COMPARING THE PERFORMANCE OF LU-VE UNIT COOLERS

INSTALLED IN APPLE STORAGE COLD ROOMS AT THE C.O.L – LOVERNATICO FRUIT AND VEGETABLE GROWERS CONSORTIUM IN SPORMINORE - MELINDA GROWERS' ASSOCIATION (TRENTO)

Final report

Authors:

Umberto Merlo, LU-VE SpA
Head of the Research and Development Laboratory

Giovanni Lozza, Milan Polytechnic
Full Professor of Systems for Energy and the Environment, Department of Energy

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1. INTRODUCTION

This report aims to summarise the information obtained experimentally during the conservation process of Melinda products (Golden Delicious apples). It compares two cold storage rooms from the points of view of both energy and process quality: one is fitted with a traditional unit cooler (with suction fans installed Photo 1) and the other is fitted with an innovative unit cooler (with blower fans, Photo 2).
The experimental results obtained are discussed, and are followed by a supporting theoretical investigation of the air distribution within the cell, carried out using computational fluid dynamic (CFD) methods.
The experimental study was carried out as a partnership between LU-VE, and the COL (Lovernatico Fruit and Vegetable Growers) Consortium. The former supplied the exchangers and specified the measurement system, whereas the latter, besides making the cold rooms available, supervised measurement system installation and carried out data acquisition. The numerical analysis was carried out at the LU-VE R&D laboratories. Milan Polytechnic, as part of a ten-year research partnership with LU-VE, supervised the measurement system and the CFD analysis.

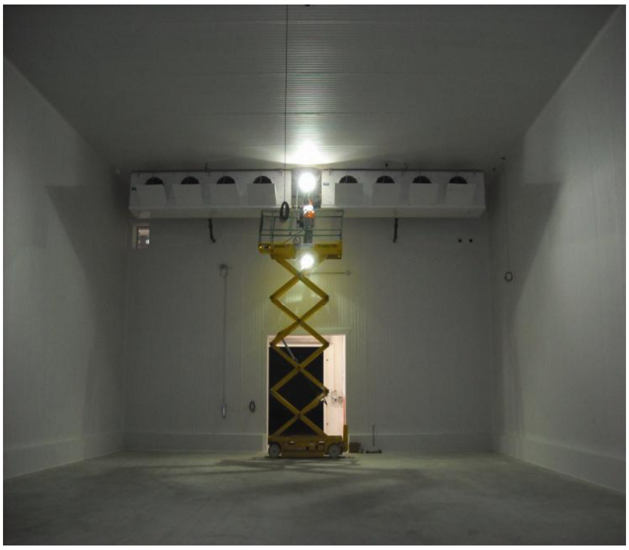


Photo 1

Photo 2

2. TRIALS AT THE COL CONSORTIUM

The comparison phase was carried out using two industrial unit coolers, both with the same heat exchanger size 18T 8R 4800A/CuAl/36N (fin geometry 55x55mm, 12.7mm tube) and fan type. They were installed in two identical cold rooms, loaded with the same quantity and type of product, and having the same operating conditions (cold room temperature =+1°C, relative humidity =85%, internal liquid input and output temperature (30% vol Ethylene Glycol) = -5/-1°C).
Besides measurement instrumentation to monitor the refrigeration plant and the air conditioning in the cold rooms, a series of instruments was installed for measuring air velocity, and heat exchanger differential pressure, as well as scales for measuring product weight loss. All the measured values were recorded in real time by the consortium-wide remote management system.
The following data was taken from the report written by Livio Fadanelli, Refrigeration and Post-Harvest Handling Technologies Manager at the CTT (Technological Transfer Centre), FEM-IASMA (Edmund Mach Foundation – San Michele all'Adige School of Agriculture).

Main specifications

Year 2010-2011
Product Variety : Golden Delicious apples
Storage: in a refrigerated environment with a Controlled Atmosphere
Storage Period: from 30.09.2010 to 05.04.2011 (up to about 180 days after harvest)

Cold room and equipment characteristics

Cold Room No. 31 - dimensions in metres: 23.3 x 12.9 x 8 (height), and capacity 620,000 kg in 2053 plastic bins. Equipped with a cooling system comprising 2 unit coolers with the following characteristics:

- finned heat exchanger - dimensions 4800x 990 x 440
- materials Cu-Al (grooved tube –louvered fins)
- Heat exchange surface 595.98 m^2
- Fin geometry 55x55 mm and tube diameter 12.75 mm
- Fin pitch 7 mm
- Qty. 4 Ø 630 mm (traditional) suction type fans, powered by EC (electronically controlled) motors each with an installed electrical power load of 3.2 kW (maximum rotation velocity 1490 rpm), regulated as follows:

a. during the initial cooling phase, at 85 % with a real electrical power draw of 2.3 kW (total 2.3 x 8 = 18.4 kW / cold room)
b. during the temperature maintenance phase, at 65 % with a real electrical power draw of 1.8 kW (total 1.8 x 8 = 14.4 kW / cold room)

Cold Room No. 30 - dimensions in metres: 23.3 x 12.9 x 8 (height), and capacity 620,000 kg in 2053 plastic bins. Equipped with a cooling system comprising 2 unit coolers with the following characteristics:

- finned heat exchanger dimensions 4800x 990 x 440
- materials Cu-Al (fluted pipe –louvered fins)
- heat exchange surface 595.98 m^2
- fin geometry 55x55 mm and tube diameter 12.75 mm
- fin pitch 7 mm
- Qty. 6 Ø 630 blower fans, powered by EC (electronically-controlled) motors, maximum rotation velocity 1140 rpm, electrical power draw 1 kW (total 1.0 x 12 = 12 kW / cold room)

The daily load, the start-up to steady state, and the scheduled ventilation procedures (times and values) were all identical for both cold rooms. Instead, the humidification parameter was regulated as required, keeping the RH (Relative Humidity) range set at between 93-95% for each of the two cold rooms. The test details were minutely defined in detail in an appropriate operating procedure.

Tests and checks

The procedure envisaged the following series of tests:

- checking the fruit core temperature on a daily basis during the cold room loading phase,
- assessing the real time weight loss, via a software system linked up to scales fitted with load cells onto which several bins of apples had been placed in each cold room.
- checking quality requirement conformity for the apples undergoing testing in the two cold rooms on three occasions, at the beginning and the end of storage, and after a ten-day shelf life,
- evaluating quality requirement conformity with an automatic Pimprenelle laboratory, on 15 representative fruit samples: average sampled fruit weight (g), pulp hardness (kg/cm2), sugar content (IR in °Brix), total sample acidity (in g/l of malic acid), succulence (% juice against total weight), quality indicator (Thiault).
- Percentage weight loss check, carried out on 7-8 bins, equal to at least 2800 kg gross weight per double weighing (beginning and end of test).

- periodic monitoring of electrical consumption and fan running hours subdivided according to the scheduled cooling and ventilation functions.
- examining the results obtained, conclusions and post-test deductions.

Storage parameters (applied to both cold rooms)

Fruit temperature: 0.9-1.4 °C, Cold room relative humidity: 93-95% Minimum ventilation: 7 hours / 24 hours CO_2% 2.2- 2.6 – O_2% 1.1-1.4

Checks during the loading / cold room phases

(monitoring 3 fruit core values at 5 points of the pile, see sequence

		Average Temperature		Hours of refrigeration		Hours of ventilation	
day	time	Cold room 30	Cold room 31	Cold room 30	Cold room 31	Cold room 30	Cold room 31
1 – 30 Sept.	15.30	5.0 (from 2.2 to 6.2 °C)	4.4 (from 1.5 to 7.3 °C)	20.20	21.08	21.16	21.33
2 – 1 Oct.		14.00 3.5 (from 1.8 to 3.8 °C)	3.1 (from 2.3 to 4.0 °C)	22.20	22.05	16.10	22.12
3 – 2 Oct.		14.30 4.2 (from 1.1 to 6.8 °C)	4.7 (from 1.8 to 7.2 °C)	16.45	22.29	18.57	19.00

Checking the quality requirements (at harvest)

Cold room	Weight Average, g	Average hardness kg/cm^2	Average IR, Brix	Acidity, g/l malic acid	% sugar	Thiault Index
30 (innov.)	216	7.0	13.2	4.4	15.6	163
31 (trad.)	205	7.1	13.5	4.6	15.1	169

Checking quality requirements (at the end of the storage period)

Cold room	Weight Average, g	Average hardness kg/cm2	Average IR, Brix	Acidity, g/l malic acid	% sugar	Thiault Index
30 (innov.)	219	5.6	13.0	4.0	15.4	157
31 (trad.)	255	5.6	13.6	4.0	14.7	164

Checking quality requirements (at the end of the storage period + 10 days shelf life at 20°C ambient temperature)

Cold room	Weight Average, g	Average hardness kg/cm^2	Average IR, Brix	Acidity, g/l malic acid	% sugar	Thiault Index
30 (innov.)	195	5.4	13.4	3.5	14.5	156
31 (trad.)	224	5.4	13.9	3.5	13.9	162

Calculated weight loss: Daily and monthly total (initial weight – final weight x 100)

Cold room 30 (innovative) Fruit temperature when opened (on 3 points) 1.46 °C No. of bins 10 (7 full + 3 empty) Initial weight, date 30.09.2010: 2830 kg(- 345 kg tare) = 2485kg final weight date 15.04.2011 2792 kg(- 345 kg tare) = 2447kg Days 197, weight loss 38 kg net **Total weight loss = 1.51 % = 0.0077 %/day = 0.23 % / month**	Cold room 31 (traditional) Fruit temperature when opened (on 3 points) 1.45 °C No. of bins 10 (7 full + 3 empty) Initial weight, date 30.09.2010: 2870 kg(- 345 kg tare) = 2525kg Final weight 05.04.2011 2824 kg(- 345 kg tare) = 2479kg Days 187, weight loss 46 kg net **Total weight loss = 1.79 % = 0.0094 %/day = 0.28 % / month**

Note: Due to unbalancing of a sample storage pile being tested, and bins rubbing against neighbouring storage piles, real time assessment of weight loss was biased. It therefore became necessary to calculate weight loss using the double weighing formula.

Calculating energy consumption

Power used, kW

Cold room no.	Period	Ventilation	Cooling	Humidification	Defrosting
30 (innov.)	Initial cooling, up to 03.10.2010	12.0	16.73	1	4.6
31 (trad.)		18.4	16.73	1	4.6
30 (innov.)	Temperature maintenance, up to 05.04.2011	12.0	16.73	1	4.6
31 (trad.)		14.4	16.73	1	4.6

(defrosting: water system pump power)

Operating duration time (Pauses: number of start ups-shut downs)

Cold room no.	Period	Ventilation	Cooling	Humidification	Defrosting
30 (innov.)	Initial cooling, up to 03.10.2010	120.27	105.90	14.70	2.91
31 (trad.)		120.79	108.50	14.43	3.19
30 (innov.)	Temperature maintenance, up to 05.04.2011	1411.20 (9658 int.)	685.05 (8589 int.)	140.22 (1012 int.)	6.11 (37 int.)
31 (trad.)		1398.24 (6871 int.)	754.35 (5878 int.)	157.59 (1187 int.)	6.59 (42 int.)

Electrical energy used, kWh

Cold room no.	Period	Ventilation	Cooling	Humidification	Defrosting
30 (innov.)	Initial cooling, up to 03.10.2010	1443.24	3042.50	14.70	13.38
31 (trad.)		2222.54	3811.60	14.43	14.67
30 (innov.)	Temperature maintenance, up to 05.04.2011	16.934.40	19.681.48	140.22	28.10
31 (trad.)		20.134.70	23.482.92	157.59	30.31

Overall energy consumption

Cold room no.	Initial cooling	Temperature maintenance	Total
30 (innov.)	4513.8	32270.4	36784.2
31 (trad.)	6063.2 (+34%)	37742.3 (+16.9%)	43805.5 (+19.1%)

Electricity costs, Euros (electrical energy cost: 0.0713 €/ kWh)

Cold room no.	Initial cooling	Temperature maintenance	Total
30 (innov.)	322	2301	2623
31 (trad.)	432	2691	3123
difference	110	390	500

Initial cooling transition phase

The above results highlight that despite occurring in a very short period of time, the initial cooling phase is of great importance for temperature maintenance, from the point of view of both energy and impact on final product quality. In this phase, the blower fan solution proved to be very efficient (a saving of 34%).

Summary graphs are shown for the period mentioned, comparing the two unit cooler solutions. In particular, we shall highlight the trend of several parameters that have a bearing on the initial transient cooling time.

Firstly, the initial load loss value of the heat exchanger units, as obtained by the differential meters installed on the equipment, is shown for the period from 29/9 to 2/10:

Cold room 30 – 139 Pa (average value)

Cold room 31 – 135 Pa (average value)

The values are very similar: since the finned stacks are geometrically identical, it can be deduced that the air velocity through the finned stack and the overall air flow rate are substantially the same, which provides support for comparison homogeneity. Despite this, the blower solution seems to provide a slightly better and more stable sample apple cooling trend, albeit one that is not very different from the standard, as shown in figure 1, which indicates the standard position apple core temperature trend.

Apple core temperature

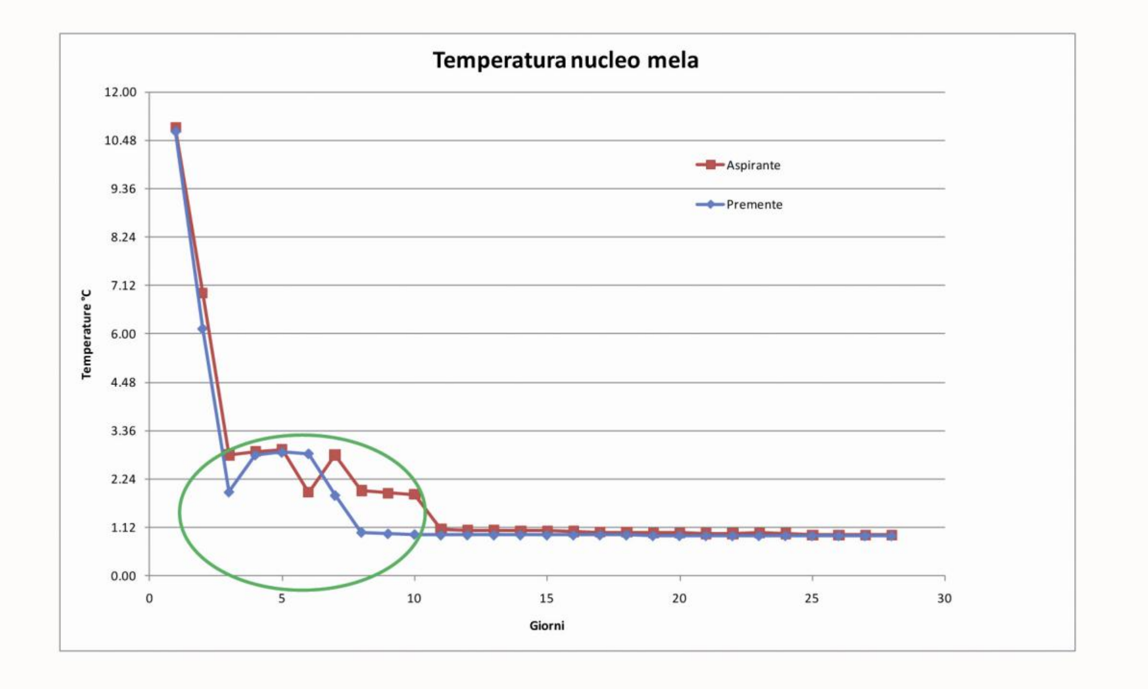
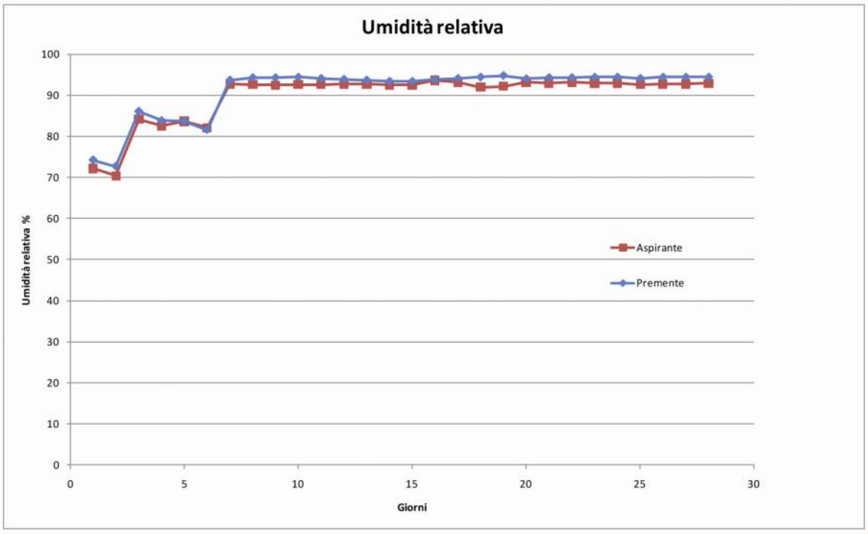


Figure 1 – Product core temperature

The chart in figure 2 shows the relative humidity trend over the same period. In this case, despite a lower number of humidifier cut-ins in the blower fan cold rooms (-11%), the value continues to coincide with the suction fan cold room one. This data pertains to the spot value measured near the entrance door, and in effect, vapour circulation and uniformity were visually ascertained to be better in the blower fan cold room than in the suction fan one. This was due to greater machine outlet velocity uniformity. In particular, in the blower fan cold room, near the water vapour inlet area, no droplets were found on the product.

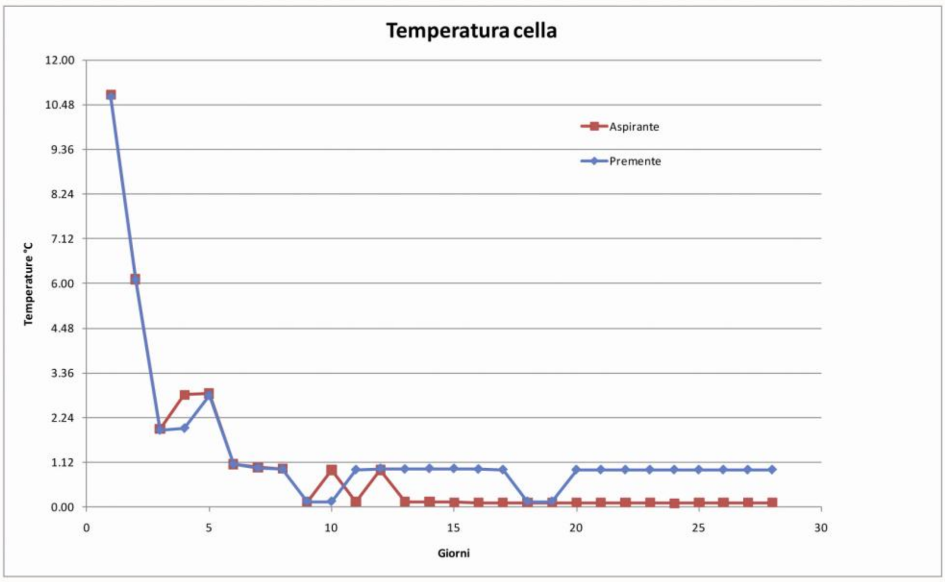


Suction fan

Blower fan

Figure 2 - Relative humidity

Figure 3 shows the cold room temperature trend in the transition period. In an analogous manner to the previous case, the two curves are almost identical. The optimum storage value (0.9 -1.4°C) was reached in about 6 days.



Suction fan

Blower fan

Figure 3 - Cold room temperature

Figure 4 shows the trend over time of cold room air oxygen content, and it is also almost identical for the two cold rooms – the optimum value is reached after about 12 days.

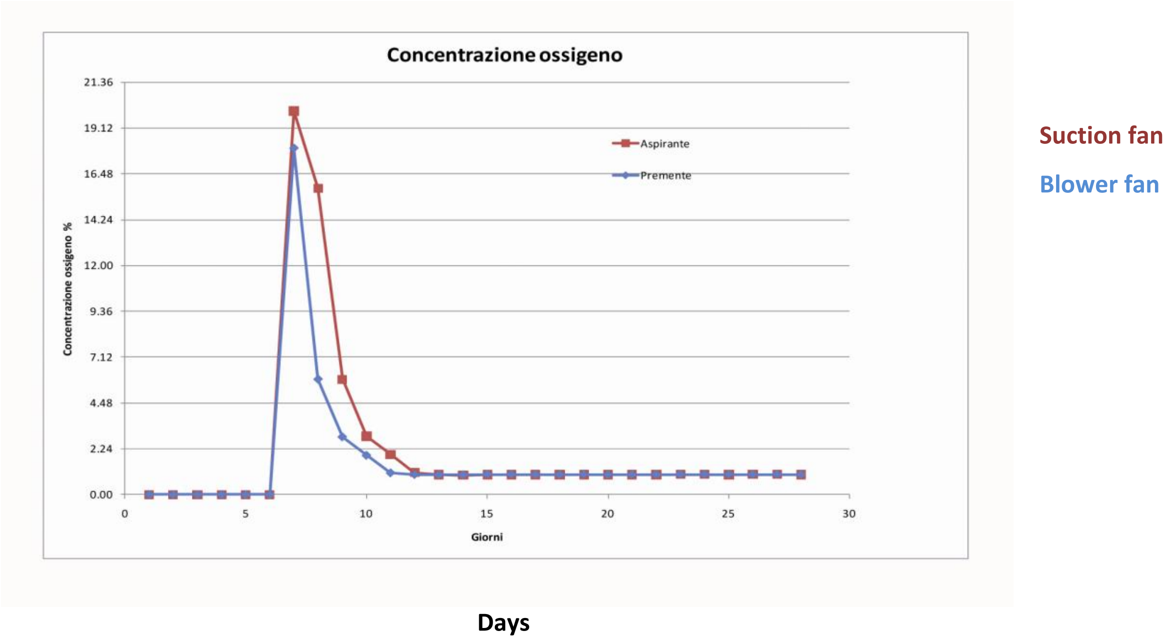


Figure 4 – Oxygen concentration

Table 1 shows the product core temperature values detected at 18 points distributed throughout the cell, after being opened at the end of storage. It was found that the average temperatures of the apples in the two cold rooms being compared were substantially identical (1.37 compared to 1.42 °C), as, too, were the standard deviations.
Figure 5 shows the positions of the apples whose temperature were measured; the first drawing is a plan view of the cold room (e.g. position A is opposite the entrance door), whereas the second shows the position of the bin number examined.

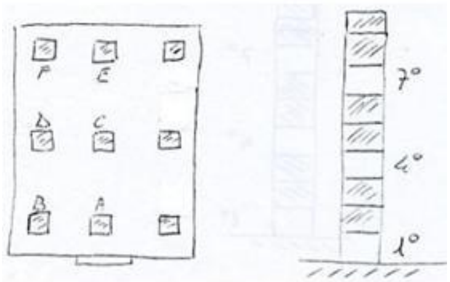


Figure 5 – Position of the apple core temperature measurement points.

On a more detailed level, it should be noted that at different levels (1^{st}, 4^{th} and 7^{th}) the variation of temperature in the two cold rooms was kept within 0.15°C. As regards temperature uniformity between the front and back ends of the cold room, although the suction fan one seemed to be slightly better, the small variations should effectively be attributed to normal measurement uncertainty (not instrumental but methodological).
An appreciable variation was encountered between the middle position of point B (1.09 compared to 1.45 °C) and point D (1.25 compared to 1.40 °C), which can be explained – as we will see better in the CFD simulations – by better air recirculation in the suction fan cold room in the side passage way above the wall base.
<u>It is important to note that the temperature value at point E is1.15°C, for the innovatory cold room, compared to 1.45°C for the traditional one. This can, to be sure, dispel any possible doubts concerning blower fan capability of delivering air flow right up to the bottom end of the cold room.</u>

				Cold room 31	
				Suction fan	
			1	4	7
A		1.55	1.83	1.53	1.29
C	cold room centre	1.59	1.73	1.53	1.52
E		1.45	1.45	1.45	1.45
B		1.09	1.11	1.08	1.09
F		1.28	1.25	1.25	1.34
D		1.25	1.26	1.26	1.24
			1.44	1.35	1.32
		Level	low	middle	top
			AVERAGE	1.37	
			STANDARD DEVIATION	0.21	
	AVERAGE				
	bottom	1.37			
	middle	1.42			
	start	1.32			

				Cold room 30	
				Blower fan	
			1	4	7
A		1.56	1.90	1.50	1.27
C	cold room centre	1.58	1.68	1.50	1.55
E		1.15	1.15	1.15	1.14
B		1.45	1.60	1.46	1.29
F		1.37	1.35	1.40	1.35
D		1.40	1.50	1.45	1.25
			1.53	1.41	1.31
		level	low	middle	top
			AVERAGE	1.42	
			STANDARD DEVIATION	0.20	
	AVERAGE				
	bottom	1.26			
	middle	1.49			
	start	1.50			

Table 1 – Product core temperatures (after cold room opening)

Finally, to conclude the analysis, two charts, 6 and 7, relating to the two cold rooms examined, highlight the air velocity readings obtained from the 8 propeller-type anemometers in 440 samplings made in the period from 1/10 to 2/11. These sensors were located at the following points:

1. bin slit (position A4 – see layout figure 5) 6
2. suction fans (wall adjacent to the entrance door)
3. side “raised base” (at a distance of 3m from the machines)
4. cold room base (positioned symmetrically in relation to the 2 (suction) intake fans).

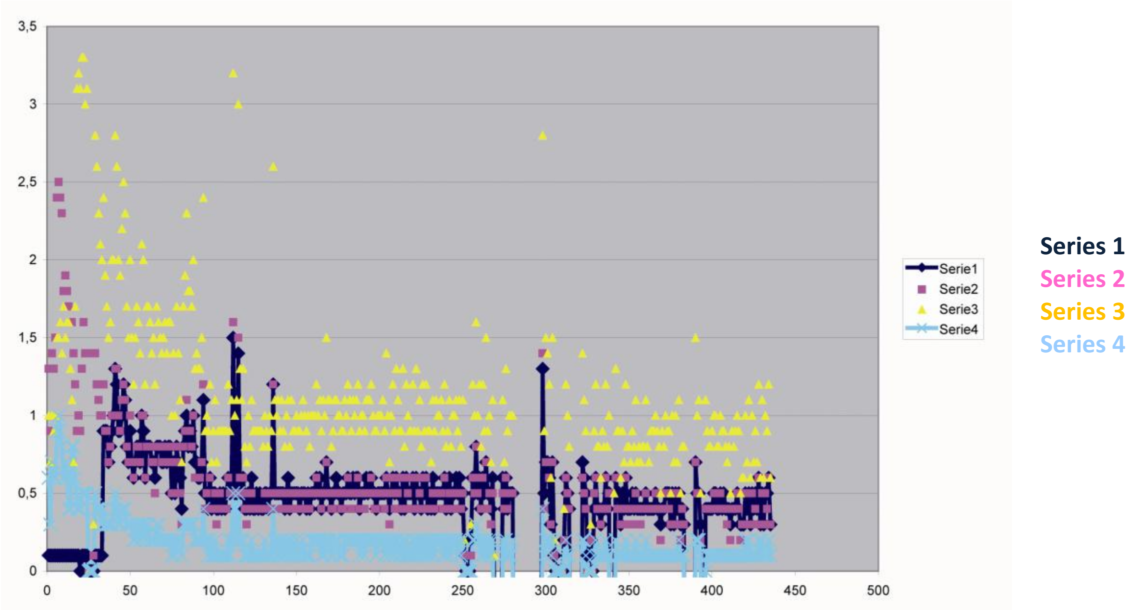


Figure 6 – Air velocity distribution in cell 31 suction fan

In positions 1-4, the following average air velocity value readings were obtained:

1: 0.5 2: 0.6 3: 1.3 4: 0.3 m/s

The most stable values are the ones pertaining to the position at the back of the cold room and to suction fans, whereas the most unstable and fluctuating ones pertain to the side “raised base” position.

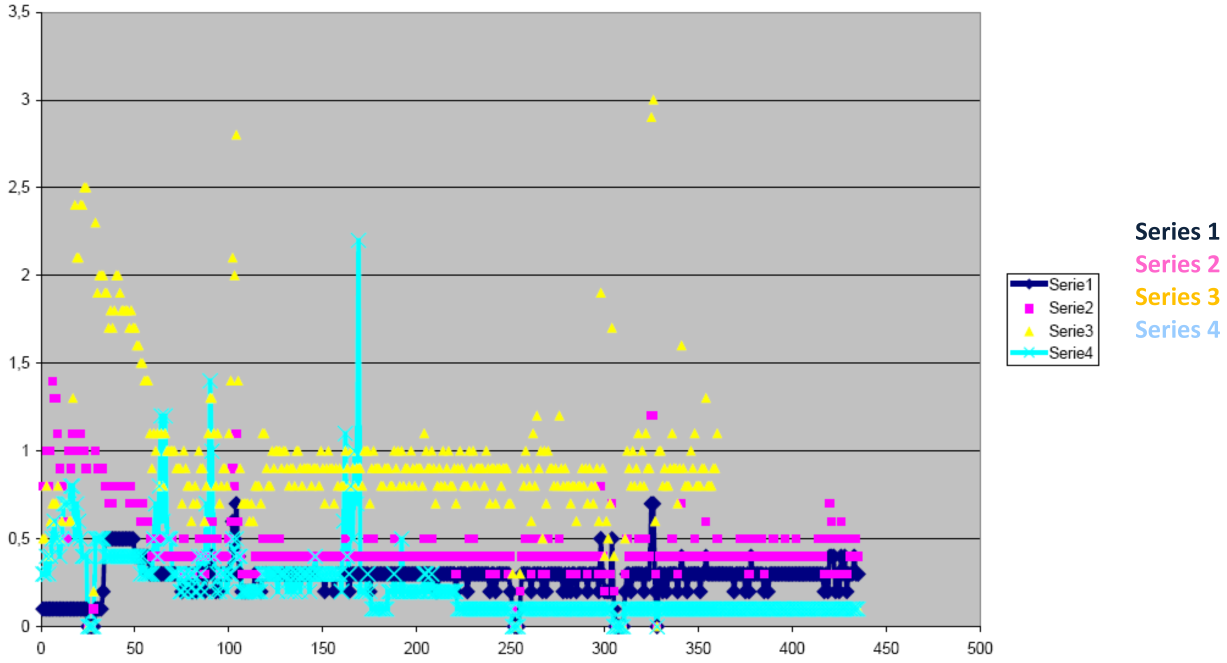


Figure 7 – Air velocity distribution in cell 30 - Blower fan

In an analogous manner to the suction fan cold room, Figure 7, shows the air velocity trend for the blower fan cold room. In particular, the average values over the same period are shown in the following:
1: 0.4 2: 0.5 3: 1.1 4: 0.3 m/s

Checking the average values over this period, it can be stated that the there was the same flow velocity in both cold rooms, that is the bins were ON AVERGAGE subjected to the same air flow. <u>In fact, as shall be seen from the CFD analysis, there are areas in the suction fan cold room, where the velocity gradients are noticeably higher compared to the blower fan cold room.</u>

3. CFD Analysis – A Fluid Dynamic Study

The CFD analysis conducted on the two cold rooms helped, firstly, to define the position of the air velocity measurement instrumentation and, secondly, to understand possible anomalies in the air distribution as well as to indicate any modifications to be carried out in the future.
The simulation was carried out making the best use possible of the symmetry of these cold rooms so as to use a reasonable number of elements that are compatible with the available hardware system potential. The number of mixed elements (triangular and hexahedral) was about 1,308,000 with the greatest concentration in the area near to unit coolers, where the pressure and velocity gradients are the greatest.
COLD ROOM WITH AN INNOVATIVE BLOWER FAN
Some initial doubt was raised about the blower fan configuration regarding the possibility that there may be poor ventilation at the bottom end of the cold room. The CFD simulations, however, always confirmed a similar velocity for both configurations. The trials confirmed these hypotheses, in which the velocity at the bottom end of the cold room proved to be almost identical (0.45 m/s).

Figure 8 shows the velocity vector trend near the bottom end of the cold room, where it can be noted that there is a discrete amount of ventilation supply to all the slits between the piled up bins (the trend described is parabolic, with maximum velocity variation of about 23%). The variation between ventilation supply velocity to the bin slits is analogous for both machine configurations.

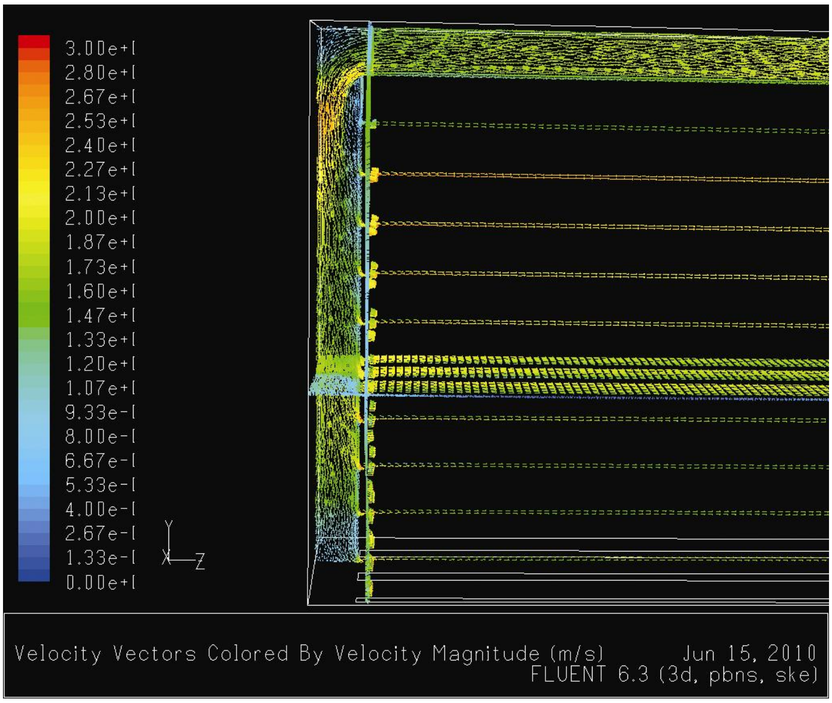


Figure 8 – Flow velocity at the bottom end of the cold room – blower fan machine

Figure 9 highlights the interesting vector flow velocity near the machines (in this specific case of a blower fan). Despite a deflector being installed, which improves the machine air throw fluid dynamics (heat exchanger units) to the channel created above the bins, it can be noted that there is a large vortex below it with air recirculation from the discharge to the fan air intake. The recirculation flow rate is estimated as being about 8%. If this flow rate were eliminated with suitable moveable flaps/closures, it would improve the return velocity in the channels between the bins, and therefore improve heat exchange efficiency between air and apples.
Figure 10 shows the velocity distribution within the entire cold room. The analysis in this figure is interesting if it is seen in comparison with figure 12, which shows the same flow velocity in relation to the traditional suction fan cold room.
The first fundamental difference is the velocity uniformity on the air jet, which is noticeably better in the case of a blower fan machine. This leads to a smaller turbulent area near to the extremes of the machines. During the humidification phase, droplets are drawn over anomalously to these areas, and there is lower energy dissipation (linked to turbulence intensity).
A common turbulent area is created at the bottom of the cold rooms in both solutions, when the air goes down the channel and meets the floor, generating pressure gradients.
These turbulences can be eliminated by fitting simple flow rectifiers (flexible vertical walls).

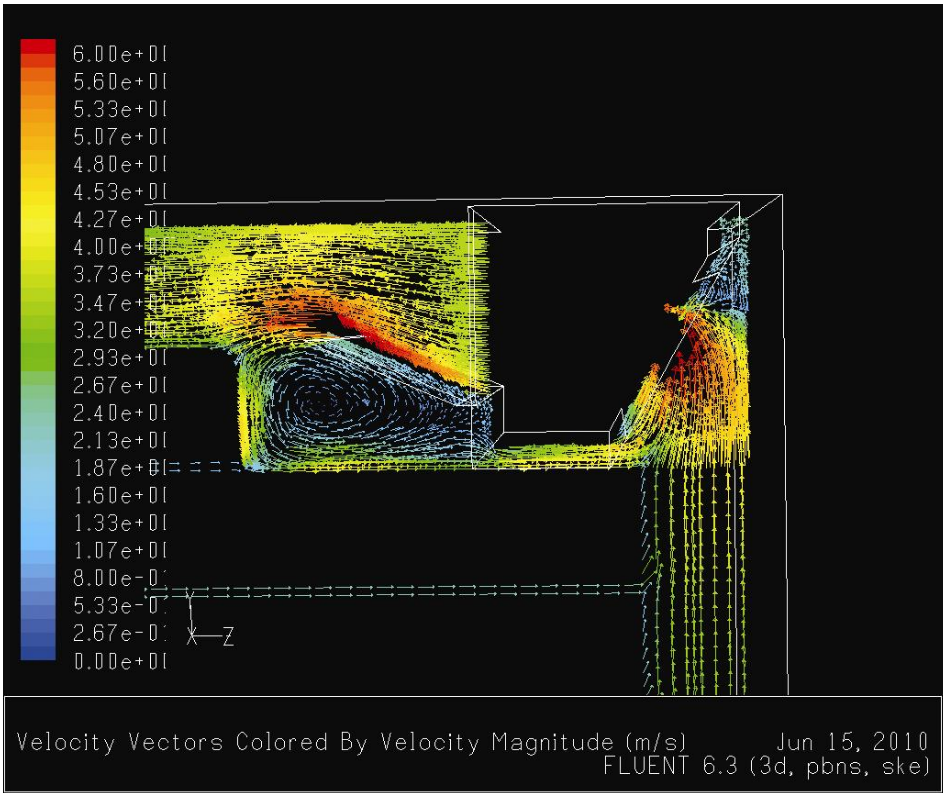


Figure 9 – Particular flow velocity near the blower fan machine

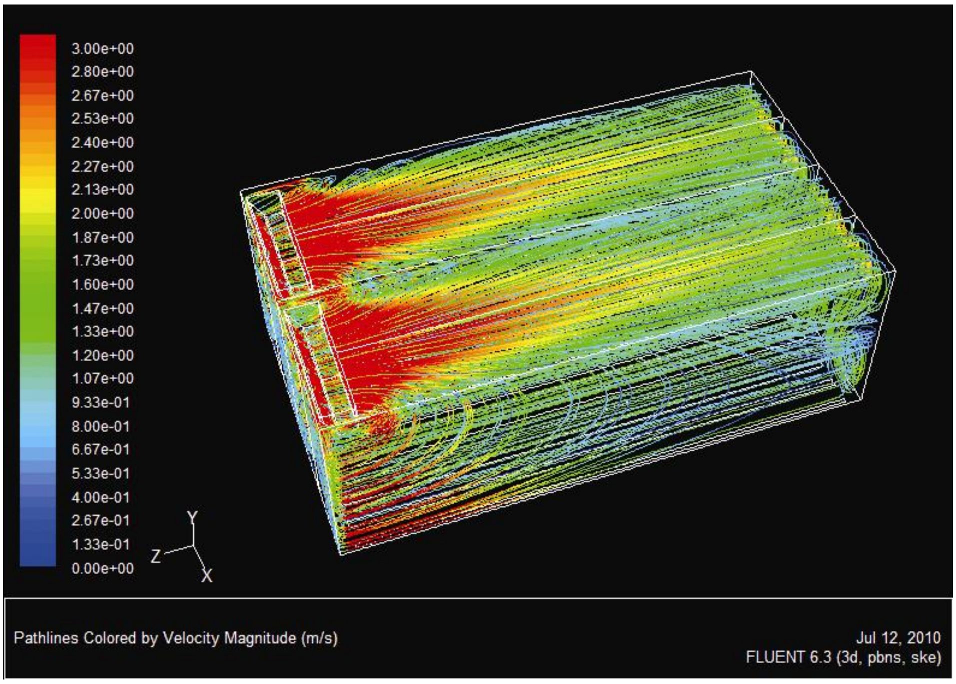


Figure 10 – Flow velocity in the blower fan machine cold room

COLD ROOM WITH A TRADITIONAL SUCTION FAN

In the same way as in figure 9, figure 11 shows the vector flow velocity near the traditional suction fan machine. The fast flow of air emitted from the fans (about 9m/s), deflected in this case, too, by the 30° inclined baffle, creates a large turbulent area immediately below the flow.
This phenomenon gives rise to energy loss and to a flow rate directly sucked in by the heat exchanger, estimated at around 11%.

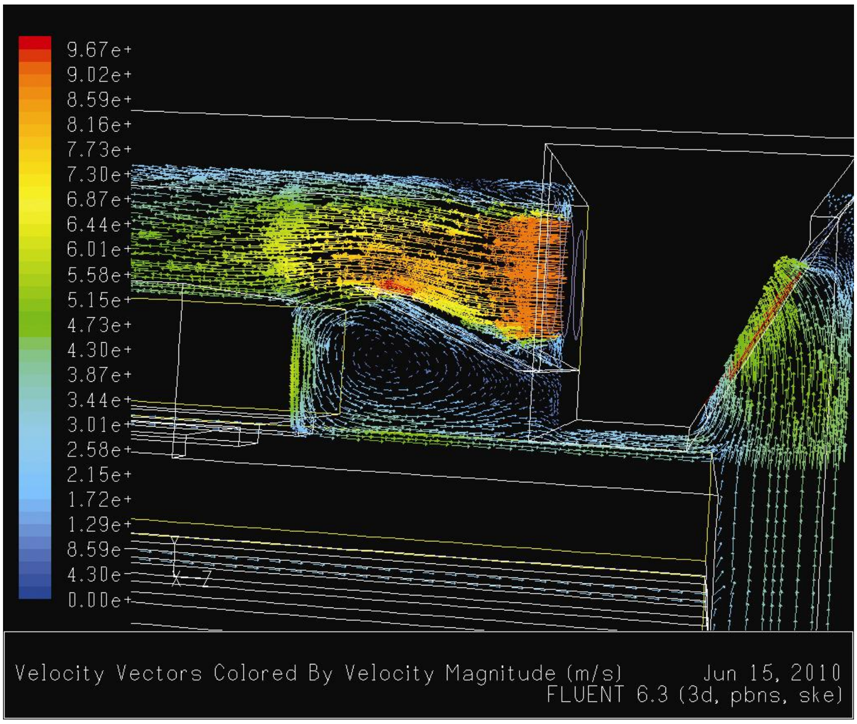


Figure 11 – Particular flow velocity near the suction fan machine

As already seen in figure 10, figure 12 shows the cold room air flows coloured on the basis of the velocity modulus. We have already discussed the significant turbulent flows that arise at the sides of the machines, and the important velocity gradients in the air launch channel. We should now also consider the air recirculation component near the walls (raised base of 120mm). The instruments gave velocity readings of about 1.3 m/s for the suction fan cold room and of 1.1m/s for the blower fan cold room.
The CFD analysis shows us that the air “return” velocity values (towards the machines) are higher in the area near to the cold room floor. The velocity vector inversion point gets continuously nearer to the floor as the bottom end of the cold room is reached. In other words, the air flow returning to the machines, in the cold room raised base area, becomes more important the closer one gets to the machines themselves. This air recirculation is estimated as being indicatively about 8% in the extraction (suction) fan cell and 6% in the blower fan one. The more uniform air flow in the blower fan solution allows air recirculation to be noticeably reduced.
This air by-pass reduces the total air flow onto the bins and would therefore reduce the convective heat exchange. It should hence be eliminated by fitting suitable flexible deflector/baffle walls (that cannot be damaged during the loading phase) positioned in two or more points, starting from the areas near to the unit coolers.
Placing these walls at the bottom end of the cold room would prove to be inefficient.

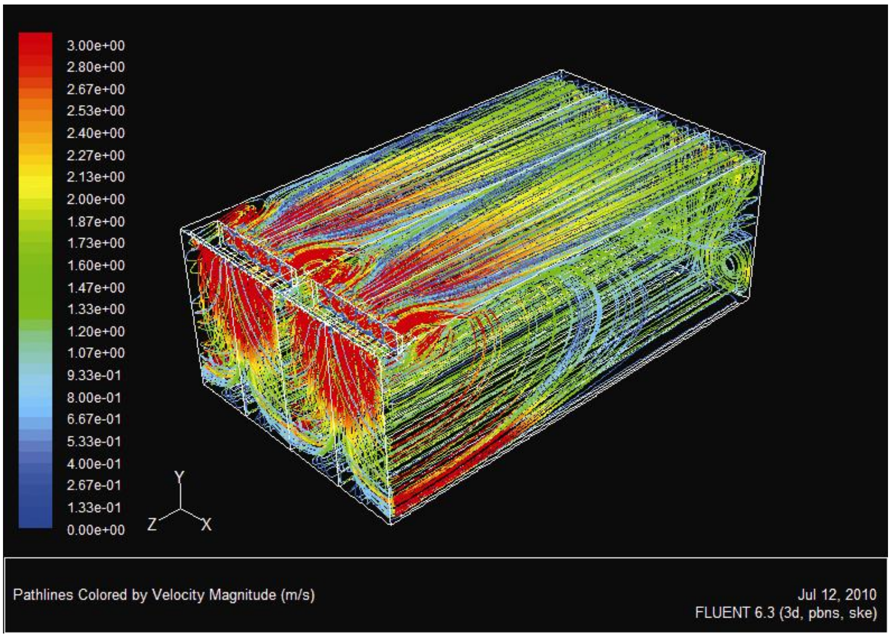


Figure 12 – Flow velocity in the suction fan machine cold room

4. Conclusions

To conclude this lengthy research and experimentation activity, the main results obtained with the innovative blower fan solution can be summarised and some further possible margins of fluid dynamic improvement can be indicated.

- It was quite evident that there was less energy consumption in the blower fan package (-19% over the year) with saving peaks during the initial cooling phase (-34%). There were annual cash savings of about €500 (for an electrical energy cost of €0.0713 KWh).
- As regards weight loss, the difference between the two cold rooms was noticeable:
    - *Total weight loss (blower fan cold room) =1.5% = 0.00771%/day= 0.23%/day*
    - *Total weight loss (extraction (suction) fan cold room) =1.79% = 0.0094%/day= 0.28%/day*

In absolute terms, the lower weight loss of the blower fan solution allowed there to be an overall fruit weight of more than 17 quintals in cold room 30 at the end of the storage period.

- The quality depreciation of the apple samples checked - both at the end of storage and after their shelf life - proved to be similar and absolutely normal.
- The lower number of humidification hours/annum required in the blower fan package cold room (11%) was somewhat evident and should be attributed to a more uniform velocity distribution at the unit cooler outlet (from the heat exchanger instead of the fan outlet nozzle).
- As regards cooling operations, these took place on fewer occasions and for an average duration of 7.7 minutes in storage cell 31, compared to 4.8 minutes and on more occasions in cold room 30 (innovative). The latter is indicative of better heat exchange and more uniform air velocity. The CFD study allows us to indicate the following fluid dynamic improvements:
- Avoid air recirculation between the expulsion outlet of the unit cooler and the air inlet (separation of the two areas). The greater the static pressure that the fan has to “overcome”, the greater the importance of this rule.
- Limit the air loops in the area near to the cold room side walls (raised base). This can be done by placing flexible deflector/baffle walls.

These improvements could increase the flow rate circulating in the cold room (i.e. the one that is really directed onto the bins) by about 15-20%.