# Experimental Performance of Carbon Dioxide Compressor with Parallel Compression

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## Astract

Carbon dioxide (CO2) was proposed in the recent years as a natural fluid to replace the HFCs in refrigeration applications. Its implementation in refrigeration, first in subcritical and recently in trans-critical systems is becoming a technology of increasing importance.

Trans-critical CO2 system presents lower COP compared to HFCs systems when the ambient temperature is high. Reciprocating compressors with two compression stages having a vapor injection port (VI) have been proposed for parallel compression to improve the efficiency of the system. Many publications have been presented to explain the application advantage but experimental data are lacking to support the theoretical analysis.

The work presented here highlight the advantage of the parallel compresion and explains the experimental tests carried-out on a reciprocating prototype with carbon dioxide working with two compression stages (parallel compression) and finally discusses the experimental results. The compressor is a semi-hermetic four cylinders compressor with one compression chamber (cylinder) dedicated to the parallel compression. The tests was performed on hot gas by-pass cycle at -10°C evaporating temperature varying the intermediate and discharge pressures.

The experimental result shows the performance at different operating conditions varying the intermediate pressure. It highlights the influence of the intermediate pressure on the efficiency of the compressor and the system. The intermediate pressure influences the volumetric efficiency of the compressor and consequently the COP of the system.

Finally, a system efficiency comparison between this configuration and dedicated compressor for parallel compression have been evaluated.

**Key Words:**
Refrigerant, CO2, Transcritic, Parallel compression

## 1. Introduction

Due to environmental concerns, HFCs refrigerants are constrained as their global warming potential (GWP) is high. Natural refrigerant presenting low GWP like CO2 are becoming potential candidates in refrigeration and heat pump applications. Refrigerating systems using CO2 as working fluid present a good COP and comparable to HFCs at low temperature (LT). Meanwhile at medium temperature (MT), the efficiency is strongly depending on ambient temperature. At higher ambient temperature a CO2 system operates in a transcritical cycle and becomes less efficient compared to the subcritical cycle. System with parallel compression was proposed as a cycle variance to improve the efficiency when operating in transcritical mode.

A reciprocating prototype designed for parallel compression has been built and tested. The present work evaluates the system performance when parallel compression is used either with a single compressor designed for parallel compression or with a dedicated compressor for each compression stage.

## 2. Refrigeration Systems With Carbon Dioxide

Carbon dioxide was a known refrigerant and was used before 1930's in the refrigeration systems as working fluid in compression type refrigerants machines. Carbon dioxide operating in transcritical cycle presents poor efficiency. Several systems variances were proposed to improve the performance. The figure 1 shows the two main cycles frequently used in early 1930's using two stages compression (Windhausen) and the parallel compression (Vorhees). The principle of such systems is to separate the vapor at an intermediate stage and compress it to the discharge pressure.

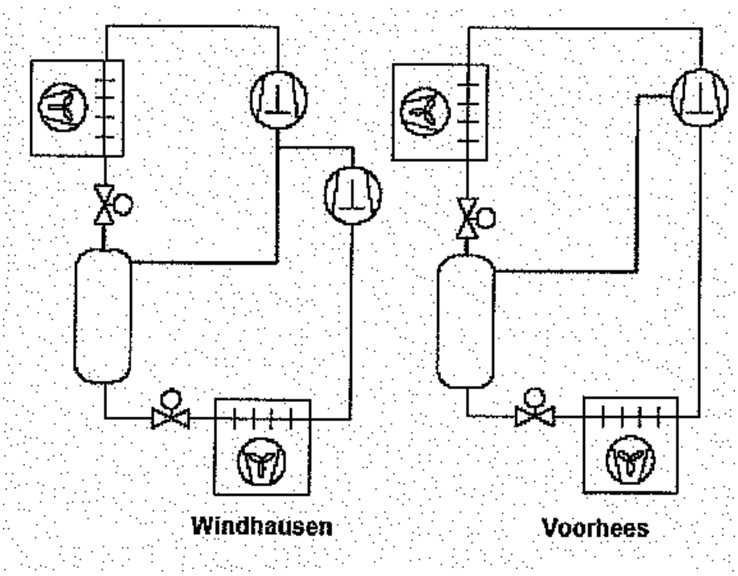


***Figure 1:*** *Main cycles used in 1930's with CO2 transcritical*

Carbon dioxide was abandoned after the second war with the introduction of CFCs and HCFCs. Recently due to the warming concern, natural refrigerant having low GWP like CO2, Hydrocarbons and Ammonia are re-considered as potential working fluids to substitute HFCs. Carbon dioxide was introduced first in subcritical applications where it has shown its competiveness and recently its introduction in trans-critical systems is becoming more and more important. Today two main trans-critical cycles are used with CO2: the so-called Flash gas bypass (FGB) and the parallel compression (Voorhees cycle).

### 2.1. Flash Gas bypass Cycle (FGB)

Figure 2 shows the schematic layout of the flash gas by pass (FGB). The compressed gas flows to the gas cooler and expands in a receiver at an intermediate pressure. The liquid flows to the evaporators and back to the compressor, meanwhile the vapor released in the separator is expanded to the suction line. With such system all components after the first expansion are operating at a low pressure and we can summarize the advantages in:

- less costly low pressure components
- easier on-site installation handling lower pressure joints
- heat transfer improvement in the evaporator
- better working of the expansion valve with liquid upstream

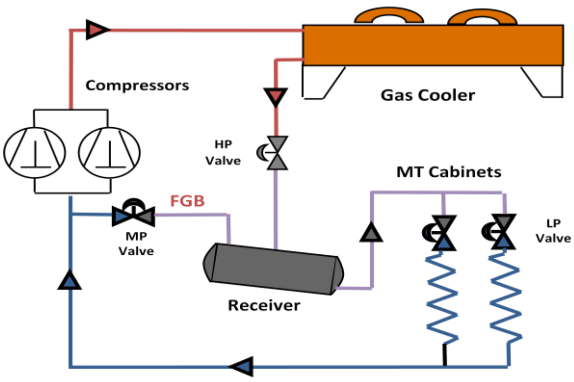


***Figure 2 :*** *Flash gas bypass system*

## 2.2. Parallel Compression (Voorhees cycle)

The parallel compression is a variant of Voorhees cycle already used in early 1900's. The vapor coming from the gas cooler is expanded at an intermediate pressure and compressed from this pressure to the discharge pressure. The parallel Compression could be either using a single compressor for both stages as reported in figure 3a or dedicated compressors as reported in figure 3b.

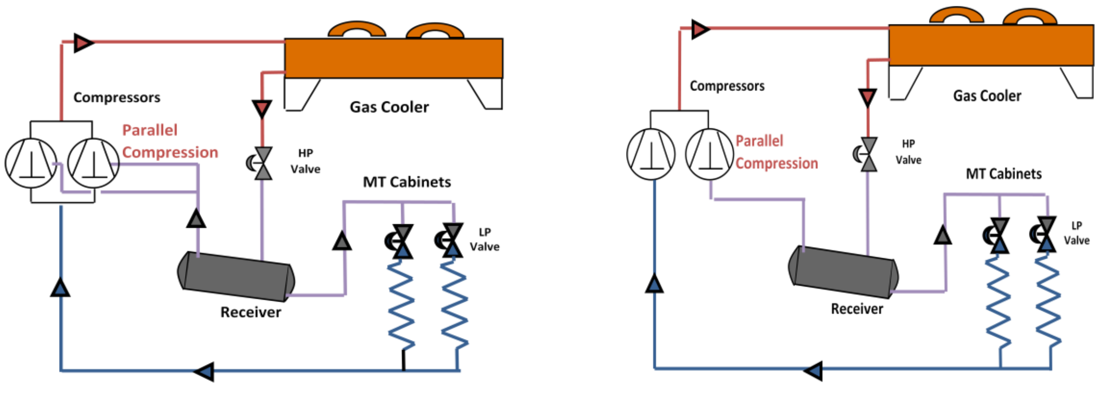


***Figure 3*** *Parallel compression cycle*

Parallel compression offers better performance because of less compression work in comparison to the flash gas bypass cycle.

# 3. System Performance with Parallel Compression

## 3.1. Intermediate Pressure Evaluation

Considering the parallel compression cycle reported in Figure 4, for given conditions suction pressure $p_1$, discharge pressure $p_2$ and the outlet gas cooler $T_3$; the mass flow ratio between vapor and liquid is governed by solving the following equations:

- Heat balance in the liquid separator:

$$\frac{\dot{M}liquid}{\dot{M}vapor} = \frac{h_7 - h_4}{h_4 - h_6}$$

- Displacement ratio Rm between suction pressure p1 and intermediate pressure pi:

$$Rm = \frac{\dot{V}i}{\dot{V}1} = \frac{\eta v_8 \cdot \rho_8 \cdot V_8}{\eta v_1 \cdot \rho_1 \cdot V_1}$$

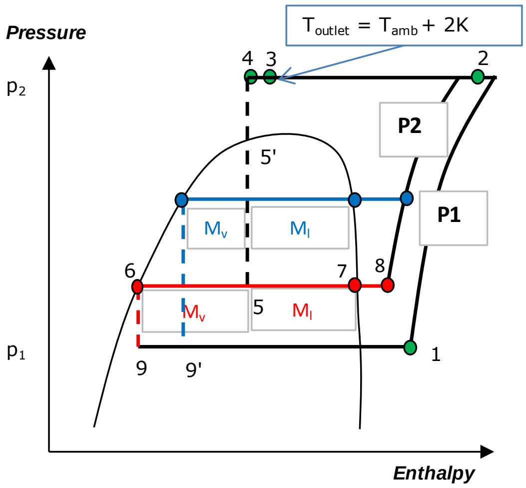


***Figure 4:*** *Parallel compression cycle*

The intermediate pressure is determined solving the above system for a given refrigerant. Three configurations have been simulated varying the displacement ratio $R_m$ between suction and intermediate stages. The figure 5 reports the intermediate pressure variation against the outlet gas cooler temperature. The volumetric efficiencies $\eta v_1$ and $\eta v_8$ of the compressors used in the equations are given from the single transcritical compressor measurements. At given outlet gas temperature the intermediate pressure increases when lowering the displacement ratio Rm.

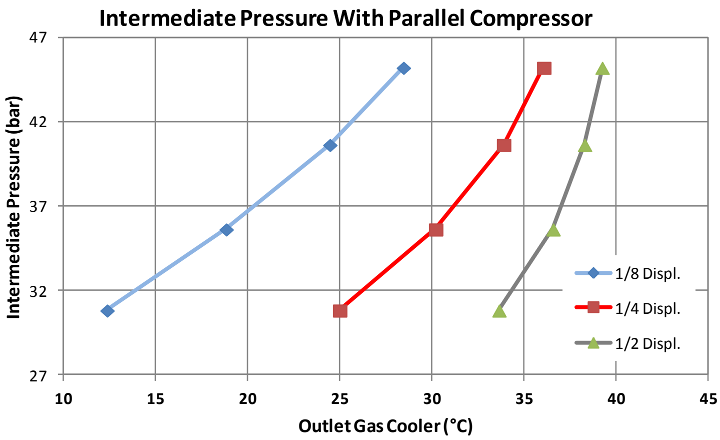


***Figure 5:*** *CO2 Intermediate pressure vs. Rm and outlet gas temperature*

Considering the range of ambient temperature up to 38°C, the configuration Rm = ¼ should be the most appropriate to keep the intermediate pressure in the range of 40 bar. When ambient temperature > 38°C, the configuration Rm = ½ is required to lower the intermediate pressure.

## 3.2. System Performance

The figure 6 is showing the system performance varying the displacement ratio Rm. The System COP is calculated as the ratio between the cooling capacity Qc and the compression power of the first stage P1 and intermediate stage P2:

$$COP = \frac{Qc}{P1 + P2}$$

The input powers are calculated considering the isentropic efficiencies obtained from the experimental tests of a single CO2 transcritical prototype without the vapor injection. Figure 6 shows clearly that the system performance improves with the parallel compression compared to standard CO2 trans-critical system without parallel compression.

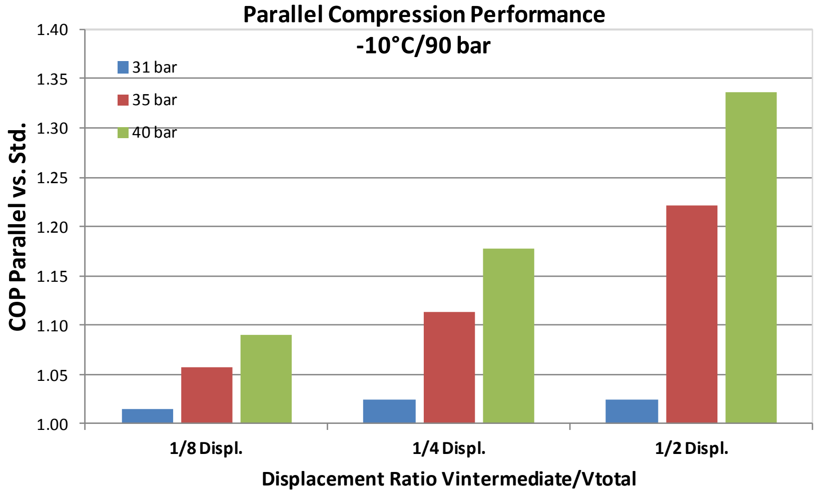


***Figure 6:*** *System Performance with parallel injection*

Using system parallel compressor with dedicated compressors for each stage as shown in figure 3b, the performance of CO2 transcritical system will improve in accordance to the system design and the outlet gas temperature. A system having a single compressor with integrated parallel compression should have the same efficiency if the efficiencies of the compressor are the same.

## 4. Parallel Compression with single Compressor

A system with parallel compression can be designed either with separate compressors for each stage or with a dedicated compressor with an integrated parallel compression realized by using some cylinders to the second compression stage.
The present experimental work is aimed to verify the performance with a reciprocating compressor running with 2 compression stages. The prototype is a four cylinders CO2 trans-critical reciprocating compressor having a 9.5 $m^3/h$ displacement. One of the four cylinders is connected to the intermediate stage and three cylinders take the vapor from the suction side. A special cylinder head was designed and built with a vapor injection port (VI port) for the integrated parallel compression. The low pressure (LP) plenum and the intermediate pressure plenum (IP) of the cylinder head are separated by an internal wall. The LP and IP pchambers are respectively connected to the evaporator and the receiver. In total the compressor has 3 cylinders in communication to suction (evaporators) and 1 cylinder connected to the receiver.

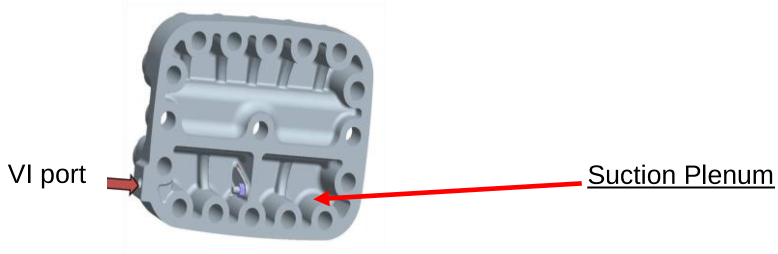


***Figure 7:*** *Cylinder head used for parallel compression with VI Port*

The two plenums are in communication through a reed valve fixed on the separated wall as shown in figure 7. The reed valve is closed when the vapor is injected through the VI port, meanwhile in case no injection occurs the valve is open and the suction vapor flows through it to the cylinder dedicated to the parallel compression.

## 4.1. Tests and Results

### 4.1.1. Test Rig Description

Figure 4 is showing the hot gas by-pass stand used for the parallel compression test.

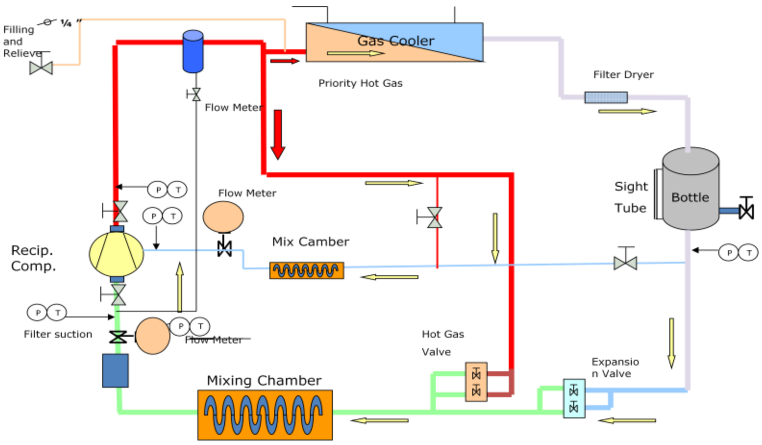


***Figure 8:*** *Test Rig for CO2 parallel compression*

The compressed gas flow is cooled in a gas cooler, then separated in 2 parallel flows at different pressures. The low pressure p1 is connected to the suction of the compressor LP and the intermediate pressure pi is connected to the Vapor injection port. Therefore in case no vapor injection occurs thru the VI port all the 4 cylinders compress the gas from LP to the discharge pressure. When the vapor injection is occurring, 3 cylinders are sucking vapor form LP side and one cylinder from IP side. Two Coriolis flow meters are used to measure the mass flow LP and IP sides.

### 4.1.2. Experimental Results

The tests have been carried-out at constant suction pressure $p_1$ = 26.5 bar corresponding to -10°C evaporating temperature and two discharge pressures $p_2$ equal to 90 bar and 110 bar. The two dotted lines in the figure 9 represent the volumetric efficiency of the standard compressor (without EVI cylinder head) at 90 and 110 bar.
The compressor has been first tested without vapor injection. The volumetric efficiency (dark mark in the figure) decreases by 4% and 6% respectively at 90 bar and 110 bar. These losses are mainly due to the pressure drop through the additional valve positioned between the two plenums.

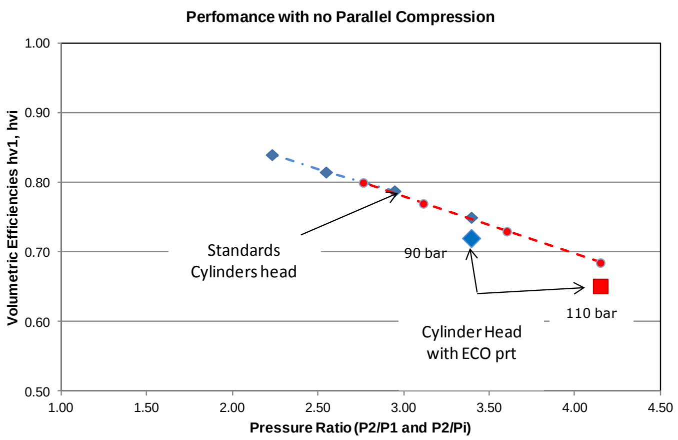


***Figure 9:*** *Volumetric efficiency with no Vapor injection*

The second set of tests was carried-out injecting vapor with constant superheat (10 K) thru the VI port. The intermediate pressure $p_i$ has been varied in the range of 26.5 to 50 bar keeping constant the suction and discharge pressure. The masses flows of the two stages have been measured and consequently the volumetric efficiency is calculated. Figure 11 represent the volumetric efficiencies $\eta_{vi}$ (referred to IP side) and $\eta_{v1}$ (referred to LP side) against $p_i$ variation.

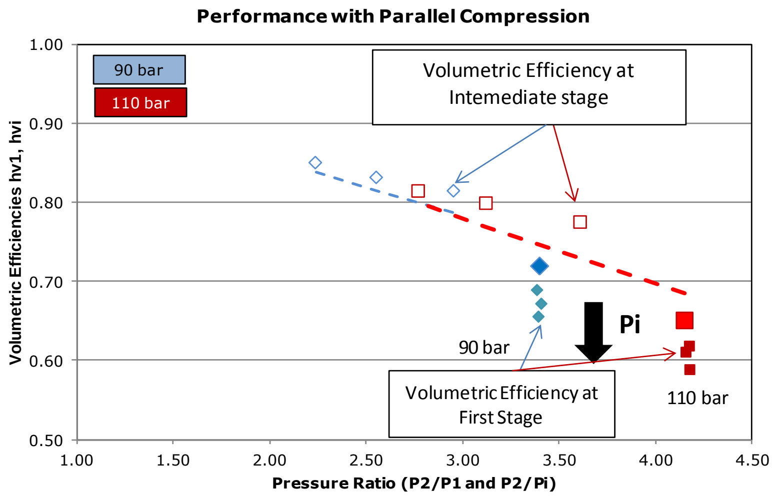


***Figure 10:*** *Volumetric Efficiencies vs. intermediate pressure with Vapor injection*

Ideally $\eta_{v1}$ and $\eta_{vi}$ should belong to the dotted line and equal respectively (to 0.75 and 0.68 at 90 bar and 100 bar). The dark mark at constant pressure ratio represents the volumetric efficiency referred to LP side. The figure shows clearly that the volumetric efficiency decreases as the intermediate pressure increases. Theoretically $\eta_{v1}$ should not been influenced by the intermediate pressure.

Varying the intermediate pressure from 26.5 to 50 bar at constant discharge pressure, the volumetric efficiency $\eta_{vi}$ of the intermediate stage increased for the intermediate stage as represented in Figure 5. Leakage and heat transfer could explain the opposite trend of $\eta_{vi}$ and $\eta_{v1}$. In fact, the vapor flowing at the intermediate pressure has higher temperature compared to the suction vapor, which imply a heat transfer from intermediate to suction side. As consequence the suction density decreases and the intermediate vapor density increases.

Leakage between the intermediate and suction could explain the opposite trend of the volumetric efficiencies too. In fact, when leakage occurs, the mass flows through the ECO port increases as well as the volumetric efficiency $\eta_{vi}$. On the other side, the mass flow at the suction side decreases as well as the volumetric efficiency $\eta_{v1}$. Additional tests were executed to detect possible leaks between the internal plenums of the cylinder head. The intermediate plenum of the ECO cylinder head has been pressurized and the pressure in the suction plenum chamber has been measured. No leak has been detected during this test. Leakage could occur through the piston rings of the intermediate compression chamber during suction phase. Motor slippage (higher torque) could also contribute to the volumetric efficiency degradation.

During the tests high vibration was observed at 45 and 50 bar. The vibrations are mainly caused by the high torque variation during the shaft revolution.

## 4.2. System Performance with Reciprocating VI Compressor

The system performance with parallel compression is evaluated elaborating the measurements of the reciprocating VI prototype. Figure 11 shows the system performance using the experimental efficiencies measured on the compressor. The cooling capacity increases with the high intermediate pressure. At 40 bar intermediate pressure, the cooling capacity is 10 % higher compared to the standard cycle without parallel compression.

Meanwhile the COP reaches a maximum of 2% at 90 bar and 5 % at 110 bar. The expected COP at 40 bar intermediate pressure and 90 bar is more than 15 %.

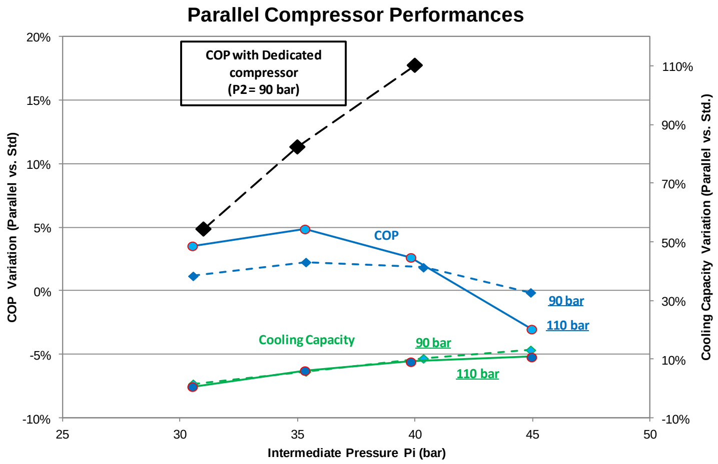


***Figure 11:*** *System Performance with VI Compressor*

## 5. Conclusion

A Carbon dioxide cycle with parallel compression has been evaluated using the experimental performances of single trans-critical compressor. Parallel compression shows clearly the efficiency increases in comparison to the standard cycle or cycle with the flash gas bypass.
A CO2 parallel compressor with dedicated cylinder for vapor injection was build and tested. The compressor shows a degradation in performance when the intermediate pressure increases. The volumetric efficiency of the compressor is affected by the variation of the intermediate pressure in the receiver at decreases by 15% compared to the compressor without VI. The cycle performances are calculated using the experimental results. The COP increases up to 2 % at 90 bar where it increases by 15 % if a dedicated compressor is used for the parallel compression. In addition, high vibrations have been observed during the tests when the intermediate pressure increases. This presents a severe practical constrain to the application of this scheme to achieve a higher system COP. Meanwhile a dedicated compressor used for the parallel compressor shows a clear advantage. In addition the control of the intermediate pressure should be easier when this configuration is applied with modulated control.

## Nomenclature

| | | |
|---|---|---|
| VI | | Vapor injection for parallel compression |
| GWP | : | Global Warming potential |
| $\dot{M}$ | : | Mass flow |
| Rm | : | Mass flow ratio between Intermediate and suction stages |
| T3 | : | Outlet gas cooler temperature |
| $\dot{V}_1$ | : | Displacement at suction stage |
| $\dot{V}_i$ | : | Displacement at intermediate stage |
| $p_1$ | : | Suction pressure |
| $p_i$ | : | Intermediate pressure for the parallel compression |
| $p_2$ | : | Discharge pressure |
| $\eta_v$ | : | Volumetric efficiency |

## Subscripts

| | | |
|---|---|---|
| 1 | : | suction side of the compressor |
| 2 | : | discharge side of the compressor |
| i | : | intermediate stage for parallel compression |
| 3 | : | outlet gas cooler |

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