Two-Stage Compression With Vapor Injection: A Study On a Disregarded Solution in CO_2 Booster Applications for Supermarket Refrigeration

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Introduction

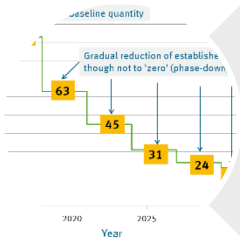
Supermarkets represent 3-4% of the total energy consumption in developed countries. [1] [2]

High specific energy demand $300 - 600\ kWh \cdot m^2$ per year , approximately between 35-50% of the electricity is required to run refrigerating equipment. [3]

In spite of enormous $GWP_{100\,years} = 3943$, R404a is still widely used in the European sector, with estimated annual leak charge of 15-20%. R22 is still the most employed working fluid in commercial units, with annual leakages of about 30%. [4]

40% of refrigerant green house gases come from the food retail sector, specially from supermarkets. [5]

Introduction



Rediscovery of CO_2 is consequence of EU F-Gas Regulation 517/2014 which seeks to lower considerably average GWP.

Being $\boldsymbol{CO_2}$**(R744) recognized as the most promising working fluid for supermarket applications**, commercial transcritical R744 refrigeration systems have emerged as leading hydrofluorocarbon (HFC)-free technologies.

R744 is not prone to be phased out if further restrictions come.

State of the Art – Current trends

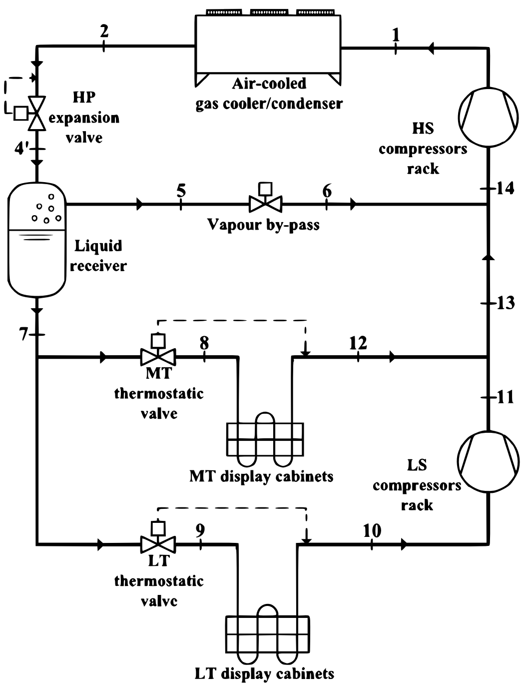


Figure: Booster System (1st gen.). [9]

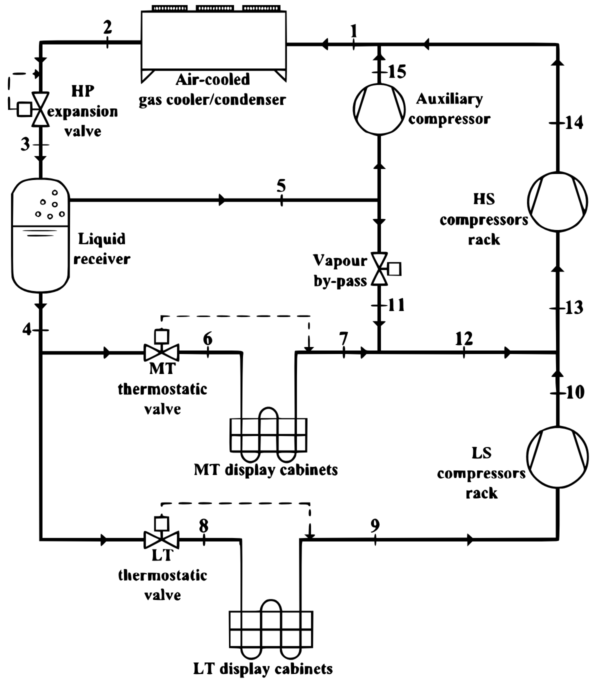


Figure: Parallel Compression System (2nd gen.). [10]

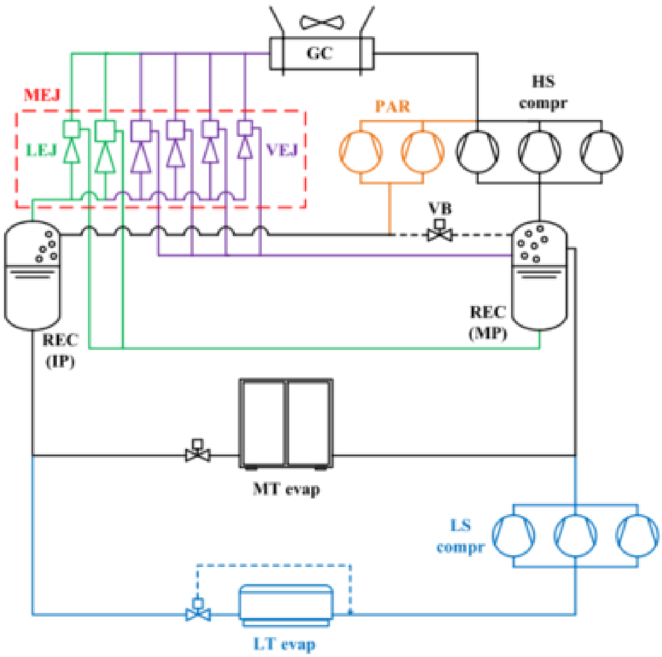


Figure: Multi-ejector System (3rd gen.) [11]

Knowledge gap – 2-Stage compression solutions

- Two-Stage rotary compressors already exist in the market for CO2 application.
- Naturally, the technology allows for 3 main pressure levels.
- A **vapor injection line can be added in the intermediate pressure**.

Some literature is published studying:

- Volume ratio optimization (between 1st and 2nd stage of compression).
- Integration with mechanical subcooling.
- Possible combination with expanders.

But:

- **No integration studied in a Booster-like system architecture .**
- **No design and operational optimization** studied in this configuration (i.e, flash tank pressure influence in a SCVI layout).
- **No direct comparison with existing developments** of R744 systems has been performed (1st, 2nd and 3rd generation)

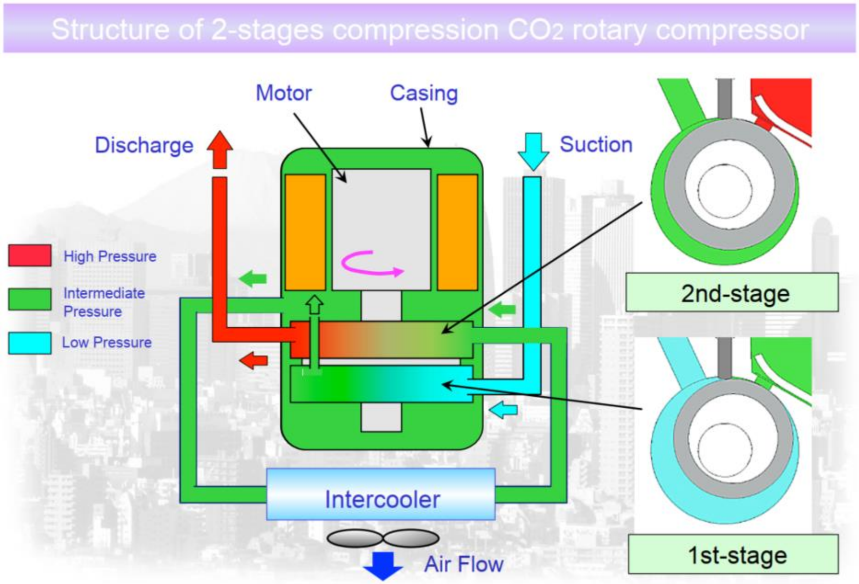


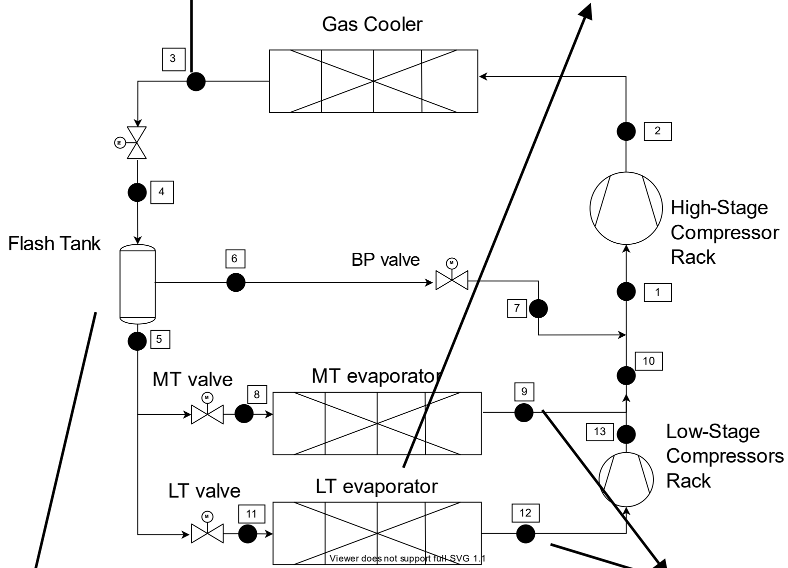
Figure: Two-Stage rotary compressor didactic scheme. Casing at medium pressure. [12]

STUDIED SYSTEM ARCHITECTURES MODELS

System Architectures

Constant temperature approach considered

Constant evaporation temperatures considered



Constant superheat

Figure: Booster Layout (1st gen)

Ideal gas-liquid separation

Constant efficiency

BP valve is considered closed

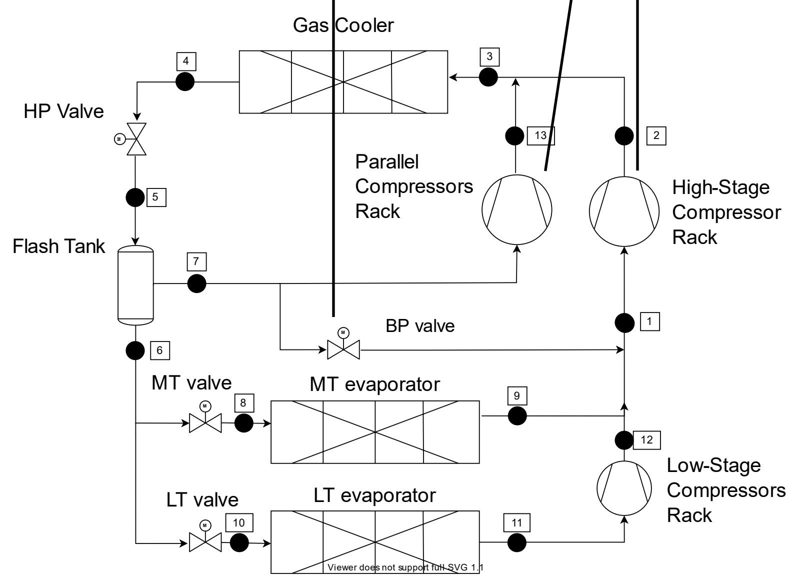


Figure: Parallel Compression Layout (2nd gen)

System Architectures

BP valve is considered closed

Constant saturation temperature lift

Ideal gas-liquid separation

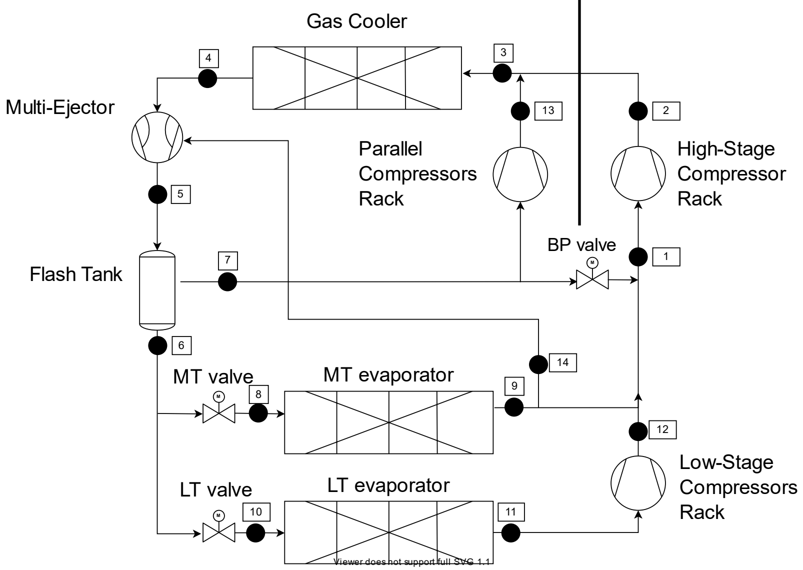


Figure: Gas Ejector - Parallel Compression (GEPC) Layout (3rd gen)

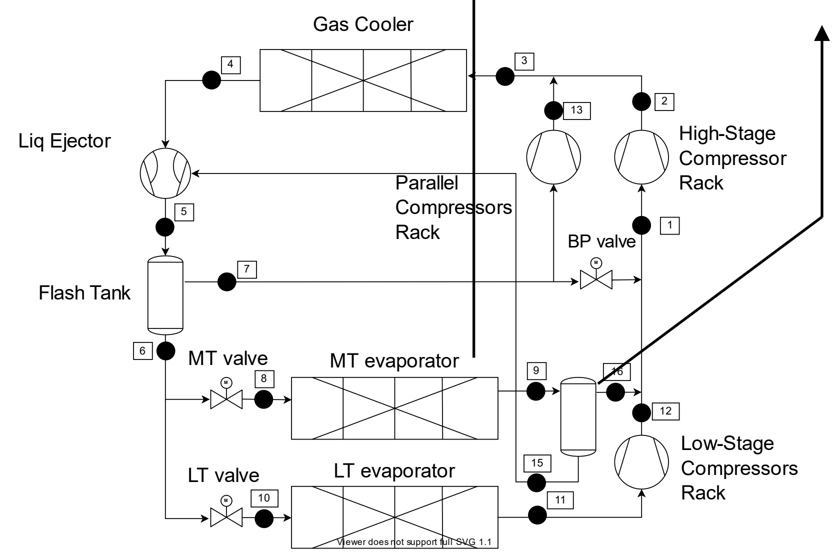


Figure: Liquid Ejector – Parallel Compression (LEPC) Layout (3rd gen)

System Architectures

No injection valve considered

Floating pressure

Same rotating speed for both stages

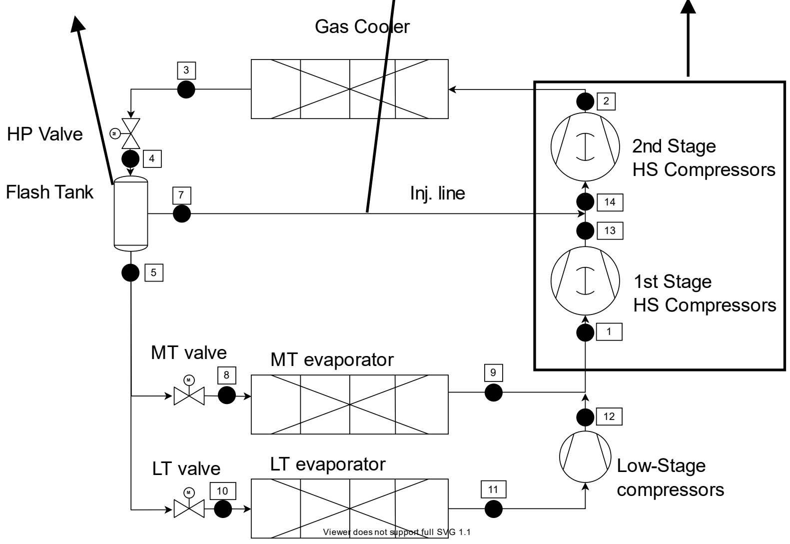


Figure: Flash Tank Vapor Injection (FTVI) Layout

Constant superheat at Economizer/ Sub-Cooler exit

BP valve to control Flash Tank Pressure

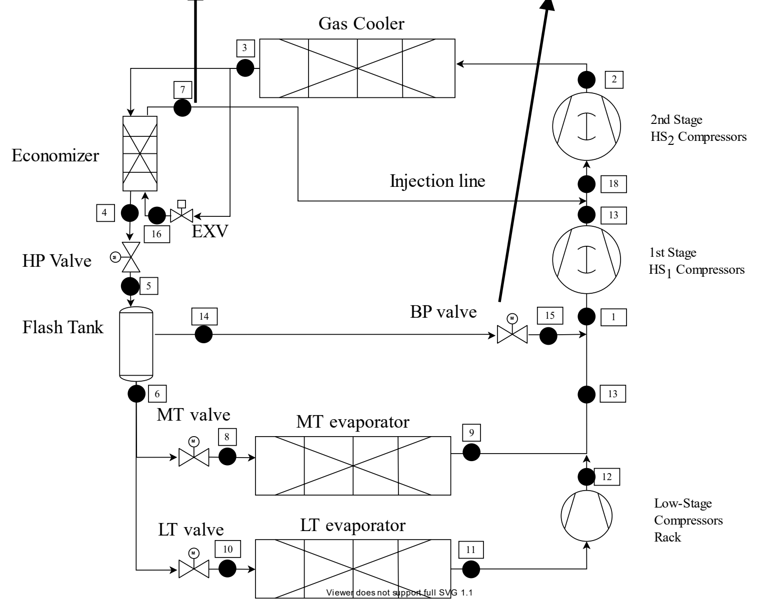


Figure: Sub-Cooled Vapor Injection (SCVI) Layout

MODELLING ASSUMPTIONS

Modelling assumptions

Table: Baseline system variables and parameters

$T_{ev,MT}$	SH_{MT}	$T_{ev,MT,liqEj}$	$T_{ev,LT}$	SH_{LT}	SC_{GC}[a]	$\varepsilon_{s,HS}, \varepsilon_{s,PC}$	$\varepsilon_{s,LS}$	$\eta_{ej,gas}$	$\eta_{ej,liq}$
-10 °C	10 K	-7 °C	-35 °C	8 K	3 K	0.6	0.7	0.2	0.02

[a] Sub-cooling of the gas cooler when acting as a condenser

$$\varphi_{ej} = \frac{\dot{m}_{suc}}{\dot{m}_{mot}}$$

$$\eta_{ej} = \varphi_{ej} * \frac{\dot{w}_{rec}}{\dot{w}_{pot}}$$

$$COP = \frac{\dot{Q}_{MT} + \dot{Q}_{LT}}{\dot{W}_{compressors}}$$

$$V_r = \frac{V_{displ,HS,2}}{V_{displ,HS,1}}$$

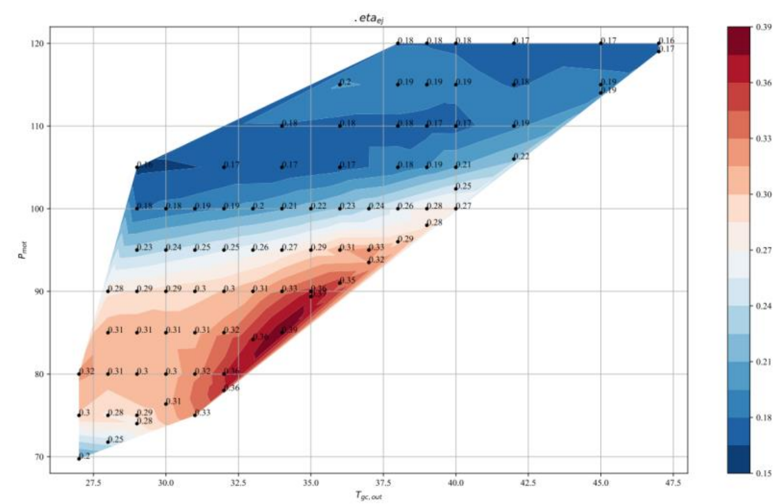


Figure: Ejector efficiency envelope for $P_{FT} = 35\ bar$

Systems optimization

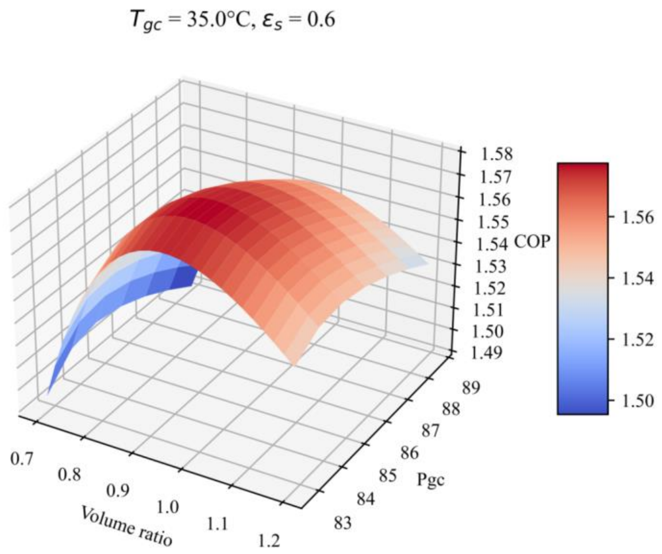


Figure: FTVI system COP optimization by P_{GC} and V_r parametric analysis

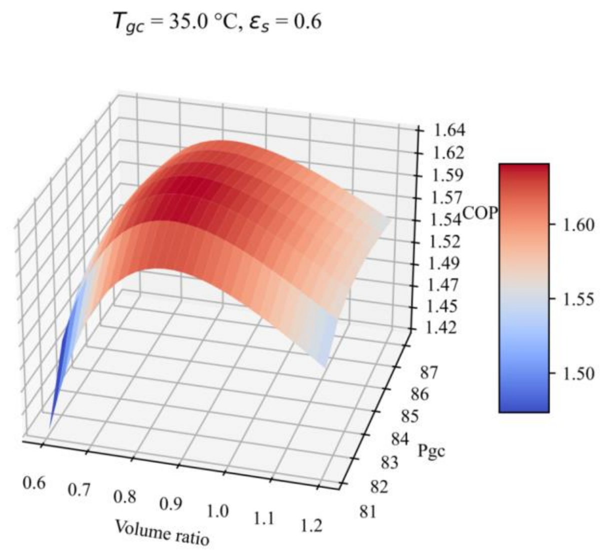


Figure: SCVI system COP optimization by P_{GC} and V_r parametric analysis

- SCVI system presents lower optimum V_r
- At higher $T_{GC,out}$, higher optimal V_r is found in the transcritical region.
- Excepting the case of heavy $P_{GC,opt}$ underestimation, COP es more sensitive to having losses by a V_r distancing from its optimal than a gas cooler pressure suboptimal operation.

Systems optimization

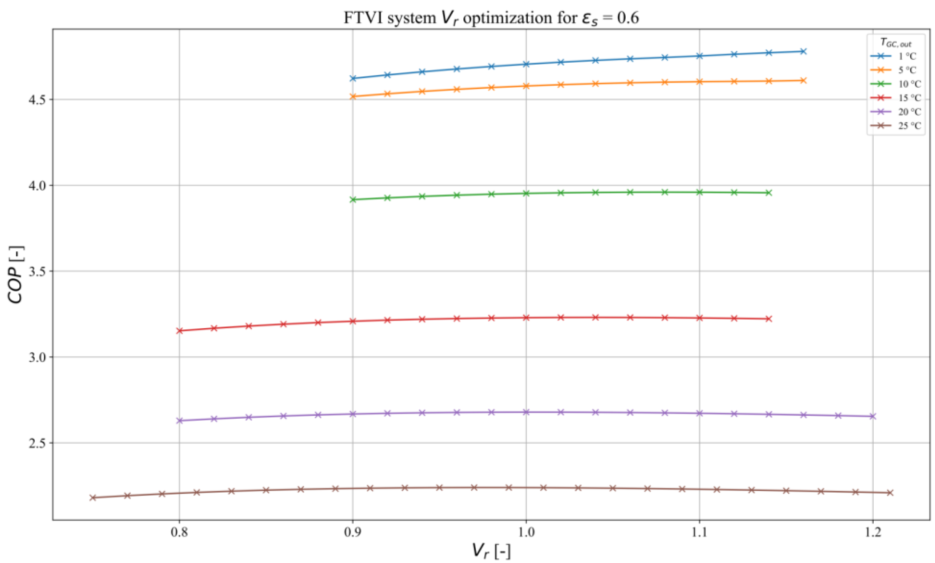


Figure: FTVI system V_r optimization in the subcritical region

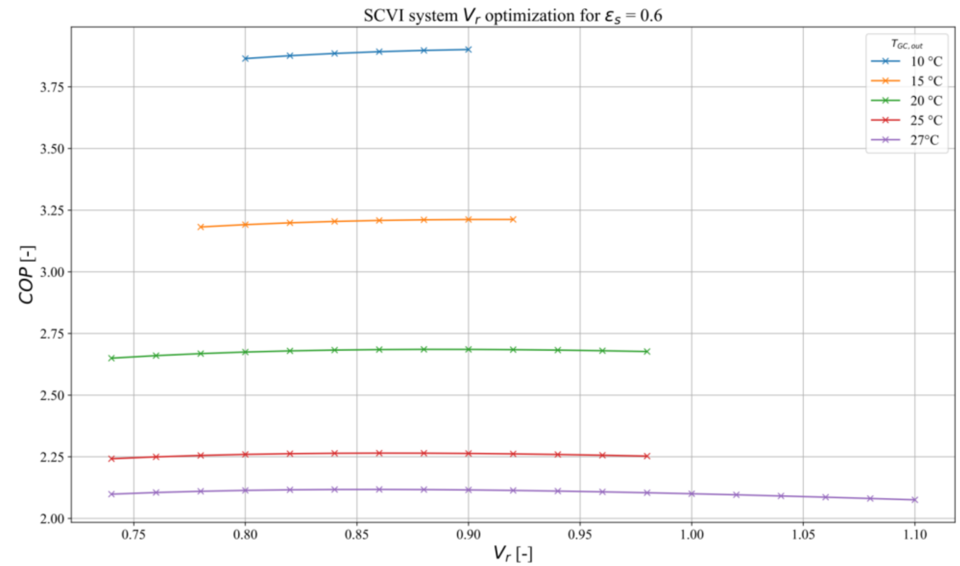


Figure: SCVI system V_r optimization in the subcritical region

- SCVI system presents lower optimum V_r
- At lower $T_{GC,out}$, higher optimal V_r is found in the subcritical region.
- Considering the subcritical and transcritical V_r optimization results, values of 0,95 and 0,85 are chosen for the FTVI and SCVI systems, respectively.

Systems optimization

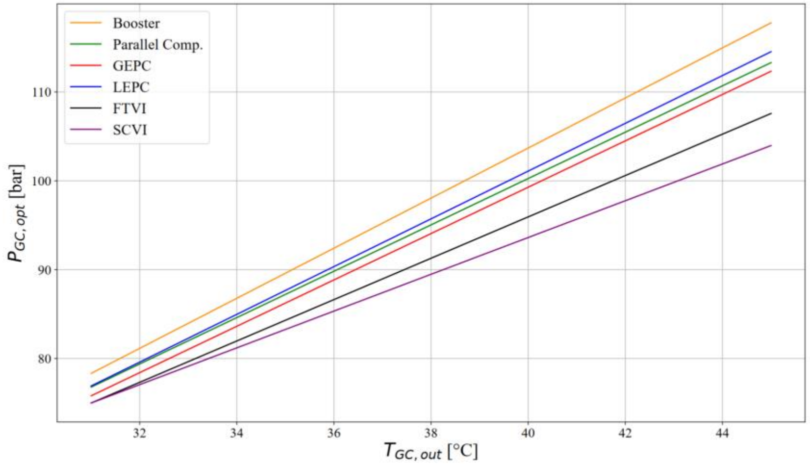


Figure: Optimal pressure lineal functions for all studied systems.

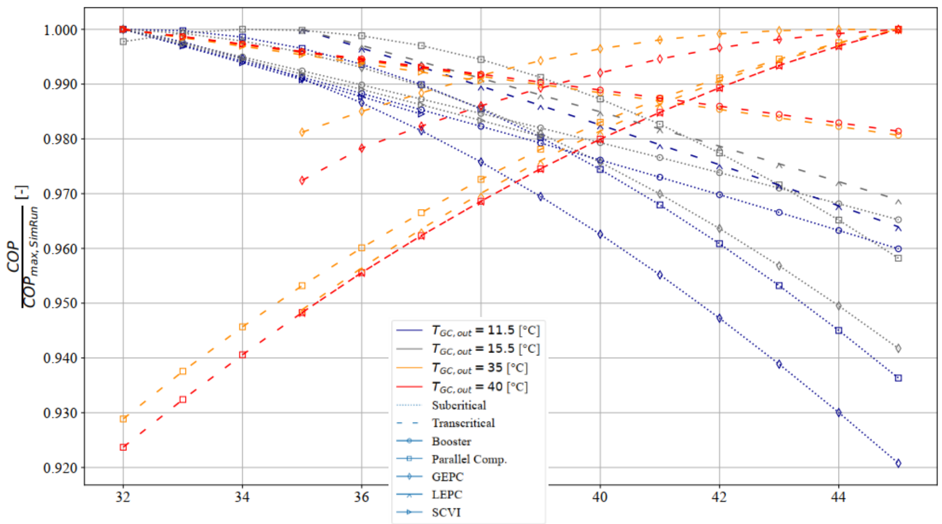


Figure: Systems COP variation with P_{FT}

- FTVI and SCVI systems have the lowest optimal high pressure
- Flash Tank optimization has also been performed for every system, excepting the FTVI layout (because of the floating pressure).
- Results suggest that systems without parallel compression find their maximum COP at the lowest P_{FT} explored, including the SCVI system. Systems with parallel compressors find an optimal point at higher pressures for higher $T_{GC,out}$.

SEASONAL COP CALCULATIONS

Seasonal COP calculations

- Three climates considered: Helsinki, Strasbourg and Athens.
- Distinction between Open Hours and Closed Hours in the supermarket is done:
  - Open Hours : 8 a.m to 7 p.m excluding sundays.
  - Closed Hours: remaining hours
  - In Closed Hours, MT cooling demand is set to its minimum.
- In subcritical conditions, a minimum high pressure of 45 bar is considered, so it does not become unrealistically close to the FT pressure. Sub-cooling increases for this operating points.

- $\dot{Q}_{LT} = 24\ kW$
- $$\dot{Q}_{MT} = \begin{cases} 72 & kW, \quad \text{if } T_{amb} > 6°, \\ 5 \cdot T_{amb} + 40 & kW, \quad \text{if } 0° \leq T_{amb} \leq 6°, \\ 40 & kW, \quad \text{if } T_{amb} < 0° \text{ or Closed Hours} \end{cases}$$

Seasonal COP calculations

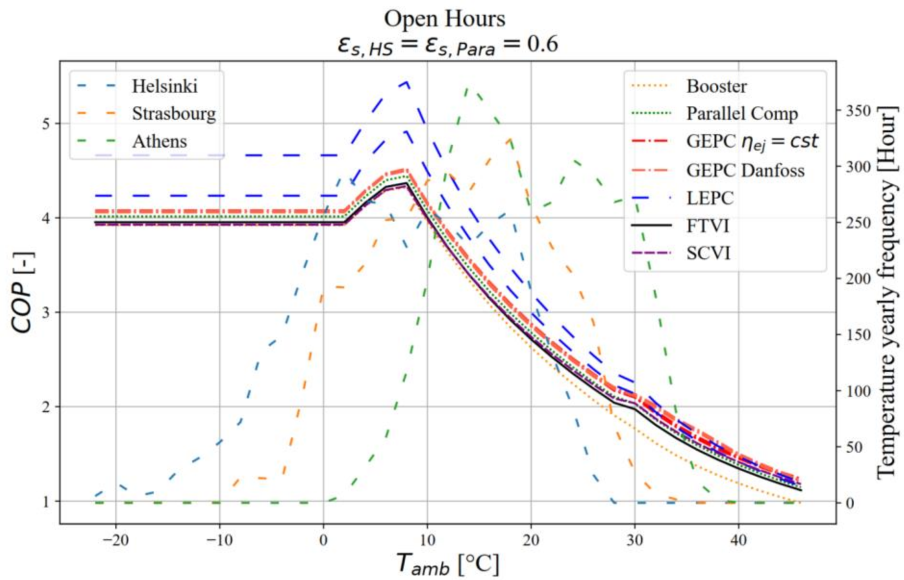


Figure: COP decay with T_{amb} for each system at Open Hours, $\varepsilon_{s,HS} = \varepsilon_{s,Para} = 0,6$

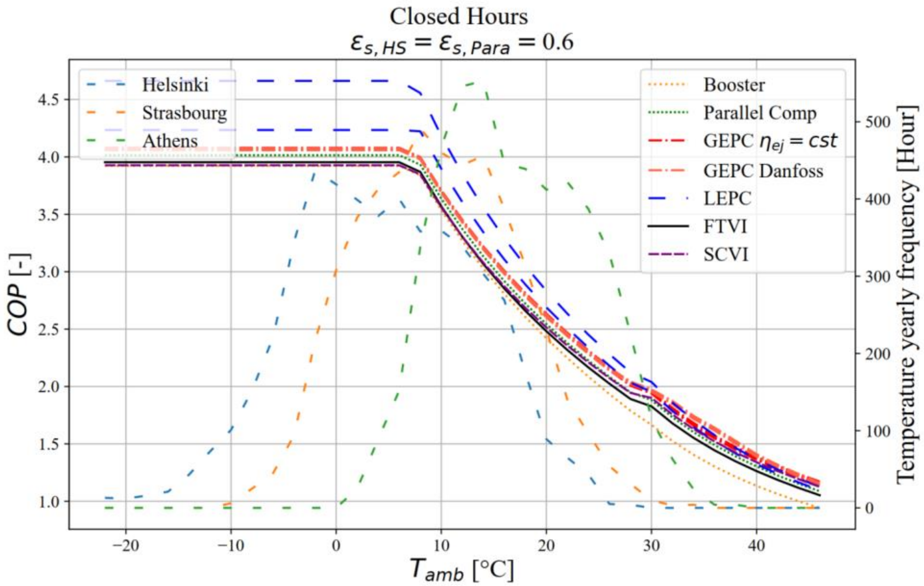


Figure: COP decay with T_{amb} for each system at Closed Hours, $\varepsilon_{s,HS} = \varepsilon_{s,Para} = 0,6$

Seasonal COP calculations

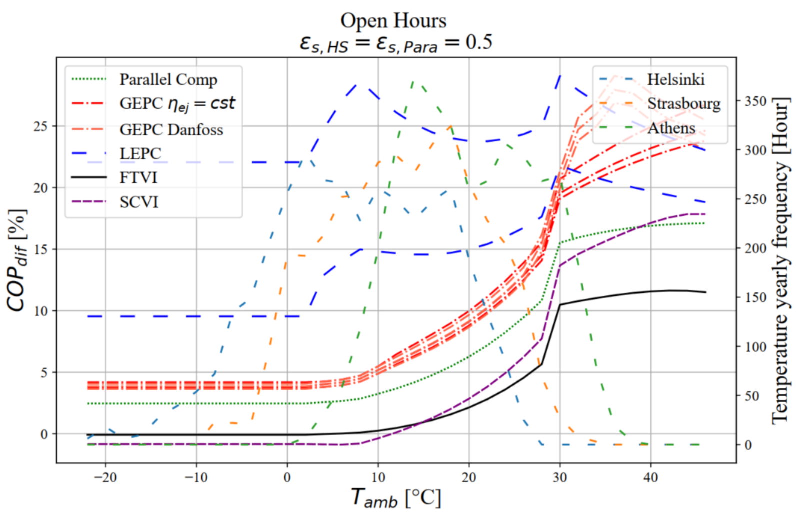


Figure: COP improvement over the Booster cycle for each system at Open Hours, , $\varepsilon_{s,HS} = \varepsilon_{s,Para} = 0{,}5$

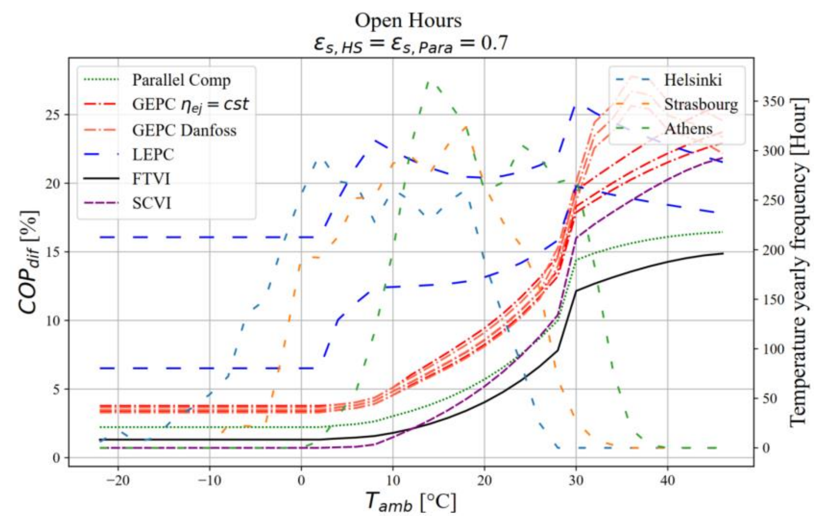


Figure: COP improvement over the Booster cycle for each system at Open Hours, , $\varepsilon_{s,HS} = \varepsilon_{s,Para} = 0{,}7$

Table: Seasonal Performance indicators and sensitivity analysis.

			Helsinki		Strasbourg		Athens	
			ε_s[a]	sec[b]	ε_s	sec	ε_s	sec
SCOP	**Ideal**	**Booster**	3.03|**3.50**|3.93	-|**3.50**|-	2.74|**3.17**|3.58	2.74|**3.17**|3.58	2.18|**2.54**|2.89	-|**2.54**|-
		Parallel Comp.	3.14|**3.62**|4.06	-|**3.62**|-	2.86|**3.31**|3.73	-|**2.86**|-	2.33|**2.72**|3.08	-|**2.72**|-
		Gas Ejector	3.20|**3.68**|4.13	3.68|**3.68**|3.70	2.93|**3.38**|3.81	2.93|**3.38**|3.81	2.40|**2.79**|3.16	2.78|**2.79**|2.81
		Gas Ej. (Danfoss)	3.20|**3.68**|4.13	3.67|**3.68**|3.69	3.37|**3.38**|3.40	2.41|**2.80**|3.18	2.41|**2.80**|3.18	2.79|**2.80**|2.82
		Liquid Ejector	3.46|**3.94**|4.38	**3.94**|4.24	3.13|**3.59**|4.01	**3.59**|3.85	2.50|**2.90**|3.27	**2.90**|3.08
		FTVI	3.05|**3.55**|4.01	-|**3.55**|-	2.24|**2.65**|3.03	-|**3.24**|-	2.24|**2.65**|3.03	-|**2.65**|-
		SCVI	3.04|**3.54**|4.01	-|**3.54**|-	2.78|**3.25**|3.70	-|**3.24**|-	2.27|**2.69**|3.08	-|**2.69**|-
	Realistic	**Parallel Comp.**	3.03|**3.50**|3.93	-|**3.50**|-	2.75|**3.18**|3.59	-|**3.18**|-	2.24|**2.61**|2.97	-|**2.61**|-
		Gas Ejector	3.03|**3.50**|3.93	-|**3.50**|-	2.75|**3.19**|3.60	-|**3.19**|-	2.26|**2.63**|2.99	2.63|**2.63**|2.64
		Gas Ej. (Danfoss)	3.03|**3.50**|3.93	3.50|**3.50**|3.50	2.75|**3.19**|3.60	3.19|**3.19**|3.19	2.26|**2.64**|3.00	2.64|**2.64**|2.65
		Liquid Ejector	3.13|**3.59**|4.00	-|**3.59**|3.90	2.80|**3.22**|3.62	**3.22**|3.49	2.26|**2.63**|2.97	**2.63**|2.82
Energy Savings	**Ideal**	**Parallel Comp.**	3.49|**3.32**|3.17	-|**3.49**|-	4.42|**4.20**|4.01	-|**4.20**|-	6.77|**6.46**|6.18	-|**6.464**|-
		Gas Ejector	5.25|**5.04**|4.84	4.80|**5.04**|5.50	6.44|**6.20**|5.97	5.99|**6.20**|6.74	9.25|**8.94**|9.32	8.67|**8.94**|9.62
		Gas Ej. (Danfoss)	5.27|**5.06**|4.86	4.79|**5.06**|5.33	6.50|**6.26**|6.03	5.94|**6.26**|6.58	9.64|**9.32**|9.02	8.91|**9.32**|9.74
		Liquid Ejector	12.3|**11.4**|10.4	**11.4**|17.6	12.6|**11.6**|10.8	**11.6**|17.5	13.1|**12.3**|11.5	**12.3**|17.4
		FTVI	0.53|**1.46**|2.03	-|**1.46**|-	1.17|**2.14**|2.73	-|**2.14**|-	3.06|**4.08**|4.67	-|**4.08**|-
		SCVI	0.29|**1.36**|2.03	-|**1.36**|-	1.34|**2.46**|3.16	-|**2.46**|-	4.18|**5.35**|6.08	-|**5.35**|-
	Realistic	**Parallel Comp.**	0.01|**0.01**|0.01	-|**0.01**|-	0.40|**0.38**|0.36	-|**0.38**|-	2.83|**2.70**|2.58	-|**2.70**|-
		Gas Ejector	0.01|**0.01**|0.01	0.01|**0.01**|0.01	0.50|**0.48**|0.46	0.47|**0.48**|0.51	3.55|**3.41**|3.28	3.33|**3.41**|3.61
		Gas Ej. (Danfoss)	0.01|**0.01**|0.01	0.01|**0.01**|0.01	0.54|**0.52**|0.50	0.50|**0.52**|0.54	3.91|**3.77**|3.64	3.64|**3.77**|3.90
		Liquid Ejector	3.33|**2.64**|1.93	**2.60**|10.4	2.18|**1.57**|1.00	**1.57**|9.14	3.77|**3.22**|2.69	**3.22**|9.78

[a] HS and PC compressors isentropic efficiency sensitivity analysis set to 0.5, 0.6, and 0.7.

[b] Secondary sensitivity analysis. For GEPC systems, it is the ejector efficiency. For the LEPC systems, it is the MT evaporating temp.

Conclusions

- The potential of a somewhat disregarded R744 system solution, FTVI and SCVI in a booster-like configuration, has been studied by means of simulation models. Models are primarily oriented at comparing the thermodynamic cycles.

- COP results are used for SCOP prediction for the climate of Helsinki, Strasbourg and Athens. Two scenarios are presented: "Ideal", where parallel compression racks do not have operating limits, and "Realistic" where they can not operate below $T_{amb} = 27\,°C$.

- Sensitivity analyses on some of the different adopted hypotheses have been conducted to understand if they could affect the final conclusions. High-stage and parallel compressors isentropic efficiency, and the gas ejector efficiencies, have the most influence. Notably, for higher compressors' efficiency, systems with PC decrease its energy savings with respect to the baseline Booster system, whilst the VI systems increase theirs.

- The "Ideal" scenario describes both VI systems as inferior to the 2nd and 3rd generation systems for all 3 studied climates. However, once the "realistic" scenario restrictions are applied VI systems present the highest energy savings. In Athens, FTVI and SCVI layouts present 4.08% and 5.35% energy savings with respect to the baseline Booster system, whilst the GEPC, presents up to 3.77%.

- More research is still to be done to understand the benefits and challenges of FTVI and SCVI in booster-like applications. Possible combinations with ejectors and parallel compressors have also not been explored.

THANK YOU