Energy consumption of **industrial-size refrigeration systems**

In this paper, based on the original work by Thomas Lund, Morten Juel Skovrup, and Mads Holst, we are analyzing and comparing the energy consumption of common industrial size systems:

1. Transcritical R744–direct expansion (DX)
2. Transcritical R744–pumped
3. Two-stage R717–pumped
4. R744/R717 cascade–pumped
5. Two-stage R507–pumped

The study includes different variations in system design and ambient conditions, with each system tested in four different sizes, using both a 'standard' and an 'optimized' approach. The comparison is based on simulation models, including real compressor data, complex heat exchanger models, and real ambient data.

These optimizations include parallel compression, hybrid coolers, ejectors, and economizers where applicable. Additionally, we have conducted calculations across various climate conditions, ranging from cold to hot climates.

Since the comparison results depend on the inputs to the simulation software, this paper offers an overview of how the choices made in the simulation software impact the overall results.

1. Introduction to data collection methods

Much has already been said when it comes to industrial refrigeration and the various systems available. Here, data is key, and with the figures below, our aim is to provide reliable data that can translate to qualified business decisions.

However, while it is essential to compare operating data from actual systems in operation, it is important to note that data collected in this manner is susceptible to various factors that can make it less reliable.

These influencing factors include operational variations, the allocation of low-temperature (LT) and medium-temperature (MT) loads, ambient conditions, as well as the design, construction, and maintenance of the systems.

Simulation software (specifically, Pack Calculation Pro ver 4.20 by IPU) was used to collect the data to reflect a real-world environment to mitigate these factors. This includes compressor efficiencies, ambient data, complex condenser/gas cooler models, and control strategies, which are used to paint a more robust picture of the difference between system types under uniform conditions and constraints, ensuring equitable consideration of their advantages and disadvantages.

2.
Calculation parameters

All five systems are calculated in both a 'standard' and an 'optimized' configuration.

Types of systems explored:

R744 TC DX: Transcritical R744, DX operation
R744 TC FL: Transcritical R744, flooded operation
R717 2ST: Two-stage R717, flooded operation
R744/R717: R744/R717 cascade, flooded operation
R507 2ST: Two-stage R507, flooded operation

Transcritical configurations
When it comes to transcritical systems, the 'standard' configuration entails that the system is equipped with a simple booster unit, as this was the system originally quoted by the supplier. On the other hand, the 'optimized' systems incorporate hybrid (adiabatic) gas coolers, parallel compression (DX only), and gas ejectors.

The remaining system types
For the remaining systems, the distinction between 'standard' and 'optimized' configurations is primarily related to the utilization of economizers and a reduced temperature difference in the cascade cooler.

While some might assume that the optimization process would involve reduced temperature differences in air coolers and condensers/gas coolers, we have gone in a different direction, as this alteration can be done across all system types with minimal impact. Instead, we opted to size heat exchangers uniformly.

Temperature range and region
To accurately reflect real-world scenarios, ambient condition data collected from Oslo, Frankfurt, and Rome are used, providing us with insights into performance throughout a range of colder and warmer climates.

Table 1. Mean ambient temperatures for chosen locations

	DRY BULB TEMPERATURE			WET BULB TEMPERATURE		
	Min	Max	Average	Min	Max	Average
Rome	-4.0°C	31.8°C	15.8°C	-6.0°C	25.8°C	13.4°C
Frankfurt	-8.9°C	33.6°C	10.1°C	-9.3°C	22.4°C	7.7°C
Oslo	-17.0°C	28.2°C	6.7°C	-17.2°C	20.5°C	4.4°C

Cooling loads–the common denominator
To ensure consistency, the duty delivered across all systems is the same and split into a low temperature (LT) and a medium temperature (MT) load.

LT cooling load delivered at an air temperature of -2°C
MT cooling load delivered at an air temperature of -25°C

In the bulk of the calculations, the MT cooling load is three times that of the LT cooling load. However, the measures of the reverse split, e.g., LT cooling load = 3 MT cooling loads, have been performed in a few of the cases to highlight the significance of the LT/MT split.

Initially, the system types are calculated in 4 different sizes, with LT/MT cooling loads at:
- 50/150kW
- 150/450kW
- 300/900kW
- 900/2700kW

While the simulation software allows for variable cooling load simulations, we have set a constant cooling load to avoid obscuring the inherent behavior of the refrigeration system, which can occur when introducing any functionality dependent on time and/or ambient temperatures.

Variable load profile

Typically, two distinct mechanisms are employed to create a variable load profile: changes in ambient temperature and production cycles. In many production cycles, full production occurs during the day, with reduced activity at night. Consequently, whether driven by ambient temperature or production cycles, the system's capacity is decreased during colder periods of the day or year.

Maintaining a constant capacity results in the refrigeration systems operating more frequently during colder periods. Here, transcritical R744 systems shine as its efficiency loss at high temperatures is greater compared to other systems.

With the room air temperatures fixed at -2°C and -25°C for MT and LT, the evaporating temperatures have been derived from the following assumptions:

- Flooded evaporator temperature difference to air inlet at 8K
- DX evaporator temperature difference to air inlet at 10K with 7K superheat
- Suction line pressure loss for R717 and R507 at 1K
- Suction line pressure loss for R744 at 0K

Note: It is worth noting that DX evaporators exhibit a larger temperature difference to air inlet than flooded evaporators. This disparity stems from the necessity to generate superheat. Based on discussions with industry professionals, the lowest safe superheat value is set to 7K, as the additional 2K temperature difference represents a rather moderate penalty for running DX.

Condensers and gas coolers

Condensers/gas coolers were sized according to the following:

- **Evaporative condensers**
  Temperature difference to wet bulb temperature is 7K at maximum design capacity and at maximum wet bulb temperature in the actual location.

- **Dry gas coolers**
  2K temperature outlet temperature difference to dry bulb in transcritical operation. In subcritical operation, a temperature difference of 5K to dry bulb is used for condensing at 25°C. Subsequent talks with people who have measured gas coolers in subcritical operations have revealed that this is very generous, as the observed temperature difference is much higher.

- **Hybrid gas coolers**
  Same temperature differences as for dry gas coolers, but with a 75% adiabatic temperature efficiency applied.

- **Cascade coolers**
  Initially calculated with a temperature difference of 5K, which has been reduced to 3K for the optimized systems.

Gas coolers have been designed with a relatively low-temperature difference in subcritical operation because they are running at less than the maximum load.

Compressor selection

Compressors for the transcritical systems were selected by a supplier and are commercially available from major manufacturers. The simulation software provided the performance correlations for R744 compressors, initially sourced from a supplier's software. For the R717, R507, and R744 compressors in the cascade system, their performance data was obtained from supplier calculation programs and integrated into the simulation software.

Table 2. Selected compressor types

Load	R744 TC DX	R744 TC FL	R717 2ST	R744/R717	R507 2ST
50/150	Recip/Recip	Recip/Recip	Recip/Recip	Recip/Recip	Recip/Recip
150/450	Recip/Recip	Recip/Recip	Recip/Recip	Recip/Recip	Recip/Screw
300/900	Recip/Recip	Recip/Recip	Screw/Screw	Recip/Screw	Screw/Screw
900/2700	Recip/Recip	Recip/Recip	Screw/Screw	Recip/Screw	Screw/Screw

Considering heat recovery

Finally, heat recovery has been left out of this study. The simulation program primarily focuses on system control to deliver the desired cooling capacity. Incorporating heat recovery could result in a suboptimal refrigeration cycle, hindering any economic benefits.

In addition, the varying constraints on heat recovery systems, including temperature ranges and required capacities, differ significantly among different installations. Accounting for these variations would introduce a multitude of complexities that could obscure the results rather than enhance clarity.

3.
Standard system calculations

Initially, calculations encompassed all system sizes and types in all locations, yielding a substantial volume of data. The primary metric is the annual power usage, measured in MWh, and serves as the core result, as seen in Table 3. Additionally, the associated CO values are presented in Table 4.

Table 3. Yearly power consumption

	Load (LT/MT in kW)	R744 TC DX MWh	R744 TC FL MWh	R717 2ST MWh	R744/R717 MWh	R507 2ST MWh
Rome	50/150	781	705	500	535	518
	150/450	2079	1948	1503	1650	1595
	300/900	4181	3926	2891	3150	3140
	900/2700	12608	11966	8542	9424	9400
Frankfurt	50/150	613	561	448	485	459
	150/450	1621	1494	1344	1490	1377
	300/900	3265	3046	2575	2820	2723
	900/2700	9833	9301	7597	8397	8169
Oslo	50/150	544	507	428	463	439
	150/450	1432	1313	1284	1431	1302
	300/900	2884	2674	2457	2699	2579
	900/2700	8691	8179	7243	8035	7242

Table 4 →

Table 4. Yearly COP

COP	Load (LT/MT in kW)	R744 TC DX	R744 TC FL	R717 2ST	R744/R717	R507 2ST
Rome	50/150	2,24	2,49	3,50	3,28	3,38
	150/450	2,53	2,70	3,50	3,18	3,30
	300/900	2,51	2,68	3,64	3,34	3,35
	900/2700	2,50	2,64	3,69	3,35	3,35
Frankfurt	50/150	2,86	3,12	3,91	3,61	3,81
	150/450	3,24	3,52	3,91	3,53	3,82
	300/900	3,22	3,45	4,08	3,73	3,86
	900/2700	3,21	3,39	4,15	3,76	3,86
Oslo	50/150	3,22	3,46	4,09	3,78	3,99
	150/450	3,67	4,00	4,09	3,67	4,04
	300/900	3,64	3,93	4,28	3,90	4,08
	900/2700	3,63	3,86	4,35	3,92	4,35

Key observations

At first glance, these COP reveal several trends. For one, the COP in transcritical systems remains relatively constant across different sizes, except 50/150kW. This variation could be attributed to an unfortunate compressor type selection made by the supplier, making the three larger sizes the only representative examples in the transcritical category.

As for the remaining systems, there is a slight upward trend in COP as system size increases, although not substantial. Consequently, the focus was shifted onto the 300/900kW system exclusively to reduce the workload and improve clarity.

Another observation is that in all situations, the two-stage R717 system delivers the best COP. However, a few points merit consideration regarding this result.

The LT room temperature of -25°C results in evaporating temperatures of -33°C in the R744/R717 cascade system. This is only on the fringe of where a cascade system starts to become competitive compared to a two-stage R717 system. As the evaporating temperature decreases, for instance, with more heavy-duty freezing applications, the cascade system eventually surpasses the efficiency of a two-stage R717 system.

Regarding the R507 system, it is included primarily for reference, but it closely trails the two-stage R717 and cascade system in terms of efficiency, although slightly behind.

Nevertheless, as the main focus is energy efficiency, it is more sensible to look at the annual COP average, using the total delivered cooling effect from the individual system sizes.

Power consumption

When examining Table 5, the 300/900kW systems in colder climates, e.g., Oslo, the transcritical systems come close to the efficiency of the two-stage R717, with the flooded version being 9% higher in power consumption and the DX version approximately 17% higher.

However, in warmer climates, e.g., Rome, the transcritical systems exhibit far higher power consumption at 32% and 45%. Clearly, the transcritical systems face challenges in warmer climates, which is to be expected. When comparing them to Oslo as the reference point, it becomes evident how these systems respond to changes in ambient temperature.

Table 5. Power consumption relative to two-stage R717

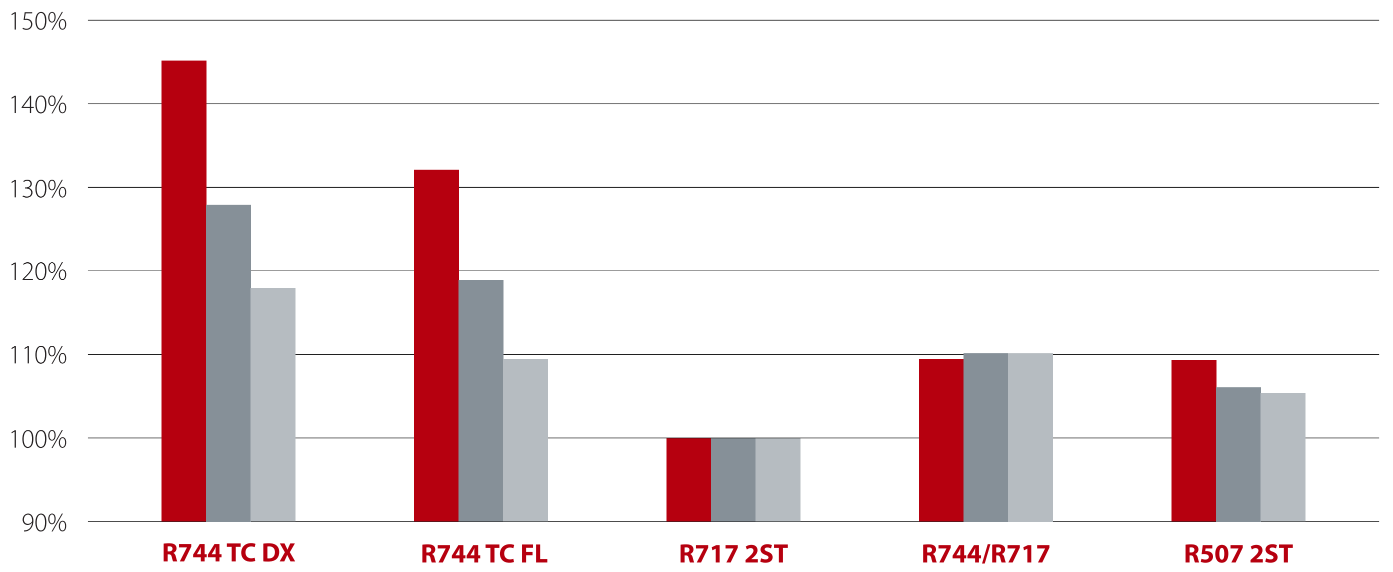
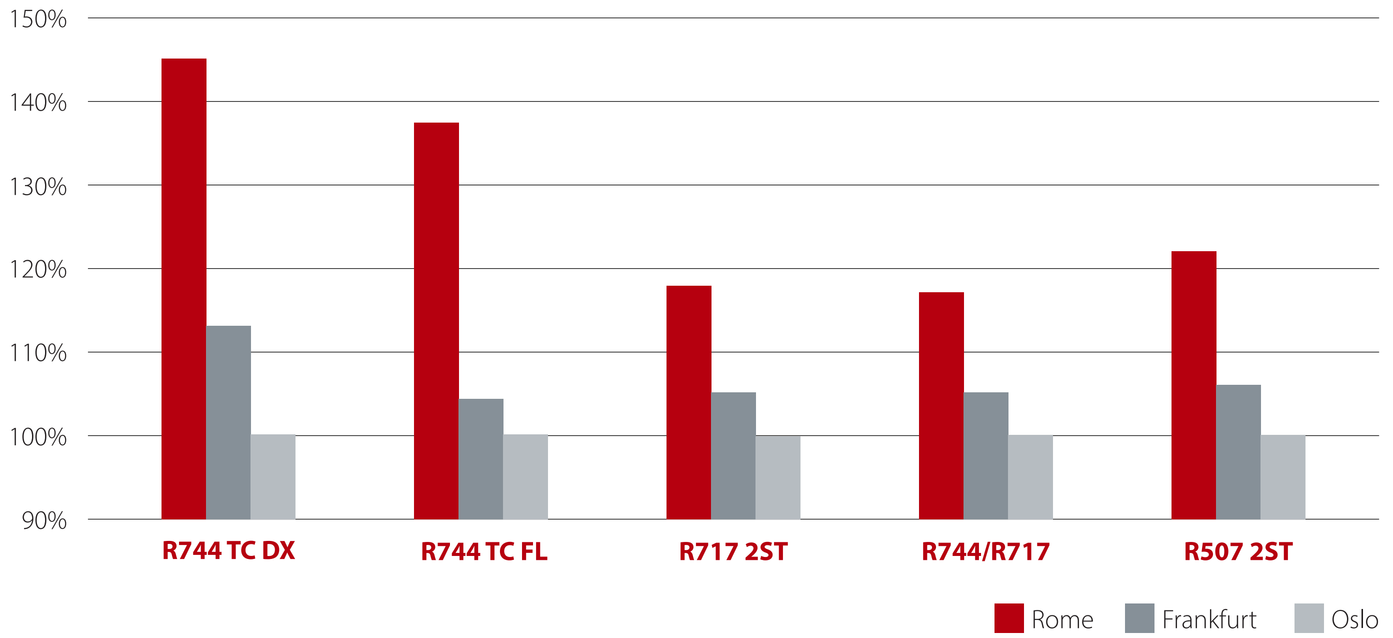


Table 6. Power consumption relative to Oslo

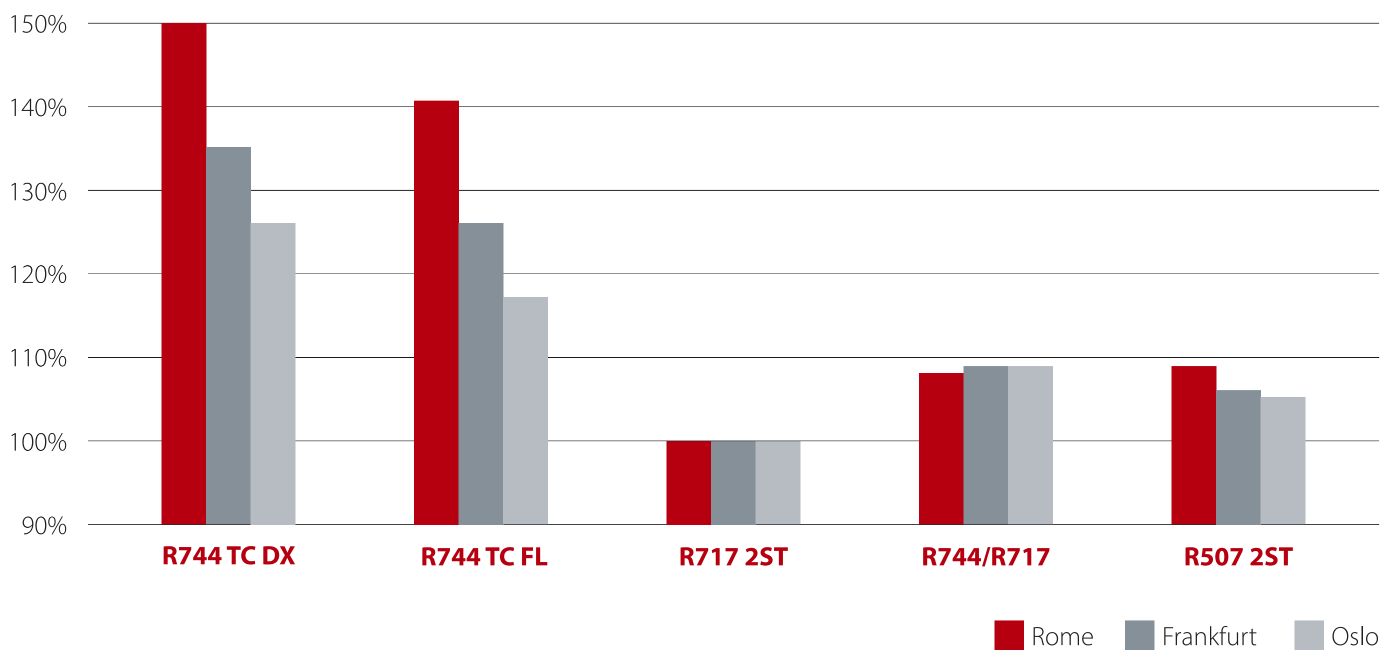


The influence of load profile

As previously mentioned, the load profile was a constant in these calculations. And looking at the relative power consumption compared to two-stage R717 in Table 7, two-stage R717 was again the most efficient.

Nevertheless, a sample load profile was used within the simulation program to provide further insight into how the load profile impacts the results. Load profiles can have a variety of forms, but in this sample, the cooling load was adjusted to vary from our baseline capacity at maximum dry bulb temperature at the given location down to 35% of this at a temperature equivalent to the room temperature. The variation measured was linear, and the delivered cooling effect remained constant across system types but varied with location.

Table 7. Power consumption relative to two-stage R717- varying load profile



Generally, the transcritical systems benefitted in the region of 5 to 10% from the constant load profile, although this depends on the actual configuration of the load profile. The individual systems' response to colder climates (Oslo) was slightly better than warmer climates (Rome), with 4 to 5% lower energy consumption with the constant load profile.

LT and MT loads explored

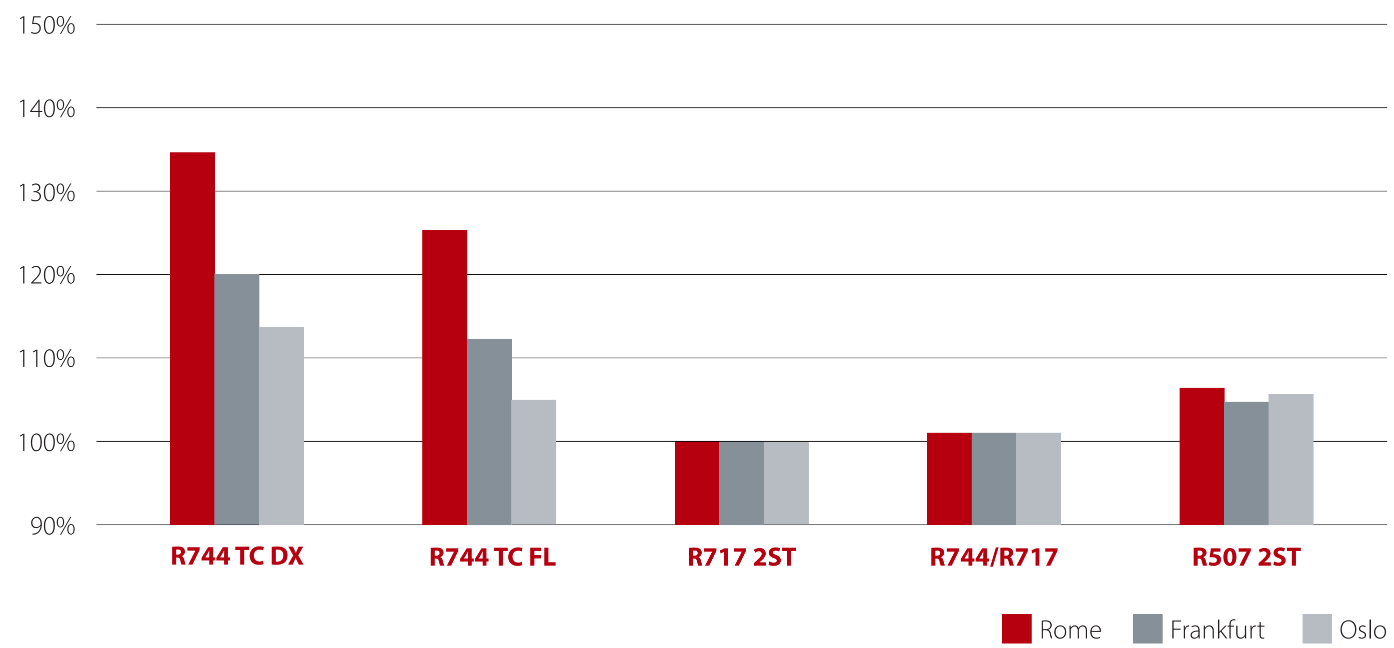
Finally, the split between LT and MT load was examined using the initial constant load profile. The recalculation of the 300/900kW system to 900/300kW provided the following power consumption in MWh, as demonstrated in Table 8.

Again, as shown in Table 9, the two-stage R717 was the most efficient, although the cascade system came in close.

Table 8. Yearly power consumption - reverse LT / MT split

	R744 TC DX	R744 TC FL	R717 2ST	R744/R717	R507 2ST
Rome	5616	5242	4182	4242	4446
Frankfurt	4610	4282	3835	3886	3989
Oslo	4193	3878	3705	3755	3831

Table 9. Power consumption relative to two-stage R717 - reverse LT / MT split



Although the transcritical system imporved performance, the picture remains much the same. The gain seen in the transcritical systems can be attributed to the LT compressors playing a larger role in the total consumption, with the superior COP of the R744 compressors in LT operation accounting for this shift.

4.
Optimized system calculations

As we now start to explore the findings for the optimized systems, it's important first to clarify the distinctions between an optimized system and a standard one in this study. Additionally, we'll address some key considerations to keep in mind before delving into the observations and results.

The changes from standard systems (only 300/900kW)

- Transcritical R744 DX systems: Added parallel compression, hybrid coolers, and gas ejectors.
- Transcritical R744 flooded systems: Added hybrid coolers and gas ejectors.
- Two stage R717 systems: Economizers added to screw compressors.
- R744/R717 cascade systems: Economizers added to screw compressors, cascade temperature difference reduced from 5K to 3K.
- Two stage R507 systems: Economizers added to screw compressors.

Considerations for optimized systems

Parallel compression is a feature of the simulation software, primarily utilizing MT compressor types.

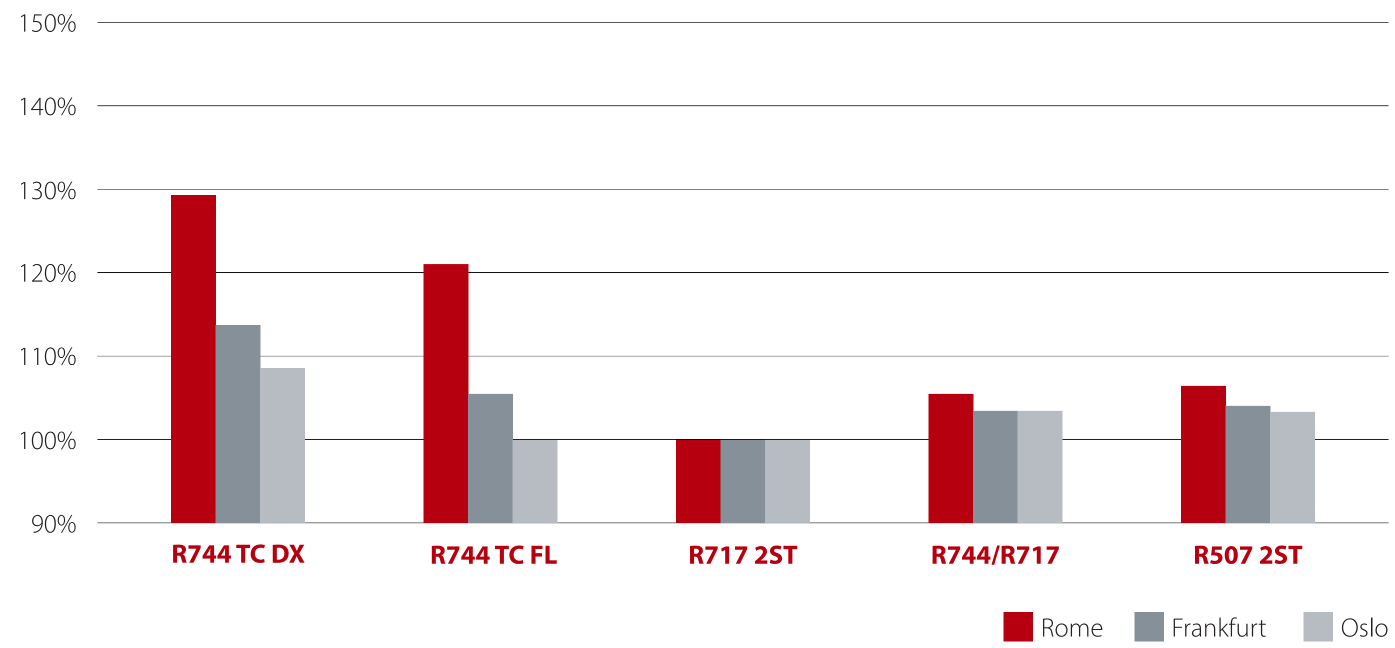
1. Hybrid coolers, also a feature in the simulation software, were dimensioned as evaporative condensers, using an air temperature attained with 75% efficiency of the adiabatic process.
2. Based on discussions with commercial refrigeration experts, the general rule of a 7% reduction in MT compressor power when the gas cooler outlet exceeded 22°C was deducted. The calculation output provides an hour-by-hour tabulation of system operations, with data rows reflecting reduced MT compressor power consumption when the gas cooler outlet exceeded 22°C, resulting in a new annual sum. Gas ejectors are not a feature of the simulation software.
3. Economizers are not a feature of the simulation software. Compressors were calculated under actual operating conditions and then compared with and without open economizers. The resulting COP gain was utilized to reduce compressor power consumption.
4. The cascade temperature difference, an adjustable parameter in the software, was modified.
5. The calculation software does not allow calculations with the combination of flooded R744 transcritical system and parallel compression. Hence, the same improvement in percent as found in the DX system, has been applied to the flooded transcritical system.
6. In Table 10, the resulting power consumption is presented in MWh.
7. Table 11 uses the two-stage R717 system as the reference point for data evaluation.

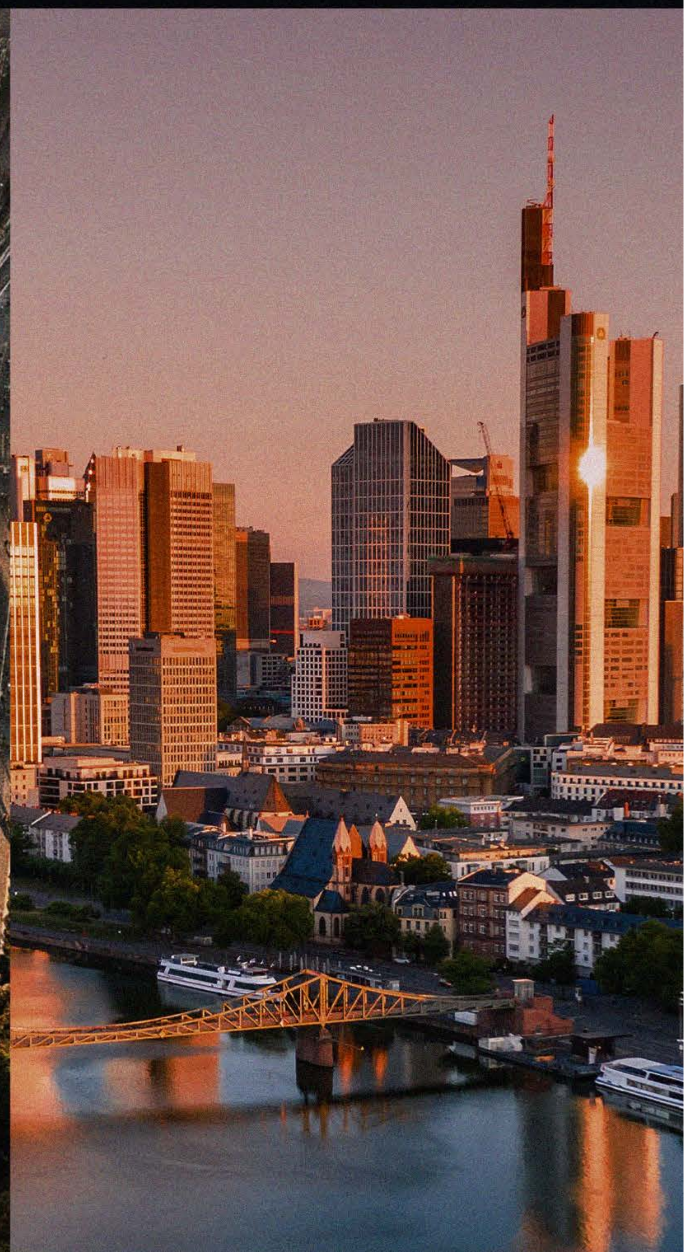
Table 10. Yearly power consumption. Optimized systems

	R744 TC DX	R744 TC FL	R717 2ST	R744/R717	R507 2ST
Rome	3578	3362	2778	2915	2933
Frankfurt	2818	2622	2494	2578	2582
Oslo	2565	2376	2384	2456	2455

Again, in relative terms with two stage R717 as a reference

Table 11. Power consumption relative to two-stage R717. Optimized systems - TC FL approximated





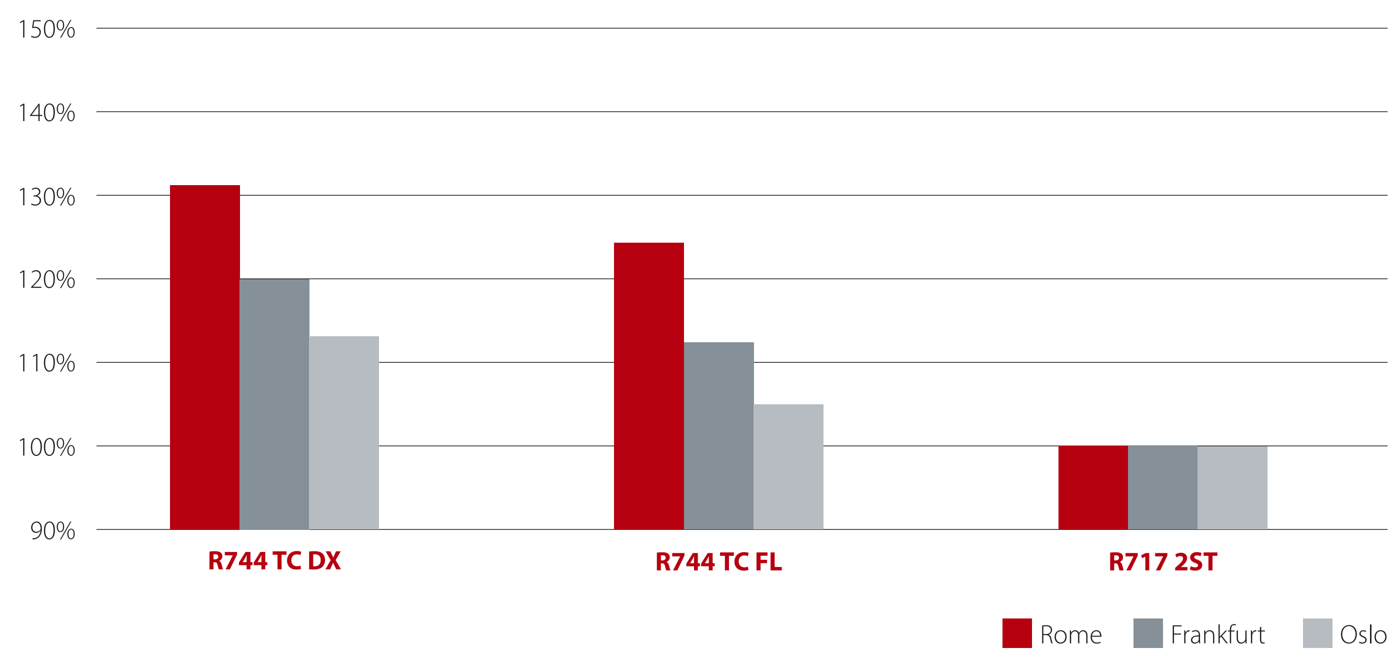
It's clear that transcritical systems show more significant gains compared to traditional systems, helping narrow the gap between the two.

As a variation of this result, the transcritical systems were calculated without hybrid coolers while employing all other optimizations. This decision was based on two considerations:

- First, the operating costs of a hybrid cooler are significantly higher than those of a dry gas cooler.
- Secondly, lowering the condensing/gas cooler outlet temperature reduced the duration where the ejector was active.

Although the economic calculations did not yield clear results, the full advantage of not running hybrid coolers may not be immediately apparent. Nevertheless, the calculated power relative to the R717 two-stage system is presented in the table and figure below.

Table 12. Power consumption relative to two-stage R717. Optimized systems - dry cooler



5.

Concluding thoughts

When analyzing the efficiency data, it is evident that no matter how much a transcritical R744 system is optimized, it cannot compete with a two-stage R717 system. However, it does come relatively close in colder climates. In general, a transcritical R744 system exhibits a 5 to 10% higher energy consumption in colder climates, which increases to around 30% higher in relatively warmer climates.

Cascade systems typically exhibit energy consumption approximately 5% higher than that of the two-stage R717 system. However, it's important to note that the temperature range investigated in this study does not favor the cascade system. In systems with a lower evaporating temperature on the LT (low-temperature) side, the cascade system may eventually become more efficient as R717's efficiency declines rapidly. The two-stage R507 system, included primarily for comparison purposes, generally consumes more power than the two-stage R717, and in the best-case scenarios for transcritical R744, it is roughly equivalent in power consumption.

In the past decade, there have been remarkable advancements in transcritical system efficiency, which is quite impressive. Simultaneously, the R717 (ammonia) proponents have placed significant emphasis on reducing charge, as this is a challenge associated with R717 systems. Consequently, the landscape is continually evolving, and we hope this paper will clarify the current status of at least certain aspects of this dynamic field.

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