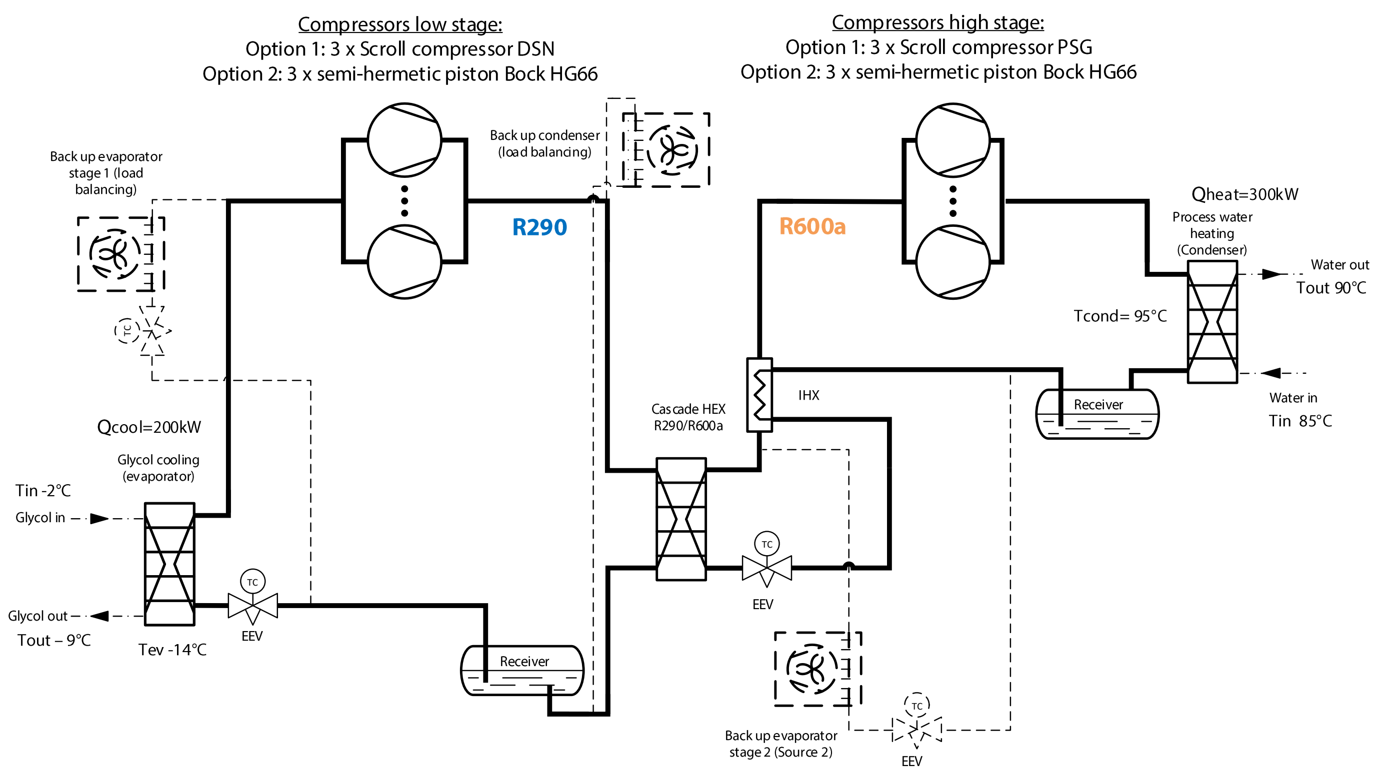

Figure 2: Generic and simplified cascade system layout applicable for cooling and process heat in micro/medium size breweries

The layout in Figure 2 is generic and is a cascade of two systems: cooling of glycol (1st stage) with R290, and heating of process heating of hot water to 90 °C (2nd stage) with R600a. The layout is simplified, and for real control of such a system it is required to: balance the load between cooling and heating as there almost always will be a mismatch. Hence back up condenser (sink) for Stage 1 needs to be installed. Similarly, on Stage 2, an additional back up evaporator (source) need to be considered for when there is insufficient waste heat from the cooling system or cooling is not required from the process side. It is important that cooling and heating are decoupled. In real systems, there are multiple sources, which also requires implementation of buffer tanks for a better balance between sources.

This is the layout which is used for simulations in this paper, which are described in the following paragraphes

2.3. System description

Reference scenario: Piston compressors. Cascade system shown in Figure 2 with semi-hermetic piston compressors. Each stage is made out of rack of three HG66e/1540-4 S HC compressors (Danfoss Bock, 2024). Each one of them has 133.8 m^3/h displacement and can operate both with R290 and R600a.

Alternative: Scrolls – same layout as above, also equivalent displacement using R290 on first stage, and R600a on second stage. Both scrolls are using similar platforms, but with specific design adaptation depending on the refrigerant and the application. Those scroll compressors are called DSN (for R290 stage) and PSG (for R600a stage) and are actually in development at Danfoss Commercial Compressor.

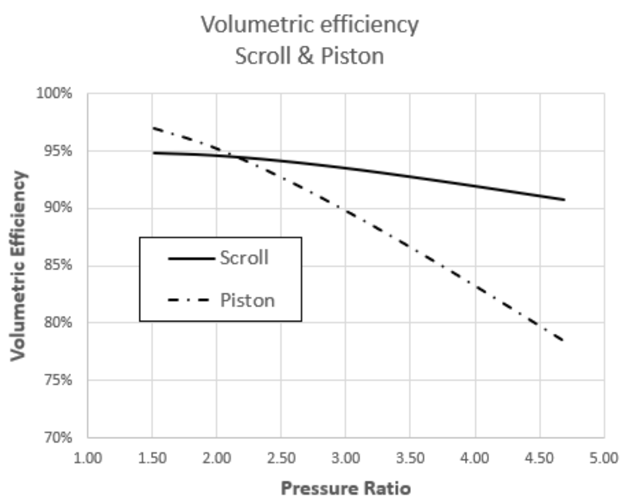

Figure 3: Volumetric efficiency Scroll & Piston

Figure 3 shows volumetric efficiencies of piston and scroll compressors at different pressure ratios. Piston compressors are particularly efficient at high pressure ratios as the volume ratio of the compression chamber is the ratio of the displacement over the dead volume of the cylinders (by construction, as low as possible). The drawback of this technology is that having a non-zero dead volume prevents a significant amount of fresh fluid to get into the piston chamber at each cycle as the pressure increases on the discharge side (dead volume is expanding back while the piston goes back in the bottom position). The result is a reduction in volumetric efficiency of the compressor as the pressure ratio increases, like we also witness on the scroll, but with a bigger penalty on capacity. Summary, if the machine operates at high pressure ratio, one needs to oversize the piston displacement to match the equivalent capacity of a scroll.

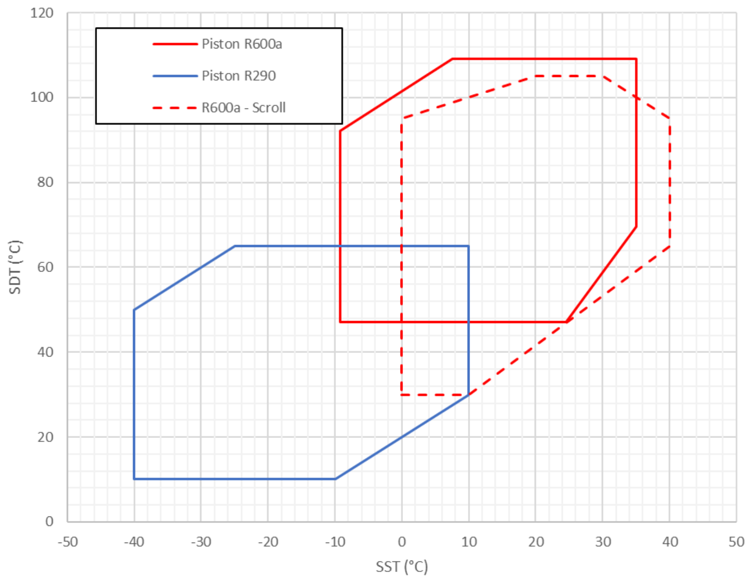

Figure 4: R290 + R600a piston compressor operating range of HG66e/1540-4 S HC by (Danfoss Bock, 2024)

As it can be seen from the piston compressor envelopes in Figure 4 above, R290 and R600a are two refrigerants with very different thermodynamic properties: R290 is classified as "Medium pressure" and operates well below condensing temperature of 85 °C and down to minimum evaporating temperature of -30 °C. On the other side, R600a is classified as a "low pressure" refrigerant and operates better at higher condensing temperature (typically up to 100 °C), but at a high evaporating temperature (> -10 °C). This last statement being directly related to boiling temperature of the refrigerant. All in all: Critical temperature sets the maximum condensing temperature so is a direct proxy of max supplied heat temperature at reach with the refrigerant. Boiling temperature at 1 bara is on the other hand a direct proxy of the minimum sink temperature. Typically, 1.3 bara is the minimum pressure compressor supplier keep as a reference for minimum evaporating temperature with volumetric compressor. The shortened top right corner for scroll compressor R600a envelope is related to motor tripping: at max power output and high evaporating temperature, the suction cooled scroll is limited in maximum current. On the other hand, the piston has higher efficiency and can traditionally be oversized in motor, letting it be more flexible in the maximum reachable envelope.

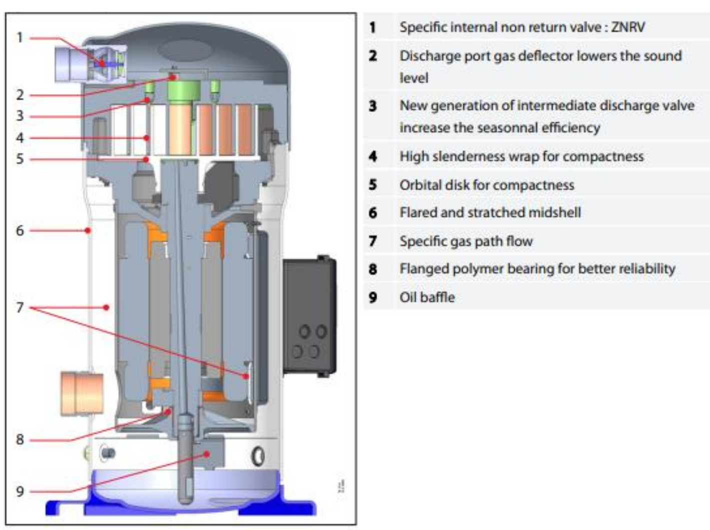

Figure 5: DSG compressor cut (base of the new PSG scroll)

Figure 5 above shows a cut of a DSG scroll compressor, which is base platform of the PSG scroll development, and share some of the same technology features. PSG has been specifically designed for High Temperature heat pump applications. Key drivers for the design are: lowering vibration and noise, improving reliability and ensuring the motor to be properly cooled down while operating at high suction temperature (~50 °C). Capacity target is in between 70-100kW per compressor (operating SST/SDT 20/90).

2.4. Cascade system with Semi-Hermetic piston compressors

The two cascade systems principle was shown earlier in Figure 2 are made out of three compressors on the 1st and three compressors on the second stage. the reference case with pistons reaches 300 kW heat supply assuming glycol loop -9 °C/-2 °C on first stage evaporator, and 90 °C supplied water on second stage. A sensitivity analysis is performed on the water supply side, trying to optimize the heating COP by adjusting the cascade HEX temperature. Approach on that heat exchanger is assumed to be 5 K. The results are depicted below and serve as a benchmark solution for the cascade system.

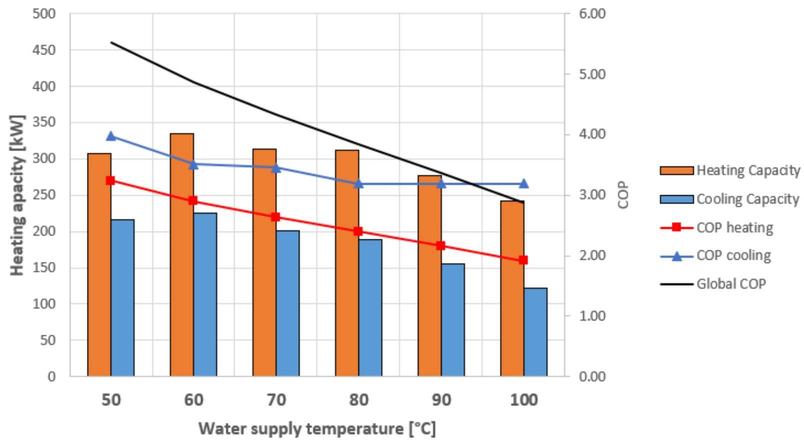

Figure 6: Cascade system with semi-hermetic piston compressors on each stage - Performance status

Simulation results in Figure 6 show that it is possible to keep heat supplied in between 240 kW to 320 kW, depending on the heat temperature delivered, while producing from 120 kW to 230 kW of cooling capacity at the same time on the glycol loop on the first stage evaporator. The degradation of the heating capacity at 100 °C supply is due to the fact one compressor on the first stage has to be switched off to keep the second stage compressor running within their prescribed envelope. As the heat supply temperature increases, the cascade HEX optimal temperature increases, while shifting the evaporating temperature of the second stage toward the maximum allowed temperature. At some point, second stage cannot handle extra power provided on the cascade HEX and the optimal configuration requires to switch off one compressor on the first stage (while still modulating on one of the two remaining). This is depicted on Figure 7 below:

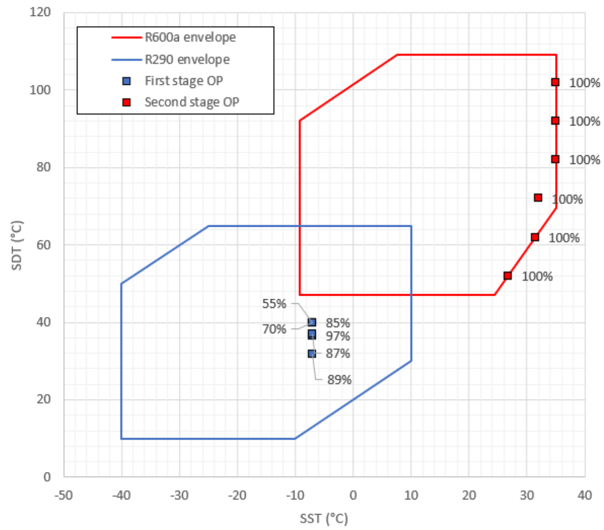

Figure 7: Operating points of the piston compressors and modulation ratio (labels indicate the modulation ratio of the compressors)

From Figure 7 it becomes clear that the more heat is needed on the second stage, the more convenient it is to get compressors able to use heat sink as high in temperature as possible. For heat supply at 80 °C and above, maximum evaporating temperature of the R600a stage should be above 35 °C to let the system find optimal balance. Oversizing of the motor is needed, while specific oil might also be required to ensure proper lubrication at very low viscosity level that are usually found in this saturated suction temperature (SST) range.

Finally, the lift split in between stage 1 and stage 2 is as follows:

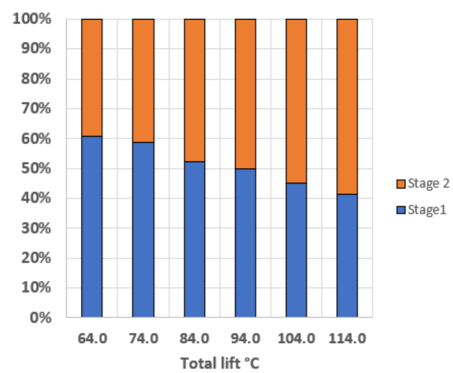

Figure 8 Lift split Stage1 / Stage2

Figure 8 shows the lift split between stage 1 and 2. The variation of lift depending on temperature supply is mainly on second stage while the first stage operates quite steadily. Lift provided on first stage is 40 K to 50 K for an ideal configuration while lift on second stage goes from 25 K to 70 K. The physics behind is directly related to the power output on the second stage: heating capacity is directly linked to the suction pressure level of the second stage. The higher the pressure on second stage, the higher the capacity delivered, and the higher the COP. This result is primarily due to the goal of the optimization which was to maximize the heating COP delivered while providing cooling, not the other way around. Interesting fact is that optimum case for heating COP and system efficiency of the cascade system (embedding Cooling and Heating) leads to the same conditions.

As a conclusion, on that type of cascade system with steady conditions at the boundary of the stage 1 evaporator, it seems relevant to note:

- Capacity modulation is relevant only on first stage

- It is important to get compressor with the highest evaporating temperature possible on stage 2 to maximize heating capacity and COP

- Optimization on the Heating COP leads to the optimum energy use of the system

2.5. Cascade system with scroll compressor

Equivalent system scrolls compressors has been simulated. Stage 1 is made out of a trio manifold of R290 compressors of 134 m^3/h displacement (theoretical compressors with adjusted displacement, to match the semis one). Stage 2 is made with equivalent displacement scrolls, designed for High Temperature heat pump application called PSG800 compressors (real compressors currently in development), assembled in trio configuration A simple COP optimization would lead to the following plot (equivalent to Figure 6):.

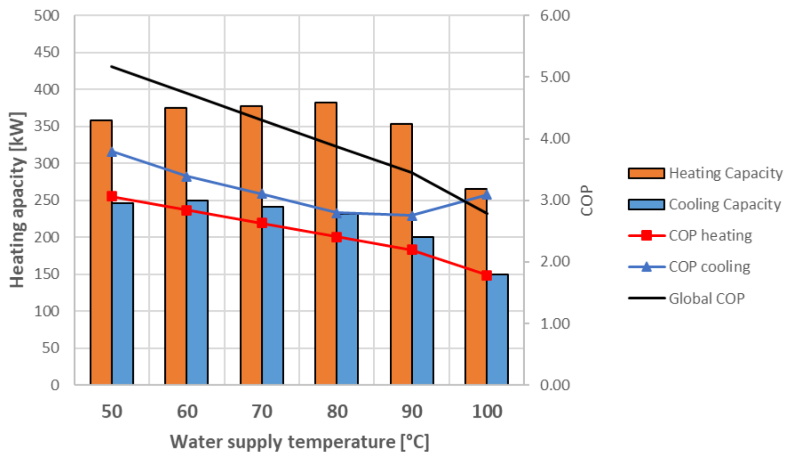

Figure 9: Performance scroll cascade system R290/R600a (3 compressors each stage)

Then, several optimizations are run:

- Optimization 1: Maximize heating capacity on the cascade system

- Optimization 2: Maximize COP while maintaining minimum 300 kW heating capacity output

Comparisons of the two optimization processes are summarized in figures 9 and 10 below:

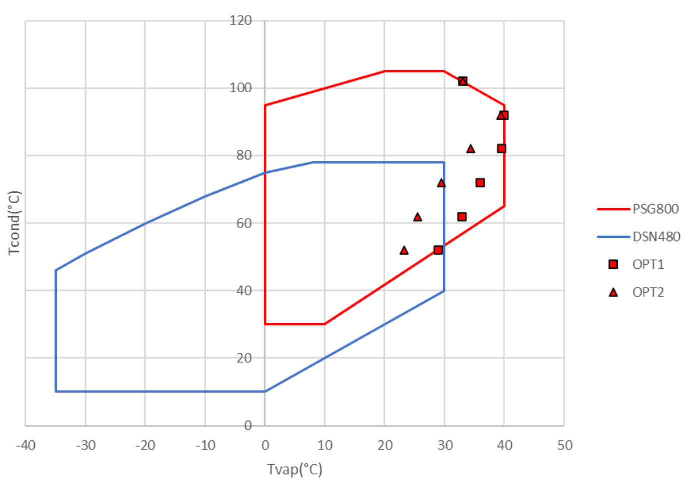

Figure 10: Operating points of the scroll compressors. OPT1 for maximizing Heating Capacity. OPT2 for optimizing COP under Heating capacity constraint.

Figure 10 shows that optimizing heating capacity leads to increasing cascade HEX temperature and working as much as possible at the maximum evaporating temperature the stage 2 can tolerate. At 100 °C output, the stage 2 compressor envelope prevents the system to reach the maximum capacity output.

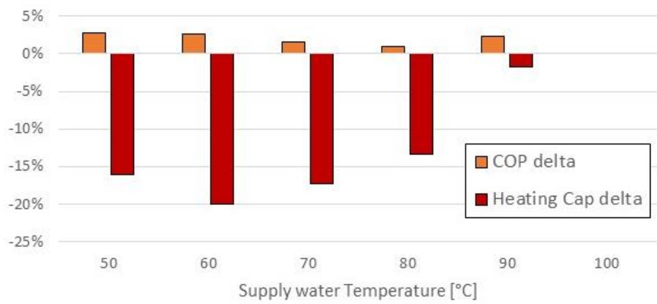

Figure 11: COP optimization vs heating capacity optimization

According to Figure 11, optimizing COP leads to ~2% improvement compared to a pure heating capacity improvement. Meanwhile, optimizing heating capacity on second stage leads to 15% to 20% difference in capacity output. There is no difference at 100°C output as both optimizations are limited by the compressor envelope restriction and cannot supply the 300kW output minimum required. Whether one would choose COP improvement over Heating Capacity improvement is a matter of how the machine will operate in its environment. For a given installed capacity (displacement then), COP improvement leads to a decrease in capacity of the unit. So, to improve the OPEX, one should increase the CAPEX of the unit. It is then a balance between OPEX & CAPEX, and then depends on the process and the final supplied water temperature. The brewery process requires two operations supplying heat below 75°C, while only one process supply heat up to 95°C. Then it would seem better optimize (economically) the system for the < 80°C supplied temperature. In that case, if you need to oversize the machine to get the best COP available, the amortization of the unit compared to Heating Capacity optimized one is in the range of 15 to 20 years (depending on the cost of energy). So it seems much more valuable, in that case, to favor heating capacity over COP.

In terms of modulation, there is no inverter on the scrolls. One scroll over 3 compressor is simply cycling on stage 1. On stage 2, like for the pistons, there is no need to cycle. Stage 2 can work at full load in all simulated configurations. The cutting of the top right corner of the scroll envelope is preventing substantial benefit when going above 90 °C heat supply, that shows how important that is for a compressor working in a cascade system like this to be able to cover the wider range possible, both in terms of evaporating, but also condensing. A specific effort in design (motor cooling) should open the door to an even better optimization. I

2.6. Final comparison Pistons vs Scrolls

Finally, even if theoretically, scrolls and pistons share the same displacement, there are some important differences to notice. Figure 11 below illustrates a comparison of the two heating capacity optimizations that has been ran on both type of compressors.

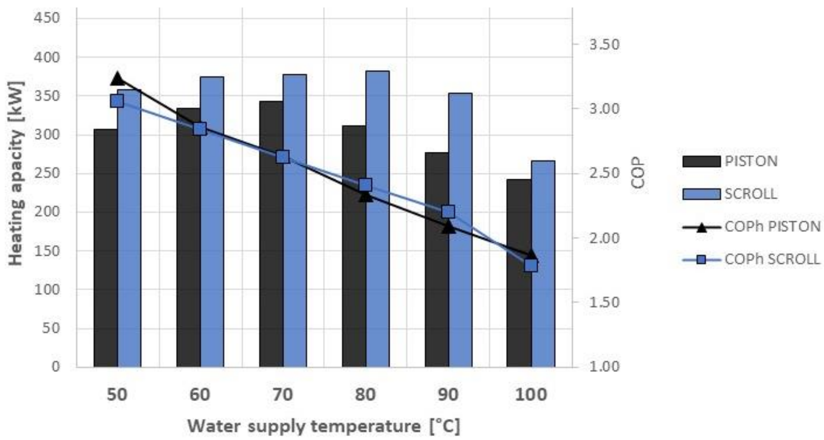

Figure 12: Heating capacity and COP comparison piston and scroll compressor; 3 compressors on each stage

There is no significant difference in the COPs. Scrolls and Pistons can compete within ±5% on the whole Heat supply range 50 °C to 100 °C. However, looking at the Heating Capacity output, there is a substantial difference. As introduced in 2.3, the reduction of volumetric efficiency as discharge pressure increases penalizes the piston: at 80 °C output, there is 23% output capacity deficit compared to the scrolls option, 27% at 90 °C output. As a result, the piston system has to be oversized in displacement to match the scroll one. First consequence is on applied costs of the solution: scroll solution will have a lower investment cost than piston based on kW heat output. Hence one can expect the return on investment to be significantly faster with scrolls than pistons, where heat supply remains below 100 °C. However, when it comes to service, semi-hermetic piston offers a value the hermetic scroll cannot yet match. In the end, both solutions are available, can meet the requirements of a medium size food & beverage processes and provide the industry with a set of solutions that can make its operations more profitable, while being more environmentally friendly.

3. CONCLUSIONS AND FUTURE WORK

There is a significant amount of applications in industrial processes which can be addressed with already proven and available heat pump technologies delivering hot water up to 100 °C. A low hanging fruit for decarbonization is Food & Beverage sector, which require both cooling and heating, such as Breweries, where the process side is cooled down and heated up multiple times during the brewing process. In fact, if certain challenges with integration and control of the heat pumps are overcome, most of the process heat needs in that application can be already met. In 2022 there were registered slightly below 9,000 microbreweries in Europe, where around 6,000 microbreweries solely in the EU (Statista, 2024). They require cooling and heating capacities below 1 MWth.

From the analysis in this paper, it becomes clear that with f.ex. Hydrocarbons or low-GWP refrigerants, these applications can be successfully addressed by a cascade for process cooling and heating using the traditional semi-hermetic piston compressors or a newly developed large scroll compressor solutions platform with compressor optimized for either lower or higher stage.

Key findings on this specific system in steady conditions:

- Capacity modulation is relevant only on the 2nd stage with either pistons or scrolls. In the case of scrolls, a trio manifold will match the capacity modulation of a piston equipped with an inverter.
- Total system COPs are very similar in both pistons and scrolls – system efficiency varies only within 5%
- Pistons compressors are more efficient when increasing Pressure ratio above 4:1.
- From installation cost perspective Scrolls have numerous advantages:
    - No oil separator

- No oil receiver
- Simple oil management (no active system)

- Choice of Piston or scrolls depends then on the mission profile of the system (maximum lift required, load factor, etc…), and ROI calculation, including commissioning, service and end of life management.

In future, the authors will focus their work on also adding a low pressure steam generating heat pump, studying ROI between different heat pump technologies, compressor development challenges for high temperature HPs.

4. ACKNOWLEDGEMENTS

The authors would like to extress their special thanks for continuous support and great collaboration to the teams of Rodrigo Link (Danfoss scroll compressors) and Manuel Froschle (Danfoss Bock piston compressors).

NOMENCLATURE

COP	Coefficient of performance	HP	Heat pump
NH_3	Ammonia (R717)	IRF	Industrial refrigeration system
R600a	Isobutane	F&B	Food and Beverage
R290	Propane	HEX	Heat exchanger
TCO	Total costs of ownership	SST	Saturated suction temperature
ROI	Return of investment	SDT	Saturated discharge temperature

REFERENCES

Arpagaus, C. (2024, 15 03). *High Temperature Heat Pumps Update.* Retrieved from https://www.a2ep.org.au/post/6-mar-2024-high-temperature-heat-pumps-update-webinar

Boer, R. d., Marina, A., Zühlsdorf, B., Arpagaus, C., Bantle, M., Wilk, V., . . . Benson, J. (2020). *Strengthening Industrial Heat Pump Innovation: Decarbonizing Industrial Heat.*

Danfoss Bock. (2024, January 18). Retrieved from Danfoss Bock seme-hermetic compressors for stationary applications: https://vap.bock.de/stationaryapplication/Pages/Index.aspx

EU-Council. (2023, August 01). *European Green Deal*. Retrieved from https://www.consilium.europa.eu/en/: https://www.consilium.europa.eu/en/policies/green-deal/

IEA HPT. (2023). *High-Temperature Heat Pumps.* Borås: Heat Pump Centre c/o RISE – Research Institutes of Sweden .

Lund, T., Skovrup, M. J., & Holst, M. (2019). Comparing energy consumption and life cycle cost of industrial size refrigeraton systems. *Proceedings of the 25th IIR International Congress of Refrigeration.* Montreal: IRC.

Pachai, A., Hafner, A., & Arpagaus, C. (2023). High-temperature working fluids for heat pumps –. *10th Conference on Ammonia and CO2* (pp. 62-71). Ohrid: International Institut of Refrigeration.

Rangelov, I., & Lund, T. (2024). High Temperature Heat Pump mapping for Industrial applications. *High Temperature Heat Pump Symposium.* Copenhagen: DTI.

Reinholdt, L. (2023). TECHNICAL BUSINESS CASE BEHIND. *Køle- og varmepumpeforum 2023* (pp. 92-100). Copenhagen: DTI.

Statista. (2024, 04 23). *https://www.statista.com/statistics/757816/number-of-microbreweries-in-europe/*. Retrieved from https://www.statista.com/statistics/757816/number-of-microbreweries-in-europe/