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NEW HEAT EXCHANGERS FOR HIGH EFFICIENCY CO_2 PLANTS

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ABSTRACT

CO2 refrigeration systems are now in widespread use in many applications, especially in the retail sector.

In fact CO2 is an excellent, environmentally-friendly refrigerant which requires very efficient heat exchangers to reach high performance. LU-VE recently made developments with the aim of further improving the design of the finned heat exchanger, using also new technology with a very compact heat transfer matrix with tube diameter of only 5mm. The new heat exchanger configuration is designed both for gas coolers and unit coolers. The paper describes the activity of designing and testing the new geometry inside the CO2 test plant of the company.

The new geometry provides an excellent balance between performance and reduced quantity of refrigerant in the circuit.

A further development was made in the heat pump sector, in particular for supermarket applications. The decision to integrate the refrigeration and heating systems in these buildings is becoming increasingly common. However, in order to guarantee a sufficient quantity of heat to warm the supermarket even on the coldest days, with temperature below -10°C, an additional evaporator, positioned outside, must be added. This involves a special machine for which LU-VE has developed innovative technologies, capable of achieving very high levels of heat transfer efficiency, as described in this article.

Keywords: CO2, gas coolers, heat exchangers, evaporators, heat pumps.

1. INTRODUCTION

Environmental sustainability has become a key element in our society. Also within the refrigeration and air conditioning sectors there has been a great deal of discussion about this topic. At first, attention was concentrated on ozone depleting refrigerants, then the focus turned to greenhouse effect reduction, following the TEWI approach (i.e. combining direct and indirect emissions). More recently, the use of natural refrigerants, especially CO2 is becoming one of the main development drivers.

Another key parameter is the reduction of refrigerant charge.

Over the last few years both authors have therefore carried out wide-ranging research activities in different directions, analysing all the possible alternatives and finally developing a completely new coil geometry, based on copper aluminium technology with tube diameter of only 5mm. There are several advantages to such a solution; this paper explains the R&D activity regarding the new geometry, its main features, and the application of this technology to the development of an innovative heat pump developed within the European funded project named NxtHPG.

1.1 New coil geometry

In order to satisfy the request for high heat exchange efficiency and low refrigerant charge, the new MINICHANNEL coil geometry has been developed as shown in Figure 1. It is very compact and reaches very high density of capacity/fin surface.

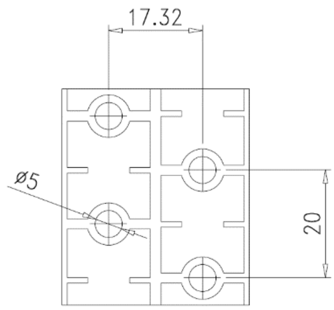


Figure 1. New MINICHANNEL coil geometry

The fins have special corrugations that combine with internal grooved tubes increasing the internal surface > 1.8 to give very high performance.

The coil headers can be chosen according to the refrigerant flow, without any particular mechanical limitation and consequently their volume can be reduced.

2. CFD APPROCH

The traditional approach followed by heat exchanger designers was traditionally focused on the selection of the global coil characteristics: tube diameter and length, tube and fin spacing, thickness and row number, in the attempt to obtain the best compromise between heat transfer performance, industrial costs and fan characteristics. In the past, the main choices and the solutions adopted were primarily based on experience and empirical correlations, derived from experimental tests [1], [2]. Less attention was given to the true core of the heat exchanger and to the behaviour of the complex flow field crossing the coil. This empirical approach is appropriate when used for coils of simple geometry, such as those with plain tubes and fins, but it is not warranted when applied to the design of modern, state-of-the-art coils, using rippled tubes and fins with sophisticated shapes.

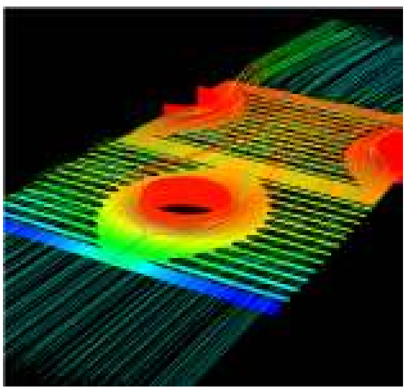


Figure 2. CFD simulations

CFD has always been of great help in the heat transfer field of application [3]. Nowadays, the great advances of computational techniques and the availability of more and more accurate and flexible numerical models, together with the growth of competences and know-how of researchers and engineers in the field of CFD, make the implementation of new strategies for advanced heat exchanger design feasible and convenient [4], [5], [6]. The approach described in this paper is based upon the massive use of numerical simulations with the goal of discovering the profound details of the fluid flow in order to gain a major understanding (i.e. based on the principles of fluid dynamics) of their heat performance and pressure losses. CFD is combined with an extensive experimental approach. In this frame, each of the two approaches gives a fundamental contribution: the former is able to support engineers by providing a fast in-depth analysis of the flow field for selecting the best fin shape; the latter is of key relevancy for measuring the predicted coil overall performance, as well as for validating numerical calculations.

More than 30 different fin configurations were investigated. A set of 2D calculations, coupled with wind-tunnel experiments, was carried out at various air velocities and fin pitches in order to compare the numerical and the experimental trends for each configuration.

This intensive activity gave valuable suggestions about the calibration of the computational tools and the influence of the fin shape on heat transfer performance. The results of the research activity demonstrate the new frontier opened by the optimization process for the fin shape and prove how this feature is relevant in enhancing overall coil performance, thus confirming that CFD is able to effectively support advanced heat exchangers design.

3. EXPERIMENTATION AND SOFTWARE CALIBRATION

The new MINICHANNEL geometry was initially tested using conventional R507A. Tests were carried out inside a calorimetric room, according to the international standard ENV 327. The testing equipment is composed of an internal room where the product (in this case the condenser) to be tested is placed and kept at constant temperature, thanks to a cooling system that gives the same capacity of measuring condenser. The external room ensures that the internal one has minimum thermal losses. The thermal capacity is measured twice: the first time by directly measuring the condenser capacity, a second time by measuring the capacity on the internal cooling system. The two capacities at constant room temperature can have a max difference of ±4%. The refrigerant used inside is R507A. The room temperature during test was 25±0.5°C and the condensing temperature varied from 35°C to 50°C, in all test conditions. Precision of temperature sensors is ±0.2°C, ±0.02 bar for pressure sensors and ±2.0% for flowmeters.


Figure 3. Picture of calorimetric testing room

Many tests were carried out to define performance under the most varied operating conditions (e.g. at different frontal air speed, mass velocity of the inside fluid, condensing temperature and air temperature at the inlet of the heat exchanger). All these tests helped to calibrate the calculation code in order to be able to estimate the performance of every unit with new geometry in the most varied operating conditions.

The Figure 4a graph shows the comparison of experimental data of thermal capacity with data determined using the calculation code; as can be seen, the deviation is very small, less than 4%. This confirms the reliability of the calculations in different operating conditions.

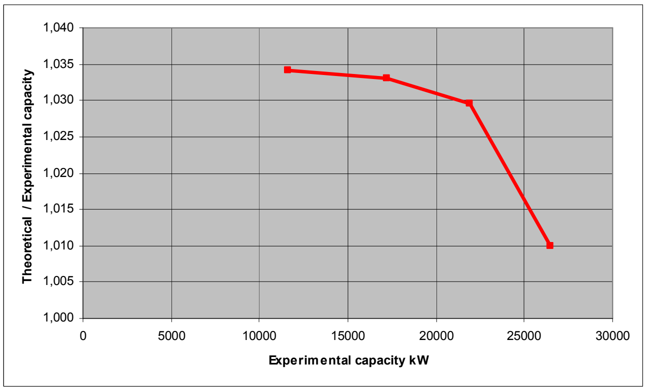


Figure 4a. Deviation between calculated and measured capacity

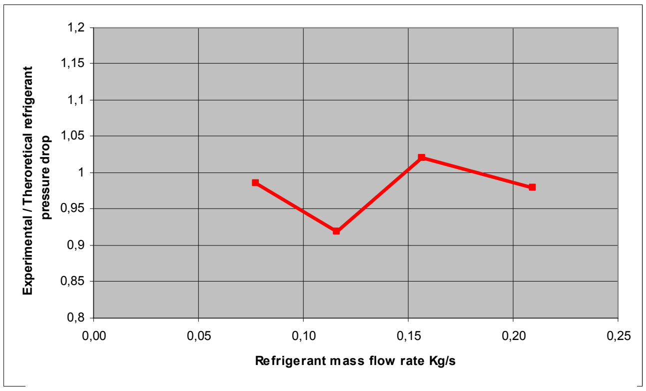


Figure 4b. Deviation between calculated and measured refrigerant pressure drop (standard tube)

The Figure 4b graph also indicates the comparison of experimental data with calculation data concerning refrigerant side pressure loss (inside tube) as a function of the mass flow rate.

Normally accepted tolerances are of the order of ±20%; in these experimental cases the error is comfortably within this interval (±8%).

This experimental campaign enabled the generation of a solid database for the further development of the MINICHANNEL technology.

4. APPLICATION OF NEW COIL MINICHANNEL TO NxtHPG PROJECT

The NxtHPG project provided an opportunity to extend the application of MINICHANNEL technology to CO2 fluid. This is a European project dedicated to the development of innovative heat pumps and it has the following objectives:

1) The development of a set of safe, reliable, and high efficiency heat pumps working with natural refrigerants (hydrocarbons and CO2) of capacity in the range from 40 to 100 kW;
2) Reach higher efficiency (10 - 20% SPF improvement) and lower carbon footprint (20% lower TEWI) than the current state of the art of HFCs/HFOs or sorption heat pump technologies;
3) Keep the cost very similar or only slightly higher (10%) in a way that the better environmental performance clearly compensates for the extra cost.

The project (financed by the EU) is coordinated by the Polytechnic University of Valencia and entails the participation of 5 universities, 1 research institute and 6 companies from the HVAC&R sector, involving a total of 6 countries.

Five heat pump prototypes were designed, 3 operating on HCs and 2 on CO2. This article illustrates the detail of the results from Case 5.

4.1 Case 5 of NxtHPG project; first test campaign

We can now describe the Case 5 heat pump in detail. This is a water-air machine, operating on CO2 and intended for the production of hot water at high temperature (80°C), designed to replace gas boilers for domestic heating. Figure 5 shows a diagram of the machine, while figure 6 shows the architecture and the layout of the principle components. Under design conditions with ambient temperature of 0°C, it can supply hot water at 80°C with thermal power of 50kW. The inlet temperature of the hot water to the machine is 40°C.

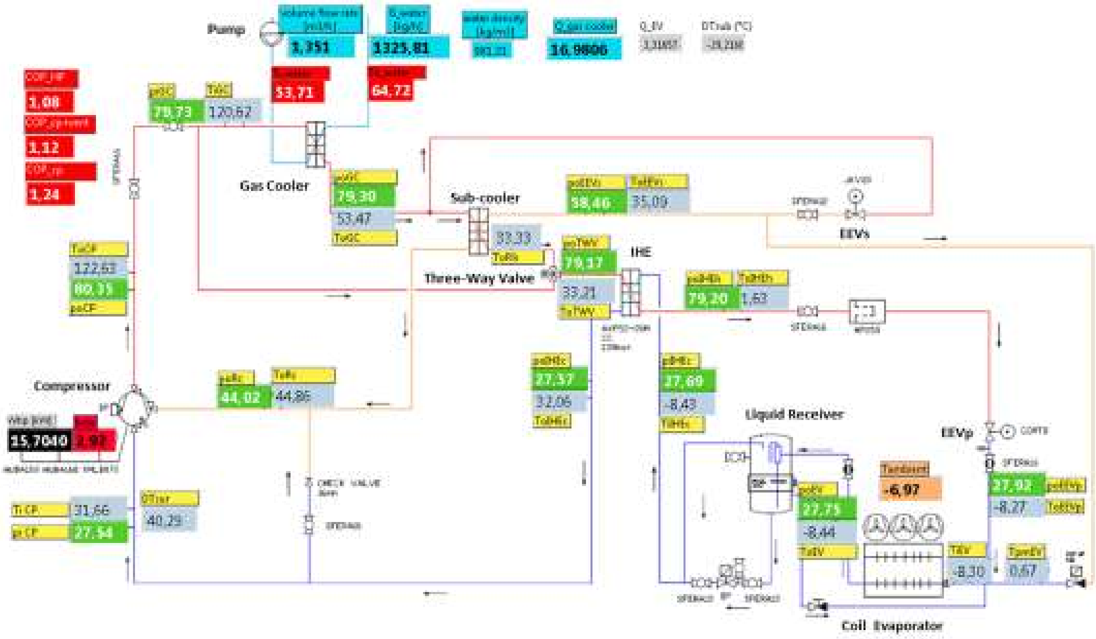


Figure 5: layout and instrumentation of case 5

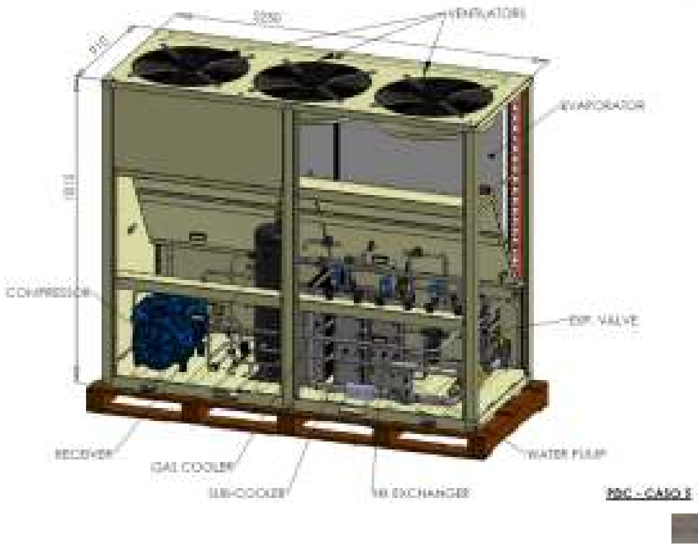


Figure 6: Picture of heat pump of case 5 **Figure 7: heat pump inside the testing room**

The heat pump was tested in the ENEA Laboratory in Rome, Italy, inside a calorimetric room (figure 7). Figure 8 shows the characteristics of the test plant.

ENEA Calorimeter Features	
Internal dimensions	4,70 m [L] x 5,50 m [P] x 4,60 m [H]
Useful area	26 m²
Useful Volume	120 m³
Temperature control	-15°C ÷ 35°C
Relative humidity control	10% ÷ 95% (managed from 10°C to 35°C)
Air speed	< 1 m/s (UNI EN 14511-3:2011 Appendix A –A.1.2)
Maximum absorbed electrical power	80 kW
Maximum dissipated cool power	50 kW

Figure 8: characteristics of the ENEA calorimetric room

To measure the operating conditions of the heat pump as the principle operating parameters varied, a grid was identified with 53 points. The ambient temperature was made to vary from -15°C to +35°C. The outlet temperature of the hot water from the machine was made to vary from +52 to +82°C, while the return water temperature to the machine was between +20°C and +55°C. In this way a very wide function field was obtained, ideal for thoroughly evaluating the performance of the machine. Figure 9 below shows the result of the COP of the machine as a function of the ambient temperature for various temperatures of inlet hot water to the same machine.

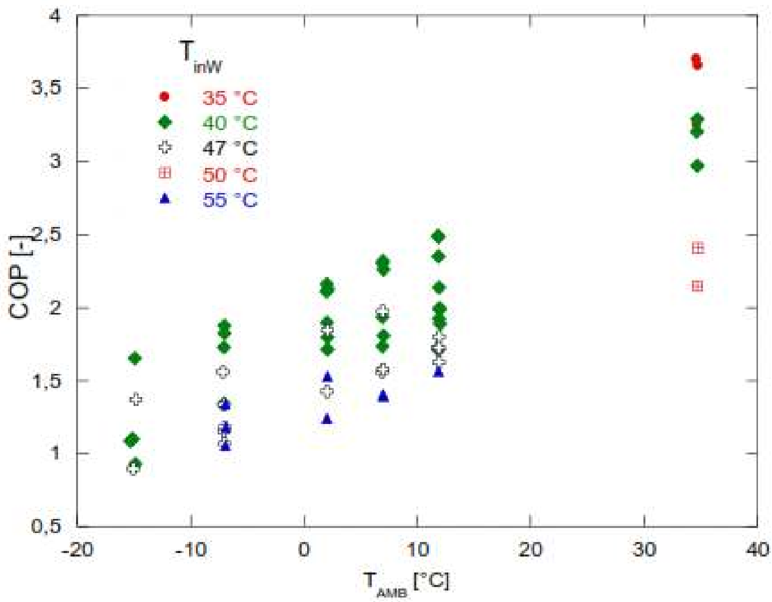


Figure 9: test results; COP as function of ambient temperatures, for different water inlet temperatures

We can now move on to an examination of a CO2 evaporator, concentrating our main attention on it. This is a finned heat exchanger produced using MINICHANNEL technology (namely tubes with a diameter of 5mm.), as illustrated in paragraph 1.1. Figure 10 shows the said exchanger.

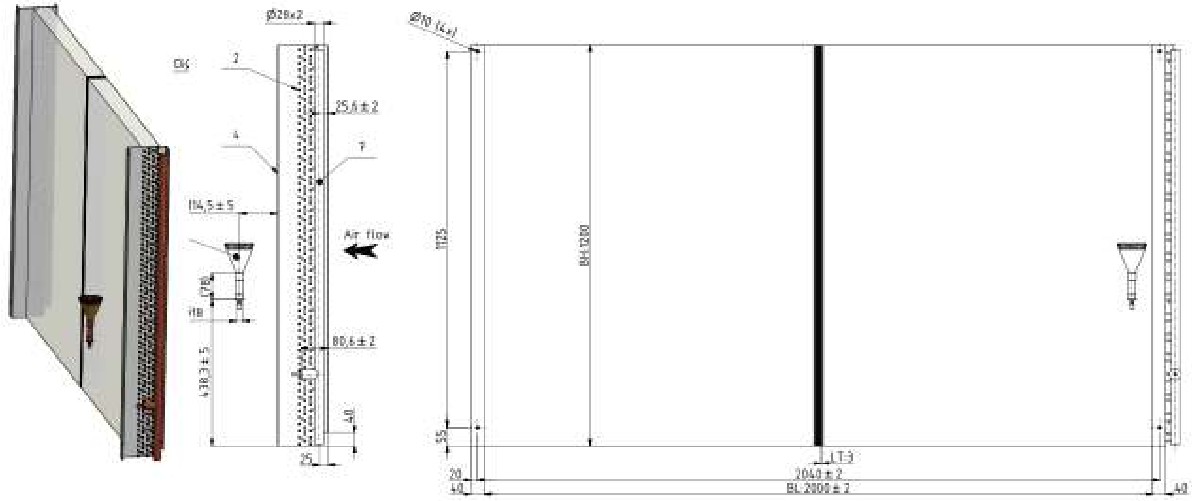


Figure 10. CO_2 evaporator, Case 5. The coil is made up of 60 tubes with 5mm diameter, 4 rows, 40 circuits. The finned coil length is 2000mm

Starting from the measurements made of the entire machine (to which were then added measurements of air quantity) we obtained the experimental measurement of the capacity of the evaporator. We therefor simulated the performance of exchanger using company software, obtaining the deviation between expected data and test data. The verification showed a deviation between experimental data and simulated data generally within 10%, certainly very positive data (figure 11). Figure 12 shows instead the pressure drop of the exchanger as a function of refrigerant flow rate; the red dot represents the data calculated theoretically in design conditions, which is well in line with the experimental data.

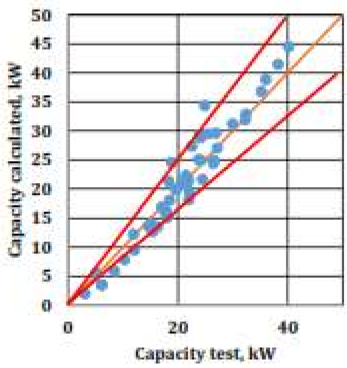


Figure 11: Comparison between experimental and calculated thermal power, Case 5.

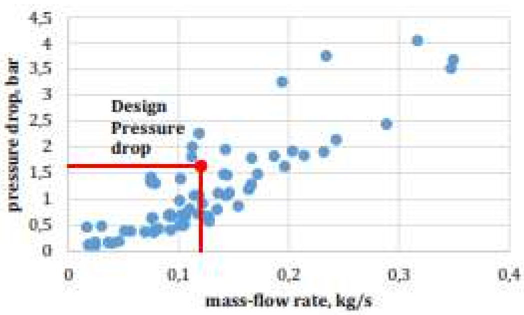


Figure 12: mass-flow rate VS pressure drop inside the whole coil (distributor+tubes+header), Case 5.

4.2 Case 5 of NxtHPG project; CFD simulation

It is then interesting to notice that the layout of the whole system may cause poor distribution of air on the coil, which could lead to a reduction of the effective air flow-rate. This effect is generally normal when the heat exchanger is set in the position of Case 5 (Figure 13). Indeed on the upper-side part of the coil, the flow must be disadvantaged (the fan-shroud actually covers part of the tubes). This fact is further shown by thermo-camera images (Figure 13b).

Figure 13.Coil position inside heat pump: (a) Thermocouple position, (b) Thermo-camera image

In order to better understand the effect of poor air distribution on the exchanger, with consequent poor distribution of refrigerant, the company carried out a CFD study. This study then compared the present solution with an alternative one in which the fan shroud no longer faces the inside of the machine but instead faces outwards from it.

In Figure 14 the total pressure fields for the tested solutions are presented. The pressure fields are very different.

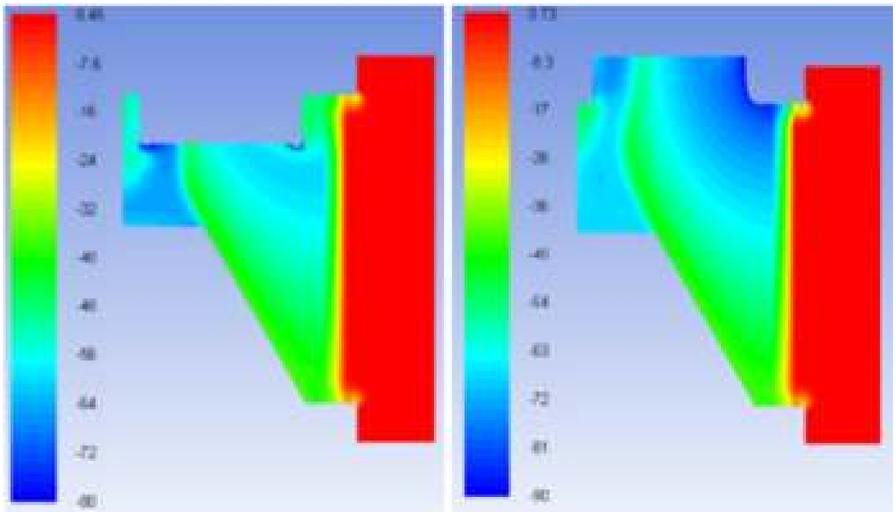


Figure 14. Pressure field for the original solution (sx) and for the modified solution (dx)

In Figure 15 the velocity contour is presented; the velocities are lower for the original solution, however a greater acceleration at the entrance of the fan is visible due to the greater curvature. This section is characterized by the greatest dissipations leading to a decrease in the mass flow-rate. Indeed the total value of mass flow-rate for the original solution is 4.0 kg/s and 4.44 kg/s for the modified solution, making an increase of 11%.

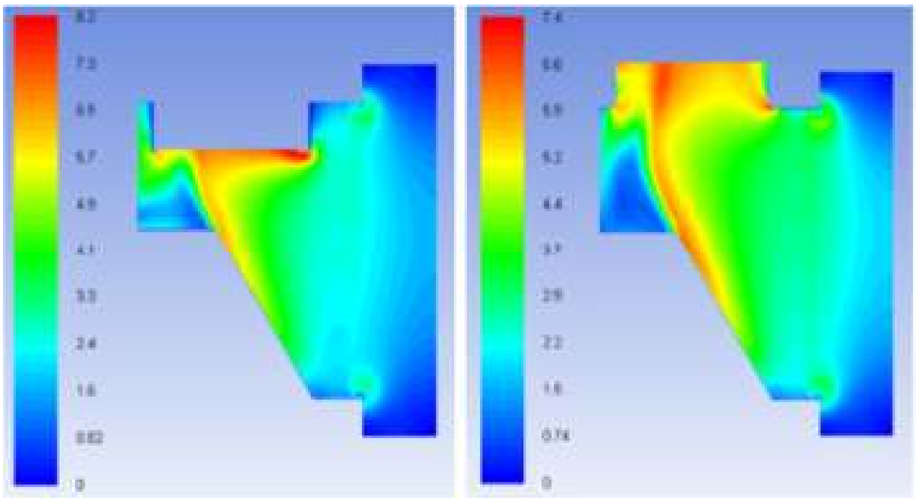
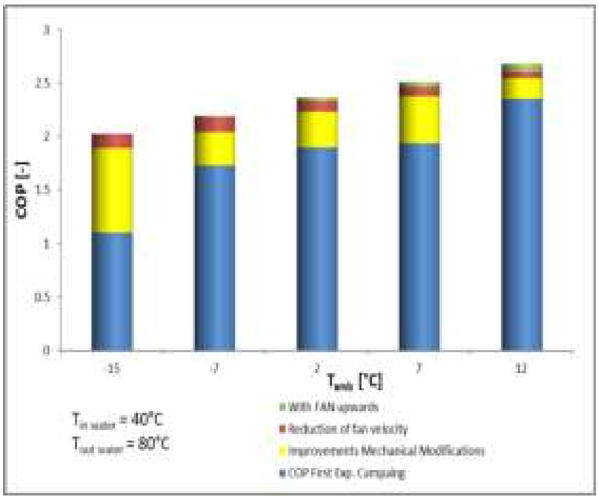


Figure15. Velocity field for the original solution (sx) and for the modified solution (dx)

4.3 Case 5 of NxtHPG project; second test campaign

After the first series of tests, many improvements were made to the heat pump, involving various components including compressors, some plate exchangers, the control algorithm and the position of the fan shroud for the ventilation of the evaporator. The finned heat exchanger on the other hand remained unchanged. In addition, the original fans of the evaporator were replaced with analogous models fitted with EC motors, operating at a velocity reduced by 20% in order to save energy.

Figure 6 shows the final result of the study with improvement of the COP, divided between 3 contributors: mechanical (compressor, plate heat exchanger); EC fans; fan shroud directed upwards. The positive results connected to this last aspect confirmed the forecasts done beforehand using the CFD method.



Ambient temperature [°C]	-15	-7	2	7	12
COP First Exp. Campaign	1.1	1.73	1.89	1.93	2.35
Improvements Mechanical Modifications	66.9%	17.9%	17.7%	22.6%	8.1%
Reduction of fan velocity	6.8%	6.8%	4.4%	3.6%	2.4%
With FAN upwards	-0.1%	0.7%	1.4%	1.9%	2.8%
FINAL COP	2.10	2.19	2.36	2.50	2.67
Total improvement	78.2%	26.8%	24.6%	29.5%	13.95%

Figure 16: COP improvements in the second testing campaign

5. CONCLUSIONS

This article has shown the development of the new MINICHANNEL heat exchanger geometry featuring tube diameter of only 5mm. This solution permits high efficiency heat transfer combined with reduced internal volumes of the exchanger itself.

This particular configuration was applied to the NxtHPG project, which is aimed at the development of a new generation of CO2 heat pumps. The performance of the innovative heat exchanger was very good and, thanks to an accurate study of the fluid dynamics of the machine using CFD methods, it was possible to make a decisive contribution to achieving high performance in the heat pump.

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