The use of propane as refrigerant in air conditioning applications. The outcome of an international research project: LIFE- ZeroGWP

Webinar, June 22, 2021, 10.00 a.m.

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Cultura e Tecnica per Energia Uomo e Ambiente

Recent developments and research perspectives about flammable refrigerants

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OUTLINE

- **Hydrocarbons** such as low GWP substitutes of R410A
  - Optimization of an **ideal vapor compression cycle** working with R290
  - Optimization of a **real vapor compression cycle** working with R290
  - Optimization of **heat exchangers** working with R290

- Numerical simulations of **propane leakage scenarios** in closed environments

  - **Validation** against experimental results
  - **Comparison** between new experimental data and numerical results
  - Prediction of **different scenarios**

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NIST thermodynamic screening

100,000,000 chemical compounds

FILTERS:

Molecules with 15 or fewer atoms

Atoms of C, H, N, O, S, F, Cl, Br [Midgley, 1937]

			H 1.008
C 12.011	N 14.007	O 15.999	F 18.998
		S 32.065	Cl 35.453
			Br 79.904

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NIST thermodynamic screening

STARTING FROM 56,203 COMPOUNDS	
FILTERS	remaining compounds
GWP < 200	52,265
toxicity	30,135
flammability LFL > 0.3 kg/m^3	20,277
Critical temp: 300 K < T_{crit} < 550 K	1,728
Stability	1,234
Thermodynamic efficiency	62

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SELECTED MOLECULES

HFCs
many in class A1 ASHRAE; high GWP

HCs (& related compounds)
flammable, class A3 ASHRAE

NH_3
excellent thermodynamic properties ASHRAE B2L

CO_2
ASHRAE A1 High operating pressures, hypercritical cycle

HFOs (& related compounds)
new, not fully characterized only some currently available on the market

Hydrocarbons as refrigerants

1. **R290** propane
2. **R600a** isobutane
3. **R1270** propilene

Pure fluids. Commercially available.

1. R432 A,B,C (R1270/R-E170)
2. R436 A,B (R290/R600a)
3. R441A (R170/R290/R600a/R600)

Mixtures. Commercialy barely available

Thermophysical properties comparison:

1. Liquid thermal conductivity

Fluid	Temp. [°C]	λ_{liq} [W/m K]
R290	50	0.083
	10	0.101
R600a	50	0.081
	10	0.095
R410A	50	0.076
	10	0.098

Liquid thermal conductivity has a fundamental role in the two phase heat transfer, especially during condensation.

Thermophysical properties comparison:

2. Liquid density

Fluid	Temp. [°C]	ρ_{liq} $[kg/m^3]$
R290	50	448.6
R410A	50	906.8

Lower liquid density implicates a lower refrigerant charge once the total volume is fixes

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Thermophysical properties comparison:

3. Latent heat

Fluid	Temp. [°C]	r [kJ/kg]
R290	10	360.3
R600a	10	344.6
R410A	10	208.6

Thanks to the higher latent heat, HCs present a lower mass flux at the same cooling capacity.

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- The major limitation in use HCs is surely the **high flammability** that requires a special design of the system components and, above all, **refrigerant charge minimization**.

- Compact air conditioning systems without external remote heat exchangers

(as for example double duct air conditioners) offer promising opportunities for refrigerant charge minimization, given the relatively shortness of refrigerant lines.

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In compact air conditioning systems, given the relatively small charge achievable, the **indirect environmental impact**, due to CO_2 emissions generated by energy consumption in use, is much larger than the **direct effect** consequent to the refrigerant release in ambient.

Therefore, it is mandatory to minimize the refrigerant charge but also **maximize the energy efficiency** of the air conditioner

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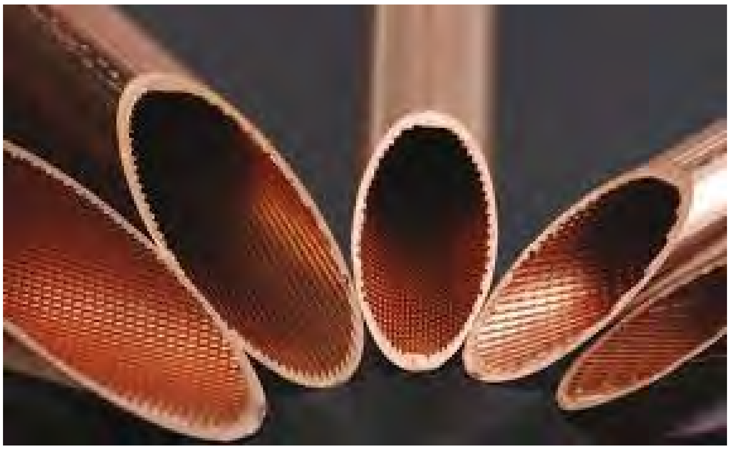
The **heat exchangers design** has a strong impact on the heat exchangers efficiency

Consequently they have a marked effect on the energetic efficiency of the air conditioning machines



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IDEAL VAPOR COMPRESSION REFRIGERATION CYCLE OPTIMIZATION: R290 vs R410A

The cycle chosen is meant to be representative of typical comfort cooling operating conditions:

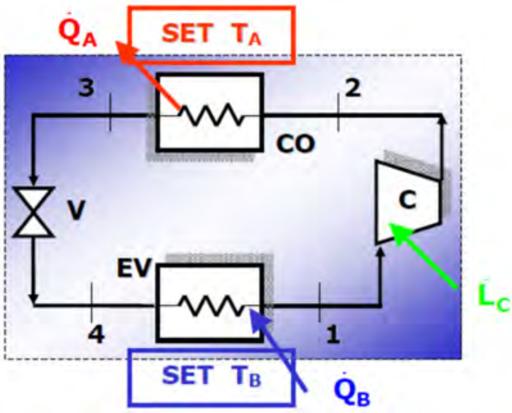
constant condensation temperature of 50° C,

an evaporation temperature of 10° C,

5 K vapor superheat;

no condensate subcooling,

0.7 isoentropic compression efficiency



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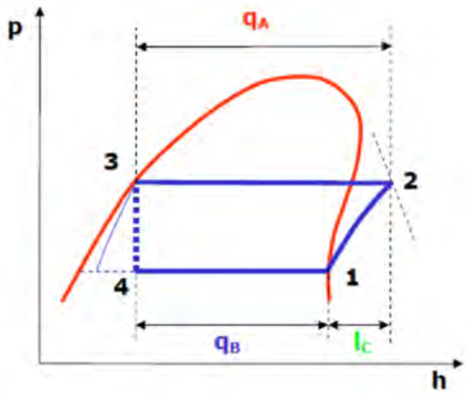
$$COP = \frac{\dot{Q}_B}{L_C}$$

	VCC (kJ/m³)	COP (-)	Sat. liquid density (kg/m³)
R410A	5879	3.52	906.8
R290	3430	3.80	448.9
Variation	-42%	+8%	-50%



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1. R290 **volumetric cooling capacity** is about 42% lower. In fact, the R410A suction vapor is about 3 times denser while the refrigeration capacity per mass of R290 is about 1.8 times the R410A one.

2. the **compression work** is some 40 % lower with R410A. Considering that R290 has 1.8 times larger refrigeration capacity per mass for R290, the COP with the HC is +8%.

3. the **lower liquid density** of the HC permits to markedly reduce the refrigerant mass inventory in the circuit. However, in order to minimize the charge, it is mandatory to reduce the finned coil tubes diameter.

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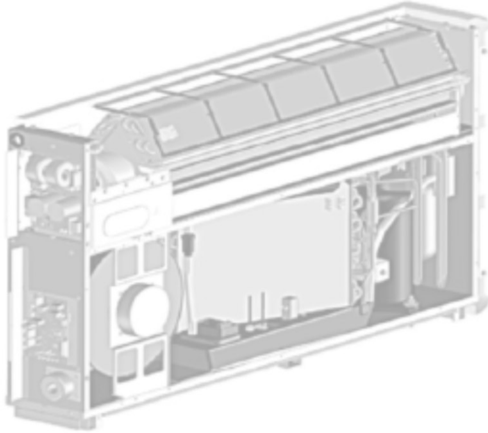
REAL VAPOR COMPRESSION REFRIGERATION CYCLE OPTIMIZATION: R290 vs R410A

An off-the-shelf "Double Duct" room Air Conditioner/heat pump (DD-AC) operating with R410A was taken as benchmark



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Finned coil heat exchanger geometry (R410A)

		Outdoor air coil (OAC)	Indoor air coil (IAC)
Tube arrangement		Staggered	Staggered
Tube material		Cu	Cu
Finned coil length	(m)	0.3	0.7
Outside tube diameter	(mm)	7	7
Inside tube diameter	(mm)	6.3	6.3
Longitudinal tube spacing	(mm)	12.7	12.7
Transverse tube spacing	(mm)	21.0	21,0
Fin geometry (material)		louvered (Al)	Louvered (Al)
Fin spacing	(mm)	1.4	1.3
Fin thickness	(mm)	0.1	0.1
Number of rows		4	2

As design criteria for the development of it was decided to keep the **R290 unit as close as possible to the off-the-shelf R410A appliance**.

the external overall dimensions, the fans and the air ducts were kept unchanged, as well as the overall heat exchangers dimensions.

In order to reduce the refrigerant hold-up, commercially **available copper microfin tubes** with 5 mm nominal OD were adopted for R290 instead of the 7 mm OD ones, used for R410A.

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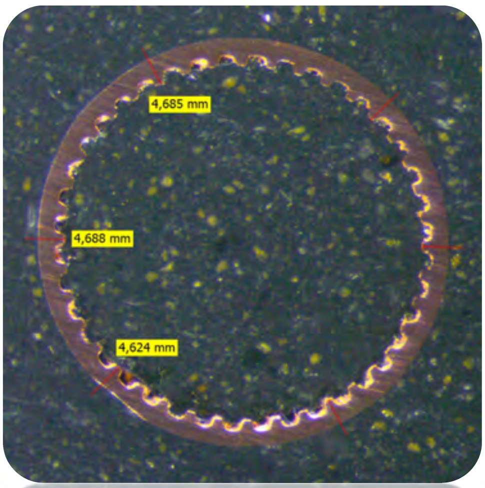
Photo of the microfin tube used for the experiments



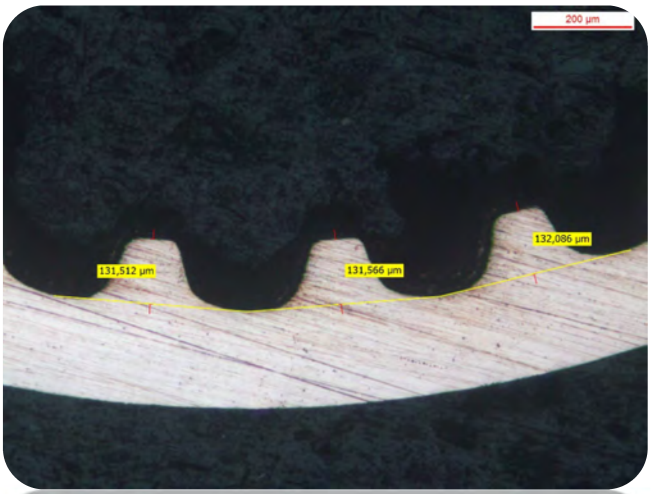


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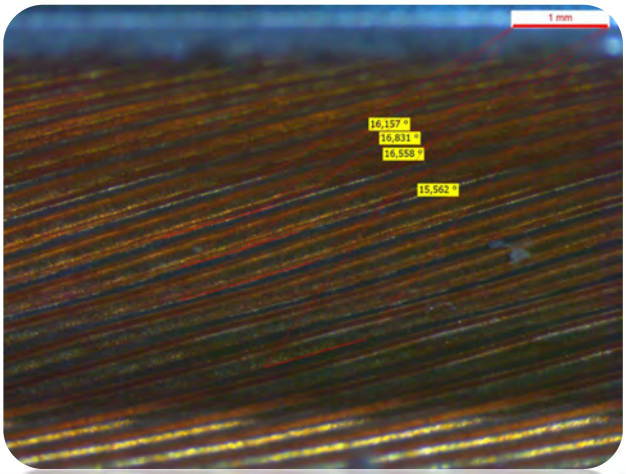
Photo of the microfin tube used for the experiments



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Thermodynamic analysis

Performance potential of different refrigerants expressed as a combination of two temperature differences

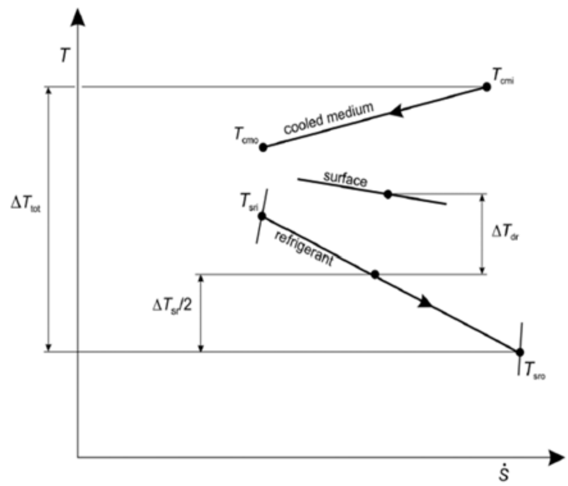
$$TTP = \Delta T_{dr} + 0.5\,\Delta T_{sr}$$

saturation temperature drop
due to the refrigerant pressure drop

ΔT_{dr}
driving temperature difference $\Delta T_{dr} = \dfrac{m\,\Delta h_{lg}}{\alpha}$

$\Delta T_{sr} = t_{sat}(p_{in}) - t_{sat}(p_{out})$

The optimal circuit length is the one leading to minimum TTP



Correlations used for the evaporator and the condenser

Evaporator refrigerant-side heat transfer coefficient	Cavallini et al (2006)
Condenser refrigerant-side heat transfer coefficient	Cavallini et al. (2009)
Refrigerant-side pressure drop	Cavallini et al. (2000)
Refrigerant void fraction	Rouhani and Axelsson (1970)
U-bend pressure drop	Equivalent length = 30D
Air-side heat transfer coefficient	Manufacturer proprietary data

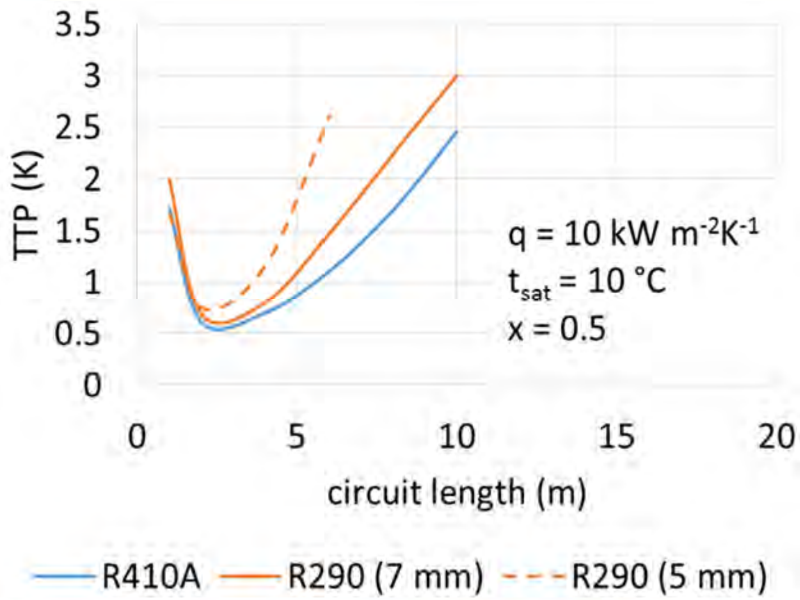
The **measured microfin dimensions** have been used for the calculation

Thermodynamic analysis: TTP approach

The TTP approach is used according to Cavallini et al (2010) for condensation and according to Brown et al. (2014) for boiling.

It is mandatory to choose a proper circuit length: the optimal length is similar for R410A and R290, but propane performance is more sensible to the length.

The circuit for evaporator should theoretically be shorter that for the condenser.



Total Temperature Penalization (TTP) as a function of the circuit length for 10 °C of saturation temperature

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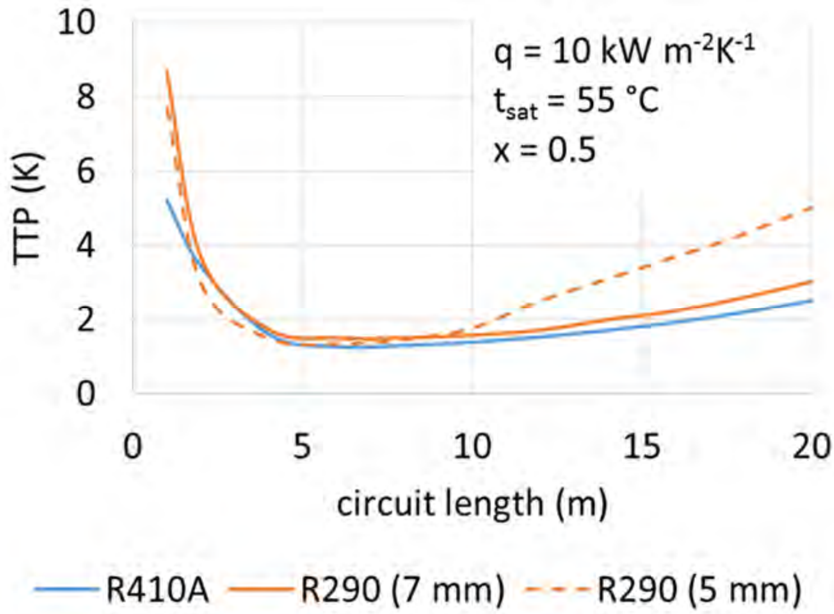

Total Temperature Penalization (TTP) as a function of the circuit length for 55 °C of saturation temperature

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Detailed Heat Exchangers design analysis

A **simulation code** of the whole chiller has been implemented based on the previously outlined simulation tools for the condenser and the evaporator.

The compressor is characterized through its **experimental compression efficiency** (based on actual electric power consumption) and volumetric flow rate polynomials

The present simulation tool was **validated on the experimental performance** data of the benchmark R410A unit.

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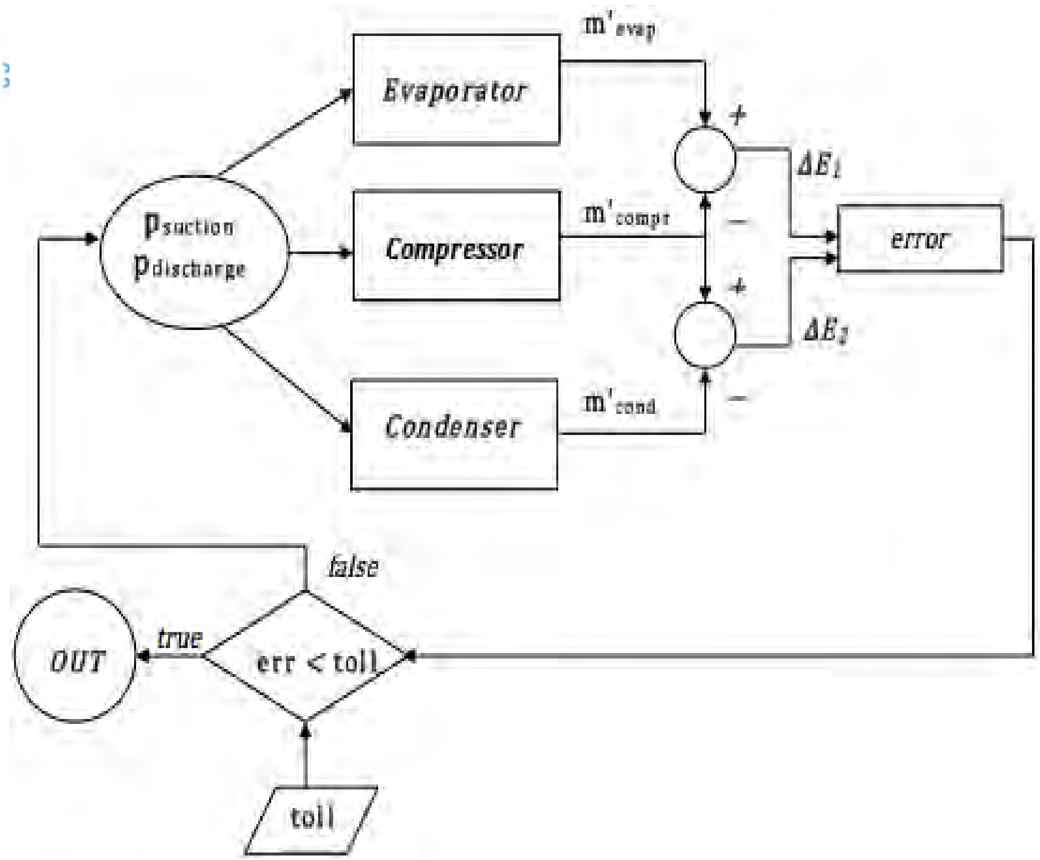
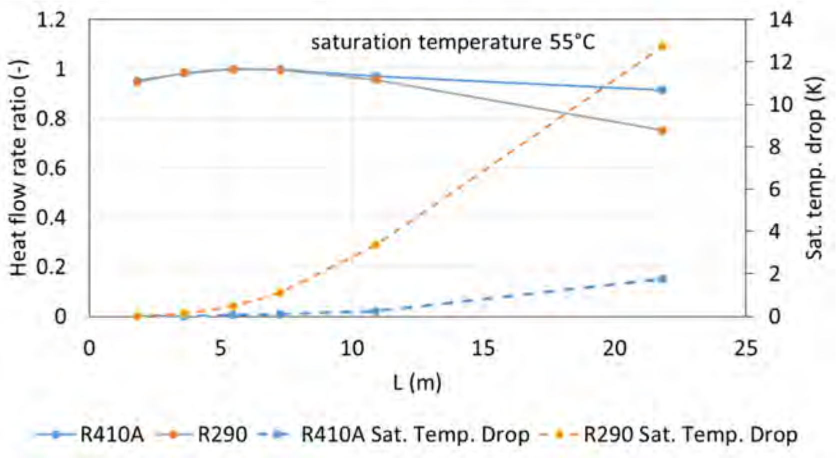


Heat flow rate ratio: Ratio of Exchanged heat flow rate vs. MAX exchanged heat flow rate

Saturation temperature drop of **OAC** (cooling mode)

the **number of circuits in the OAC** was increased from 3 (used for R410A) to 4 for R290



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R290 would require a shorter length circuit in comparison to R410A.

However, one should also consider that the two-phase fluid distribution in finned coil evaporators can be a heavy constraint for the designer.

For this reason, since there is limited difference in the heat capacity between 2 m and 4 m for the IAC, **a configuration with 4 circuits** (i.e., length of the single circuit about 4 m) was adopted for both R410A and R290.

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Subcooling

Subcooling occurs by flooding the terminal region of a condenser, with the consequence of an increase in the condensation pressure

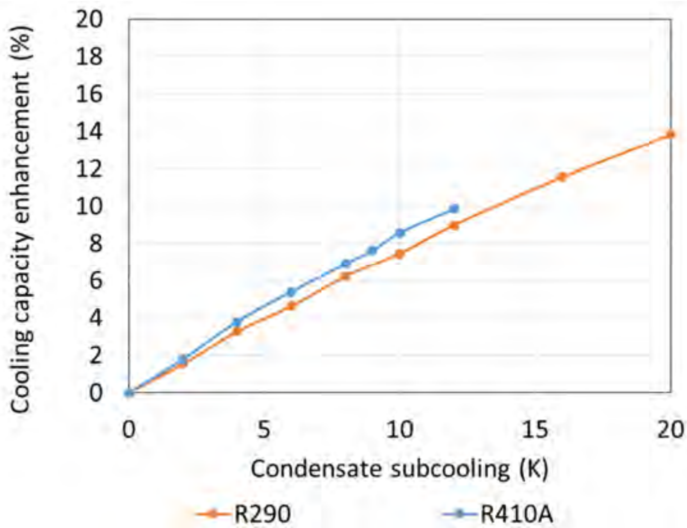
It is common opinion that it brings about an improvement of COP

From a theoretical point of view, this advantage can mostly be attributed to the reduction of **exergy loss in throttling process** together with a reduction of the heat transfer irreversibilities

A larger benefit can be expected for propane having larger throttling losses than R410A

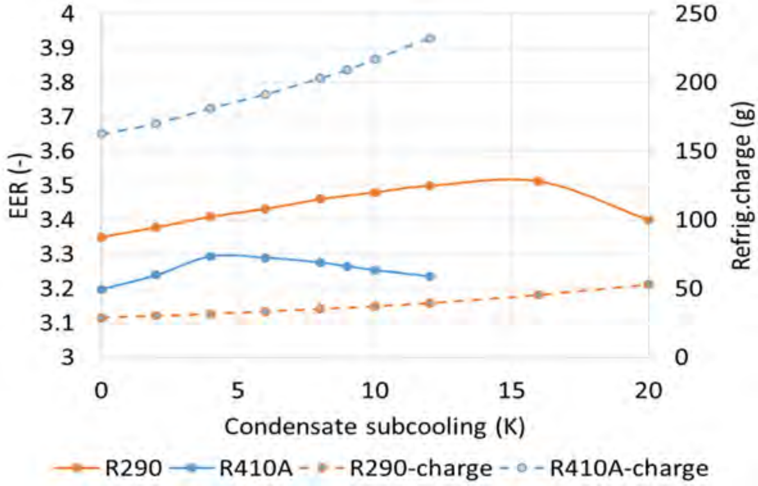
Subcooling effect on the cooling capacity

Propane benefits more for the larger subcooling, obtaining higher EER with the same cooling capacity of the R410A appliance



Subcooling effect on EER and refr. charge

Effect of condensate subcooling on the EER, and on the mass of refrigerant inventory in the condenser (cooling mode)



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CONCLUSIONS OF THE REAL VAPOR COMPRESSION REFRIGERATION CYCLE OPTIMIZATION

- A detailed **thermodynamic analysis** was implemented to evaluate the circuit length influence on the global heat transfer performance;

- A **simulation code of the whole chiller** to find out the optimal circuit length has been implemented and validated on the experimental performance data of the benchmark R410A unit;

- The **subcooling effect** on the cooling capacity and on the EER was analyzed.

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Numerical simulations of propane leakage in closed environments

Numerical simulations allows you to **save time and money** if you have a powerful enough computer and you have the sufficient know-how.

It becomes possible to simulate **numerous scenarios and numerous working conditions** by the only use of a computer and a single person implementing the code.

Achieving the same results experimentally, is a task that involves long times, high costs, adequate spaces, dedicated experimental material and competent personnel.

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Numerical simulations of propane leakage in closed environments

The commercial code Ansys fluent 18.2 was used to implement all the numerical simulations.

The firts simulations aimed at investigating the **best physical models** required to simulate the propane diffusion in a close environment.

Secondly, the simulation was **calibrated against some experimental data** available in the literature.

Finally, the calibrated simulation was used to **predict different scenarios.**

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The 3D simulation of the whole room turned out to **be very time demanding** in terms of computational power required.

For this reason, all the 3D simulations have been carried out in 1:10 scale.

But, by working only on a 1:10 scale, it is not possible to validate the numerical model against the experimental data acquired on a real-scale environment

So, it was necessary to demonstrate that a 1:10 scale simulation results are superimposable to a full scale simulation results.

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The study was carried out on 2D geometries, since a 2D simulation requires less execution time and/or less computational effort than a 3D one.

The advantage of this procedure is the reduction of the calculation time. In fact, the simulation reaches convergence in a very rapid time.
So it is possible to do many trials in a short time.

Once the most suitable mathematical models have been identified, it is possible to invest time and calculation efforts in a 3D simulation

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2D 1:10 scale model

	2D full scale	2D 1:10 scale
Room dimension	5.4 x 2.6 m	0.54 x 0.26 m
Inlet dimension	0.05 m	0.005 m
Inlet height from the ground	0.6 m	0.06 m
Propane velocity	0.0178 m/s	0.00178 m/s
Propane amount	34 g	0.34 g
Leakage time	20 s	20 s
Propane concentration	0.00243 kg/m^3	0.00243 kg/m^3
Elements number	62267	62333
Max element size	0.03 m	0.003 m

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2D full scale simulation

2D 1:10 scale simulation

It can be concluded that the **results obtained from a scale simulation are comparable with those obtained from a full scale simulation.**
Based on the results obtained here, the study was focused only on 1:10 scale geometries.

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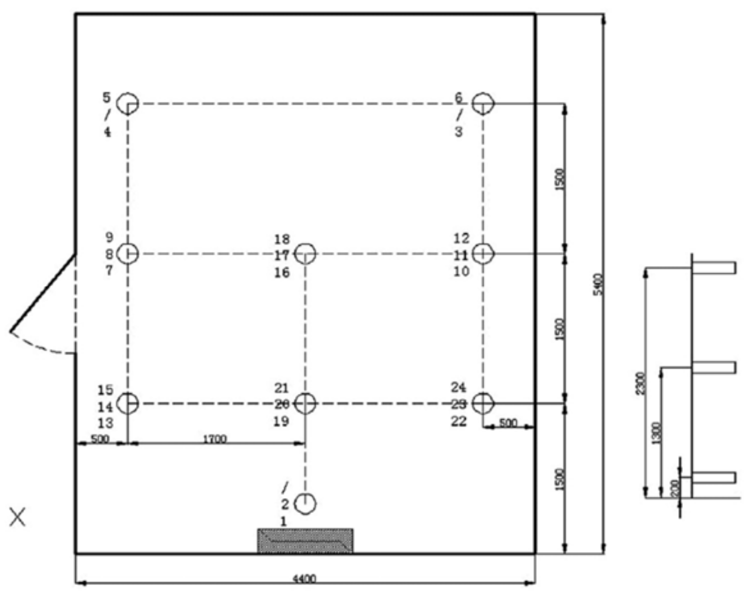
The numerical model **was validated against experimental data** available in the open Literature published by Maojuan et al. in 2018.

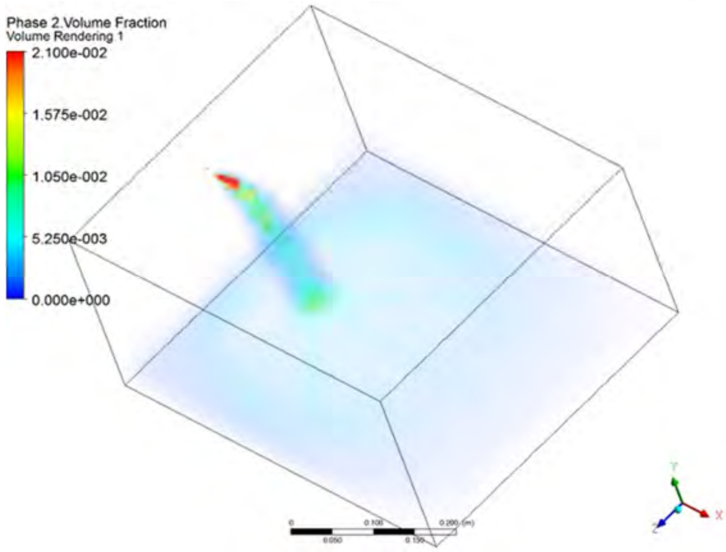
(Maojuan et al., International Journal of Refrigeration vol. 90, 2018, pp. 163–173).

The experimental data refers to a 4.4 m x 5.4 m x 2.6 m hermetically closed room and take into account the introduction of 328 g of propane from a 6 mm diameter opening positioned at 1.8 m height at a constant speed for 4 min.

The Authors experimentally measured the concentration of propane in various locations during the test.



The simulation implemented in UNIPD has been calibrated in terms of "**phase interaction**" and "**molecular diffusion coefficient**" in order to estimate propane concentration values close to those experimentally measured.



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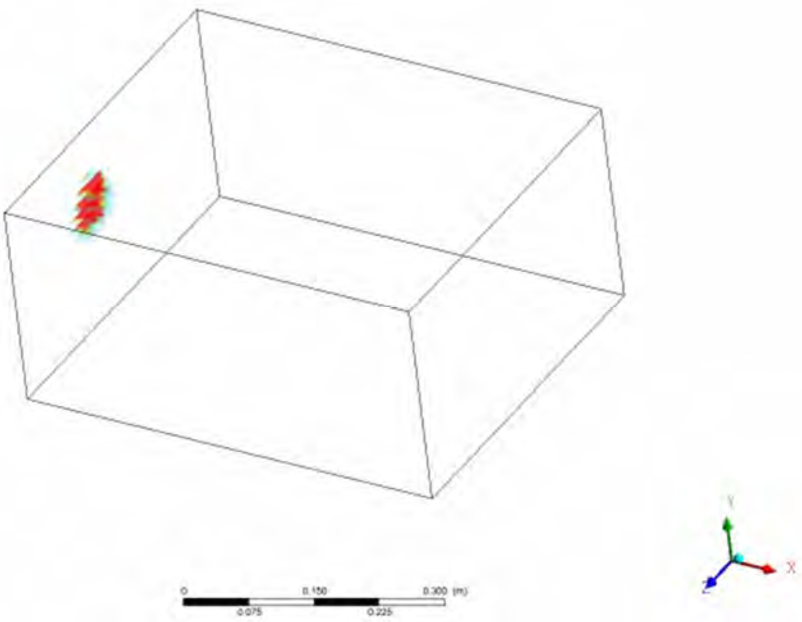

VIDEO:

Propane volume fraction during the leakage time (2 min)

Red color: 100% LEL

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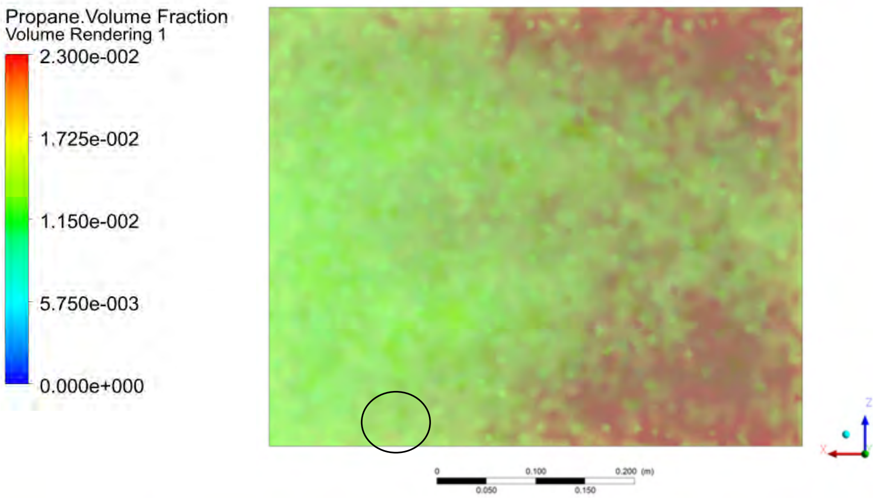
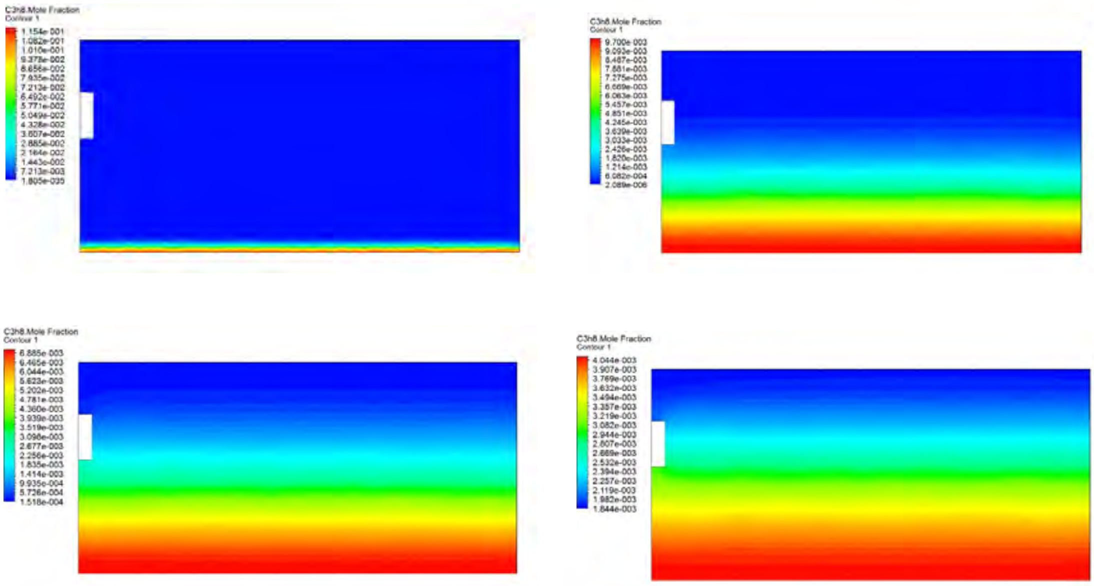
The average propane concentration at 20 cm from the floor at the end of the simulation was numerically evaluated equal to 1.66 vol%, in line with what was experimentally measured.



DIFFUSIVE PHENOMENA

In a time span of several tens of minutes, propane, which fell to the ground due to the force of gravity, begins to mix with the ambient air due to the diffusive mechanisms.

If you want to analyze the system for longer, it becomes necessary to introduce the analysis of **diffusion processes.**



The experimental tests of Maojuan et al. (2018) were taken as a reference for the validation of the model.

The analysis was carried out over the duration of **approximately 1 hour**.

The **«Species Transport»** model was used to study the diffusion process, and the option Energy Source was switch on.

The **mixture model** was taken into account.
A mixture of propane and air was created. Both the fluids were considered as ideal gasses.

The kinetic theory was adopted to calculate the mass diffusivity coefficient.
While thermal conductivity and viscosity were kept constant.

COMPARISON BETWEEN SIMULATION RESULTS AND NEW EXPERIMENTAL DATA

DATA COLLECTION @ INNOVA

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- Different amounts of propane (150 g, 70 g, 20 g) were inserted inside a room simulating various leakage cases.
- Fan: ON-OFF
- Leakage from: evaporator, condenser, etc.
- 5 propane detectors were used in different configurations

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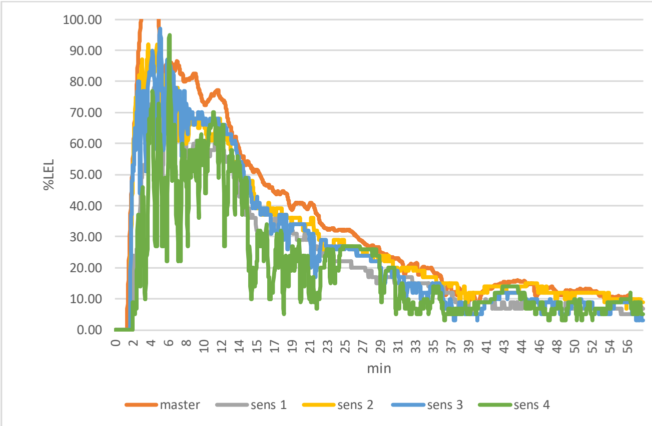
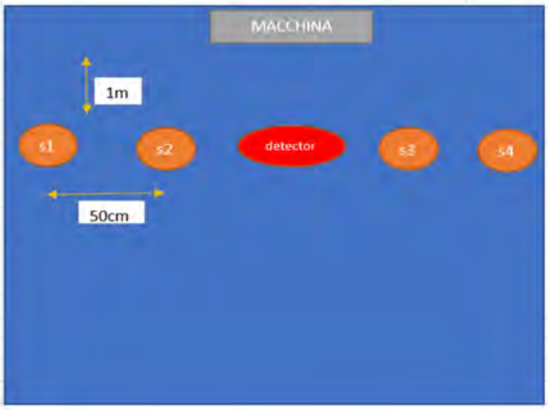
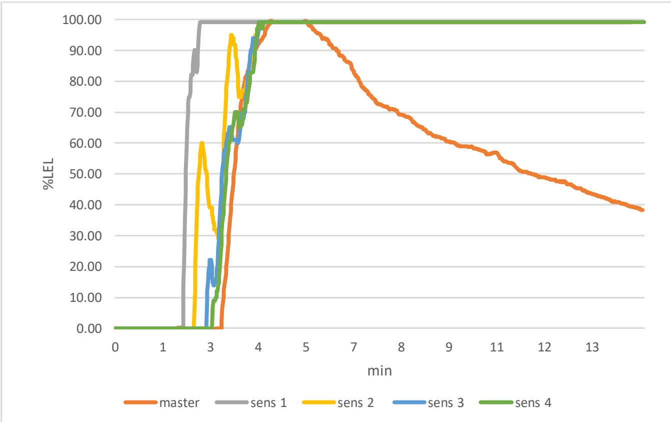

%LEL
min
master
sens 1
sens 2
sens 3
sens 4


MACCHINA
1m
s1
s2
detector
s3
s4
50cm


%LEL
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sens 1
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sens 3
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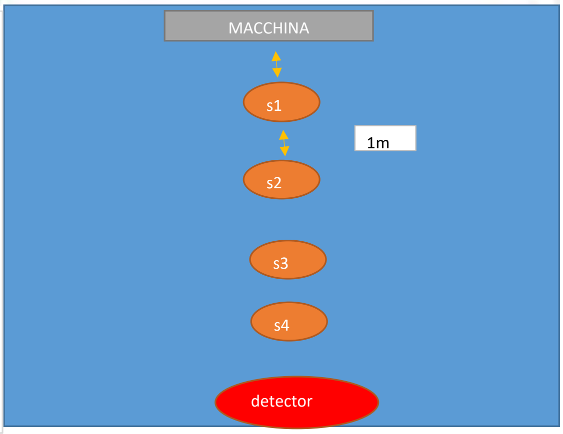
MACCHINA
s1
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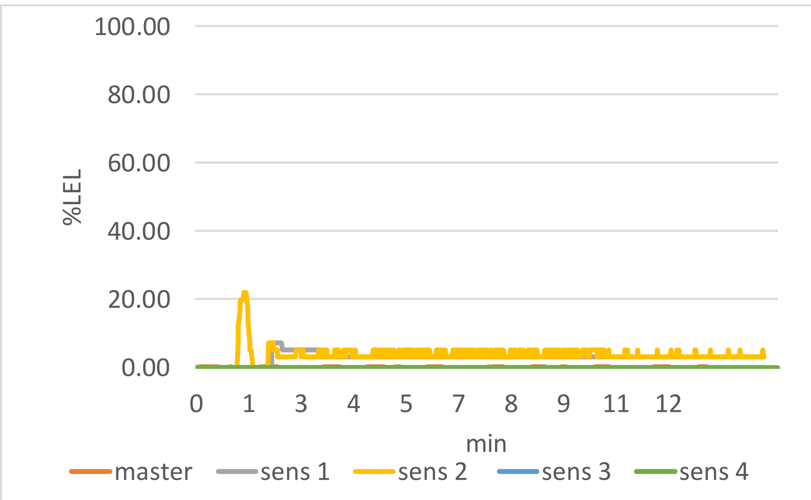
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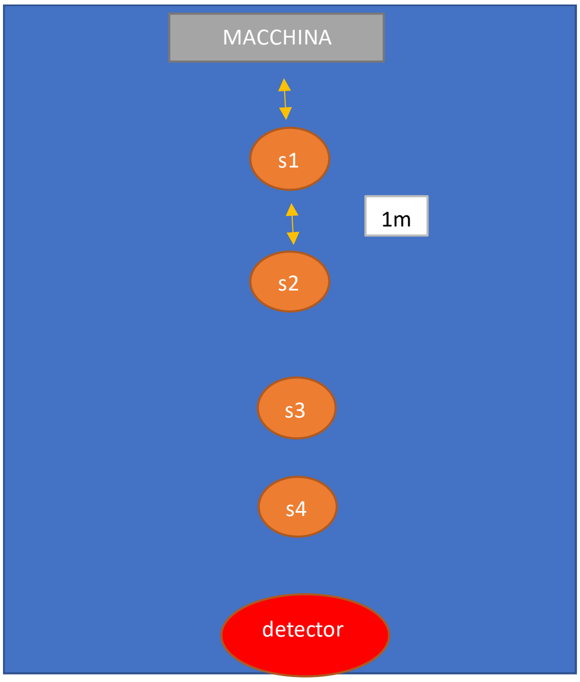
MACCHINA
s1
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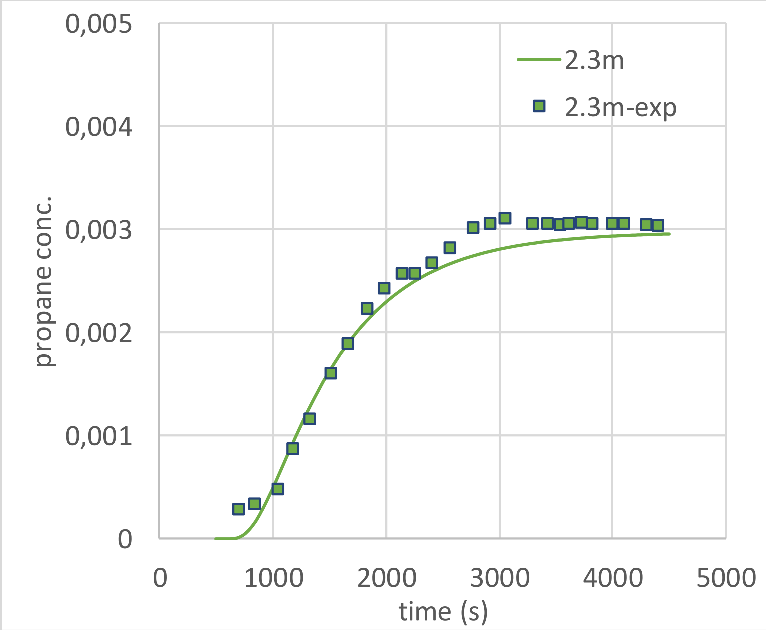
s4
detector



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Prediction of different leakaging scenarios

Once calibrated, the numerical model was used to predict different scenarios.

A simulation was carried out by keeping **the same refrigerant leakage amount (328 g) but halving the leakage time: 2 min instead of 4 min.**

The leakage flow rate and, thus, the inlet propane speed increased.

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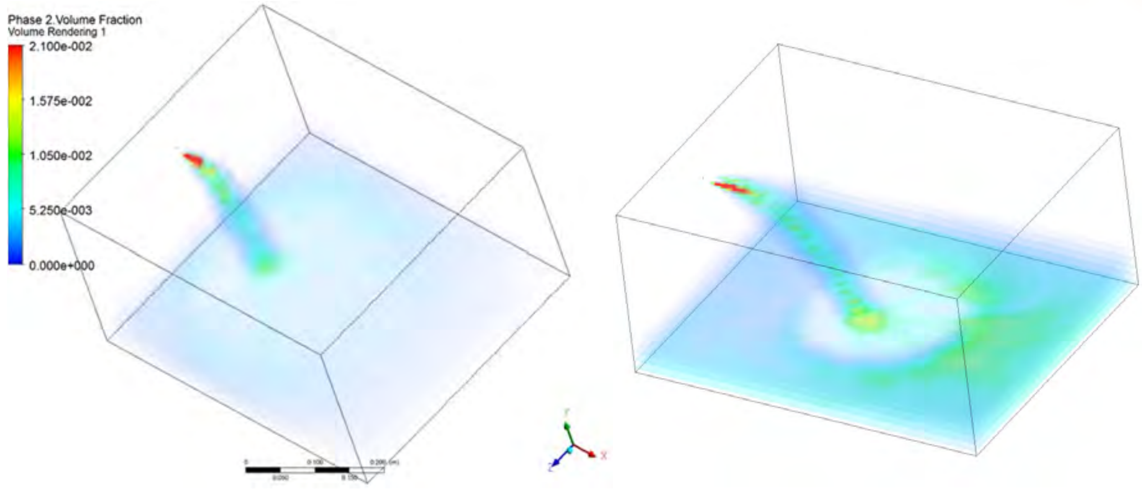

4 min leakage time

2 min leakage time

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The refrigerant entered deeper into the environment due to its higher flow rate.

Since all the refrigerant gas was injected in half the time**, the refrigerant did not have enough time to mix with the ambient air**, so the concentration on the floor at the end of the leakage time is higher.

Nevertheless, the maximum concentration remained lower that the flammable threshold.

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A simulation was conducted to investigate the **location of the leak effect.**

In this case the leak location was moved down to recreate the situation of a floor-mounted machine.

The leak was set at 30 cm from the ground.

All the refrigerant (328 g) was discharged into the environment in 2 min.

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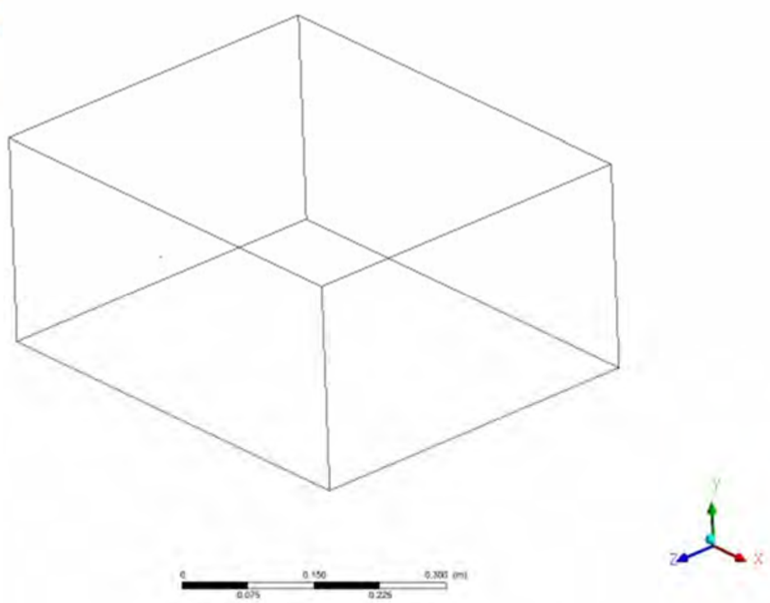

VIDEO:

Propane volume fraction during the leakage time (2 min)

Red color: 100% LEL

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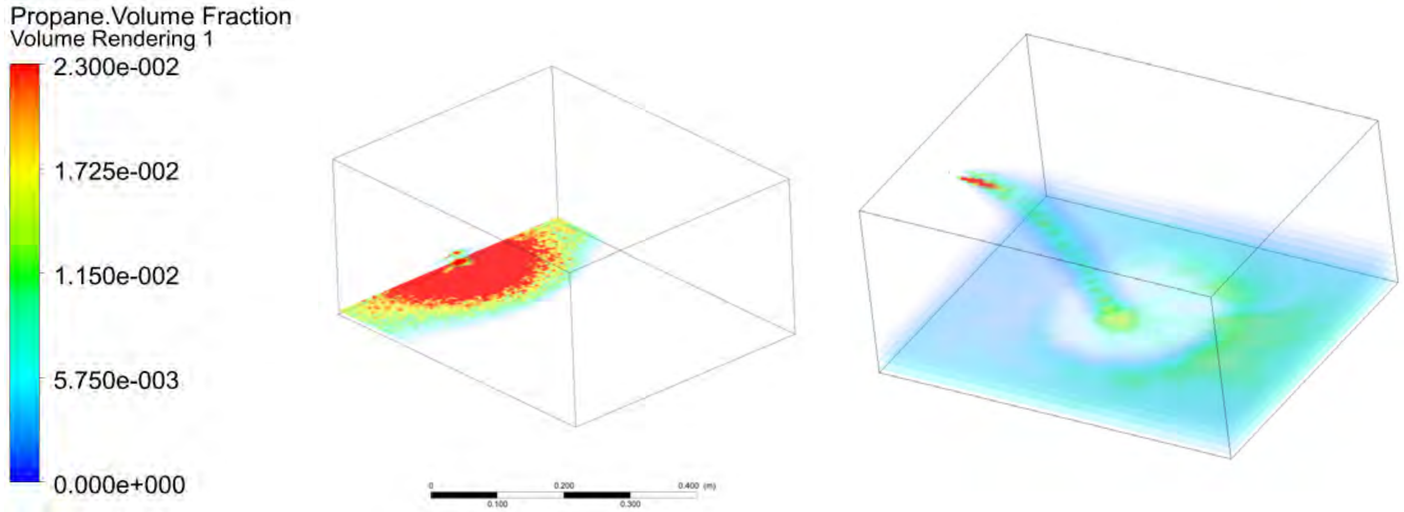


After 2 min

After 2 min

The jet range was much shorter than the previous case.

So, the **refrigerant did not have the possibility to mix with the ambient air**.

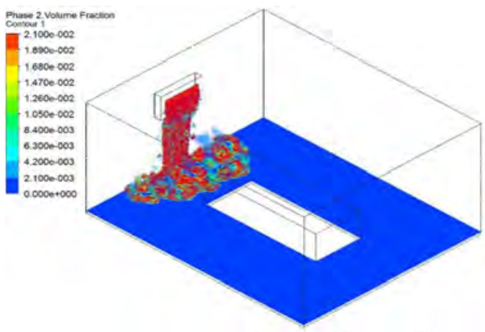
In this case, the area around the leakage point reached the flammability conditions.

To limit this drawback, the circulation of the air should be improved, for instance by switching on a fan.

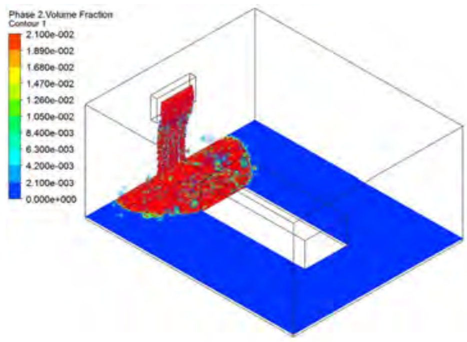
In another cases study, a table was placed inside the room to simulate if the presence of obstacles (such as furniture, tables, etc.) can change the diffusion of propane in the environment.

Here is the case of a 110 x 270 x 70 cm sized table placed at the center of the room.

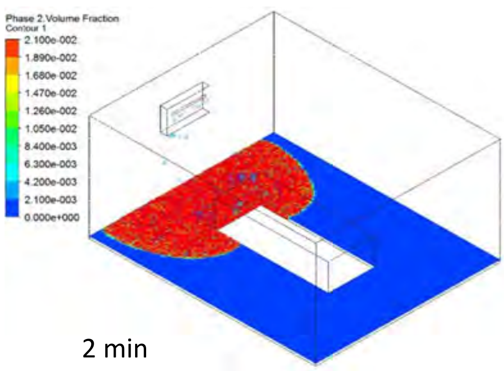
All the refrigerant (328 g) was discharged into the environment in 2 min.



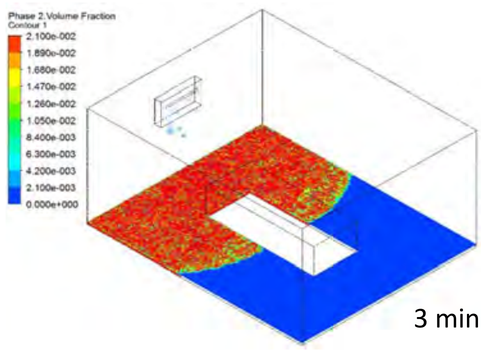
15 s



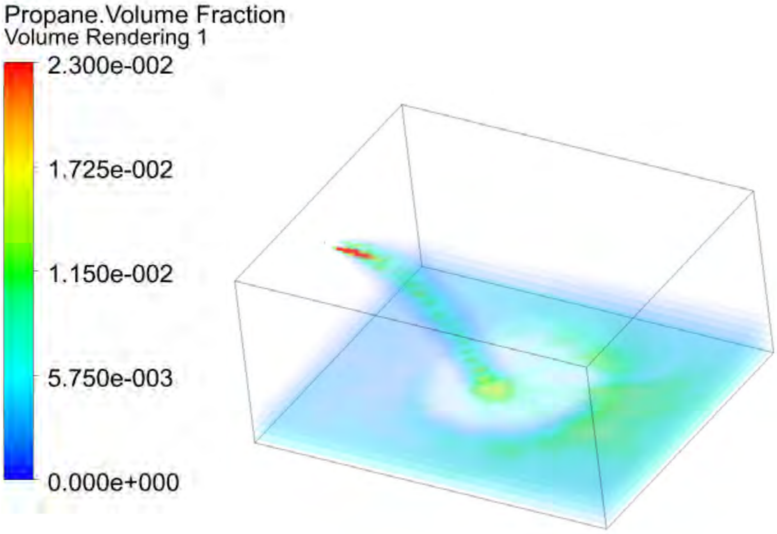
40 s



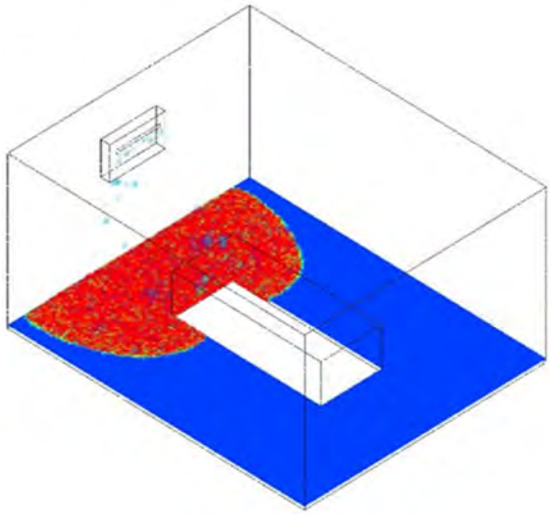
2 min



3 min



2 min



2 min

CONCLUSIONS

A numerical model saves time and obtains information on multiple operating conditions

- a new model has been implemented to study different possible leakage scenarios in closed environments
- the model was **validated** against experimental data available in the open literature
- the model was also **calibrated** against new experimental data

- the model was **used to predict new operating conditions** (different flow rate, leakage time, propane amount, leakage position, presence of obstacles, etc.)
- the model allows to identify the **most demanding conditions** in terms of safety

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Recent developments and research perspectives about flammable refrigerants

THANK YOU!

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