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DEVELOPMENT AND TESTING OF SUSTAINABLE REFRIGERATION PLANTS

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ABSTRACT

Although Ozone Depleting Substances (ODS) were banned with the Montreal Protocol in 1987, current refrigeration plants can not be considered sustainable for the environment. ODS have been in fact substituted by gases with high Global Warming Potential (GWP).

Among many alternatives, inverse Joule Brayton air cycle had been already implemented and tested for refrigeration purposes. In the open cycles described in the available literature, the operating fluid (air) is firstly compressed by a bootstrap (volumetric) compressor and then processed by a second (centrifugal) compressor and cooled; then, it is expanded in a turbine which drives the centrifugal compressor and discharges a cold flow which can be used (directly or indirectly) for refrigeration purposes.

In this work, an inverse Joule Brayton air cycle has been studied with the employment of turbocharger units. Experimental tests have been performed in order to reproduce the state-of-the-art with a small automotive turbocharger unit. Measurements show Coefficient Of Performance (COP) smaller than unit together with minimum turbine exit temperature equal to -10°C. This is due to low components efficiency: the analysis of turbine and turbocompressor maps highlights a non-optimal coupling between them.

Secondly, basing on these considerations, two new air cycle layouts are proposed and analyzed. Calculations performed by means of a thermodynamic model show that higher COP and lower cycle minimum temperature can be achieved with the proposed new cycles by means of better turbine and turbocompressor matching and bigger turbocharger units with higher components efficiency.

INTRODUCTION

Refrigeration plants are crucial components in our daily life, since they are used for food preservation, air conditioning and many industrial applications like, for example, medical and pharmaceutical ones. Apart from cryogenic applications, almost the totality of refrigeration plants are based on vapour compression cycles. Historically, these cycles used refrigerants that were found to be Ozone Depleting Substances (ODS) and were banned in 1987 with the Montreal Protocol. However, ODS gases like clorofluorocarbons (CFC) and hydroclorofluorocarbons (HCFC) have been substituted by global warming potential (GWP) gases as hydrofluorocarbons (HFC) and perfluorocarbons (PFC).

Common Name	Formula	Chemical Name	GWP*
HFC-23	CHF_3	trifluoromethane	11,700
HFC-32	CH_2F_2	difluoromethane	650
HFC-125	C_2HF_5	pentafluoroethane	2,800
HFC-134a	$C_2H_2F_4$	1,1,1,2-tetrafluoroethane	1,300
HFC-143a	$C_2H_3F_3$	1,1,1-trifluoroethane	3,800
HFC-152a	$C_2H_4F_2$	1,1-difluoroethane	140
HFC-227ea	C_3HF_7	1,1,1,2,3,3,3-heptafluoropropane	2,900
HFC-236fa	$C_3H_2F_6$	1,1,1,3,3,3-hexafluoropropane	6,300
PFC-116	C_2F_6	hexafluoroethane (perfluoroethane)	9,200
PFC-14	CF_4	tetrafluoromethane (perfluromethane)	6,500
PFC-410	C_4F_{10}	perfluorobutane	7,000

Fig. 1 – Global Warming Potential (GWP) of the most common refrigerants, according to IPCC [1].

Figure 1 shows the GWP of the most used gases in refrigeration plants; such gases are constantly released in the atmosphere through plant leakages. Additionally, the Environmental Protection Agency (EPA) of the United States estimates that emissions of GWP gases will nearly double in the next ten years [2], as clearly reported in Fig. 2.

Exhibit ES-1: Baseline World ODS Substitute High GWP Gas Emissions (MMTCE)				
Source of High GWP Gases Used as ODS Substitutes	2005	2010	2015	2020
Refrigeration and Air-Conditioning	59.92	94.50	128.73	158.80
Partially and Fully Fluorinated Solvents	1.07	1.14	1.20	1.26
Aerosols	12.26	14.44	15.92	17.60
Foams	2.94	5.30	7.76	16.38
Fire Extinguishing	2.26	2.26	2.65	3.19
TOTAL	**78.45**	**117.66**	**156.26**	**197.23**

Fig. 2 – GWP gases emissions in the next years, according to EPA [2].

The previous data indicate that vapour compression cycles are still far from being considered sustainable for the environment, so that the search of alternative and environmentally benign refrigeration concepts is still going on. A complete review of the different alternative refrigeration technologies can be found in [3] and exceeds the scope of this work; however, in that review the air cycle is recognized as one of the most promising long-term solution for sustainable refrigeration at acceptable cost.
The idea of using air as operating fluid for refrigeration purposes dates back to the second half of XIX century, when the first machines were patented. Starting from these early applications, the idea of compressing, cooling and finally expanding air in an open cycle to produce low temperatures has been repetitively proposed and applied, for example in aircrafts and trains cabin conditioning. In the late years, however, environmental reasons have led to a renewed interest towards the potential of this completely sustainable technology in freezing and chilling. In 1998, Fleming et al. [4, 5], investigated the competitive position of the air cycle for heating and cooling applications by means of the construction of a pilot plant and computer-based design tools. They came to the conclusion that the application of the air cycle should be carefully chosen due to generally low cycle and components efficiency. Two years later, Institute of Environmental Sciences, Energy Research and Process Innovation (The Netherlands), TNO, released an exhaustive report about their air cycle research project [6]. The authors of the report firstly analyzed the air cycle capabilities (also including existing commercial air cycles) and then built and tested a prototype designed for Heating, Ventilating and Air Conditioning in Buildings (HVAC). They demonstrated the feasibility of the air cycle as well as the simplicity of this technology, which turns in reliability, cheapness and easiness of maintenance. The TNO pilot plant for freezing application (see Fig. 3) processes air in open and recuperated cycle: a two stage intercooled compression produces slightly pressurized air which is refrigerated by the recuperator and then expanded in the turbocharger turbine. Cold air resulting from expansion is used to refrigerate a freezing room, whereas discharging air is delivered to the recuperator.

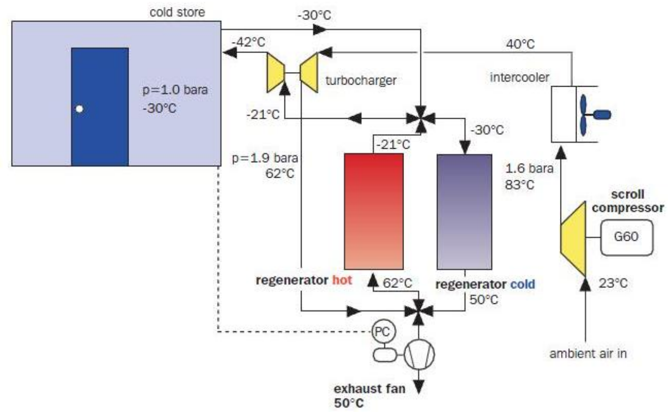

Fig. 3 – TNO air cycle pilot plant [7].

The prototype can produce air at -50°C with a Coefficient Of Performance of 0.72 [6, 7]; however, the authors calculated that air cycles can be competitive with vapour compression cycles if components with higher efficiency are used (particularly turbine). Also the recuperator has a large influence on the plant performance and had to be designed ad hoc. Air cycle potential has been assessed for road transport as well, especially food [8], and judged promising, although with a low COP derived from the available technologies.
Spence et al. [9, 10] built a demonstrator with almost the same layout of the TNO pilot plant but capable of fitting the envelope of existing trailer refrigeration units for road transport. They obtained an acceptable refrigeration performance, but with excessive fuel consumption, especially at full load. However, moving from the experimental data, the authors were able to identify the major deficiencies (turbomachinery and heat exchangers efficiencies) and then numerically predicted that an optimised air cycle can "potentially match the overall fuel consumption of the vapour-cycle transport refrigeration unit, while delivering the benefit of a completely refrigerant free system" [10], especially at part load.
Recently, the air cycle has been applied to a combined cooling-dehumidifying system [11], resulting in performances greater than the conventional desiccant systems.
Aim of this paper is to enhance the state of the art about air cycle technology with the proposal of two new layouts able to increase the efficiency of the components, i.e. improving the limiting factor of the past designs.
The next section describes the test bench employed to perform an experimental campaign aiming at understanding the main problems affecting the performance of the aforementioned open air cycles. Moving from this analysis, the innovative layouts are proposed in the subsequent section and their potential assessed with thermodynamic calculations. Finally, some conclusions are drawn about the work done.

EXPERIMENTAL TEST BENCH AND PRELIMINARY RESULTS.

The air cycle employed to perform experimental tests is sketched in Fig. 4: ambient air is compressed by a bootstrap compressor (a Roots blower has been employed due to its availability); then a fan cooler restores ambient temperature before entering the centrifugal compressor of an automotive turbocharger unit. Another fan cooler refrigerates warm air resulting from the second compression; compressed air at ambient temperature is finally expanded and discharged at low temperature.

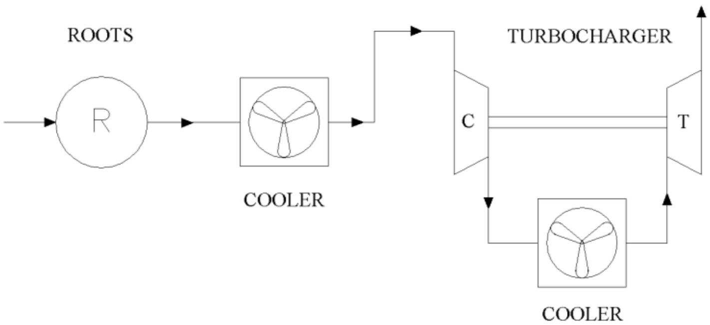

Fig. 4 – Test bench layout.

This scheme is simpler than those analyzed in the aforementioned works since there is no regeneration. Although this configuration limits the temperature of the cold flow at the end of the expansion, it should be recalled that the scope of the preliminary experimental campaign was only to detect the main deficiencies of the air cycle and not to test an optimized refrigeration plant. Therefore, the chosen configuration has been considered suitable and preferred to more complex layouts due to its simplicity. Figure 5 shows a view of the test bench, where compressed air from the Roots blower (nor represented) comes from the left. The turbocharger unit employed in the present study is the wastegated Garret GT 1541 (with blocked wastegate valve), usually used for rather small automotive applications (mounted on 1.3 l engines).

Fig. 5 – Test bench: turbocharger and fan coolers.

Almost all authors studying the air cycle have recognized turbomachinery efficiencies as the limiting factor affecting the overall performance [4-10]. One of the main objects of our analysis is to measure the efficiencies of the centrifugal compressor and of the turbine. The polytropic efficiencies of the two components are calculated with the following equations:

$$\eta_{yC} = \frac{\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{T_2}{T_1}\right)} \cdot \frac{k-1}{k} \tag{1}$$

$$\eta_{yT} = \frac{\ln\left(\frac{T_2}{T_1}\right)}{\ln\left(\frac{p_2}{p_1}\right)} \cdot \frac{k}{k-1} \tag{2}$$

where “1” and “2” subscript represent inlet and outlet conditions, respectively, and k is the ratio of specific heats. Since heat transfer is neglected in Eq. 1 and 2, the test bench has been insulated to reduce heat losses; the experimental equipment is made of thermocouples (type K and T), pressure transducers, and flow rate turbine meter (Elster Instromet SM-RI-X). Measurements are displayed in Fig. 6 and clearly indicate that the turbine is expanding air with low efficiency (ranging from about 0.3 to 0.4), whereas the compressor performs better, especially at large mass flows.

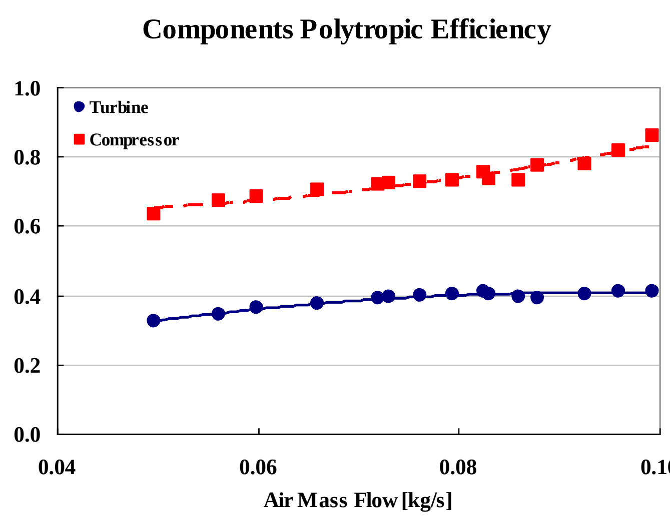

Fig. 6 – Compressor and turbine polytropic efficiencies.

Certainly, such a low turbine efficiency could not be accepted since this results in a lower power extracted from the working fluid, thus higher temperature at the end of the expansion and low compression rate. Moreover, the maximum achievable

(isentropic) efficiency certified by the manufacturer is 0.65, so turbine effectively works far from its potential best condition.
Figure 7 represents the effects of a reduced turbine efficiency: the temperature at the end of the expansion is always higher than -10°C, even at high air flows, and the overall pressure ratio feeding the turbine is relatively small due to poor pressure increase in the centrifugal compressor stage.

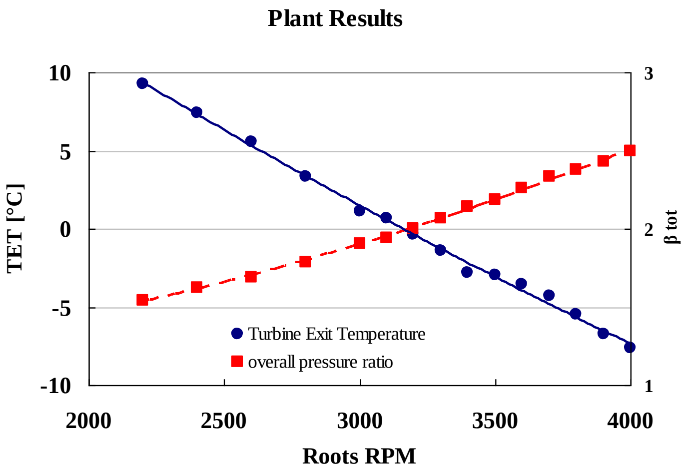

Fig. 7 – Measured turbomachinery efficiencies.

These considerations indicate that more suitable operating conditions should be chosen in order to obtain higher turbomachinery efficiencies and then better cycle performance. In other words, choosing turbochargers with high prescribed efficiency is not sufficient: optimal operating conditions are also mandatory; therefore, the optimal coupling between centrifugal compressor and turbine should be imposed during the refrigeration plant design process. This is different from simply adding a "plug-in" automotive-derived turbocharger designed to elaborate the same mass flow, due to different thermodynamic conditions. Since operating temperatures and pressures differ from those found on automotive applications in fact, the turbocharger equilibrium condition in the air cycle is likely to fall far away from the optimum planned by the manufacturer. Although completely purpose-made turbocharger designs are not available, there is a potential for a more convenient matching between off-the-shelf turbomachinery and air cycle typical operating conditions.
Finally, we want to underline that bearings employed in the automotive turbochargers do not guarantee that the delivered air is absolutely oil-free; this suggests to abandon the idea of employing the cold air directly for food refrigeration or freezing, as well as for other applications where air is breathed
In the next section two different solutions able to increase the turbomachinery efficiencies of the turbocharger is proposed and analyzed in detail.

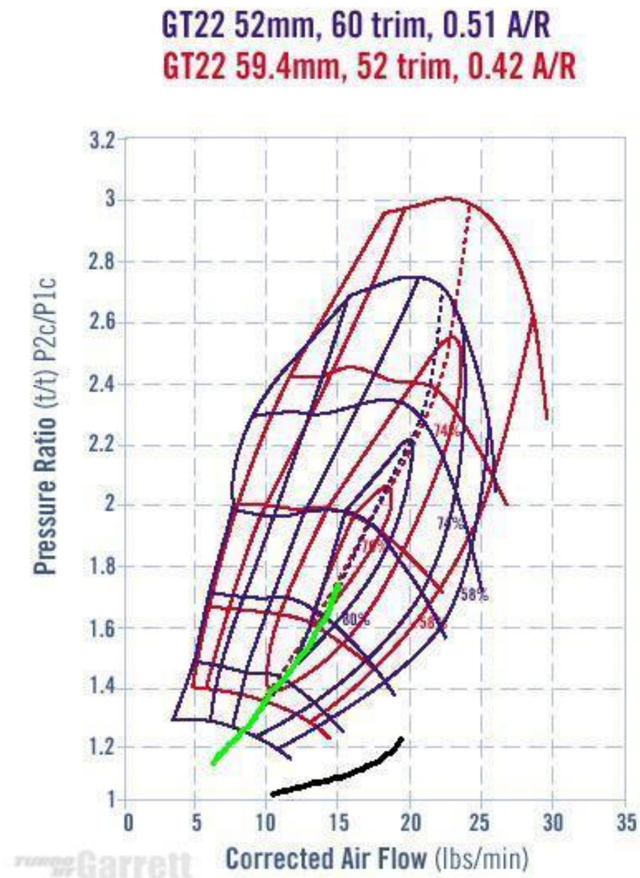

Fig. 8 – Centrifugal compressor map, courtesy Garret.

INNOVATIVE AIR CYCLE LAYOUTS

A possible amelioration of the air cycle is driven by two design aspects:

- an improved matching between plant operating conditions and turbocharger mechanical balance;
- the employment of larger turbochargers (capable of higher turbomachinery efficiencies) for assigned mass flow rate.

The first design driver aims at guaranteeing that the chosen turbocharger actually works at its highest potential. Given component maps, this can be achieved imposing suitable corrected mass flows and pressure ratios during the design process. An example of this procedure is explained in Fig. 8, where the compressor map of the Garrett GT22 turbocharger is shown. The line of maximum efficiency (highlighted in green) describes the ordered pairs of corrected mass flow and pressure ratios to be imposed in our design.
However, there are infinite ordered pairs that can be chosen: the best is the one which maximizes the efficiency of both centrifugal compressor and turbine. There are also some constraints to be considered:

- turbocharger shaft steady-state balance (the power delivered by the turbine equals the power required by the compressor plus the power dissipated in the bearings);
- feasible pressure rise in the bootstrap compressors, which must be operated with allowable pressure ratios;
- some temperatures are fixed by the presence of heat exchangers, e.g., the lowest temperature at recuperator inlet or at turbine inlet (if the recuperator is not employed) is ambient temperature.

The above mentioned design procedure is iterative: once some parameters have been fixed (i.e.: mass flow rate, ambient temperature, etc.) a control variable is iteratively changed until a satisfied matching is achieved. Thermodynamic transformations within plant components can be parameterized by means of 1D equations and turbocharger maps.

The second design driver aims at selecting an appropriate turbocharger with the greatest turbomachinery efficiencies: due to manufacturing tolerances, the efficiency increases as the geometric dimensions increase: very large turbochargers can easily reach compressor efficiency above 80% and turbine efficiency above 70%. However, larger turbochargers require larger corrected mass flow rates, which in turns leads to a larger plant flow rate and so enlarged boot-strap compressor and heat exchangers; thus, the applicability of an air-cycle refrigeration plant could be limited to very large plants.

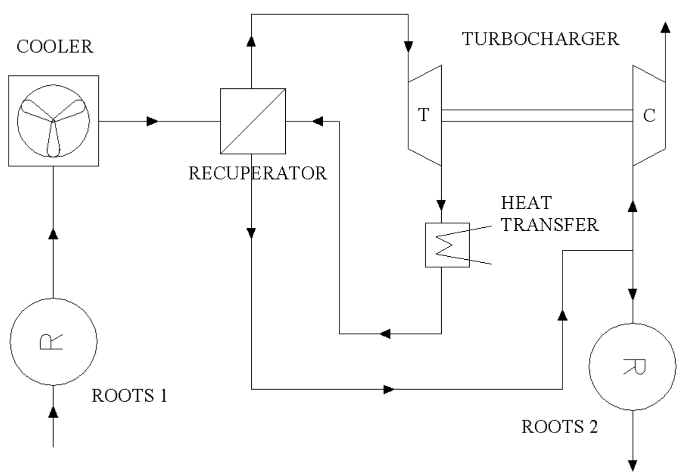

Fig. 9 – First layout proposal.

Keeping in mind the two design drivers described above, two new air cycle layouts are proposed, see Fig. 9 and Fig. 10. In the first layout, ambient air is slightly compressed (by a Roots blower or even by a fan), cooled by a cooler using ambient air and then in a recuperator; finally it is expanded in a turbine. At this stage, the very cold air at a pressure lower than ambient can be used for refrigeration purposes (generally indicated in Fig. 9 as "User") and then (in the recuperator) to lower turbine inlet temperature. Finally, ambient pressure is restored by means of another compression. However, the centrifugal compressor is not able of operating both at maximum efficiency and on the entire flow rate delivered by the turbine; for this reason, a second bootstrap compressor (it will also be referred to as complementary compressor) is inserted in parallel with the centrifugal compressor. Therefore the compressor operation is moved towards a lower corrected flow rate and a higher pressure ratio. The proposed layout enables to employ very large turbochargers (with the aforesaid advantages) with relatively small bootstrap compressors. In addition, it can be designed to optimize the matching between turbomachinery performance maps and the operating conditions typical of an air cycle refrigeration plant. The plant can be also operated in closed-cycle mode, by inserting an after-cooler between the exit of the second compression and the intake of the first bootstrap compressor. This would also allow to employ a different, already dehumidified, gas.

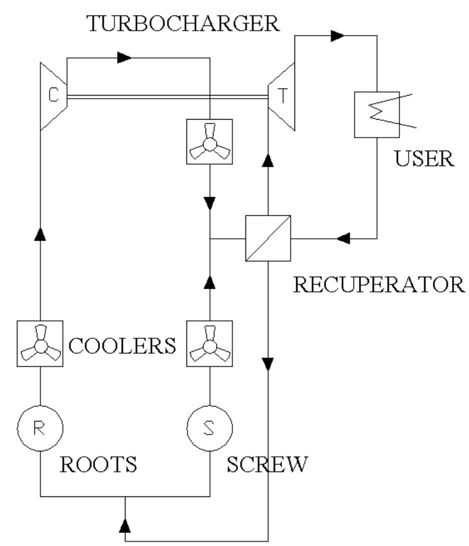

Fig. 10 – Second layout proposal.

In the second layout (Fig. 10) there are two compression branches: in the first one the gas follows the usual path (Roots compressor, cooler, centrifugal compressor), whereas in the second one a screw compressor is responsible for the entire pressure increase, as visible in Fig. 11, which provides the T-s diagram of the proposed second layout. The two path join together before entering the recuperator; air is then expanded in the turbine and can be used for refrigeration purposes. The cycle is closed with another passage in the recuperator, where the gas temperature is restored at ambient value and can be processed again. With the aid of the screw compressor, large turbochargers can be also employed in this layout, with the main advantage of a simplified plant management in comparison to the previous one. For a constant expansion rate in fact, the gas pressure is everywhere greater than ambient condition in the second layout, with evident advantages for piping sealing and overall plant cost. In addition, the second layout do not require a complementary vacuum compressor, which can be expensive due to special internal leakages and sealing. The second layout is simpler than the first one also because it requires only one inverter instead of two, as in the first proposal. When a mass flow regulation is required in fact, the ratio of the two volumetric flow rate elaborated by the two Roots in the first layout is not the same of the design point, since the minimum pressure varies. This means that the two Roots compressors have to be controlled by two different inverters, at increased plant cost. On the other hand, when off-design condition are required with the second layout, the ratio of the two volumetric flow rate is constant and equal to design point, since the inlet pressure is the constant (ambient pressure); consequently, a single inverter can be used.

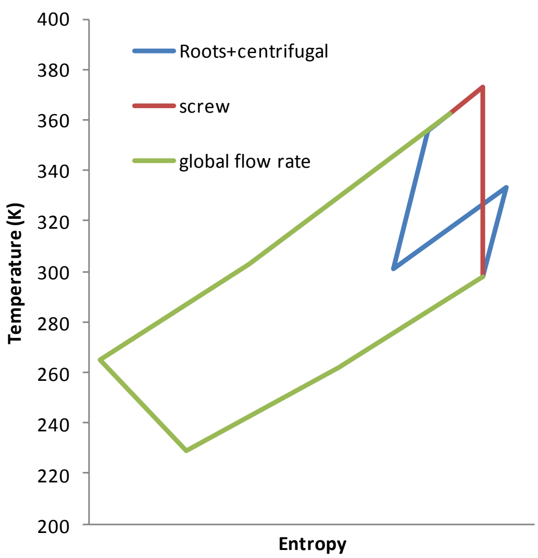

Fig. 11 – T-s diagram for the second layout.

The previous comparison of the two layouts is based on qualitative considerations; in order to properly assess the quantitative advantages of each plant, the aforementioned design algorithm has been applied to the two plant layouts and implemented in computer routines; some significant results are reported in the next section employing the Garret GT22 turbocharger.

COMPUTED RESULTS

The assessment of the proposed air cycle layouts and design improvements has been made: Fig. 12 compares the cooling capacity that can be obtained for different Turbine Exit Temperature (TET), i.e.: the minimum temperature achievable in the refrigeration plant. As clearly shown in the plot, the second layout has a larger cooling capacity at all TETs; that is, the second layout performs better over the whole operation range. This is highlighted also in Fig. 13, where COP is plotted against the minimum temperature.

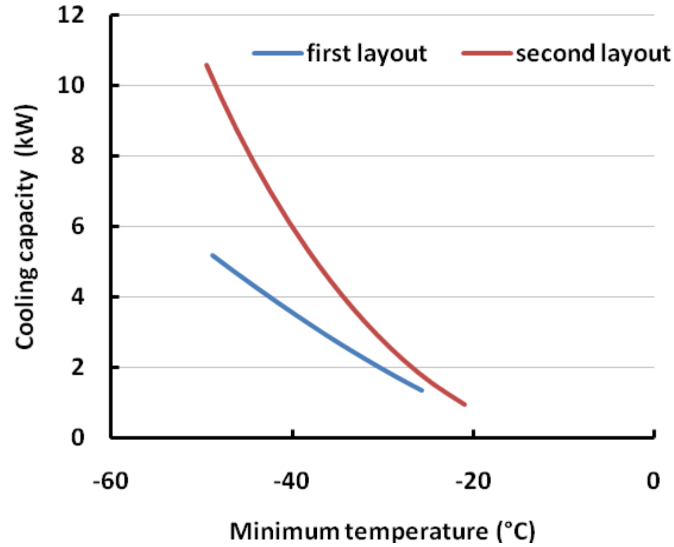

Fig. 12 – Layouts comparison (I).

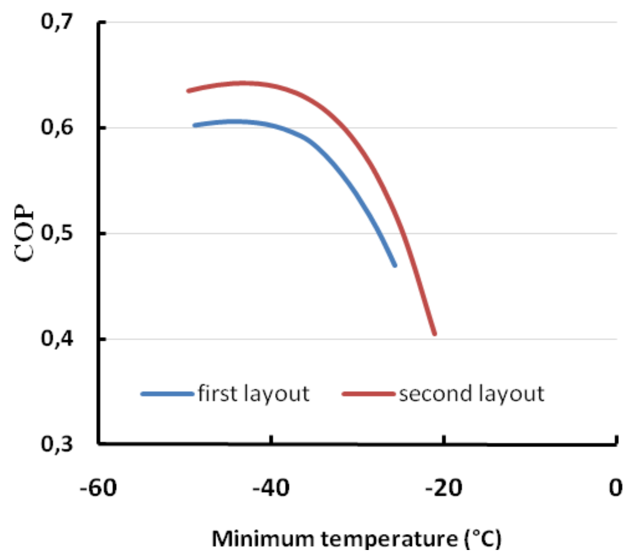

Fig. 13 – Layouts comparison (II).

The COP is defined as follows:

$$COP = \frac{\dot{m} C_p (T_U - TET)}{P_{BC1} + P_{BC2}} \quad (3)$$

where $\dot{m}$ is the air mass flow rate, C_p is the air specific heat at constant pressure, T_U is the temperature at the end of the heat transfer to the user, (recuperator inlet temperature on the lower pressure side), and (P_{BC1} + P_{BC2}) is the mechanical power required by the two bootstrap compressors. The COP of the second layout is always greater than the first layout, stating that not only it can extract more heat, but is also more efficient.

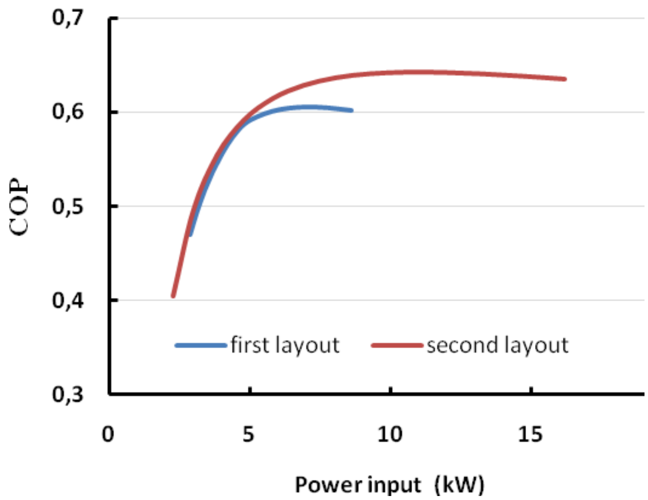

Fig. 14 – Layouts comparison (III).

From the aforementioned considerations then, it should be concluded that the second proposal is better than the first one, especially considering the plant cost discussed in the previous section. Nevertheless, Fig. 14 suggests that for a given absorbed mechanical power (corresponding to electric power input), the COP gain is very small using the second layout than the first; moreover, larger COP can be obtained only with the second layout, but at the price of a large energy expense. An economic trade-off between operating cost and initial investment should then be considered, particularly for small refrigeration plants.
Fig. 15 provides a comparison between the performance of the second scheme proposed in this paper and the performance of a standard plant based on the recuperative inverse Joule-Brayton cycle with intercooled compression, using only one bootstrap compressor. Both plants are supposed to employ the GT22 Garret turbocharger and the same recuperator. The black line in Fig. 8 represents the operating line of the state-of-the-art plant; as clearly visible in Fig. 8 and also demonstrated by the experimental tests presented above, the state-of-the-art plant, using only one bootstrap compressor, does not allow to operate the compressor at high efficiency.
Generally, it could seem that the overall COP of the two layouts is excessively lower than a common vapour compression plant, but some considerations have to be made:

- the minimum temperatures achievable with the proposed air cycle are lower than those achievable with vapour compression plants. For example, the described plant can be used for extremely fast freezing, and thus could be useful in the food industry;
- larger turbochargers with higher turbomachinery efficiency are available, so the plant COP can further be increased as well as the TET.

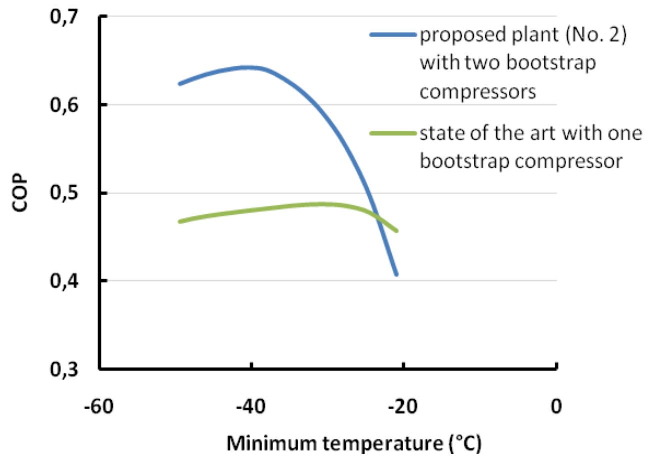

Fig. 15 – Performance comparison between the second layout and a state-of-the-art inverse Joule-Brayton plant.

As an example, Fig. 16 shows how the COP of the second layout can be improved when using a centrifugal compressor with higher efficiency (the efficiency of the GT22 compressor has been increased of 5% and 10%, respectively); as well, Fig. 17 shows the performance improvement associated to an increasing turbine efficiency: it is noteworthy that a higher turbine efficiency affects the COP twice, since it allows both to improve the mechanical behaviour of the turbocharger and to increase the extracted thermal power. Since higher turbocharger efficiencies characterize larger turbocharger units, we can conclude that the proposed layout becomes more attractive for larger size plants.

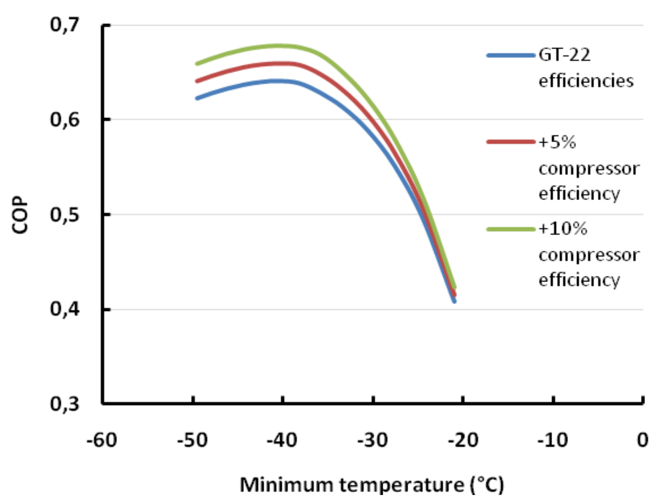

Fig. 16 – Performance of the second layout for higher values of centrifugal compressor efficiency.

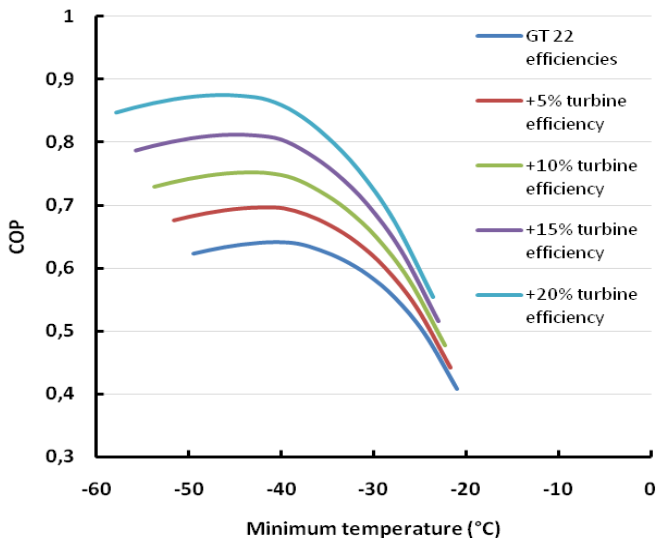

Fig. 17 – Performance of the second layout for higher values of turbine efficiency.

Further investigations should focus on these aspects together with the experimental validation of the computed air cycle.

CONCLUSIONS

Refrigeration plants currently in use employ Global Warming Potential (GWP) gases working in vapour compression cycles. Sustainable future plants should avoid the utilization of such refrigerants, and on-going researches are trying to develop some valid alternatives. Among them, air cycle has been recognized as the most promising technology, even if limited by poor turbomachinery efficiencies.

Starting from past works available in the literature, in the present work a test bench has been employed in order to assess the air cycle main features. It is basically an open cycle with a two stage, intercooled compression but without heat recovery. Performed measurements revealed poor performance (both in terms of COP and TET) due to low turbomachinery efficiency.

The authors highlighted the non optimal matching between refrigeration operating conditions and automotive-derived turbocharger design and proposed two new air cycle layouts. The proposals are based on two design drivers: 1) an imposed optimal matching between turbomachinery maximum efficiency conditions and plant operating conditions, and 2) the employment of larger turbochargers (able to achieve higher efficiencies).

The innovative layouts have been simulated by means of 1D thermodynamic equations and their performance compared. The second layout, with two separate compression branches, has a better performance than the first one both in terms of cooling capacity and COP, and should also be preferred in relation to its reduced cost. However, further economic analysis are necessary to assess the impact of operational cost as well, particularly for small refrigeration plants.

Incoming studies should also focus on COP improvement and experimental validation of the proposed layouts.

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NOMENCLATURE

Symbol	Quantity	SI Unit
COP	Coefficient of performance	
C_p	Air specific heat	J kg^{-1} K^{-1}
k	Adiabatic index	
$\dot{m}$	Air mass flow rate	kg/s
p	pressure	bar
P_{BC}	Bootstrap compressor power	W
T	temperature	K
TET	Turbine exit temperature	K
TIT	Turbine inlet temperature	K
T_U	User final temperature	K
η_y	Polytropic efficiency	

REFERENCES

[1] International Panel on Climate Change (IPCC), Special Report on Safeguarding the Ozone Layer and the Global Climate System, IPCC/TEAP, 2005, Cambridge University Press, UK.

[2] U.S. Environmental Protection Agency (EPA), Analysis of Costs to Abate International Ozone-Depleting Substance Substitute Emissions, EPA 430-R-04-006, June 2004.

[3] J. S. Lewis, I. Chaer and S. A. Tassou, Reviews of Alternative Refrigeration Technologies, Centre for Energy and Built Environment Research, School of Engineering and Design, Brunel Universvity, July 2007, available at www.grimsby.ac.uk/documents/defra/tech-newrefrigetechs.pdf.

[4] J.S. Fleming, B.J.C. van der Wekken, J.A. McGovern, R.J.M. van Gerwen, Air cycle cooling and heating – Part 1: A realistic appraisal and a chosen application, *International Journal of Energy Research* 22 (1998), **7**, 639–655.

[5] J.S. Fleming, B.J.C. van der Wekken, J.A. McGovern, R.J.M. van Gerwen, Air cycle cooling and heating – Part 2: A mathematical model for the transient behaviour of

fixed matrix regenerators, *International Journal of Energy Research* 22 (1998), **5**, 463–476.

[6] J. E. Verschoor (editor), Guidelines for the Application and Design of Air Cycle Systems for Heating, Ventilating and Air Conditioning in Buildings, TNO-MEP, Appeldorn, The Netherlands, 2001.

[7] TNO, Cooling, Freezing and Heating with the Air Cycle, Documentation Sheet, TNO Environment, Energy and Process Innovation, Department of Refrigeration and Heat Pump Technology, 2003, Available at: www.mep.tno.nl/Informatiebladen_eng/002e.pdf/

[8] S. A. Tassau, G. De-Lille, Y.T. Ge, Food transport refrigeration – Approaches to reduce energy consumption and environmental impacts of road transport, *Applied Thermal Engineering* 29 (2009) 1467–1477.

[9] S.W.T. Spence, W.J. Doran, D.W. Artt, Design, construction and testing of an aircycle refrigeration system for road transport, *International Journal of Refrigeration* 27 (2004) 503–510.

[10] S.W.T. Spence, W.J. Doran, D.W. Artt, G. McCullough, Performance analysis of a feasible air-cycle refrigeration system for road transport, *International Journal of Refrigeration* 27 (2005) 381–388.

[11] S.S. Elsayed, T. Miyazaki, Y. Hamamoto, A. Akisawa, T. Kashiwagi, Performance analysis of air cycle refrigerator integrated desiccant for cooling and dehumidifying warehouse, *International Journal of Refrigeration* 31 (2008) 189–196.