TEWI and energy assessment of integrated CO2 refrigeration, heating and cooling technology

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Abstract. In order to meet with the regulations proposed by the Intergovernmental Panel of Climate Change to control emissions from fluorinated gases, an EU funded project (LIFE) has been engaged in. Through this project, the reduction of CO2 emissions by using an integrated refrigeration, heating and cooling system, in real shops across Europe and the impact of the raw materials used will be investigated. In a theoretical evaluation, the Seasonal Energy Performance Ratio and Total Equivalent Warming Impact of the unit is compared to an R-410A unit using test measurements. Although the CO2 unit has a lower Seasonal Energy Performance Ratio, the Total Equivalent Warming Impact was calculated to be lower in comparison to the R-410A unit over a period of 10 years. These measurements were also used to discuss the importance of heat recovery by comparing the unit to a non-integrated refrigeration, heating and cooling system. The energy assessment of the unit at a real installation in a supermarket in Europe has been presented on a monthly basis. This assessment involves the use of a compressor curve method to estimate mass flow and as a result, the delivered energies in the absence of expensive flow meters. The precision of such a method has been discussed. To conclude, challenges concerning the technology and important results and conclusions have been discussed.

Keywords. CO2 emissions, heat recovery, refrigeration, energy efficiency, TEWI.
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1. Introduction

1.1 LIFE Project

A LIFE project has been engaged in, in line with the F-gas regulations proposed by the IPCC [1]. This project aims at presenting an integrated refrigeration, space heating and cooling solution to the commercial market with a natural refrigerant i.e. CO2. The main objectives of this project are to demonstrate the unit in supermarkets across various countries in the EU, develop a CO2 cassette indoor unit, investigate the integration of a thermal storage, improving safety standards and energy efficiency regulations, training of service engineers, installers, designers and dissemination of information. This paper discusses the theoretical climate impact of the unit as compared to an R-410A unit through TEWI calculations and the impact of heat recovery on the SEPR. Real site data from one of the installations also presents an energy assessment of the unit, with an introduction to a mass flow estimation method to calculate the delivered energies. Feedback and comments received on a previous paper have been taken into account and applied wherever applicable in this follow-up paper.

1.2 Combined Refrigeration, Heating and Cooling unit

The CO2 unit under consideration is an integrated refrigeration, heating and cooling system with CO2 as a refrigerant, with the possibility of heat recovery from the refrigeration cabinets for indoor space heating. The unit also has the ability to operate as a heat pump in case of additional space heating demand. This can be achieved by two independent low-stage swing compressors; one for refrigeration and the other for air conditioning or heating, with a common high-stage compressor. The indoor units can switch between cooling or heating depending on the demand with the help of solenoid valves. The indoor units are either ducted or in a cassette form. The schematic of the CO2 unit is shown in figure [1].

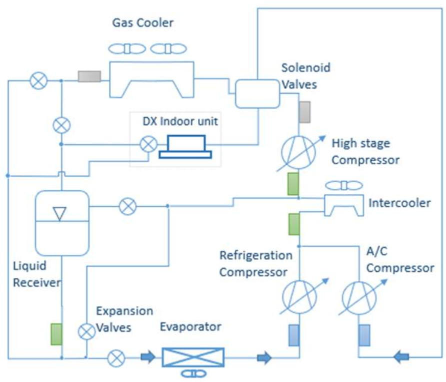

Fig. 1 – Schematic of integrated CO2 unit

2. Methods

2.1 TEWI Analysis

The climate impact of the unit was studied using a theoretical evaluation of the TEWI in comparison with a R-410A unit. The equation used for this is as shown in Equation (1).

$$TEWI = (GWP \cdot L_{annual} \cdot n) + GWP \cdot m \cdot (1 - \alpha_{recovery}) + (E_{annual} \cdot \beta \cdot n) \qquad (1)$$

This equation comprises direct and indirect emissions of the technology over the lifetime considered. The first two terms in the equation show the direct emissions caused as a result of the Global Warming Potential of the refrigerant, the annual leakage rate, refrigerant charge and the recovery rate of the system. The last term represents the indirect emissions as a result of the annual energy consumption and emission factor based on the energy mix of the region. The values used for the parameters in the equation are given below in table [1].

Tab. 1 – Parameters for TEWI calculation

Parameter	R410A	R744	Unit
GWP	2088[3]	1	
L_{annual}	2 %	2 %	
n	10	10	years
m	20	30	kg
$\alpha_{recovery}$	50 %	50 %	
E_{annual}	23062	24119	kWh/year

2.2 Impact of HR on SEPR

As described in the previous section, the integrated system delivers refrigeration, cooling and heating with one outdoor unit with the possibility of heat recovery. A non-integrated system, on the other hand, needs independent outdoor units to perform the same functions. Figure [2] shows the difference in configuration of the two systems. This analysis compares the Seasonal Energy Performance Ratio of both systems and evaluates the impact of heat recovery in integrated systems. The SEPR was calculated based on temperature bins from EN13215 and EN14825 [4][5]. The capacity and COP values from test rooms were used at various condition points to calculate the SEPR.

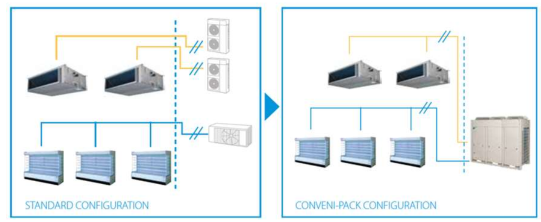

Fig. 2 – Difference between non-integrated (left) and integrated (right) systems

Figure [3] and Figure [4] show the P-design points used for these calculations. In order to account for heat recovery in an integrated unit, a common P-design point at 5 was chosen instead of 7̊C or 2̊C, as seen for heating in a non-integrated system.

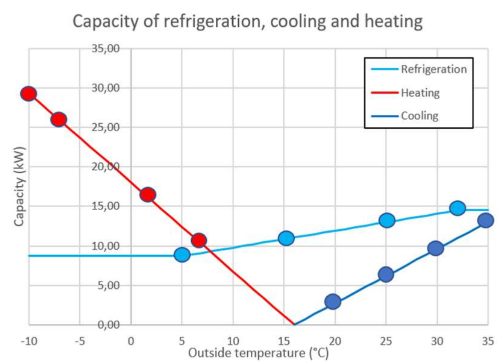

Fig. 3 – P-design points for non-integrated system

Similarly, for cooling, a common P-design point at 32̊C was chosen. This was done to accommodate for the lack of a methodology to calculate the SEPR of such combined refrigeration, heating and cooling equipment.

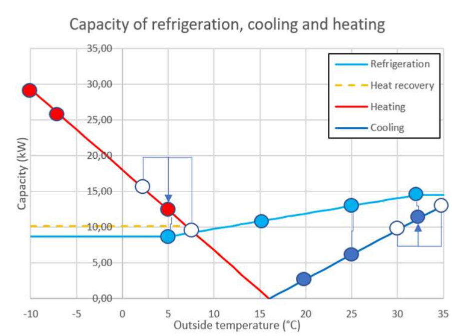

Fig. 4 – P-design points for integrated system

2.3 Compressor Curve method

As mentioned in the introduction to the project, it is the aim to demonstrate the technology in various sites across EU. Along with the demonstration, it was also the objective to monitor the operational data of the units to evaluate the delivered energies, i.e. refrigeration, heating and cooling; and consumed energies. While the consumed energies were measured using energy meters, the delivered energies were calculated using the product of enthalpies and estimated mass flows as shown in the succeeding section. The mass flows were estimated based on the compressor speed, the density of the refrigerant at the suction side and a constant representing the swept volume of the compressor. As the CO2 Coreolis mass flow meters were expensive, they were only installed at two local sites to validate the method. Separate constants were derived for refrigeration, heating and cooling respectively. The precision of the method will be discussed in the results section for each function.

2.4 Energy assessment

As mentioned in the previous section, delivered energies were calculated using operational data from each site, site 1 situated in Czechia and site 2 in Germany. This, along with the measured consumed energy was used to calculate the operational COP of the units. Following are the formulae used to calculate delivered refrigeration, cooling and heating energies. The refrigeration capacity was calculated as the product of the estimated refrigerant mass flow through the cabinets and the difference in the enthalpy between the outlet and inlet of the cabinets respectively, as shown in equation (2). The inlet enthalpy, $h_{ref,in}$, was calculated using the receiver pressure and the liquid temperature of the refrigerant. The outlet enthalpy, $h_{ref,out}$, was calculated using the suction pressure and temperature.

$$Ref_{cap} = m_{ref} \cdot \left(h_{ref,out} - h_{ref,in}\right) \qquad (2)$$

The heating capacity was calculated as shown in equation (3). The inlet heating enthalpy, $h_{heat,in}$, was calculated using the discharge pressure and temperature of the refrigerant after the high stage compressor. The outlet enthalpy, $h_{heat,out}$, was calculated using the discharge pressure and the liquid outlet temperature after passing through the indoor units.

$$Heat_{cap} = m_{ac} \cdot \left(h_{heat,in} - h_{heat,out}\right) \qquad (3)$$

Similarly, the cooling capacity delivered was calculated using receiver pressure and the liquid refrigerant temperature for the enthalpy on the inlet side and the suction pressure and temperature for the enthalpy on the outlet side of the indoor units. This is shown in equation (4).

$$Cool_{cap} = m_{ac} \cdot \left(h_{ac,in} - h_{ac,out}\right) \qquad (4)$$

3. Results

3.1 TEWI Analysis

The TEWI analysis was performed using the parameters mentioned in Table 1. As can be seen in figure (5), contributions as a result of direct and indirect emissions have been shown with a stacked bar graph for both refrigerants, R410A and R744. From the table, it could be seen that the annual energy consumption of the R744 system is higher than that of the R410A system, thus, resulting in slightly higher indirect emissions from the former. But this is offset by the negligible direct emissions from the R744 system, owing to a GWP of 1 as compared to significantly higher direct emissions from the R410A unit owing to a GWP of 2088. The total difference in the emissions results in the R744 unit emitting 37% less as compared to the R410A unit.

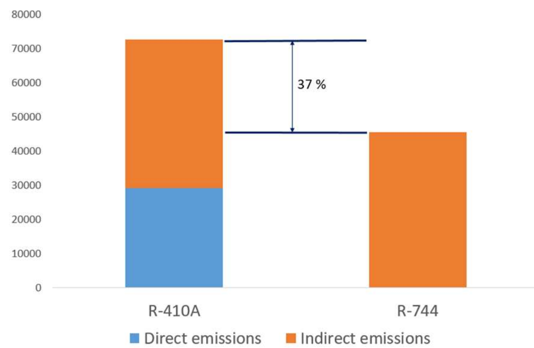

Fig. 5 – TEWI comparison between R-410A & R-744

Although product specifications indicate an annual leakage rate of about 1 % for both systems, a leakage rate of 2 % was chosen for the analysis to conform wih existing research literature [6]. It was observed that with a leakage rate of 1%, the R744 system emitted about 34 % lower as compared to the R410A unit over their lifetime. This clearly indicates that the R744 unit becomes more attractive as the leakage rate is increased.

3.2 Impact of HR on SEPR

Based on the methodology described in section 2.2, the distribution of consumed energy was plotted as a function of ambient temperature, as shown in figure (6) and figure (7) for a non-integrated and integrated system respectively. It can be observed that the total energy consumption for the non-integrated system shows a peak between ambient temperatures of 0̊C and 5̊C due to the high heating demand which needs to be supplied by the independent heating system. This peak can be observed to be shaved off in the integrated unit, as the heat demand during this range was satisfied by the heat recovered from the refrigeration cabinets. This results in a 43% increase in the SEPR for the integrated unit as can be seen from table [2].

Tab. 2 – SEPR values

	Non-integrated system	Integrated system
SEPR	3,4	4,9

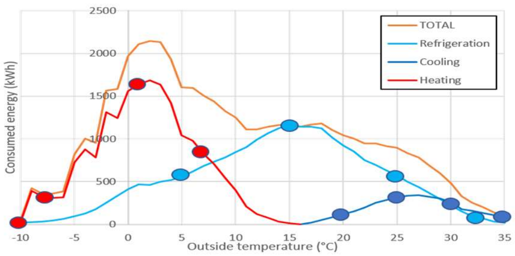

Fig. 6 – Consumed energy distribution for non-integrated CO2 system

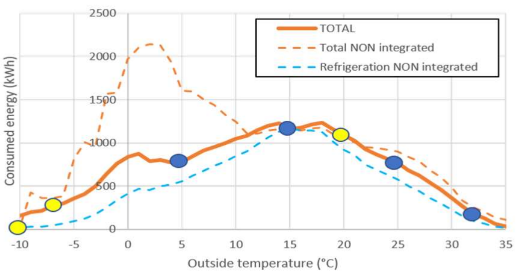

Fig. 7 – Consumed energy distribution for integrated CO2 unit

3.3 Precision of Compressor Curve method

This section discusses the precision of the compressor curve method for each function i.e. refrigeration, cooling and heating respectively. For each function, the precision has been described on an hourly and daily basis respectively. The x-axis on the below graphs indicates the deviation of the calculated capacity from the capacity measured using flow meters. Thus, a value of -10 indicates that the capacity calculated using the CC method is lower than the measured capacity by 5 to 10 %. The Y-axis indicates the occurrence of the deviation. In figure (8), the precision of the refrigeration capacity on an hourly basis is described by the blue line and on a daily basis by the red line. For both resolutions, it can be observed that majority of the datapoints occur within +-5% of the measured values, about 70% on an hourly resolution and more than 80% on a daily resolution.

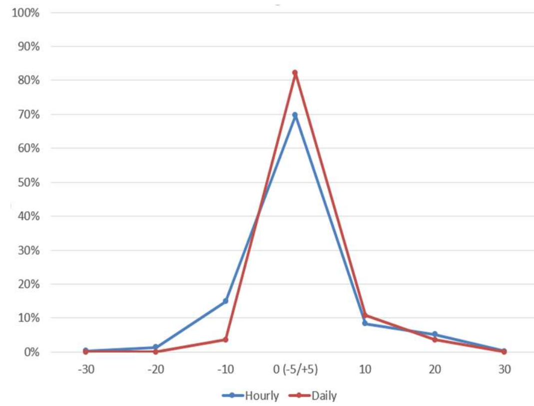

Fig. 8 – Precision of CC method for refrigeration

The following graph in figure (9) describes the deviation of the heating capacity from the measured values. It can be observed that the calculated heating capacity is more deviant from the measured capacity than what was observed for refrigeration and, as will be seen below, for cooling. This can be attributed to the complexity arising due to the presence of four different heating operation modes in the unit. These include two heat recovery modes and two heat pump modes with or without refrigeration. A large part of the deviation arises from the heat recovery mode, wherein not all of the refrigerant mass flow coming from the refrigeration cabinets is delivered to the indoor units after the second stage compressor. Thus, without flow meters, it was not suitable to assign a constant purely based on compressor speed.

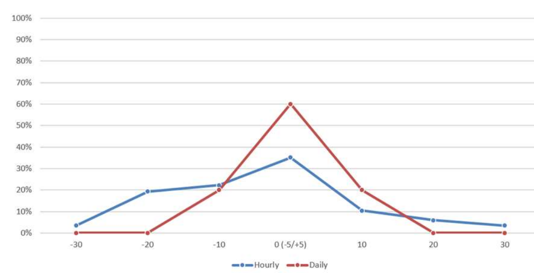

Fig. 9 – Precision of CC method for heating

For cooling capacity, although the percentage of values occurring within 5% of the measured values is lower, it can be observed from figure (10) that the majority of the deviation is either less than or equal to the measured value. On an hourly basis, a deviation of 0 to -10% occurs more than 80% of the duration and this occurrence reaches 100% when a daily resolution is considered. This shows that the compressor curve method has been adapted to under-estimate the capacities rather than over-estimate them.

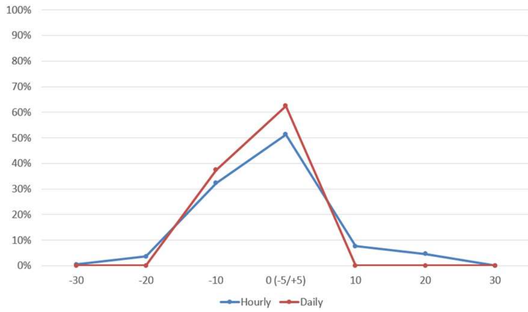

Fig. 10 – Precision of CC method for cooling

3.4 Energy Assessment

In this section, a monthly energy overview is presented for two independent sites. In the following graphs, the delivered refrigeration, heating energy through heat pump operation, heating energy through heat recovery and cooling energy is shown for both sites. From figure (11), it can be observed that there was a high demand for heating throughout the year. This demand was to a large extent satisfied using heat recovered from the refrigeration cabinets. Besides the bars labelled as 'HR Energy' in the graphs, a large portion of the 'HP Energy' delivered was also recovered from the cabinets in addition to the heat extracted from the ambient. Site 2, in comparison, was observed to have a lower heating demand as compared to site 1 as shown in figure (12). A large amount of cooling energy was also delivered in this site during the summer. This was due to a low indoor setpoint temperature of 17°C.

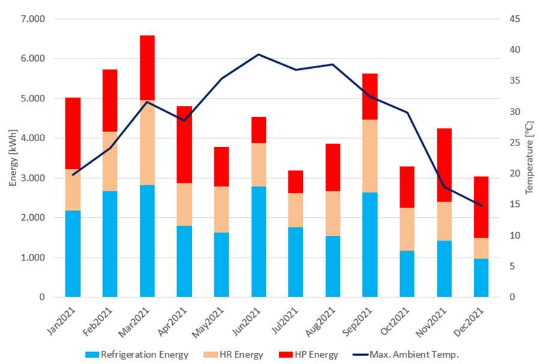

Fig. 11 – Energy overview of LIFE project site 1

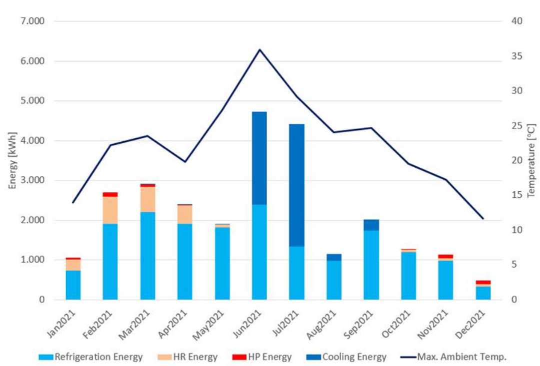

Fig. 12 – Energy overview of LIFE project site 2

As opposed to site 2, site 1 showed no cooling demand even during the summer when the ambient temperature reached a maximum of about 39 °C. This has been highlighted using figure (13), where a distribution of the operation modes of the unit at site 1 during the summer months of June, July, August and September has been shown. It can be seen that the unit ran as a heat pump for about 27% of the duration, besides the 67% of heat recovery operation observed. This was because of the layout of the store and the type of refrigeration cabinets used. While site 2 had almost entirely closed door cabinets, site 1 was installed with open cabinets. To add to that, the total area of the store was only 100 m², thus the indoor units were required to heat up the space which was being cooled by the open cabinets. This shows how important a role the layout, store design and selection of equipment play in efficient performance of commercial units.

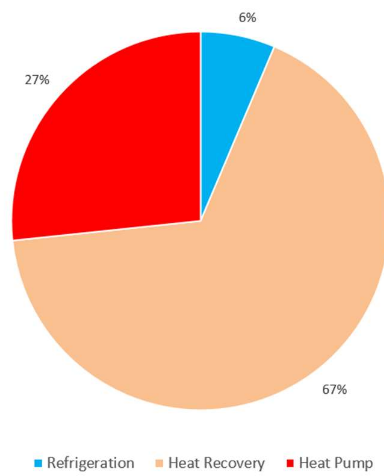

Fig. 13 – Operation mode distribution of site 1 during summer

The SEPR for site 1 was calculated to be about 7% lower than that of site 2. This can be attributed to the large amount of heat pump operation observed in site 1. Additionally, both sites showed a large occurrence of operation in a heat recovery mode. It was observed that there was a significant difference in power consumption during this operation mode.

4. Challenges

Due to the low critical temperature and high operating pressures of CO_2, implementing such solutions for cooling in regions with a warm climate is a challenge and an important obstacle in facilitating the spread of this technology further. To counter this, propositions such as an adiabatic cooling system for the gas cooler and thermal storage are currently being tested. The difficulty in the selection and availability of components such as valves, suitable refrigeration cabinets for the operating pressures also needs to be addressed. As mentioned in sections 2.2 and 3.2, no methodology currently exists in calculating SEPR of such combined refrigeration, space heating and cooling systems with the appropriate consideration of heat recovery. Efforts are being taken currently to establish such a methodology with a third-party research institution and validation from experts and scholars within the field. It is the aim to propose this methodology for inclusion in future standards concerning similar technologies. Monitoring and recording real operational data is invaluable and equally challenging from the point of view of logistics, economics and maintenance. The CC method, as discussed in the previous sections, has been proposed as one solution to tackle this challenge, by eliminating the need for installation of expensive and often intrusive CO2 Coreolis flow meters. Although it has been observed to have a high precision for lower resolutions, a more sophisticated algorithm would be needed for a higher precision at higher resolutions to account for the dynamic operation of the unit and the error between multiple sources of data. As the data is currently being recorded every 15" at every site, handling such large amounts of data has also been found to be an important aspect in implementing such projects. Finally, with the growth in CO2 technologies in the market, there is a constant need for a knowledge transfer between engineers, technicians, installers and researchers. It is the aim of this project to share as much information as possible to improve the state of the art and help in further development.

5. Conclusions

Through a theoretical analysis of the technology, it was observed that the R744 unit has a significantly lower TEWI than that of a R410A unit over a lifetime of 10 years. It was also seen that this difference increases with an increase in the annual leakage rate, owing to the large difference in GWP of the two refrigerants. The presence of heat recovery was found to have a positive impact on the SEPR of a combined refrigeration, heating and cooling unit as compared to a system with independent units for the same. The compressor curve method, devised to estimate the mass flow of the refrigerant in the absence of expensive flow meters, was found to improve in precision as the resolution of the data was lowered. This was found to be positive for the calculation of SEPR of such systems, wherein the calculated values were found to be within 5% of the measured values and adapted to under-estimate rather than over-estimate delivered capacities. Based on the assessment of the real site data from the two sites, the importance of store layout, selection of equipment and indoor setpoint can be noted. At site 1, even at a high ambient temperature of 39 °C, a need for heat pump operation was observed due to the presence of open refrigeration cabinets and a small store area. On the other hand, site 2 showed a high cooling demand during the summer due to a low setpoint of 17 °C indoors. Site 1 showed a lower SEPR than site 2 due to higher operating pressures during the heat recovery modes and a large occurrence of heat pump operation throughout the year.

6. Acknowledgement

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7. References

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