

UNIVERSITÀ DEGLI STUDI DEL MOLISE



Department of Agricultural, Environmental and Food Sciences

PhD Course in:

AGRICULTURE TECHNOLOGY AND BIOTECHNOLOGY

(CURRICULUM: SUSTAINABLE PLANT PRODUCTION AND PROTECTION)

(CYCLE XXX)

Related disciplinary scientific section: AGR/09 (Meccanica Agraria)

PhD thesis

CONTROLLED MECHANICAL VENTILATION TO REDUCE PRIMARY ENERGY CONSUMPTION IN AIR CONDITIONING OF GREENHOUSES

Coordinator of the PhD Course: Prof. Giuseppe Maiorano

Supervisor: Prof. Antonio De Cristofaro

Co-Supervisor: Prof. Ing. Pasquale Catalano

PhD Student: Claudio Perone

153741

NOMENCLATURE

A	area [m^2]
BLDC	brushless direct current
CE	confined environment
COP	coefficient of performance [$\text{W}_{\text{th}}/\text{W}_{\text{e}}$]
E	primary energy [Wh]
EAHP	exhaust air heat pump
EE	external environment
HPT	high pressure transducer
i	index
LPT	low pressure transducer
n	speed of compressor [rps] / number of air change per hour [1/h]
P	heat/power
q	heat gains / losses [W]
T	temperature [$^\circ\text{C}$]
U	Thermal transmittance [$\text{W}/(\text{m}^2\text{K})$]
V	volumetric flow rate [m^3/h]
V	volume [m^3]

Greek symbols

Δ	difference
ε	efficiency
η	temperature ratio
ρ	density

Subscripts

a	air
AI	air in
AO	air out
c	condensation
DB	dead band
DIS	discharge
DIS_ST	static discharge
e	electric / evaporation
E	exhaust
EAHP	exhaust air heat pump
f	floor
h	heating system
HP	heat pump
i	i-th element

inf	infiltration
m	motor
net	net value
O	outdoor
R	recovery / refrigerant
R_ST	static recovery
ref	refrigerant
ren	renewal
S	supply
s	system / specific / sensible
S_ST	static supply
SET	set-point
SET_calc	calculated set-point
sh_d	delivery superheat
sh_s	suction superheat
SIVeMeC	SIVeMeC prototype
SO	solar
sub_s	summer subcooling
sub_w	winter subcooling
T	total
Umax	maximum useful
vi	ventilation in
vo	ventilation out
w	wall
Superscripts	
1	without electrical absorption of fans
2	overall system

ABSTRACT

In order to ensure optimal growing conditions inside greenhouse it becomes necessary a very close control of the internal climate conditions. In the first section, the available conditioning plant solutions are described. However, these systems generally require high investment costs. In addition, also high operational costs are required for an efficient solution without reducing yield crop or quality. Therefore, the winter conditioning of the internal air of a greenhouse occurs by means of fossil fuels. The use of a mechanical ventilation system contributes to a proper control of temperature, relative humidity and CO₂ rate. However, the literature about the application of mechanical ventilation with heat recovery applied in greenhouses conditioning is very poor. For this purpose a research is being carried out.

In section 2 a prototype of a mechanical ventilation unit and two climate rooms, for the reproduction of external and internal (built in laboratory) conditions, are described. The recovery unit is equipped with a heat pump and is able to increase the thermal energy recovered by the flow of exhaust air and through a high efficiency heat exchanger.

A first study was carried on to evaluate the energy performances of the system during the control of temperature in winter season. Tests reported in section 3 were performed at different temperature values of simulated outdoor air T_o (-5 °C, 0 °C, 5 °C and 10 °C) and a fixed (reference) internal simulated greenhouse temperature (20 °C). Each trial was performed with a ventilation flow rate of 535 m³/h. The resulting Coefficient Of Performance of the overall system (COPs) is 9.50 at 0 °C, 8.86 at 5 °C and 6.62 at 10 °C respectively. It has to be highlighted that during the trials carried out at -5°C the compressor behaved as an on-off type. This is due to a safety mechanism for the defrost of the evaporator. In addition, the ventilation flow rate was reduced to avoid a too low value of the supply air temperature. For the other trials (T_o = 0 °C or 5 °C), the overall COPs decreases when the external temperature increases, due to a lower difference between external and indoor air enthalpy.

To study a real case the mechanical ventilation unit was also installed at service of a greenhouse at Vivaio Verde Molise, Termoli – Italy. The experimental apparatus, described in detail in section 4, consists of the mechanical ventilation system, a perforated duct for air distribution, a fog system to adjust humidity and a supervision system to acquire the field data. Another dedicated supervision system allows measuring and collecting all the parameters of the prototype, such as thermophysical parameters of the airflow, thermophysical parameters of the refrigerant circuit of the heat pump, status and alarms of the unit.

First tests, carried out on temperature control in winter season, are analysed in section 5. They show that the indoor air temperature (set at 27 °C) is suitable regulated by driving the unit with the reference probe installed on the recovery side. Only an offset of few Celsius degree is observed due to duct heat loss and the recovery grid placed on one side. Moreover, the mechanical ventilation system had also shown notable energy performance: COPs (mean value) of 5.4 and 5.7 at outdoor air temperature of 18.0 °C and 15.7 °C respectively.

Finally, section 6 displays the main conclusions of the present work.

ACKNOWLEDGMENTS

Firstly, my great gratitude goes to my two supervisors, Prof. Antonio De Cristofaro and Prof. Pasquale Catalano, for their guidance and for their valuable suggestions that made the development of this work possible.

I must thank too the agricultural engineering working group, in particular Prof. Ferruccio Giametta for assisting me during my period abroad at Kyoto University.

Special thanks are also due to Prof. Flavio Fucci for welcoming me in the agricultural engineering working group and for his always-precious teachings.

I would also like to extend my appreciation to Clifel S.r.l. for having funded a substantial part of the research within the POR FESR Program Molise 2007–2013 Activity I.2.1.

Finally I want to thank my family who has always supported me. They always provided to me the assistance and the strength throughout my studies.

A loving thank is for my daughter Danila, my darling, to whom this work is dedicate. She gave me one more reason to face the challenges encountered on my path.

LIST OF FIGURES

Figure 1: classification of air-conditioning systems [2].....	3
Figure 2: air-condensing unit.	18
Figure 3: air-evaporator unit.	18
Figure 4: immersed electrodes humidifier.	19
Figure 5: connection duct from external environment to the suction side of the system.	20
Figure 6: fan-coil reproducing summer temperature values.	20
Figure 7: heat pump external units.....	21
Figure 8: confined environment split unit.....	21
Figure 9: radiator with thermostatic valve.	22
Figure 10: immersed electrodes humidifier in the confined environment.	22
Figure 11: climatic rooms a) E.E. climatic room with the plenum room, b) C.E. climatic room....	23
Figure 12: general control panel.	23
Figure 13: electric power panel and refrigeration plant panels.....	24
Figure 14: SIVeMeC installation and its connections with climatic rooms.	25
Figure 15: “wire sheet” programming for the control logic design.	27
Figure 16: user-interface for the managing of the climatic rooms conditions.	28
Figure 17: SIVeMeC user-interface for control and managing.	29
Figure 18: SIVeMeC heat-pump user-interface.....	29
Figure 19: schematic laboratory user-interface page.	30
Figure 20: history page to select the variables of interest for historicization.	31
Figure 21: SIVeMeC prototype.....	32
Figure 22: psychrometric transformation of renewal air (blue line) and exhaust air (red line) on Mollier diagram.....	34
Figure 23: pressure drop (PD) through the heat exchanger in function of volumetric airflow rate AV; the line PD/AV is that for 500 m ³ /h.....	35
Figure 24: heat pump cycle in winter arrangement. 1) BLDC compressor; 2) evaporator; 3) condenser; 4) electronic expansion valve; 5) inverter for compressor driving; 6) 4-way valve; 7) liquid receiver; 8) dehydrator filter; 9) sight glass; Tsh_s suction superheat temperature; HTP high pressure transducer; LPT low pressure transducer; Te evaporation temperature; Tc condensation temperature; Tsh_d delivery superheat temperature; Tsub_w subcooling temperature in winter arrangement; Tsub_s subcooling temperature in summer arrangement; μPC control panel.	36
Figure 25: compressor performance curves at 60 rps (right) and 30 rps (left).	38
Figure 26: control panel of SIVeMeC. Power+ is on the right; μPC is in the low side.	39

Figure 27: E ² V smart electronic expansion valve.....	40
Figure 28: six rows expansion coil	41
Figure 29: two rows expansion coil	42
Figure 30: technical drawing of electronic fan	43
Figure 31: fan characteristic curves.....	44
Figure 32: by-pass duct above the heat exchanger.....	45
Figure 33: remote panel PGD1.....	47
Figure 34: integrated simulator 1Tool®	48
Figure 35: 1Tool® screen. The sidebar contains the pages that make up the strategy. The "AnalogInput" page is open in the main window.....	50
Figure 36: two PID controllers in series to manage compressor, delivery air fan and exhaust air fan.....	51
Figure 37: 1Tool® part of program for compressor calling.....	52
Figure 38: 1Tool® program part for fans speed regulation by using CO ₂ probe as reference.....	52
Figure 39: on-off behaviour of the compressor (test with T _O = -5°C).....	62
Figure 40: airflow rate reduction due to the high difference between T _s e T _{SET_calc} (test with T _O = -5°C).....	62
Figure 41: temperature trend at the evaporator inlet (test with T _O = -5°C).....	62
Figure 42: terms appearing in sensible energy balance of a typical agricultural building [1].....	66
Figure 43: mass balance for a typical agriculture building [1]	68
Figure 44: dimensions of the greenhouse served by SIVeMeC.....	71
Figure 45: experimental apparatus installed at Vivaio Verde Molise, Termoli - Italy.....	73
Figure 46: perforated duct for supply air distribution.....	74
Figure 47: solenoid valve.....	75
Figure 48: experimental apparatus layout and probes location.....	75
Figure 49: PlantWatch PRO®.....	76
Figure 50: control panel equipped with energy meter.....	78
Figure 51: user interface of pCOManager® for SIVeMeC monitoring.....	79
Figure 52: Test 1 – exhaust air temperature (T _E), supply air temperature (T _s) and coefficient of performance (COPs) as a function of outdoor air temperature (T _O).....	82
Figure 53: Test 1 – coefficient of performance (COPs) as a function of the compressor speed.....	83
Figure 54: Test 1 – temperature distribution inside greenhouse.....	84
Figure 55: Test 2 – exhaust air temperature (T _E), supply air temperature (T _S) and coefficient of performance (COPs) as a function of outdoor air temperature (TO).....	85

Figure 56: Test 2 – coefficient of performance (COPs) as a function of the compressor speed.	86
Figure 57: Test 2 – heat supplied by SIVeMeC (P_R) as a function of compressor speed.....	86
Figure 58: Test 2 – compressor speed as a function of exhaust air temperature T_E	87
Figure 59: Test 2 – temperature distribution inside greenhouse.....	88

LIST OF TABLES

Table 1: probe used in the laboratory.....	25
Table 2: typology and number of probes for each zone.....	25
Table 3: REK +39 technical data according to EN 308: 1997.....	33
Table 4: passive heat recovery P_{R_ST} , residual heat load P_{HP_set} , supply air temperature (T_{S_ST}) and discharge air temperature (T_{DIS_ST}) downstream the static recuperator, as a function of outdoor temperature T_O for fixed values of exhaust air temperature (T_E), temperature ratio η and both exhaust (V_E) and supply (V_S) airflow rate.	37
Table 5: technical data of DA75F0F – 11UA compressor.....	38
Table 6: technical data of direct expansion coils.	42
Table 7: average values and standard deviation of the main physical measured parameters.	59
Table 8: average values and standard deviation of the main energetic calculated parameters.	60
Table 9: Comparison of the COP of different systems.	64
Table 10: values of thermal transmittance for values of wind speed below 4 m/s.	69
Table 11: Optimum temperature for different crops [35].	71
Table 12: Heat losses by considering only transmission through structure and air infiltration.....	72
Table 13: probe used in field.....	76

INDEX

NOMENCLATURE	i
ABSTRACT.....	iii
ACKNOWLEDGMENTS	v
LIST OF FIGURES	vi
LIST OF TABLES	ix
1. INTRODUCTION	1
1.1 Research project	1
1.2 State of the art.....	2
2 PROTOTYPE OF THE MECHANICAL VENTILATION SYSTEM – SIVeMeC	12
2.1 General aspects	12
2.2 Laboratory	16
2.2.1 Climatic rooms.....	16
2.2.2 Supervisory system	23
2.3 System configuration.....	31
2.3.1 Countercurrent heat exchanger	33
2.3.2 Heat pump	35
2.3.3 Electronic fans.....	43
2.3.4 By-pass ducts	44
2.3.5 SIVeMeC	45
2.4 SIVeMeC control system	46
2.4.1 Temperature control.....	51
2.4.2 Heat pump control.....	51
2.4.3 Ventilation control	52
3 LABORATORY EXPERIMENTAL TESTS.....	54
3.1 Experimental procedures.....	54
3.2 Energy parameters	54

3.3	Results and discussion.....	58
4	GREENHOUSE EXPERIMENTAL APPARATUS	66
4.1	Greenhouse energy and mass balance	66
4.2	Experimental apparatus installation	72
4.2.1	Air and water distribution systems.....	74
4.2.2	Field probes.....	75
4.2.3	Field supervisory system.....	76
4.2.4	SIVeMeC supervisory system.....	78
5	GREENHOUSE EXPERIMENTAL TESTS	81
5.1	Experimental procedures	81
5.2	Results and discussion.....	81
6	CONCLUSIONS	89
	REFERENCES.....	91
	STANDARDS	94
	ATTACHMENT 1.....	

1. INTRODUCTION

This PhD thesis deals with the mechanical ventilation technology used to reduce the primary energy consumption in air conditioning within confined environment. This technology is well established in the civil sector but cannot be transferred to the agricultural one without an appropriate optimization. In fact, in this particular application the need to ventilate confined environments is important not only to guarantee the healthy air for the people welfare , but also for the animals welfare (in their shelters) and the growth of the protected crops in confined environments such as the greenhouses. Latter is the prevalent context in which this research can be developed, without excluding the other fields of application.

Greenhouses require an appropriate control of air temperature, relative humidity, carbon dioxide level and light. Suitable environmental condition have a direct effect on quantity, quality and timing of crops growth. The need for quantity is obvious (production increases when the temperature increases up to the point where damaging begins). Environmental conditions are critical in determining quality with many greenhouse crops. Also timing is very important for assuring profit (consider for example the market for poinsettias).

Heating, cooling, ventilation and air distribution are critical operations for environment control in greenhouses. While in cool climates it is possible to maintain suitable conditions by natural ventilation alone for plant growth during summer, mechanical ventilation is necessary for most conditions [1].

The increase of annual crop yield in a greenhouse is possible only through a very close control of the internal conditions. Currently available conditioning plant solutions require both high investment costs and high operational costs for an efficient solution without reducing yield crop or quality. Current conditioning systems applied in a greenhouse use fossil fuels, especially during winter. Ventilation systems could allow proper control of temperature, relative humidity and CO₂ rate.

1.1 Research project

Aim of the present PhD thesis was to design and realize a high efficiency *dynamic mechanical-ventilation system* (SIVeMeC – Sistema Integrato per la Ventilazione Meccanica Controllata: Integrated System for Controlled Mechanical Ventilation) integrated with a heat pump. The system works adapting itself to the real ventilation need of the confined environment.

The performance of such system in different configurations was analyzed:

- a) static heat recovery;

- b) thermodynamic heat recovery;
- c) both thermodynamic and static heat recovery;
- d) free heating/free cooling.

Two climatic rooms were realized to evaluate the energy performance of the prototype and to reproduce different operative boundary conditions before applying it in a greenhouse.

Afterwards the tests carried out in laboratory the prototype was installed at service of a mini-tunnel greenhouse located at Vivaio Verde Molise, Termoli – Italy.

Plant growth and annual crop yield in greenhouse can be increased also through a very close control of the internal conditions. However, a close control of the climate condition of a greenhouse requires the installations of new technologies that may lead to an additional energy consumption. Usually, greenhouse heating is done by firing fossil fuels in a boiler and this is disadvantageous at both the energy and environmental level.

The experimental apparatus consists of a mechanical ventilation system, a perforated duct for air distribution, a fog system to adjust humidity and a supervision system to acquire the field data. Another dedicated supervision system allows measuring and collecting all the parameters of the prototype, such as thermos-physical parameters of the airflows, thermos-physical parameters of the refrigerant circuit of the heat pump, status and alarms of the unit. In order to characterize the whole system under investigation it is possible to measure and collect temperature, relative humidity and carbon dioxide inside the greenhouse and temperature in all sections of the prototype. Other two mini-tunnel greenhouses used as references are monitored. One is heated with radiant tubes placed on the bench, and humidified with a fog system; the other is completely passively heated (without heating system) and humidified with the same fog system.

1.2 State of the art

Environmental control of greenhouses involves the managing of the main parameters that characterize the internal conditions. To this purpose, air conditioning systems are often used. Based upon their characteristics these systems are classified as heating systems, cooling systems and composite systems (Figure 1):

- Heating systems: water storage, rock bed storage, PCM (Phase Change Materials) storage, movable insulation, ground air collector and north wall.
- Cooling systems: natural and forced ventilation, shading and evaporative cooling (fan-pad, mist/fog and roof cooling).

- Composite systems: EAHES (Earth-to-Air Heat Exchanger System) and ACCFHES (Aquifer Coupled Cavity Flow Heat Exchanger System).

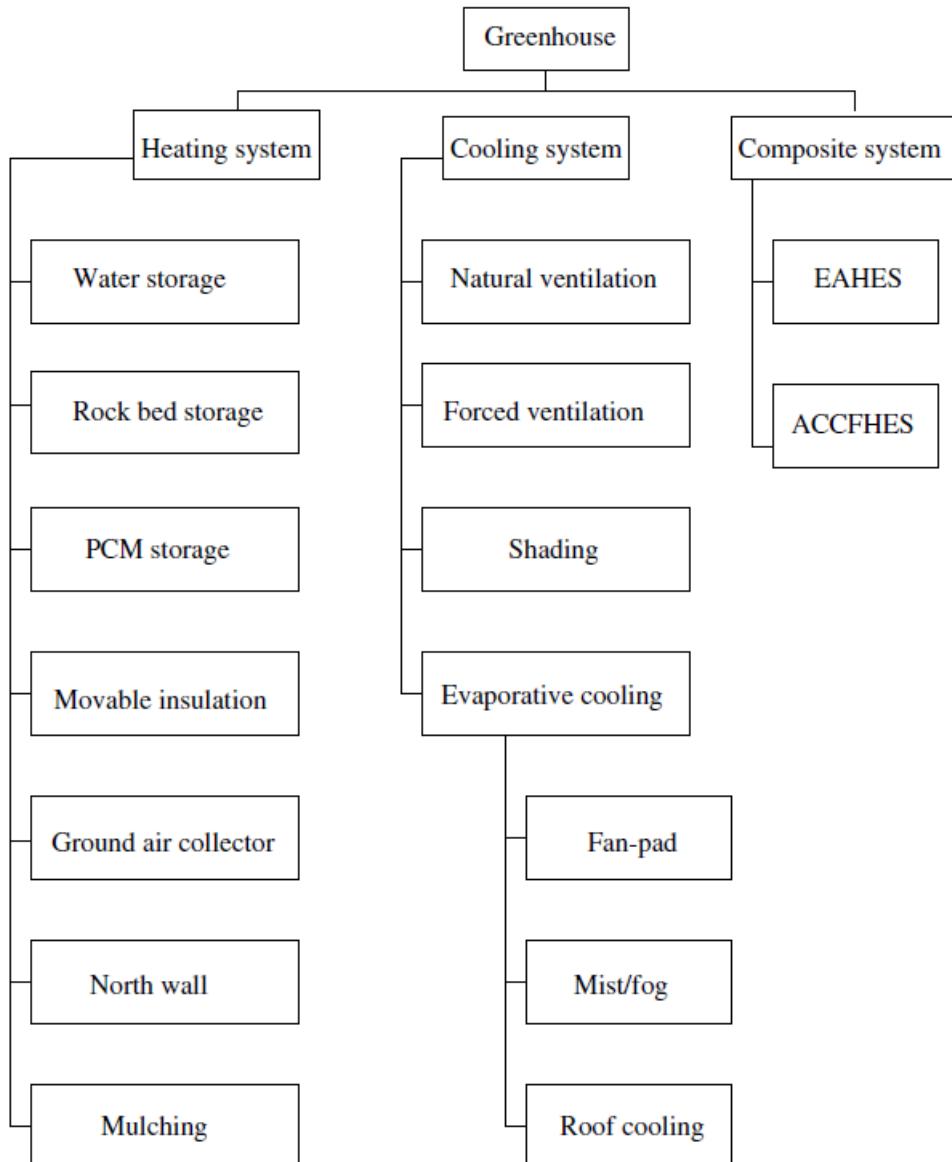


Figure 1: classification of air-conditioning systems [2].

The aim of the **heating systems** is to replace energy lost from the greenhouse when outside temperatures are lower than desired one in the greenhouse growing area. Apart from these factors overall performance of a greenhouse coupled with any heating system is influenced by several interrelated parameters such as: size of greenhouse, type of cover material used, heat storage method, quantity of material used, type of cultivation, desired day and night temperature of the inside air, location of the greenhouse and outside ambient conditions [2].

Water storage systems consist of: water filled plastic bags and ground tubes placed inside the greenhouse on the pathway between the row of plants, water tanks/barrels (sometimes black

painted) along the north side of the greenhouse that act as solar collectors and heat storage media. It was observed an increase of internal temperature of 2 – 6 °C by using plastic bags/tubes. The problem of using these systems is that about 20 – 25 % of the valuable ground surface becomes unavailable and cannot be used for cultivation [2]. Water tanks/barrels occupy lesser space inside the greenhouse as compared to ground tubes and allow an increase of internal temperature of 2 – 11 °C and can satisfy 25 – 62 % of the annual heating needs of the greenhouses at various locations.

Another popular and economical heat storage material is rock bed (pebble, gravel and bricks), in which the sensible heat is stored in an underground rock bed placed at a depth varying between 40 – 50 cm or outside the greenhouses. The air is used as the energy transport mechanism. The collected information shows that gravel having 20 – 100 mm diameter is preferred as bed material. Heat capacity of the gravel is 157.28 kJ/°C (about 220 kg) per m² of greenhouse area and can satisfy 20 – 70 % of the annual heating needs [3]. Inside air temperature ranges from 4 to 13 °C above the minimum external air temperature. The main problems of this system are related to the huge quantity of rock pile required (due to the low thermal capacity) and the pumping cost of air through the rock pile (high pressure drop created by the resistance of the rocks).

Phase change materials (PCMs) are able to store a large amount of latent heat in the phase change from solid to liquid (latent heat of fusion) at the constant pressure and phase transition temperature. A circulating fluid (air or water) extracts the latent heat from the storage unit causing the solidification of the material then the cycle starts again. The most popular PCM used for greenhouse applications is calcium chloride hexa-hydrate ($\text{CaCl}_2 - 6\text{H}_2\text{O}$), which melts at 29 °C. The representative value of the latent heat capacity of this eutectic mixture used per m² of the greenhouse ground area is approximately 2600 kJ, which is equivalent to 13.70 kg per m² [2]. The use of $\text{CaCl}_2 - 6\text{H}_2\text{O}$ can increase the internal air temperature of 2 – 8 °C than the outside one and can satisfy 22 – 75 % of the annual heating needs. The energy storage density of $\text{CaCl}_2 - 6\text{H}_2\text{O}$ is over ten times higher than the rock bed but the current price is about four to five times than the gravel system to store the same amount of energy. Moreover, this inorganic compound is corrosive to most metals and suffer from decomposition and sub-cooling, which can affect its phase change properties.

Movable insulation is a passive system able to reduce the heat loss through the boundaries of the greenhouse. The main effect of the curtain is to provide additional thermal resistance that reduces the overall rate of heat transfer to the surroundings. During daytime, the movable

insulations are removed in order to allow solar radiation to enter and closed during nighttime to reduce thermal losses. Generally, their internal installation is to be preferred because it reduces the thermal conductivity of the structure. Furthermore, if aluminized polyester sheets are used on both side of the curtain, the maximum energy saving is obtained. A study was carried out to observe the effect of different greenhouse design parameters and heating systems on the greenhouse energy saving [4]. Curtains reduce the nighttime heating requirement by 70.8 % and daily requirement by 60.6 %. However, they have poor mechanical reliability, incomplete sealing after closure and are subject to condensation damage.

North wall consists of a thick thermal mass, made of brick or cement blocks filled with concrete able to capture and retain the solar radiation transmitted through the greenhouse. North wall is therefore externally insulated and internally painted black to increase thermal storage. The heat stored during daytime is released during the night (or when the internal temperature falls below the storage temperature). A study carried out on 30 small size greenhouses (20 m^2 floor area each), using north wall heat storage made of 60 cm wide concrete blocks showed an indoor air temperature of $15 - 20^\circ\text{C}$ during winter conditions, when outside temperature was less than 10°C [5]. Another study showed that these systems can cover about 35 – 50 % of greenhouse heating needs [6].

Ground Air Collector (GAC) is a flat plate collector installed in the ground subsurface. The information about the use of GAC is limited in literature. It comprises four components: (1) glazing for providing the greenhouse effect; (2) sand or concrete bed as an absorber of solar radiation; (3) tube or passage for conducting or directing the heat transfer fluid from inlet to outlet and (4) blower for air circulation [7]. The heat stored by the sand is transferred to the pipe embedded in it. The air circulating in the pipe is heated and then, the hot air is sent to the internal of the greenhouse, while the indoor air is drawn from the greenhouse and is heated again. It was seen that GAC can raise the inside air temperature by $5 - 6^\circ\text{C}$ during extreme winter condition [8]. However, 0.73 m^2 GAC area is required per m^2 of greenhouse area [2]. It also involves different cost items: pipes, air pumping, electric motor and blower assembly.

The **cooling systems** include ventilation (natural and forced), shading and evaporative cooling (fan-pad, mist/fog and roof cooling). Ventilation consists of replacing the internal air with cool air from outside.

Natural or passive ventilation is based on differential pressure, between inside and outside, created by the wind and/or the greenhouse temperature. The external cool air enters through the

lower side openings while the internal hot air exits through the roof openings due to density difference between air masses.

In forced ventilation, systems like exhaust fans or blowers can supply high air exchange rates. Generally, ventilation can reduce the inside temperature, but the lower limit is represented by the external temperature. Moreover, in natural ventilation another critical factor is the rate of air exchange by free convection. A total vent area equivalent to 15 – 30 % of the floor area is recommended for effective cooling and the presence of the insect screen significantly reduces the airflow and increases the internal thermal gradient inside the greenhouse [9]. Although little advantage is observed with fan cooling because of increasing of the flow rates over 0.05 $\text{m}^3/\text{m}^2\cdot\text{s}$, it is still not enough to lower the internal air temperature during the summer peak [10].

Shading/reflection can be done by various methods such as paints, external shade cloths, louvers or slatted blinds, nets (of various colors), partially reflective shade screens, water film over the roof and liquid foams between the greenhouse walls [11]. Generally, the use of these systems can reduce the internal air temperature of 3 – 6 °C in mild summer conditions. On the contrary this system does not allow to reduce adequately the inside temperature in hot climatic conditions.

Evaporative cooling is based on the conversion of sensible heat into latent heat by means of evaporation of water supplied directly into the greenhouse atmosphere (mist/fog system) or through evaporative pads (fan-pad). It is observed that fan-pad system is able to reduce the internal air temperature of 4 – 6 °C if used alone and of 4 – 12 °C if used with shading [11]. The limits of this system consists of a non-uniform progressive clogging of the pad resulting in a loss of cooling performance. Furthermore, the too high humidity level could also promote the growth of microorganisms facilitating the development of diseases. Mist or fog systems generate small droplets (2 – 60 μm in diameter) with high pressure. It is observed that this system can provide more uniform temperature distributions (3 – 8 °C below ambient temperature) than fan-pad and uniform high humidity level [11]. So, problems related to microorganisms growth and diseases development are observed for the same above-mentioned reasons. Roof evaporative cooling is sprinkling of water onto a surface of the roof to form a thin layer, which results in an increase of the free water surface area and consequently of the evaporation rate. This causes the water temperature fall to the wet bulb temperature of the closely surrounded air. In a study researchers used a sprinkling system placed in the greenhouse attic above the blanket, observing 3 – 4 °C temperature reduction as compared to the control

greenhouse [12]. However, this system does not promote the growth of microorganisms and the development of diseases due to not excessive inside moisture.

In the end, other studies analyze the composite systems able to heat the greenhouse in cold seasons and cool it in hot one. These systems use the geothermal energy. Earth-to-Air Heat Exchanger System (EAHES) uses the constant temperature of the huge underground mass of earth, while Aquifer Coupled Cavity Flow Heat Exchanger System (ACCFHES) uses deep underground aquifer water. These systems are still much expensive and require further investigations.

With particular attention to ventilation, only in recent years mechanical ventilation was proposed as an alternative to natural ventilation [13]. Commonly, both indoor temperature and humidity are controlled by opening, manually or automatically, the windows alongside the greenhouse or on its rooftop. In this way, the indoor air is replaced with the outdoor one which has a lesser moisture content and temperature. In addition, the adjustment of CO₂ happens in the same way. Generally, the outdoor carbon dioxide level is higher than that of the indoor due to its consumption in the photosynthesis.

Natural ventilation is based on the pressure difference between the indoor of the greenhouse and the external environment. Consequently, its efficiency is closely linked to the climatic zone in which the greenhouse is located. In fact, the effectiveness of mass and energy exchange is related to the outside wind and internal and external temperatures. Moreover, natural ventilation efficiency is function of the design characteristic of the greenhouse. When only natural ventilation is used to control the climate condition, the greenhouse is considered an “open-system”.

In the recent years, also mechanical ventilation is considered to maintain the internal condition. The presence of fans shows a lesser dependence from the external conditions and internal buoyancy forces than natural ventilation [7]. Thus, mechanical ventilation reduces internal air stratification and vertical temperature gradients in summer. Furthermore, this system allows treating the airflow before its inlet in the greenhouse (airflow mixing, heat recovery, pre-heating, etc.). If only mechanical ventilation is used, the greenhouse is considered a “closed-system”.

From a literature search of recent studies on mechanical ventilation systems appears that only few works deal with this topic (e.g. [13], [14]). In addition it is usually accompanied by natural ventilation. As a result we have a “semi-closed” system. In these systems, generally, mechanical

ventilation is used as the primary mean limiting the use of natural ventilation at times in which is energetically and economically viable.

The following description of the two cases of study well represents the state of the art about the mechanical ventilation applied to greenhouses until today.

The first of the two systems found in literature involves an innovative ventilation concept based on intensive screening and controlled ventilation through combined mechanical and natural ventilation [14]. The experimental apparatus was installed in a warm temperate humid climate zone. The ventilated greenhouse was insulated with two thermal screens. A first screen guarantees a high thermal insulation (67 %) but a low optical transmission (25 % in sunlight and 24 % when overcast). Thus, it was used only during nighttime. The second screen was a movable AC (Anti-Condensation) foil with high optical transmission (90 % in sunlight and 90 % when overcast). The AC foil avoids the formation of droplets on the foil that fall down on the crops, but promotes the formation of a thin film of water. This water is then removed through drainage gutters embedded in the foil. It could be noted that the presence of a water film increases the thermal insulation and optical transmission of the foil.

Mechanical ventilation can allow dehumidification by avoiding the screens opening, so as to limit heat losses. As a result, the purpose of this experimental apparatus was to postpone the need of natural ventilation through mechanical dehumidification. The mechanical ventilation system was placed at the center of the outer wall and was equipped with three intake ducts for the adjustment of the supply air. One duct is located externally to the greenhouse and is used to carry the outdoor air in. The other two intake ducts are installed at the inner side of the greenhouse. Particularly, one is placed above the screens the other below them. The latter is used for recirculation of internal air while the upper one can provide drier air for dehumidification, as the air above the screen is less humid due to the condensation on the cold deck of the greenhouse (especially in winter season). The air mixing required at the inlet of the compartment is adjusted by controlling the valves installed on the intake ducts. A radial ventilator is in charge of distributing the supply air below the growing gutters through perforated air duct, with perforation every 12.5 cm. The radial fan is designed for a maximum air rate of $10 \text{ m}^3/\text{m}^2\text{h}$. Finally, the air passes through a low temperature heat exchanger before entering the greenhouse. A second stage condenser on the facility's central boiler makes the heat generation. Two tube rails and two growing tubes for each growing gutter represent the main heating system of the compartment.

A central Building Monitoring System (BMS) allowed to manage all field devices and to acquire all the data measured on a minute basis and averaged every five minutes. The mechanical ventilation system has shown an energy saving up to 12 % if compared with a reference greenhouse equipped only with natural ventilation system. The reference greenhouse was similar to the ventilated one except for the thermal screen (with higher optical transmission than that of the ventilated greenhouse, 83% in sunlight and 75% when overcast, and mediocre thermal insulation, 47%). The energy consumption reduction mainly occurred in summertime, because the presence of the pre-heater eliminated the need of the tube heating. The average windows positioning in the ventilated greenhouse was lesser than reference one demonstrating that a lower degree of humidity was possible through mechanical ventilation. Moreover, it can be argued that mechanical ventilation improves homogeneity of the indoor climate and stimulates plant activity. While the crop yield in both greenhouses remains almost identical, a small production gain in favor of the ventilate greenhouse was seen at the beginning of growing season (which is economically beneficial).

The second mechanical ventilation system found in literature was almost identical to that described above, with the exception of the presence of an air-to air heat exchanger [13]. Furthermore, this system has only two intake ducts. The first one is positioned in the inner side of the greenhouse under the thermal screens, while the second one is located at the exterior providing outdoor air supply. In addition, the performance of the thermal screens differs from the previous case. In this concept a first screen has high thermal performance and low optical transmission (72% thermal insulation, 18% direct light transmission and 17% diffuse light transmission) and a second screen has mediocre thermal performance and high optical transmission (45% thermal insulation, 84% direct light transmission and 75% diffuse light transmission).

The presence of a heat recuperator reduces energy loss when ventilating with cold outside air. The heat exchanger extracts heat from the exhaust air outgoing the greenhouse to pre-heat the cold incoming airflow from the outside. Thus, heat loss can be reduced when mechanical ventilation is used for dehumidification. In this application totally heat recuperation of 30 kWh/m², which correspond to 12 % of the ventilated greenhouse's energy consumption, was obtained. Most of this heat is recovered in spring and falls when a higher need for dehumidification coincides with relatively low outside temperature. The heat recovery decreases in summer with the rise of the external temperature, so mechanical ventilation begins to be replaced by natural ventilation. Measurements have shown that about 60 % of all heat is

recovered with an efficiency of 25 – 40 %, while the maximum recuperation efficiency of the heat exchanger amounted to 45 %. Particularly, the average value of the recuperation efficiency was 25.4 %. The data acquired showed that efficiency has a strong dependence on airflow rate (the efficiency increases when the ventilator is at maximum capacity and the outside intake is opened at 100 %). Moreover, when natural ventilation is used combined with mechanical one the efficiency decreases further. This is easily explained by the fact that indoor air parameters conform to the outside conditions by resulting in a reduction of heat potentially available for recuperation.

The mechanical ventilation system has shown an energy saving of 28 % respect to the reference greenhouse in which only natural ventilation is used. The reduction of energy consumption is mainly observed during spring and falls when there were higher levels of humidity inside the greenhouse and relatively low outside temperature. In this way, it become increasingly beneficial to use mechanical ventilation. Indeed, in the same period the reference greenhouses needed more heat to compensate the heat loss due to the frequent use of natural ventilation for dehumidification. Also in summer, when the enthalpy difference decreases, an important energy saving was observed. In fact, in the reference greenhouse a certain amount of heat is still required to remove the excess moisture from the indoor, but this heat was quickly eliminated by the windows opening. Instead, in ventilated greenhouse, humidity is removed by mechanical ventilation system and the heat generated by sunlight could be accumulated.

The main difference in energy consumption of the two cases of study is related to the operating strategy adopted. In the first study the tube rails were used principally in the winter season, when the plants are small, and the growing tubes in summer, when the plants are fully grown (common approach in horticulture). In the second case, tube rail network is used throughout the year and the growing tubes are only a back-up heat source. This means that an additional energy saving is possible through a suitable operating strategy of traditional heating systems.

It is finally important to note that the presence of a low temperature pre-heater has reduced the need for tube rail heating in both cases and has completely replaced the growing tube network in the second case. Since the use of a low temperature heater is advantageous for energy performance of the system, it is important to evaluate the possibility of using a more efficient heat source than boiler.

The lack of studies concerning heat recovery in greenhouses has led to the research activities development addressed in this manuscript. In this research an innovative mechanical ventilation system equipped with an air-to-air heat exchanger and integrated with a heat pump (SIVeMeC:

SIstema di Ventilazione Meccanica Controllata: Integrated System for Controlled Mechanical Ventilation) is realized and tested [15]. This new concept of heat recovery is optimized and installed at service of a mini-tunnel greenhouse. The main differences with the cases of analyzed studies (in particular with the second one) are: an higher efficiency of the heat exchanger, and a heat pump with BLDC compressor driven by an inverter as a heat source for the pre-heating of the supply airflow rate.

2 PROTOTYPE OF THE MECHANICAL VENTILATION SYSTEM – SIVeMeC

The research project started with a collaboration between Califel S.r.l and the Department of Agricultural, Environmental and Food Sciences inside a POR-FESR Molise 2007/2013 program - Activity I.2.1 “*Aiuti alle imprese per l’attività di ricerca industriale, sviluppo sperimentale e industrializzazione dei risultati*”.

The main objectives of that research program were:

- high efficacy: the system had to work smartly, on the basis of the active probes feedback;
- high efficiency: the system had to be equipped with high efficiency components (e.g. high efficiency heat exchanger and electronic fans);
- high reliability: the system had to be driven by a suitable controller able to regulate the unit after the elaboration of the inputs (outputs of the active probes).

The active probes are a fundamental item to overcome the limits of the currently mechanical ventilation systems on the market. Their function is to acquire in real time the values of the fundamentals parameters (thermo-hygrometric parameters, pollutant levels, energy parameters, etc.) and give a feedback. As said, the probes outputs are the controller inputs. The controller represent the “smart unit” responsible for properly adjust the system performance.

The very first stage of the research project was mainly related to the civil sector applications. However, passing through a re-design of the system, its use was extended to other fields such as agricultural, food and livestock sector. The present PhD research activity is based on these assumptions in order to apply SIVeMeC in a greenhouse.

2.1 General aspects

Most of the applications of heat recovery systems use air-to-air devices able to recover heat from exhaust air, which usually has more favourable energetic conditions than outdoor air. There are two types of recovery techniques, one defined active or thermodynamic recovery, which operates according to a thermodynamic cycle (external power supply is needed); the other defined passive recovery in which the heat transfer only occurs in the air-to-air heat exchanger and no energy consumption is needed except that for pressure drop in the heat exchanger. When the heat recovery happens without moving parts, passive heat exchanger is also said static exchanger. In the last few years, passive plus active recovery has taken hold, in which generally a heat pump retrofits a cross flow heat exchanger.

Passive heat recovery systems consist of a heat exchanger able to transfer heat and/or mass from one air stream to the other. Exhaust air from the building passes through one side of the heat exchanger, while the incoming supply air passes through the other side transferring sensible heat only, or both sensible and latent heat [16].

Several configurations of air-to-air heat recovery exchanger exist. Mardiana and Riffat [17] classified these systems on their design characteristics: fixed-plate; rotary wheel, heat pipe and run-around coil (this is a nearly passive device).

Fixed-plate configuration is the most common one and consists of thin plates (made of aluminium or stainless steel) suitably spaced and sealed to realize the internal airstreams. Commonly, airflow arrangement is cross-flow, but to enhance the efficiency counter-current flow arrangement is also used. If plate's material is moisture permeable also latent heat can be transferred. Typical efficiency of fixed-plate exchangers is 50-80 % [17].

In rotary wheel, exhaust air crosses one-half side of a rotating disc, supply air crosses the other side. A motor at no more than 15-20 rpm drives the rotor so the heat absorbed from the hot side is transferred to the cold one. If the rotor material is a hygroscopic one, it is possible to transfer moisture too. It is fairly easy and inexpensive to control heat recovery rate by adjusting the rotational speed of the rotor. Usually, temperature efficiency is above 80 %.

“Heat pipe” heat exchanger is a passive energy recovery technique like an ordinary plate-finned or steam coil, except that the pipes are not interconnected and the heat exchanger is divided into evaporator and condenser sections by a partition plate [18]. The hot air passes through the evaporator while the cold air passes through the condenser side. Heat pipes systems have a thermal efficiency 45-55 %.

In run-around coil systems, coils are connected via pipes to a loop in which a mix of water and anti-freeze fluid flows (heat transfer fluid). Heat is transferred from hot side (exhaust air) to the cold side (supply air) via the heat transfer fluid [19]. This heat recovery type has a thermal efficiency 45-65%.

Thermodynamic recovery systems in buildings usually use an air-air heat pump to recover heat from exhaust air according to refrigeration cycle (EAHP - Exhaust Air Heat Pump). Fracastoro and Serriano [20] classified EAHPs on the fraction of the exhaust air and outdoor air flowing through the two batteries (internal exchanger on supply side and outside exchanger before the exhaust outlet) and on the final use. The first classification method divides systems into “with

“all outdoor air” ones and “with partial recycling” ones. The first one has the highest efficiency because the temperature of thermal sink (e.g. outdoor air) is usually lower than temperature of thermal source (e.g. exhaust air). Instead, the second system allows increasing the amount of heat transferred to the building. Classification on the final use includes air renewal management, production of hot/cold water for air conditioning and production of Domestic Hot Water (DHW).

Riffat and Gillot [21] developed an innovative mechanical ventilation system based on the integration of revolving wire-finned pipes (which are the evaporator and condenser of a vapour compression heat pump) to facilitate air movement, heat recovery and heat pumping in a single unit.

Generally, EAHPs also contribute totally or partially to balance the heating loads of the buildings introducing renewal air at higher temperature in winter (lower in summer) than “neutral point”.

In recent years a new heat recovery concept has been developed. Such systems include both passive and active recovery techniques. Thermodynamic recovery acts in series to the passive one in which a fixed-plate or rotary wheel heat exchanger transfer the heat from the exhaust air to the supply one in winter. Nguyen et al. [22] developed an experimental heat recovery system, which allows to simulate four configurations:

Configuration A - no heat recover is used: it is the reference case;

Configuration B – it uses a separate sensible heat recovery;

Configuration C – heating and ventilation are integrated in a single-heat-recovery system

Configuration D – it is a double-heat-recovery system with sensible exchanger and heat pump (also in this case heating and ventilation are integrated in a single unit).

The study has shown that Configuration D is the most efficient.

D’Este et al. [23] compared different heat recovery systems: passive, exhaust heat pump and combined heat exchanger and heat pump. They proved that the combined system has the highest performance.

During my research activity, an innovative combined system with a counter-current flow heat exchanger retrofitted by a variable capacity heat pump is developed analysing its performance. The high efficiency heat exchanger has two main aims:

1. to reduce the compressor size;
2. to reduce the heat pump operating range so to avoid high or low pressure stops during unfavourable outdoor conditions.

The presence of passive heat recovery, unlike the only thermodynamic recovery, modifies the temperature at the inlet of the direct expansion coils. This fact has a direct effect on the heat pump coefficient of performance (COP_{HP}). Particularly a higher passive recovery means a lower COP_{HP} because of a lower evaporation temperature. Due to these considerations, we introduced a BLDC (Brushless Direct Current) compressor and two electronic fans to modulate both supply and exhaust airflow rate.

The possibility to manage independently the airflow rates allows regulating both the passive and thermodynamic recovery efficiency. Moreover, the adjustment of fans rotation speed produces a higher energy efficiency during free-cooling arrangement due to lower pressure drops (by-pass of static heat exchanger).

The BLDC compressor allows managing the heat load downstream the efficient static recovery without bypassing the superheated vapour. Indeed, heating load modulation reduces the heat exchanged by employing only a portion of the compressed vapour in the condenser but does not decrease the compressor power absorption. This fact means an energy loss (heat produced but not utilized).

In order to understand the influence of the temperature ratio η (according to EN 308: 1997) and the COP_{HP} on energy consumed for each m^3 of air, we can refer to the specific primary energy E_s (Wh/m³):

$$E_s = \frac{0,335 \cdot \{T_s - [\eta \cdot (T_e - T_o)T_o]\}}{COP_{HP} \cdot \eta_e} \quad (1)$$

$$\eta = \frac{T_s - T_o}{T_e - T_o} \quad (2)$$

where

T_s is the supply air temperature (downstream the static heat exchanger);

T_E is the exhaust air temperature (from the indoor environment);

T_O is the outdoor air temperature;

η_e is the national electrical efficiency.

Equation 1 shows that the energy necessary for the conditioning of 1 m³ of renewal air decreases when increasing of η and COP_{HP}. However, these two parameters have an opposite behaviour. In fact, when the supply air temperature at the outlet of the static heat exchanger is high (close to the design air temperature) it means a very efficient passive recovery but, on the contrary, a low evaporation temperature (because the exhaust air temperature of the static exchanger is low) and thus a lower heat pump performance.

To overcome this challenge it is important to manage the overall system by using a “smart controller” able to adjust all the physical and energetic parameters.

2.2 Laboratory

To test the performance of a heat recovery system, but more generally of a thermal machine, it is necessary to reproduce the conditions in which the system must operate. In details it should be reproduced, with an acceptable accuracy, the temperature and relative humidity of both indoor and outdoor environment.

To this purpose I designed and realized the Research and Development (R&D) Center at Califel S.r.l. factory – Vinchiaturo (CB) – (Attachment 1). The laboratory consists of two climatic rooms: one reproducing indoor conditions, the other reproducing outdoor environment. The laboratory test includes also the “engine room” in which the unit is placed, the “duct room” in which an insulated duct connects the external room to the aspiration side of the recovery unit and a “supervisory room” in which an acquisition system and a remote PC allow the supervision of the overall system.

2.2.1 Climatic rooms

The operative conditions for the experimental tests (temperature, relative humidity and, if necessary, concentrations of pollutants) on the system are reproduced in the two climatic rooms:

- a climatic room characterizing the External Environment (outside: E.E.);
- a climatic room characterizing the Confined Environment (inside: C.E.);

Machines and plants installed at the service of climatic rooms allow reproducing both the external and the confined environment conditions in the following range of operation:

- EXTERNAL ENVIRONMENT

- TEMPERATURE: -20 °C to +40 °C;
- RELATIVE HUMIDITY: 30 % to 95 %;

- CONFINED ENVIRONMENT

- TEMPERATURE: 20 ± 2 °C to 26 ± 2 °C;
- RELATIVE HUMIDITY: 30 % to 95 %.

The external environment room has an inner volume of 35 m³ insulated on the perimeter walls with a composite structure of Styrofoam (4.0 cm) and a sandwich panel with polyurethane foam. While, indoor environment has an inner volume of 40 m³ insulated with the same structure also on the roof.

The technological plants built for both external and confined environment have functional and performance features of different level. Indeed, those for external environment require higher accuracy than for confined ones because of the extreme temperature values we can reach.

Technological plants for external environment

In order to facilitate the understanding of the plant layout it is convenient to distinguish in:

- Winter conditions plant;
- Summer conditions plant.

It is important to note that some machine and plant components are used for both winter and summer conditions.

Reproducing winter conditions of different localities means ensuring to reach and keep temperature values even several degrees below zero. Differently, to have the typical winter relative humidity values it is necessary a small amount of water per kilogram of dry air.

It is crucial to keep in mind that the external conditions must be kept constant also during the working of the mechanical ventilation system. This imply the necessity of conditioning the same airflow rate suctioned by the recovery unit incoming from outdoor (in the real outdoor conditions). To clarify this concept suppose we want to make a winter test in summer season: the real outdoor air temperature is very high while our desired external environment air temperature is very low. To do that, we need to break down a huge heat load.

For this aim it was installed an air-condensing unit (Figure 2) equipped with a semi-hermetic compressor and a condenser with copper tubes and aluminium fins. Such machine is able to provide a cooling power up to 22 kW. This is the power necessary to reach the lowest desired temperature in the E.E. room (-20 °C) when the true outside temperature is 32 °C (simulated winter conditions during true summer season). In this way it is possible to reproduce winter condition in summer season by allowing tests deseasonalization.



Figure 2: air-condensing unit.



Figure 3: air-evaporator unit.

An air-evaporator unit (Figure 3) is connected to the condenser, with copper tubes (with internal grooves to facilitate the refrigerant evaporation) and aluminium fins (with wide pitch to avoid the frost formation).

The water content to adjust the relative humidity inside the external environment room is ensured by an immersed electrodes humidifier (Figure 4), retrofitted with an external proportional signal, capable of modulating the flow of steam from zero up to 8 kg/h controlling the air hygrometric content of the E.E..



Figure 4: immersed electrodes humidifier.

An insulated duct (3 cm of expanded elastomer) connects the E.E. to the suction side of the mechanical ventilation system (Figure 5).

In the external environment climatic room it is possible to reproduce also the typical summer conditions. To this aim the temperature is adjusted by a fan-coil (Figure 6) served by a boiler. A three-way solenoid valve allows modulating the water flow rate at the coil inlet based on a proportional controller.

A second way for summer temperature values reproduction is the usage of an inverter heat pump (Figure 7) with a split unit inside the external room.

The relative humidity control happens in the same way of the winter case.

By utilizing the fan-coil it is possible to balance an higher heating load than heat pump, also with critical (real) outdoor conditions. Moreover, its managing is easier than heat pump because it is necessary only to control a valve. Heat pump has its own controller. However, in the fall and spring it is energetically convenient conditioning with heat pump for both summer and winter conditions reproduction.



Figure 5: connection duct from external environment to the suction side of the system.



Figure 6: fan-coil reproducing summer temperature values.



Figure 7: heat pump external units.

Technological plants for confined environment

As above-mentioned, this climatic room does not require high-level control. The confined environment must only reproduce the typical winter and summer conditions. To do that we use an inverter heat pump (Figure 7) with an indoor split unit (Figure 8).



Figure 8: confined environment split unit.

To realize winter condition it is also possible employing a radiator regulated by a thermostatic valve (Figure 9). This heater is supplied by the same boiler that serves the fan-coil.



Figure 9: radiator with thermostatic valve.

The steam for relative humidity control is produced with an on/off immersed electrodes humidifier (Figure 10). The steam rate is set with the on board control panel at a value in the range 0.9 – 3.0 kg/h. The on/off cycles are managed through a probe installed in the confined environment.



Figure 10: immersed electrodes humidifier in the confined environment.

Figure 11 (a) shows the E.E. climatic room reproducing the external conditions. Within this environment, there is a compartment (Plenum Room, P.R.) to prevent interference between the fan section and suction line of the mechanical ventilation unit.

Figure 11 (b) shows the C.E. climate room reproducing all the previously listed conditions that characterize the Confined Environment.



Figure 11: climatic rooms a) E.E. climatic room with the plenum room, b) C.E. climatic room.

2.2.2 Supervisory system

As said above the laboratory (see Attachment 1) has also a room in which the SIVeMeC is located (“engine room”), a zone for the connection between external climatic room and the suction side of the machine (“duct room”) and “supervisory room” with the remote PC.

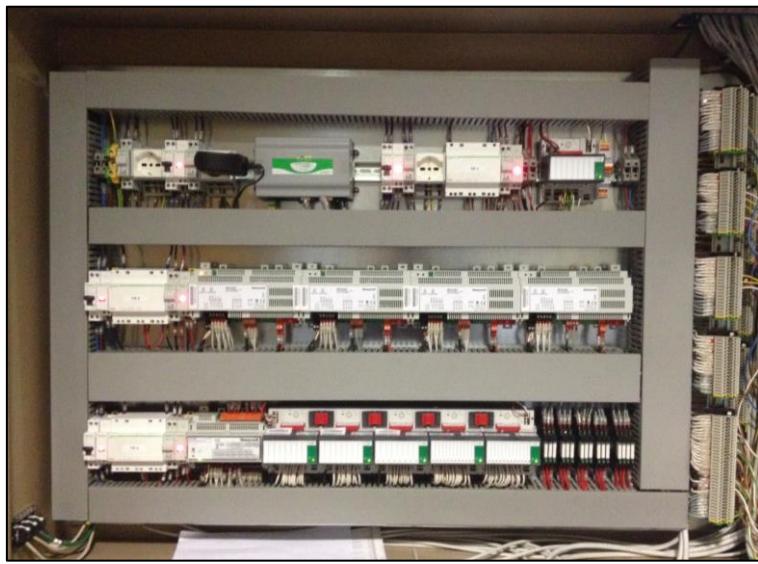


Figure 12: general control panel.

In the engine room there is the general control panel (Figure 12) where all the measuring, control, command and alarm points are connected.

All the hardware components are installed in this electric board, connecting all the probes and actuators at service of the laboratory plants. These modules have both analog and digital inputs, while the outputs, activated by suitable relays, are all digital. Probes and actuators are wired with shielded cables of appropriate section.

In the same room the electric power panel and the control panel are also wired to manage the refrigerator plant (Figure 13).

In the end, the “engine room” also includes the mechanical ventilation system installation and its connections with the two climatic rooms (Figure 14).



Figure 13: electric power panel and refrigeration plant panels.

The control of all the devices and component installed at the laboratory is made thanks to a set of probes and a supervisory system.

Table 1 shows the characteristics of the probes used for the measurements carried out during the experimental tests, while

Table 2 indicates where these probes are located and how many they are. Particularly, (see Attachment 1), each duct on each side of the recovery unit is equipped with a combined temperature – relative humidity probe and a differential pressure probe. Suction side and exhaust side are also equipped with a CO₂ probe to evaluate the carbon dioxide level of the E.E and C.E respectively and air speed probe for volumetric airflow rate evaluation. Another air speed probe is located in C.E. for velocity profile evaluation of the perforated supply air duct.



Figure 14: SIVeMeC installation and its connections with climatic rooms.

Table 1: probe used in the laboratory

<i>Probe</i>	<i>Accuracy</i>
Humidity and Temperature Room Sensors, range 5-95% -30-70°C, output 0-10Vcc	$\pm 0,2\text{K}$ at 25°C $\pm 3\%$ at 25°C 30-70% rh $\pm 5\%$ at 25°C 10-30% 70-90% rh $\pm 10\%$ at 25°C 5-10% 90-95% rh
Duct humidity-temperature sensor, range 5-95% - -30-70°C, output 0-10Vcc	$\pm 0,3\text{K}$ at 25°C $\pm 2,5\%$ at 20°C 10-95% rh
Duct CO ₂ transmitter, output. 0-10 Vcc	+ 2%
Duct speed sensor; range 0-20 m/s, output 0-10 Vcc	< 10%
Transmitter of differential pressure 0-500 Pa	$\pm 5\%$

Table 2: typology and number of probes for each zone.

ZONE	PROBE				
	Temperature	Relative Humidity	Air Speed	CO ₂	Differential Pressure
External Environment	N°1	N°1	N°1	-	-
Confined Environment	N°2	N°2	N°1	-	N°3
Engine Room	N°2	N°2	N°1	N°1	N°2
Duct Room	N°1	N°1		N°1	N°1

The values measured with the probes are the inputs of the supervisory system (COACH AX of CentraLine by Honeywell) to manage the technological plants and reproduce the boundary conditions for testing SIVeMeC prototype. The supervisor system has been completely designed and realised in the present PhD thesis.

COACH AX is a “one-tool” solution for engineering and managing all AX-based system components, which mainly are the ARENA AX supervisor, the integration controller HAWK and all I/O modules (see Figure 12). COACH AX allows building an application using the “wire sheet” format for creating the control logic and represents a tool to configure the graphical user interface for the ARENA AX supervisor and the HAWK controller as well as for the trend, schedule and alarm of all components [24].

The main features are:

- Supervisor engineering (ARENA AX) – is a BACnet certified web-based supervisor utilizing the NiagaraAX Framework for small to big size systems. It is used to supervise HVAC systems and non-HVAC systems (e.g. lighting, security, life safety) in a building or across multiple buildings. ARENAAX serves real time graphical information to standard web-browser clients and also provides server-level functions such as: centralized data logging, archiving, alarming, real time graphical displays, master scheduling, system-wide database management, and integration with enterprise software applications [25].
- Integration plant controller engineering (HAWK) – is a certified compact platform utilizing the NiagaraAX Framework. It combines integrated control, supervision, data logging, alarming, scheduling and network management functions with internet connectivity and web serving capabilities in a compact device. HAWK allows the control and management of external devices over the internet and present real-time information to users in web-based graphical views [26].
- User-specific templates – all elements (e.g. web graphics, logic, devices) within a station can be stored in a library and easily reused.
- Export tags – elements within HAWK, which should be exposed to several supervisors, can be tagged to avoid double engineering on controller and supervisor level. The tagged elements will be exposed automatically when the HAWK joins the supervisor.
- Input/output configuration modules – special views for fast configuration and start-up of LON I/O modules on the HAWK.

Each measured point is a virtual point in the ARENA AX and allows designing the control logic of the laboratory plants.

The first stage of supervisory system programming was the writing of the regulation logics of the plants at service of the climatic rooms. Particularly, it was needed to configure all the measuring points described above and wrote the algorithms for the control of desired thermo-hygrometric parameters. In Figure 15 an example is reported of the “wire sheet” to create the control logic for managing the technological plants.

In order to make user-friendly the system and simplify its handling, different user-interface pages were created. Figure 16 shows one of these pages in which all the command and measuring point of the plants are reported to reproduce the boundary conditions in both external and confined environment.

By clicking the SET button it is possible to change the design value of the correspondent parameter, while TREND button shows the real-time trend of the selected parameter. Other variables have no button on its side, in this case that parameter is a component STATUS.

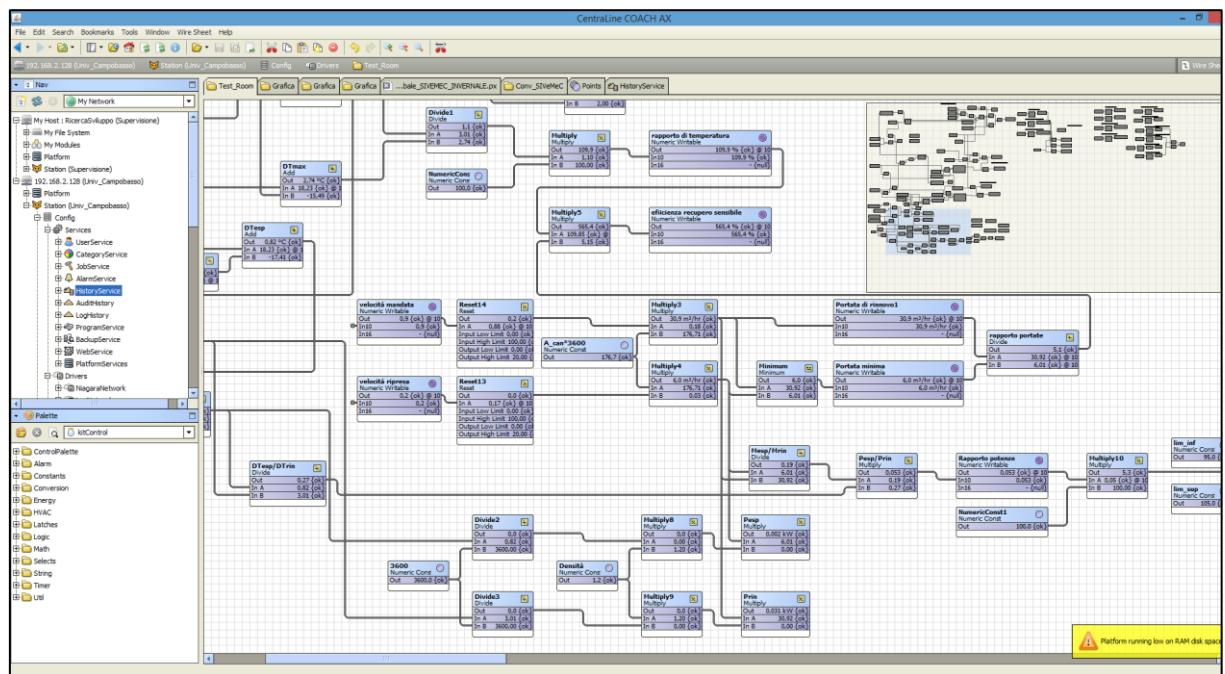


Figure 15: “wire sheet” programming for the control logic design.

The installed probes, in addition to return the desired measurements, also give an electrical output as 0 – 10 V and/or 4 – 20 mA. In this way, it is possible to design the suitable regulation logics of the actuators and those machines which have a modulating behaviour like the fan-coil solenoid valve and the immersed electrodes humidifier in the external environment climatic room.

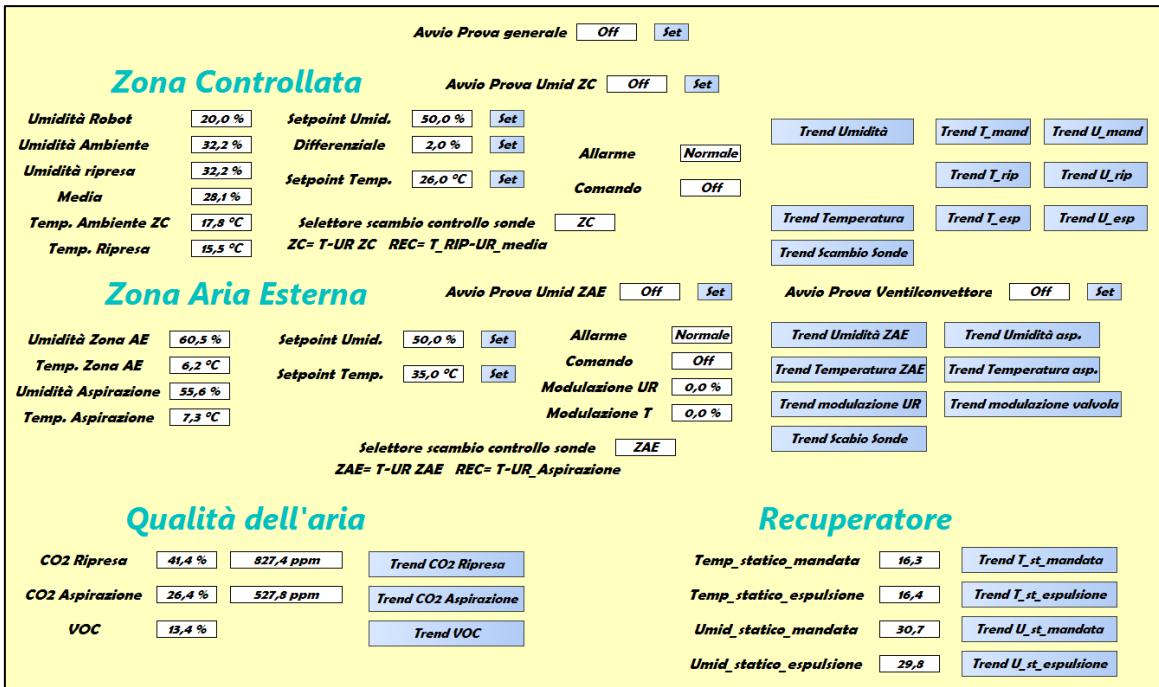


Figure 16: user-interface for the managing of the climatic rooms conditions.

Humidifier in Figure 4 can modulate the steam flow rate in the range 0 – 8 kg/h by using a proportional external signal. This regulation occurs through two probes. To better explain why we need two different sensors it is worth remembering that, when the mechanical ventilation system works, an airflow rate of the same magnitude of that suctioned by SIVeMeC is replaced from the outside into the external climatic room. However, in the early stage of testing procedure, it is not convenient turn on SIVeMeC, but it is better uniformly reproduce the desired conditions in the room and exploit its thermal inertia. Hence, in the start-up stage a T-RH (Temperature – Relative Humidity) probe located in the room is used, while, after this initial transitory, the humidifier control system is retrofitted by a T-RH installed on the suction duct (see Figure 5).

The actuator used to adjust the water flow rate supplied in the fan-coil (see Figure 6) handle a three-way valve with a proportional signal. This retrofit is done, for the same reason explained above, by the T-RH probe in the climatic room during the early stage and the suction duct T-RH probe during the second stage.

It is now clear that the control and regulation of the same device could be done with different probes depending on the test stage: generally, the probes located inside the rooms are used during transitory phase; the probe installed on ducts are used during regime phase. To do that automatically, virtual switch inside ARENA AX were programmed. The shift happens when all set conditions occur simultaneously. However, it is possible also to force it manually.

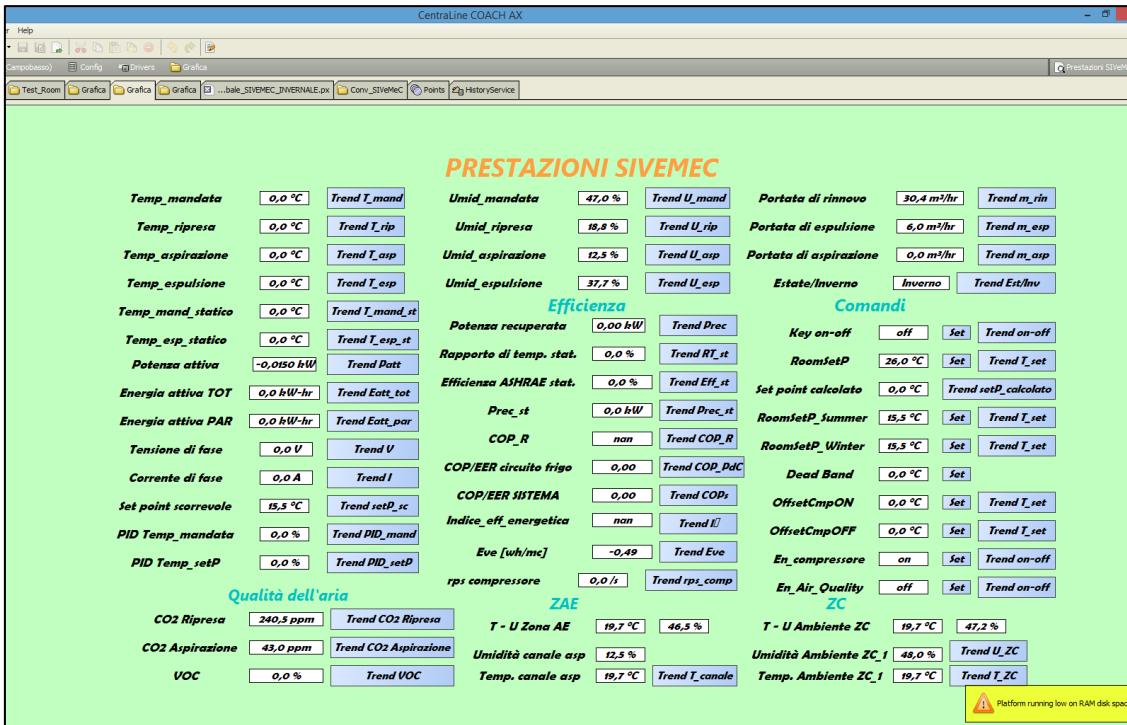


Figure 17: SIVeMeC user-interface for control and managing.

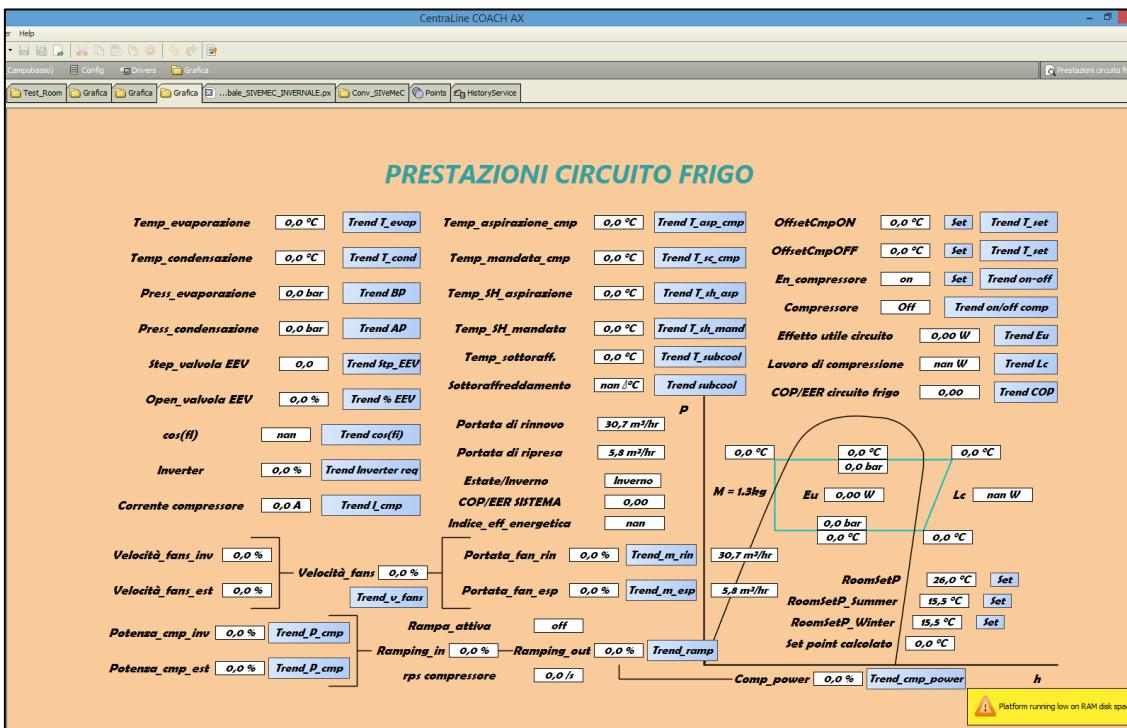


Figure 18: SIVeMeC heat-pump user-interface.

To acquire and read all the data measured from the field (laboratory probes and devices), it is possible to set what to show of each variable and the sampling time.

Others user-interface pages are developed for control and manage the principal component of SIVeMeC. To connect the prototype to the supervisory system, SIVeMeC is equipped with a BMS (Building Management System) serial board installed on its electronic board. The communication protocol is standard Modbus. In this way, all the variables for the correct working of the mechanical ventilation system (see section 2.4) could be shown into supervisory system platform. Particularly, only a set of variables of interest are selected and used for remote control of the unit. Figure 17 shows the principal variables of mechanical ventilation system during its operation. Particularly these parameters mainly describe the system focusing on the “air-side”. On the contrary, Figure 18 largely deals with heat pump and inverter operation.

To better visualize in an aggregate mode all the variables of interest and where they act in the laboratory, a graphic page is realized (Figure 19).

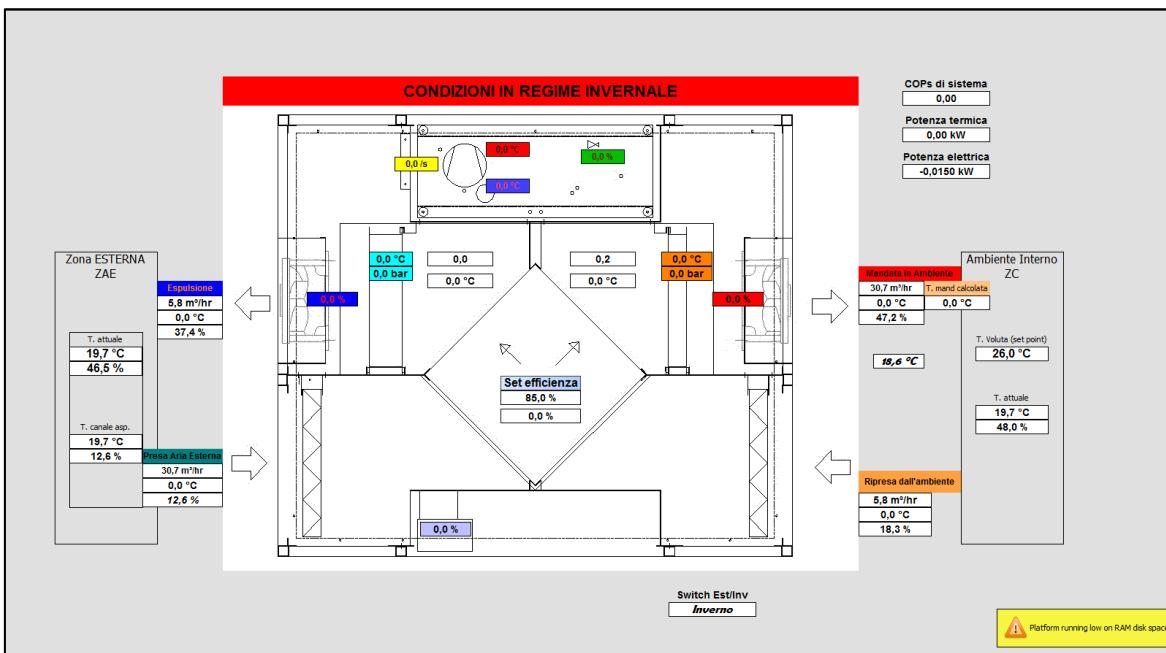


Figure 19: schematic laboratory user-interface page.

Figure 19 gives at first sight all the primary information about the mechanical ventilation system and the climatic rooms. Particularly, we can see the flow rate and thermo-hygrometric conditions of both supply and exhaust air, the compressor behaviour, the static heat exchanger efficiency, the thermo-hygrometric conditions in the climatic rooms and top-right COPs (Coefficient of Performance of the overall system: see section 3.2). Figure 19 represents a schematic user-interface page of the laboratory during winter test. An analogue page is developed for summer test.

As it easy to deduce, the supervisory system is a programmable platform and new control logics and other user-interface could be designed.

Finally, a “history page” (Figure 20) allows to select which variable we want to log. For each variable we could acquire 500 recordings, so the test length is a function of the sampling time (about 40 minutes with 5 s time step acquisition).

Point	Extension	History Name	Status	System Tags
Drivers/LinkNetwork/MOD_02_XPC10A1/points/ZAA_T_AMB	NumericInterval	Temperatura_Zona_Aria_Externa	(disabled)	
Drivers/LinkNetwork/MOD_05_XPC30500/points/ZAA_LND_AMB	NumericInterval	Umidità_Zona_Aria_Externa	(disabled)	
Drivers/Test_Room/CO2_aria_interna	NumericInterval	Sonda_CO2_Aspirazione_pmn	(disabled)	
Drivers/Test_Room/CO2_AE	NumericInterval	CO2_Aspirazione	(disabled)	
Drivers/Test_Room/LoopPoint1	Trend\$20modulazione\$20UR\$20ZAE	Trend_modulazione_UR_ZAE	(disabled)	
Drivers/Test_Room/Umidità\$0_Aspirazione	NumericInterval	U_asp	(disabled)	
Drivers/Test_Room/CO2_aria_rpresa	NumericInterval	Sonda_CO2_Ripresa_pmn	(disabled)	
Drivers/Test_Room/SM_CO2_RP	NumericInterval	CO2_Ripresa	(disabled)	
Drivers/Test_Room/Robot_ZC	UR_Point_ZC	Udcntral	(disabled)	
Drivers/Test_Room/Driver	NumericInterval	Umidità_media_Tzona_Controlloata	(disabled)	
Drivers/Test_Room/LoopPoint2	Trend\$20modulazione\$20valvola	Trend_modulazione_valvola	(disabled)	
Drivers/Test_Room/Temperatura_Aspirazione	NumericInterval	Temperatura_Aspirazione	(disabled)	
Drivers/Test_Room/NumericalVariable	NumericInterval	Umid_Zona_Controllata	(disabled)	
Drivers/Test_Room/Scambio_sonde_ventil	BooleanInterval	Scambio_Sonde_Zona_Aria_Externa	(disabled)	
Drivers/Test_Room/Scambio_sonde_zona_ctrl	BooleanInterval	Scambio_Sonde_Zona_Controllata	(disabled)	
Drivers/Test_Room/Umidità\$0_Ambiente_ZC	NumericInterval	Umidità_ZC	(disabled)	
Drivers/Test_Room/Temperatura_Ambiente_ZC	NumericInterval	Temperatura_Zona_Controllata	(disabled)	
Drivers/Test_Room/Temperatura_Canale_Ripresa	NumericInterval	Temperatura_Canale_Ripresa	(disabled)	
Drivers/Test_Room/Sonda_VOC	NumericInterval	Sonda_VOC	(disabled)	
Drivers/Test_Room/Temperatura\$0_mandata	NumericInterval	T_mand	(disabled)	
Drivers/Test_Room/Temperatura\$0rpresa	NumericInterval	T_rp	(disabled)	
Drivers/Test_Room/Temperatura\$0espulsione	NumericInterval	T_esp	(disabled)	
Drivers/Test_Room/Umidità\$020mandata	NumericInterval	U_mand	(disabled)	
Drivers/Test_Room/Umidità\$020rpresa	NumericInterval	U_rp	(disabled)	
Drivers/Test_Room/Umidità\$020espulsione	NumericInterval	U_esp	(disabled)	
Drivers/Test_Room/Temperatura\$0_mandata\$0_aspirazione	NumericInterval	T_st_esp	(disabled)	
Drivers/Test_Room/Umidità\$0_mandata\$0_aspirazione	UR_Pt	Udcntral	(disabled)	
Drivers/Test_Room/Temperatura\$0_mandata	NumericInterval	T_st_mand	(disabled)	
Drivers/Test_Room/Umidità\$0_mandata	UR_pt	U_st_mand	(disabled)	
Drivers/Test_Room/export\$20d\$20temperatura	rapporto\$20d\$20temperatura	rapporto_d_temperatura	(disabled)	
Drivers/Test_Room/Portata\$20d\$20innovo	m_rinnovo	m_rinnovo	(disabled)	
Drivers/Test_Room/Portata\$20d\$20rpresa	m_rpresa	m_rpresa	(disabled)	
Drivers/Test_Room/efficienza\$20recupero\$20sensibile	effi_rec_sens	effi_rec_sens	(disabled)	
Drivers/Test_Room/Pesp\$2Prin	Pesp\$2Prin	Pesp\$2Prin	(disabled)	
Drivers/Test_Room/Trend\$20Prin	Trend\$20Prin	Trend\$20Prin	(disabled)	
Drivers/Test_Room/Allarme\$20Trend\$20Prin	Allarme_Trend\$20Prin	Allarme_Trend\$20Prin	(disabled)	
Drivers/Test_Room/DTesp\$20DPrin	DTesp\$20DPrin	DTesp\$20DPrin	(disabled)	
Drivers/Test_Room/Trend\$20DPrin	Trend\$20DPrin	Trend\$20DPrin	(disabled)	
Drivers/Test_Room/Pesp	Trend\$20Pesp	Trend\$20Pesp	(disabled)	
Drivers/Test_Room/Temperatura_Robot_ZC	Temperature_Robot	Temperatura_Robot	(disabled)	
Drivers/Test_Room/Portata\$20d\$20aspirazione	m_aspiratore	m_asp	(disabled)	
Drivers/Test_Room/CO2_aria_interna	CO2\$20zona\$20aria\$20esterna	CO2_ZAE	(disabled)	
Drivers/Test_Room/CO2_AE	NumericInterval	CO2_Aspirazione	(disabled)	
Drivers/Test_Room/CO2_aria_rpresa1	CO2\$20zona\$20controlloata	CO2_Ripresa	(disabled)	
Drivers/Test_Room/SM_CO2_RP1	NumericInterval	CO2_Ripresa	(disabled)	
Drivers/Conv_SIveMeC/Temperatura_aspirazione	T_esp_UPC	T_esp_UPC	(disabled)	
Drivers/Conv_SIveMeC/Temperatura_mandata	T_mand_UPC	T_mand_UPC	(disabled)	
Drivers/Conv_SIveMeC/Temperatura_esterna	T_est_UPC	T_est_UPC	(disabled)	
Drivers/Conv_SIveMeC/Temperatura_rpresa	T_rp_UPC	T_rp_UPC	(disabled)	
Drivers/Conv_SIveMeC/Temperatura_statica	Tc04d\$20air	Alpha_misurazione	(disabled)	

Platform running low on RAM disk space

Figure 20: history page to select the variables of interest for historicization.

2.3 System configuration

SIVeMeC is the prototype with a static heat exchanger with the integration of a reverse-cycle heat pump machine (Figure 21) whose main components are:

- a countercurrent heat exchanger able to ensure the highest possible static recovery;
- fans which speed is modulated by an inverter;
- a compressor for modulation of the thermal load;
- an expansion valve for superheating management;
- a programmable control unit to control the summer and winter configurations.

All SIVeMeC components and their sizing occurred through a suitable energy analysis. In fact, it is not sufficient rely on single device manufacturer performance declaration. The real efficiency must be validated when each component is connect to the others in the final system. To choose the appropriate size and test its suitability for the high performance of the system several energy parameters were evaluated (see section 3.2).

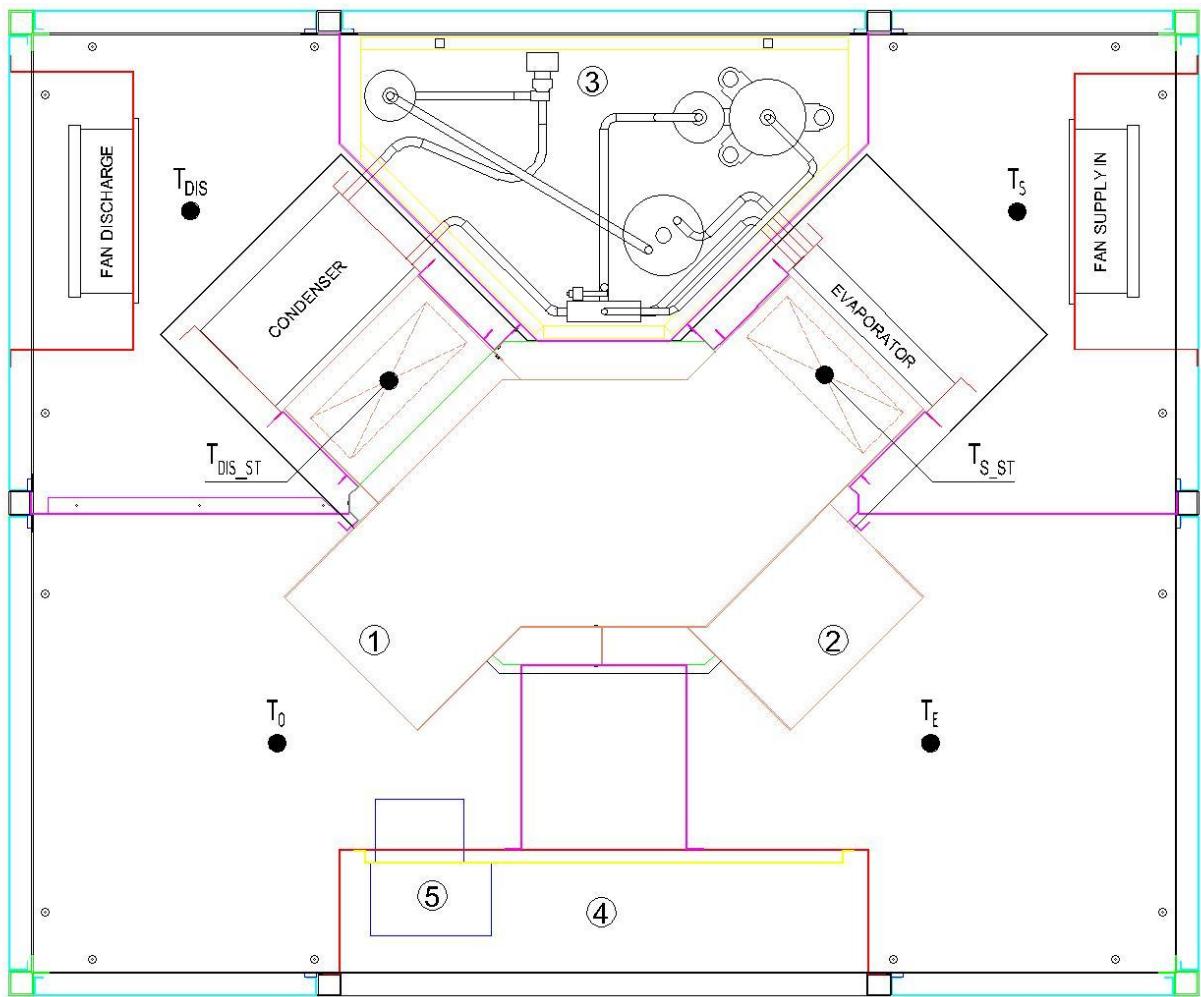


Figure 21: SIVeMeC prototype.

1. by-pass duct for supply airstream
2. by-pass duct for exhaust airstream
3. Heat pump compartment
4. Electrical panel
5. Inverter to modulate the rotary compressor

T_S supply air temperature probe

T_o outdoor air temperature probe

T_{S_ST} supply air temperature probe downstream the static recovery

T_E exhaust air temperature probe

T_{DIS} discharge air temperature probe

T_{DIS_ST} discharge air temperature probe downstream the static recovery

The evaluation of the system operation could be made thanks to a set of probes installed on board as in Figure 20 (for a full description of their function see section 2.4).

Below the description of the main system components are reported.

2.3.1 Countercurrent heat exchanger

The passive recovery in SIVeMeC application occurs through a countercurrent fixed plate exchanger. This type of heat exchanger is realized by coupling plates which alternatively allow passing exhaust and renewal air. The two flow rates are physically separated so as to avoid streams contamination. Plate material is aluminium, for its characteristic of corrosion resistance, lightness, easy workability, non-flammability and durability. The tightness over the time is pretty good and can be defected only in the case of a crack due to a high differential pressure. Another good quality is the absence of moving parts (static exchanger). Thus, energy consumption is only related to fans power absorption in order to win the pressure drop. Furthermore, its flexibility permits a wide adaptability to many applications.

The heat recovery unit selected for SIVeMeC prototype is REK +39 of Recutech s.r.o. Company. The permissible temperature of the transferred air ranges from -40 °C to +80 °C, with relative humidity from 0 % to 100 %. Table 3 shows technical data according to EN 308 in both summer and winter conditions. It is possible to note that the efficiency of the recuperator is very high and is in the range 78 – 85 %.

Table 3: REK +39 technical data according to EN 308: 1997.

Description	Summer	Winter	Description	Summer	Winter
Standard airflow volume - supply in [m ³ /h]	498,1	517,2	Standard airflow volume - exhaust in [m ³ /h]	483,9	483,9
Airflow volume - supply in [m ³ /h]	500	500	Airflow volume - exhaust in [m ³ /h]	500	500
Temperature - supply in [°C]	16	6	Temperature - exhaust in [°C]	25	25
Rel. humidity - supply in [%]	72	76	Rel. humidity - exhaust in [%]	28	28
Abs. humidity - supply in [g/kg]	8,2	4,4	Abs. humidity - exhaust in [g/kg]	5,6	5,6
Face air velocity - supply in [m/s]	1,9	1,9	Face air velocity - exhaust in [m/s]	1,9	1,9
Mass flow - supply in [kg/h]	599,8	622,7	Mass flow - exhaust in [kg/h]	582,6	582,6
Enthalpy - supply in [kJ/kg]	37	17,2	Enthalpy - exhaust in [kJ/kg]	39,3	39,3
Temp. condens. - supply in [°C]	11	2,1	Temp. condens. - exhaust in [°C]	5,2	5,2
Standard pressure drop - supply [Pa]	180,1	190,7	Standard pressure drop - exhaust [Pa]	172	172,4
Pressure drop - supply [Pa]	181,1	181,1	Pressure drop - exhaust [Pa]	181,1	181,1
Airflow volume - supply out [m ³ /h]	512,3	526,8	Airflow volume - exhaust out [m ³ /h]	487,2	472,9
Temperature - supply out [°C]	23,1	21	Temperature - exhaust out [°C]	17,4	8,9
Rel. humidity - supply out [%]	46,2	28,6	Rel. humidity - exhaust out [%]	44,7	78,1
Abs. humidity - supply out [g/kg]	8,2	4,4	Abs. humidity - exhaust out [g/kg]	5,6	5,6
Face air velocity - supply out [m/s]	1,9	2	Face air velocity - exhaust out [m/s]	1,8	1,8
Enthalpy - supply out [kJ/kg]	44,3	32,3	Enthalpy - exhaust out [kJ/kg]	31,5	22,9
Exchange efficiency - supply [%]	79,2	78,7	Exchange efficiency - exhaust [%]	84,7	85
Exchange efficiency dry - supply [%]	79,2	78,7	Exchange efficiency dry - exhaust [%]	84,7	85
Heat recovery - supply [kW]	1,2	2,6	Heat recovery - exhaust [kW]	-1,3	-2,7
Condensation - supply [l/h]	0	0	Condensation - exhaust [l/h]	0	0

The recuperator works with a balanced ratio of inward and outward airflow, but can operate also up to a maximum unbalanced ratio of +/- 50%. This allows regulating the performance of the heat static recovery by adjusting its efficiency. In fact, a decrease of the renewal flow rate by keeping fixed the exhaust one produces an increase of heat recovery efficiency (according to EN 308). It is easy to understand that in this case the heat capacity of the exhaust air is higher than the renewal one because of a greater air mass flow rate (the specific heat of the two air flows can be considered the same). On the contrary, we can obtain a reduction of heat recovery efficiency by increasing renewal air with a fixed exhaust air (for more detail see section 3.2).

Figure 22 shows the psychrometric transformation, taking place inside the heat exchanger, of exhaust and renewal air on Mollier diagram according to the nominal data provided by the manufacturer. The transformations are related to the winter conditions reported in Table 3.

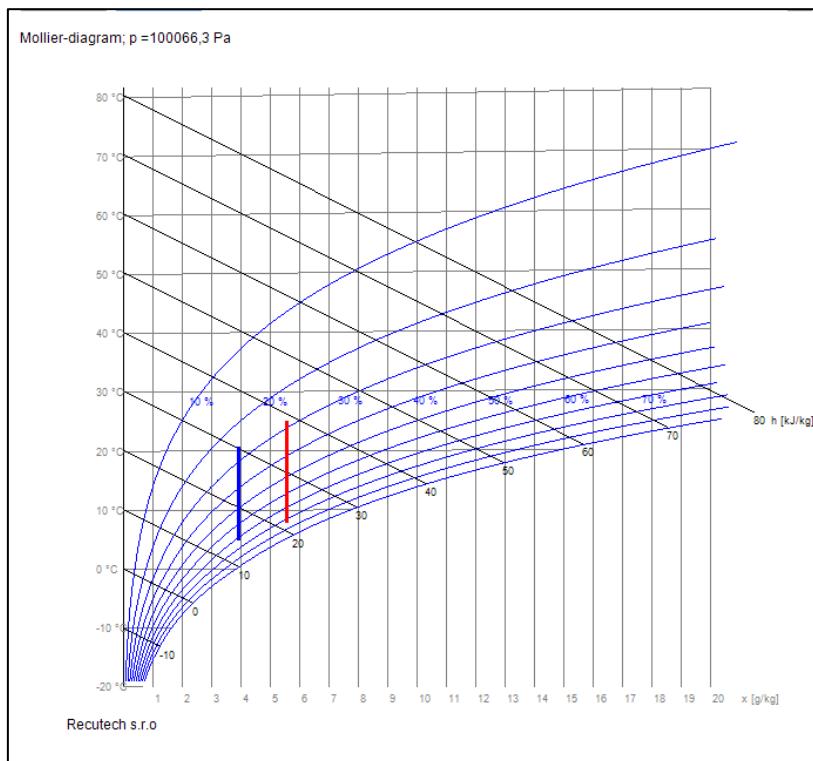


Figure 22: psychrometric transformation of renewal air (blue line) and exhaust air (red line) on Mollier diagram.

The pitch between the fins is 2.2 mm and their thickness is 0.1 mm. This configuration produces the pressure drop (Figure 23) through the heat exchanger as the flow rate varies.

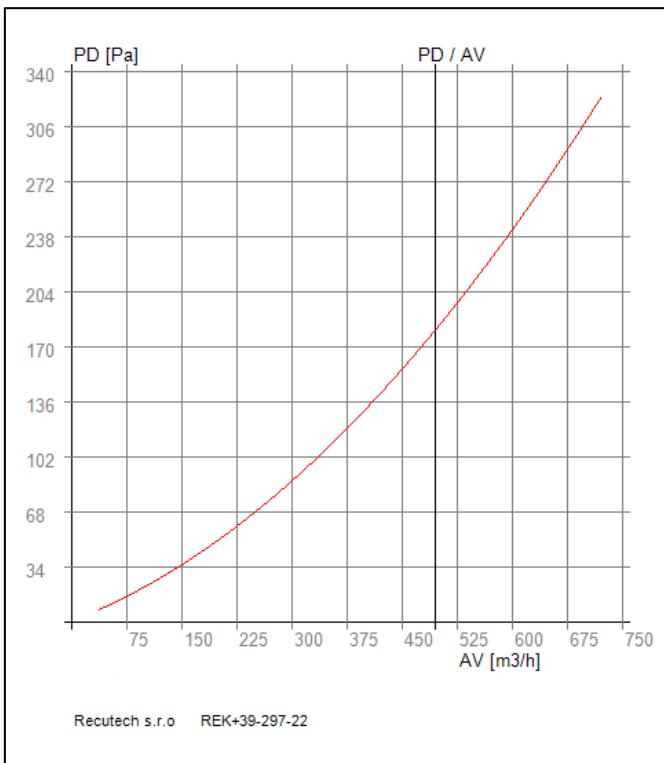


Figure 23: pressure drop (PD) through the heat exchanger in function of volumetric airflow rate AV; the line PD/AV is that for 500 m³/h.

2.3.2 Heat pump

SIVeMeC is a passive plus active heat recovery system because it acts as an air-to-air heat pump downstream a possible heat recovery through a counter flow recuperator.

Heat pumps working is based on a reversed thermodynamic cycle that is a cycle in which heat is “pumped” from an environment at lower temperature to another environment with a higher temperature. This cycle, in accordance with the second low of thermodynamic, requires energy expenditure (work). When a reverse cycle is used for remove heat from an environment, we have a refrigeration cycle. Instead, when the reverse cycle contributes to heat up an environment (with respect to the surrounding environment), we have a heat pump cycle.

Schematically, the main components of a vapor-compression heat pump are: compressor, evaporator, condenser and expansion valve. Particularly, the heat pump cycle designed for SIVeMeC unit has a BLDC compressor, two direct expansion coils (evaporator and condenser), an electronic expansion valve, a liquid receiver, a dehydrator filter, a sight glass and a 4-way inversion cycle valve.

Figure 24 shows the single-wire scheme of the heat pump cycle (principal components) installed in the mechanical ventilation system and the set of probes, very important for both cycle monitoring and heat pump driving.

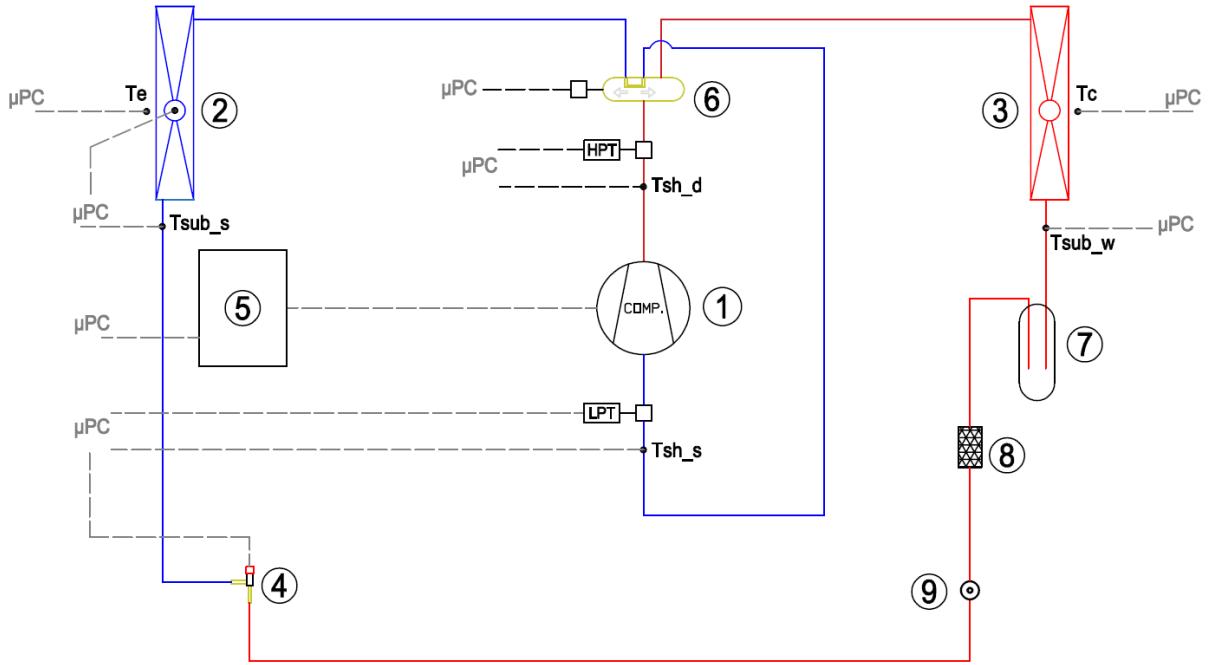


Figure 24: heat pump cycle in winter arrangement. 1) BLDC compressor; 2) evaporator; 3) condenser; 4) electronic expansion valve; 5) inverter for compressor driving; 6) 4-way valve; 7) liquid receiver; 8) dehydrator filter; 9) sight glass; Tsh_s suction superheat temperature; HTP high pressure transducer; LPT low pressure transducer; Te evaporation temperature; Tc condensation temperature; Tsh_d delivery superheat temperature; Tsub_w subcooling temperature in winter arrangement; Tsub_s subcooling temperature in summer arrangement; μPC control panel.

The BLDC compressor is the core of the system. It is a Toshiba single rotary vane compressor (DA75F0F – 11UA model). Rotary vane compressors consist of a cylindrical casing, suction and discharge opening and a rotor, which is placed eccentrically with respect to the casing. Compression occurs by refrigerant flowing into the chamber where, due to eccentric rotation, there is a reduction in the desired volume.

The compressor was chosen based on the residual heat load downstream a very efficient passive recovery. Table 4 reports the supply air temperature (T_{s_st}) and discharge air temperature (T_{dis_st}) values, downstream the static heat recovery (see Figure 21). These temperatures are calculated by assuming a passive heat recovery temperature ratio η of 80 % and a balanced airflow rate both on exhaust \dot{V}_E and supply \dot{V}_s side of 500 m³/h ($\dot{V}_E = \dot{V}_s$). The heat recovered by the static exchanger is evaluated on the basis of T_{s_st} and T_o (Outdoor air Temperature), while the residual heat load in charge of the heat pump is calculated by considering an indoor design temperature of 20 °C (T_{set}). The specific heat at constant pressure of the air (in the temperature range of interest) could be assumed equals to 1.2 kJ/kgK.

As shown in Table 4 the heat pump must provide a very small heating load (P_{HP_set}), even at very low outdoor temperature. In fact, first row of Table 4 shows that at -20 °C the heat pump

should provide only 1.34 kW downstream the static recovery. The following rows are about typical outdoor air conditions of winter, fall and spring seasons.

Table 4: passive heat recovery P_{R_ST} , residual heat load P_{HP_set} , supply air temperature (T_{S_ST}) and discharge air temperature (T_{DIS_ST}) downstream the static recuperator, as a function of outdoor temperature T_o for fixed values of exhaust air temperature (T_E), temperature ratio η and both exhaust (\dot{V}_E) and supply (\dot{V}_S) airflow rate.

T_E	T_o	η	T_{S_ST}	T_{DIS_ST}	\dot{V}_S	\dot{V}_E	P_{R_ST}	P_{HP_set}
[°C]	[°C]	[%]	[°C]	[°C]	[m ³ /h]	[m ³ /h]	[kW]	[kW]
20	-20	80	12	-12	500	500	5,36	1,34
20	-5	80	15	0	500	500	3,35	0,84
20	-4	80	15,2	0,8	500	500	3,22	0,80
20	-3	80	15,4	1,6	500	500	3,08	0,77
20	-2	80	15,6	2,4	500	500	2,95	0,74
20	-1	80	15,8	3,2	500	500	2,82	0,70
20	0	80	16	4	500	500	2,68	0,67
20	1	80	16,2	4,8	500	500	2,55	0,64
20	2	80	16,4	5,6	500	500	2,41	0,60
20	3	80	16,6	6,4	500	500	2,28	0,57
20	4	80	16,8	7,2	500	500	2,15	0,54
20	5	80	17	8	500	500	2,01	0,50
20	6	80	17,2	8,8	500	500	1,88	0,47
20	7	80	17,4	9,6	500	500	1,74	0,44
20	8	80	17,6	10,4	500	500	1,61	0,40
20	9	80	17,8	11,2	500	500	1,47	0,37
20	10	80	18	12	500	500	1,34	0,34
20	11	80	18,2	12,8	500	500	1,21	0,30
20	12	80	18,4	13,6	500	500	1,07	0,27
20	13	80	18,6	14,4	500	500	0,94	0,23
20	14	80	18,8	15,2	500	500	0,80	0,20
20	15	80	19	16	500	500	0,67	0,17
20	16	80	19,2	16,8	500	500	0,54	0,13
20	17	80	19,4	17,6	500	500	0,40	0,10
20	18	80	19,6	18,4	500	500	0,27	0,07
20	19	80	19,8	19,2	500	500	0,13	0,03

This is why (to overcome the problem of this heating load variability) a BLDC compressor, which can modulate up to 30% its load, was selected. Moreover, it is worth remembering that BLDC compressor has its best performance exactly when working at modulated load.

Table 5 shows the compressor technical data provided by the manufacturer, while Figure 25 shows the performance curves at both nominal load (60 rps) and partial load (30 rps). The full load is obtained at 100 rps. The compressor provides 2.18 kW in nominal conditions, so by

taking into account P_{HP_set} in Table 4 it is easy to note that the heat pumps works almost always at partial load, with higher performance and lower power absorption.

Table 5: technical data of DA75F0F – 11UA compressor.

Application	Inverter air conditioner	Rating conditions Revolution : 60 s ⁻¹ Condensing temp. : 42.3 °C Evaporating temp. : 2.8 °C Return gas temp. : 12.8 °C Liquid temp. : 34.3 °C Ambient temp. : 35.0 °C
Compressor type	Hermetic rotary compressor	
Refrigerant	R-410A	
Painting color	Black	
Mass (included oil)	5.4 kg	
Number of cylinder	1	
Displacement	7.49 cm ³ /rev	
Diameter of cylinder	ø37.5 mm	
Height of cylinder	16.0 mm	
Orbital radius	4.50 mm	
Oil charged	Ester oil VG74 320 ml	
Suction tube	ø12.9 mm(I.D.)	
Discharge tube	ø8.1 mm(I.D.)	
Wiring connection terminal	Receptacle #250(flag type)	
Motor data		
Motor type	DC Brushless Motor	
Number of pole	4	
Starting type	DC Inverter Starting	
Revolution range	30~100 s ⁻¹	
Nominal output	630 W	
Winding resistance	(20°C) 1.95 Ω	
Classification of insulation	E	

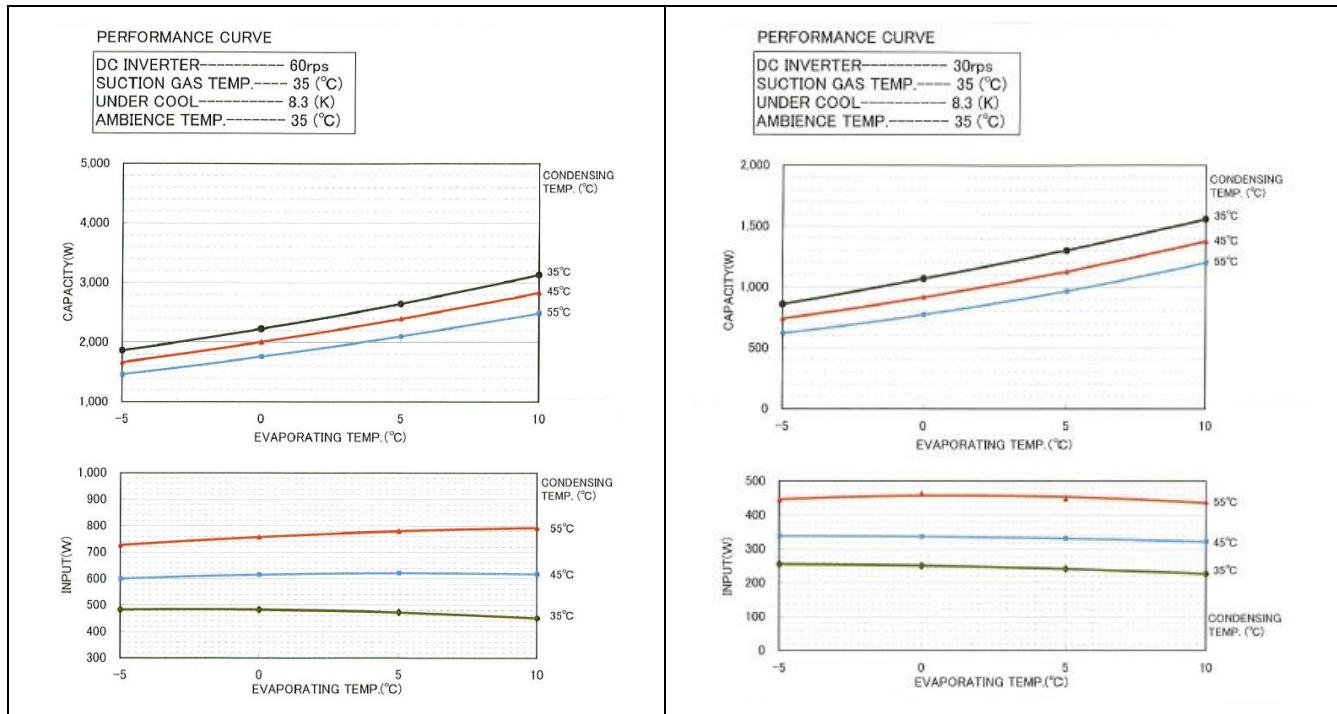


Figure 25: compressor performance curves at 60 rps (right) and 30 rps (left).

To drive the compressor a Carel S.p.A. Power+ PSDO inverter is installed. Power+ is a drive designed to control compressors with sensorless-brushless permanent magnet motors

(BLDC/BLAC) or asynchronous induction motors. It is fitted for panel installation. A CAREL pCO controller or any other controller device via RS485 serial connection using the Modbus protocol in master mode manages configuration and programming, as well as the run/stop controls and speed reference. The main characteristics are:

- operation at ambient temperatures from -20 to 60 °C;
- can be installed in residential and industrial environments;
- connection via serial network to Master programmable controller;
- network address can be configured by setting the dipswitches inside to the drive;
- can control various types of compressors;
- STO (Safe Torque Off) safety digital input for system protection;
- programmable acceleration curve to adapt to the required specifications when starting compressor;
- high switching frequency to limit motor noise;
- detailed information on drive status via numerous read-only variables;
- protection functions for the drive (short-circuit, overcurrent, earth fault, overvoltage and under-voltage on the bus, over-temperature, overload), motor/compressor (overload and limitation of current delivered, locked rotor) and system (communication failure).

The inverter is managed by the electronic board μPC installed in the control panel of the machine (Figure 26).

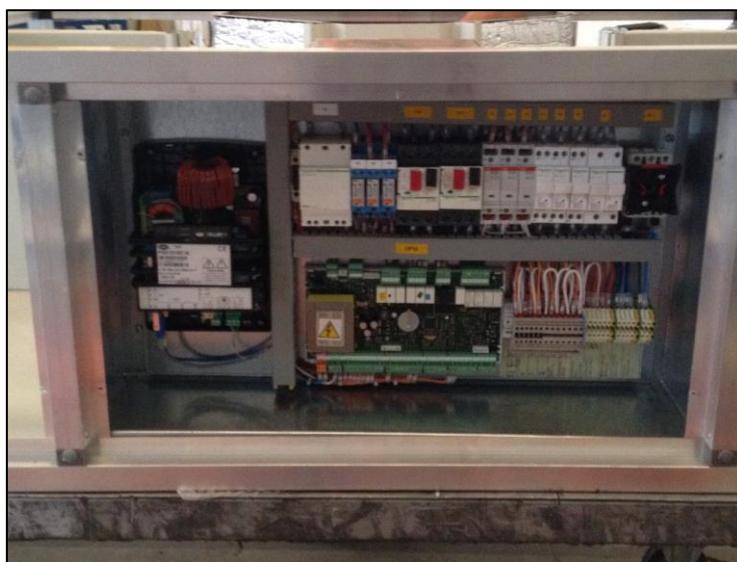


Figure 26: control panel of SIVeMeC. Power+ is on the right; μPC is in the low side.

The regulation happens through a special module inside the control logics of the electronic board. Its management occurs through several variables that give a real time monitoring of the

system. Particularly, it is possible to regulate the acceleration and deceleration ramp and all those parameter that ensure the compressor working inside the envelope curves.

An electronic expansion valve operates the control of the refrigerant flow evolving inside the heat pump cycle. The selected valve is an E²V smart (Figure 27) of Carel S.p.A. (05SSF50 model). Modulation of refrigerant flow guarantees a wide operating range, due to the combination of the fixed opening and the moving element with a travel of 15 mm driven by stepper motor. The E²V smart is able to guarantee high reliability, and ensures correct operation with fluid flow in both directions. This simplifies the refrigerant circuit for reverse-cycle operation. One of the main advantages in using expansion valve technology is the energy saving achievable due to improved efficiency of the refrigeration cycle. The possibility to operate at low condensing pressures, as well as precise superheat control, allow considerable energy saving.



Figure 27: E²V smart electronic expansion valve.

Its sizing depends on the heat capacity and evaporation and condensation temperature. As for the Power+ drive, also E²V smart is controlled by a dedicate module inside the control logics of the system. To regulate the degree of openness it is possible to operate according to three principal modes:

1. as a function of superheat temperature on the compressor suction side;
2. as a function of superheat temperature on the compressor delivery side;
3. as a function of exhaust compressor temperature.

In this way an equal-percentage regulation of the refrigerant flow, with high precision level, especially at low rates, is ensured.

The BLDC compressor control, through Power+ drive, includes some functions performed in combination with the electronic expansion valve, such as to control the maximum current limit, of low compression ratio and of low and high pressure difference.

Two direct expansion coils (evaporator and condenser) must dispose the heat loads produced by the reverse cycle. These heat exchangers are with copper tubes and aluminium fins and are designed in order to exchange also very small heat loads. In fact, due to a very high efficiency heat recovery, the residual heat load provided by the heat pump is small, also with relatively high flow rate (e.g 500 m³/h and 5 °C of temperature difference requires only 0,84 kW of heat load). To these purpose the designed direct expansion coils are that of Figure 28 and Figure 29:

- 6 rows exchanger for evaporator in winter case and condenser in summer (this is the most stressed coil);
- 2 rows exchanger for winter condenser and summer evaporator (to provide also very small heat loads).

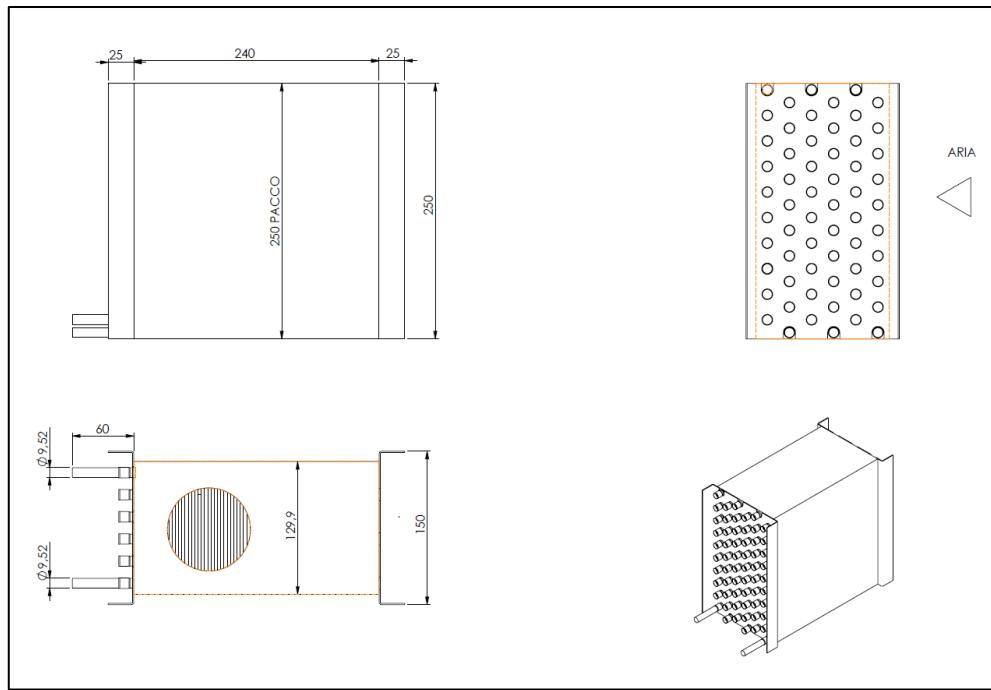


Figure 28: six rows expansion coil.

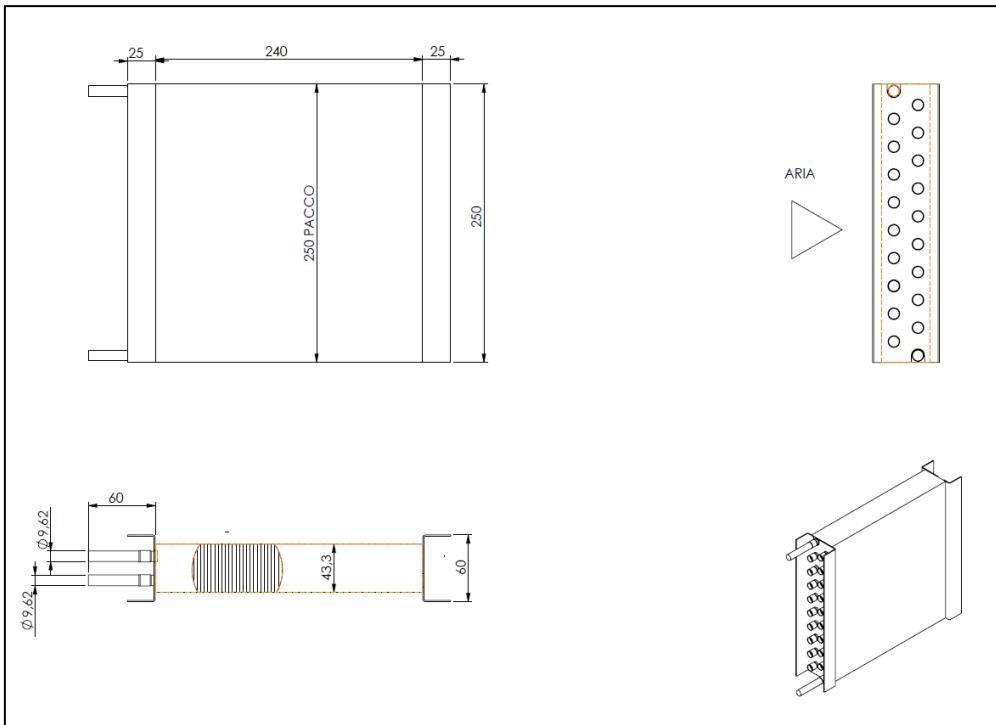


Figure 29: two rows expansion coil.

Table 6 show the technical data of both expansion coils. Particularly, it could be seen the thermal power P_T supplied by the two rows expansion coil working as condenser, at different condensation temperature T_C , and by the six rows expansion coil working as evaporator, at different evaporation temperature T_E .

Table 6: technical data of direct expansion coils.

Two rows expansion coil working as condenser						
T_C	\dot{V}	T_{AI}	T_{AO}	ΔT	\dot{m}_R	P_T
°C	m^3/h	°C	°C	°C	kg/h	kW
30	500	15	19,1	4,1	1,1	0,7
35	500	15	21,2	6,2	2,1	1,06
40	500	15	23,3	8,3	3,4	1,42
45	500	15	25,5	10,5	4,9	1,8
50	500	15	27,7	12,7	6,6	2,18
Six rows expansion coil working as evaporator						
T_E	\dot{V}	T_{AI}	T_{AO}	ΔT	\dot{m}_R	P_T
°C	m^3/h	°C	°C	°C	kg/h	kW
-1	500	2	0,6	1,4	6	0,5
-2	500	2	-0,1	2,1	9	0,38
-3	500	2	-0,9	2,9	12	0,51
-4	500	2	-1,6	3,6	15	0,64
-5	500	2	-2,3	4,3	18	0,76

2.3.3 Electronic fans

Other fundamental components, for the managing of the thermal load variability, are the electronic fans. Indeed, by modulating the airflow rate we can have a direct effect on the heating load disposed by the expansion coils.

Moreover, the static heat recovery efficiency (EN 308) η varies with the renewal airflow rate \dot{V}_S and exhaust airflow rate \dot{V}_E :

- η decreases when $\dot{V}_S > \dot{V}_E$
- η increases when $\dot{V}_S < \dot{V}_E$

Hence, the suitable sizing of the electronic fans guarantees high performance, due to a very low power absorption, and allows an extreme flexibility of working.

The selected fans (*RadiCal* model of *ebm-papst*) are centrifugal ones with backward-curved blades (Figure 30). Their features include both noise minimisation and low energy consumption. The impeller is made of fibreglass-reinforced plastic, enabling an aerodynamically optimised shape that cuts the noise level in half and reduces power requirements significantly. The model selected is K3G 190 with M3G055 – BI electric motor.

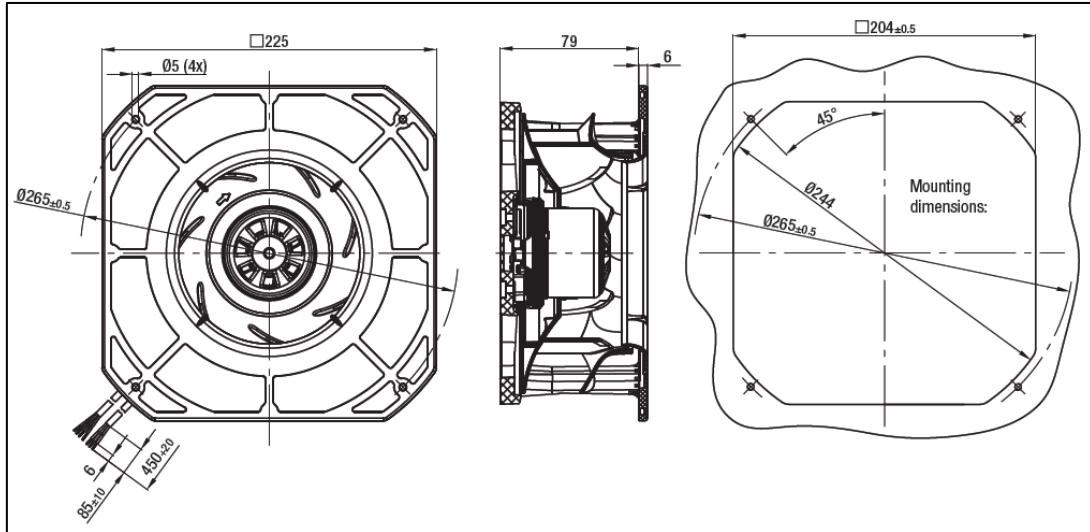


Figure 30: technical drawing of electronic fan

Figure 31 shows the characteristic curves (pressure drops vs airflow rate) of the fan. Particularly, the reference curve of this model is ‘C’. The small box at the top right reports the number of revolution per minutes of the electric motor in four working points. The selected electronic fans elaborate a volumetric airflow rate of 500 m³/h and supply a static pressure of 300 Pa by absorbing only 83 W.

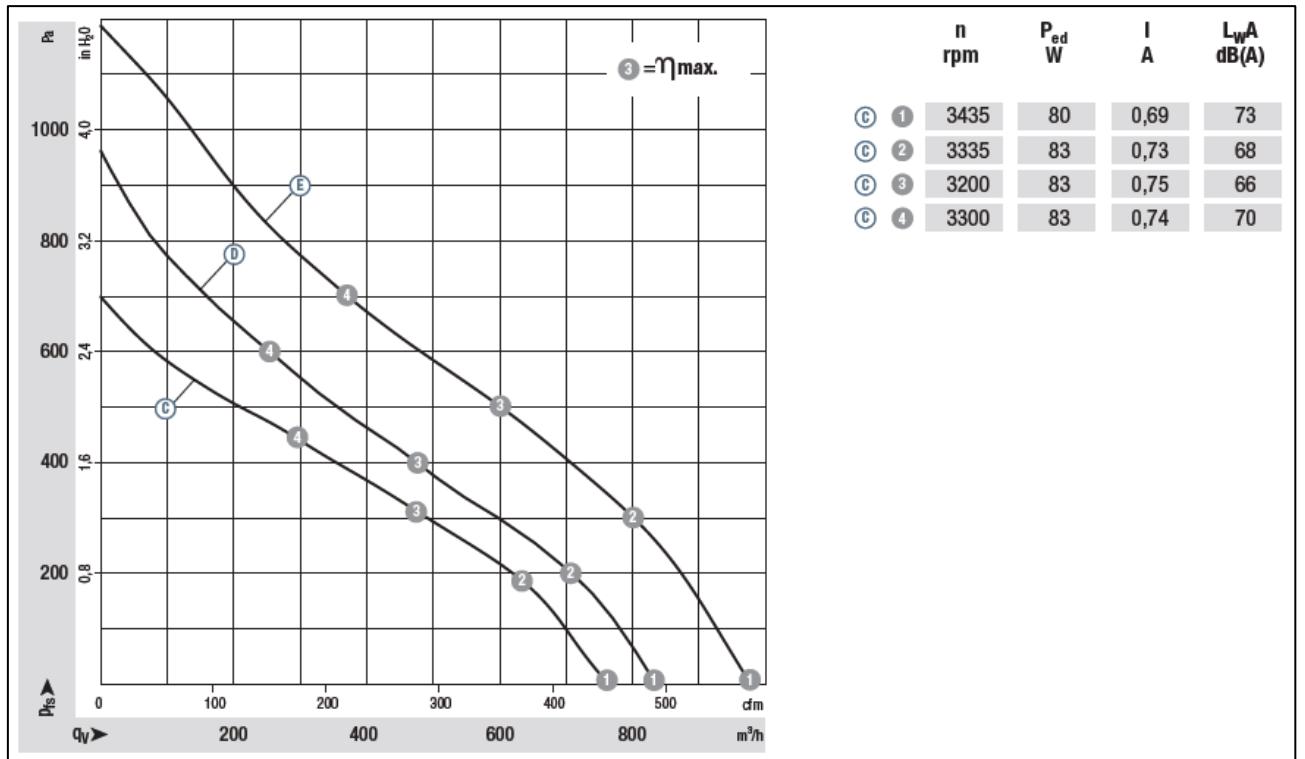


Figure 31: fan characteristic curves.

An integrated inverter modulates high efficiency electronic fan speed: the electronic motor has a permanent magnet combined with the current in the stator winding, so it generates a torque on the rotor without dissipating energy by increasing the efficiency.

2.3.4 By-pass ducts

By-pass ducts, located above the heat exchanger for passive recovery, are those components that allow the recovery unit to work according different arrangements. Figure 32 shows one of the two by-pass ducts equipped with micro-switches to control the damper opening.

As aforementioned, the system can work according different operating modes thanks to appropriate subdivisions within it determining the presence of by-pass ducts, thus to by-pass the passive heat recovery. In fact, when the by-pass dampers are closed, the system can operate with the combined action of the heat recovery and the heat pump or with the static recovery only (heat pump turned off). When the dampers are open, the air bypasses the static heat exchanger (through the by-pass ducts) and the system operates with the heat pump alone. In this latter case, if the heat pump is turned off, the system works in freeheating or freecooling mode.

Currently, dampers inside the by-pass ducts work only in open-close way. However, it is also possible to regulate the opening percentage of dampers to split each airflow stream (renewal

and exhaust) in two parts: one passing through the static heat exchanger and the other that bypasses it. As a result we have, downstream the countercurrent recuperator, the mixing of two streams whose temperature is proportional to the magnitude of the airflows previously splitted.



Figure 32: by-pass duct above the heat exchanger.

2.3.5 SIVeMeC

In the previous paragraphs all SIVeMeC components are described. Summarizing, it is equipped with a static heat exchanger and a heat pump whose main components are:

- ❖ high efficiency countercurrent heat exchanger for high passive recovery;
- ❖ Heat pump with BLDC (Brushless Direct Current) rotary compressor, an electronic expansion valve and two direct expansion coils;
- ❖ High efficiency electronic fans with speed modulated by an integrated inverter;
- ❖ Two by-pass ducts to work according different arrangements.

The refrigeration circuit is also equipped with a liquid receiver of 1.3 l (storage of liquid during load modulation), a 4-way valve for reversing of cycle, a filter dryer, liquid indicator and two direct expansion coils. The direct expansion coil, which is the evaporator during heating mode, is greater than the direct expansion coil employed as condenser. In fact, both in heating and cooling mode this is the most stressed exchanger. The proper functioning of the compressor is ensured by two pressure transducers, which prevent to operate outside the envelope curves. The mass flow rate of refrigerant is regulated by the electronic expansion valve. A temperature probe on the suction line of the compressor regulates the greater or smaller opening of the valve.

In this way the adjustment is made by the suction super-heat value (set at 10 °C). However, the regulation of the expansion valve is also possible through the superheat on the supply of the compressor through a probe installed on the discharge line. With this kind of regulation it is possible to limit the discharge temperature of the compressor, at the inlet of the condenser.

By assembling all these components, we obtained a very compact and smart solution for the ventilation of Confined Environment. This system allows managing the environment ventilation autonomously. This fact is crucial because in this way such a system is easily embeddable in existing air-conditioning plants solutions. Moreover, it could be used in new air-conditioning plants by assuring the sizing reduction of thermal machines that must provide the thermal energy.

Its extreme compactness facilitate its installation also in very small technical compartments. The three dimension of the prototype are Length – Width – Depth of 1200 – 1450 – 450 mm. Especially is noteworthy the depth dimension by considering also the presence of two by-pass ducts.

2.4 SIVeMeC control system

The managing of the mechanical ventilation system prototype takes place through a suitable software programmed and installed on the electronic board of the unit. This software allows managing of:

- Static heat exchanger;
- BLDC compressor;
- Electronic expansion valve;
- Electronic fans;
- By-pass ducts.

Moreover, it allows the communication with typical BMS (Building Management System) for the supervision of HVAC plants.

The programmable controller is μ PC medium of Carel S.p.A, with 230 V power supply. It is equipped with:

- N° 10 digital inputs;
- N° 12 digital relay outputs;
- N° 4 analog outputs 0-1 Vdc (of which two allows also 0-5 Vdc output and Pulse Width Modulation PMW output);

- N° 12 analog inputs which allows the connection with NTC 10 kΩ and 50 kΩ probes, 0-1 V active probes (with 24 Vdc power supply).

Among these twelve analog inputs:

- N° 2 allow also the installation of 4-20 mA active probes;
- N° 4 allow also the installation ratiometric pressure sensors (with 5 Vdc power supply).

The μPC controller is completely programmable with a proprietary software: *1Tool®*.

By using this software it was possible to design all the working logics of the system, including user interface masks visible on the remote panel PGD1 (Figure 33). PGD1 is a graphic 132x64-pixel display terminal with six buttons.



Figure 33: remote panel PGD1.

The application created with *1Tool®* takes the name of “solution”. Within this lives the “strategy” (or the strategies). Each project is organized in programming pages with blocks. The simplest blocks are called “atoms” and consist of elementary functions, such as the acquisition of an analog or digital input, a logical or mathematical operation, etc. The composition of several atoms form a “macroblock” (which is a true and own project). Finally, a macroblock with graphic masks takes the name of “module”. The software is equipped with libraries containing atoms, macroblocks and modules. However, it is possible to create user libraries containing macroblocks and modules designed precisely for the specific application.

The program is executed according to the page design order and, within each page, to the numbering shown above the blocks.

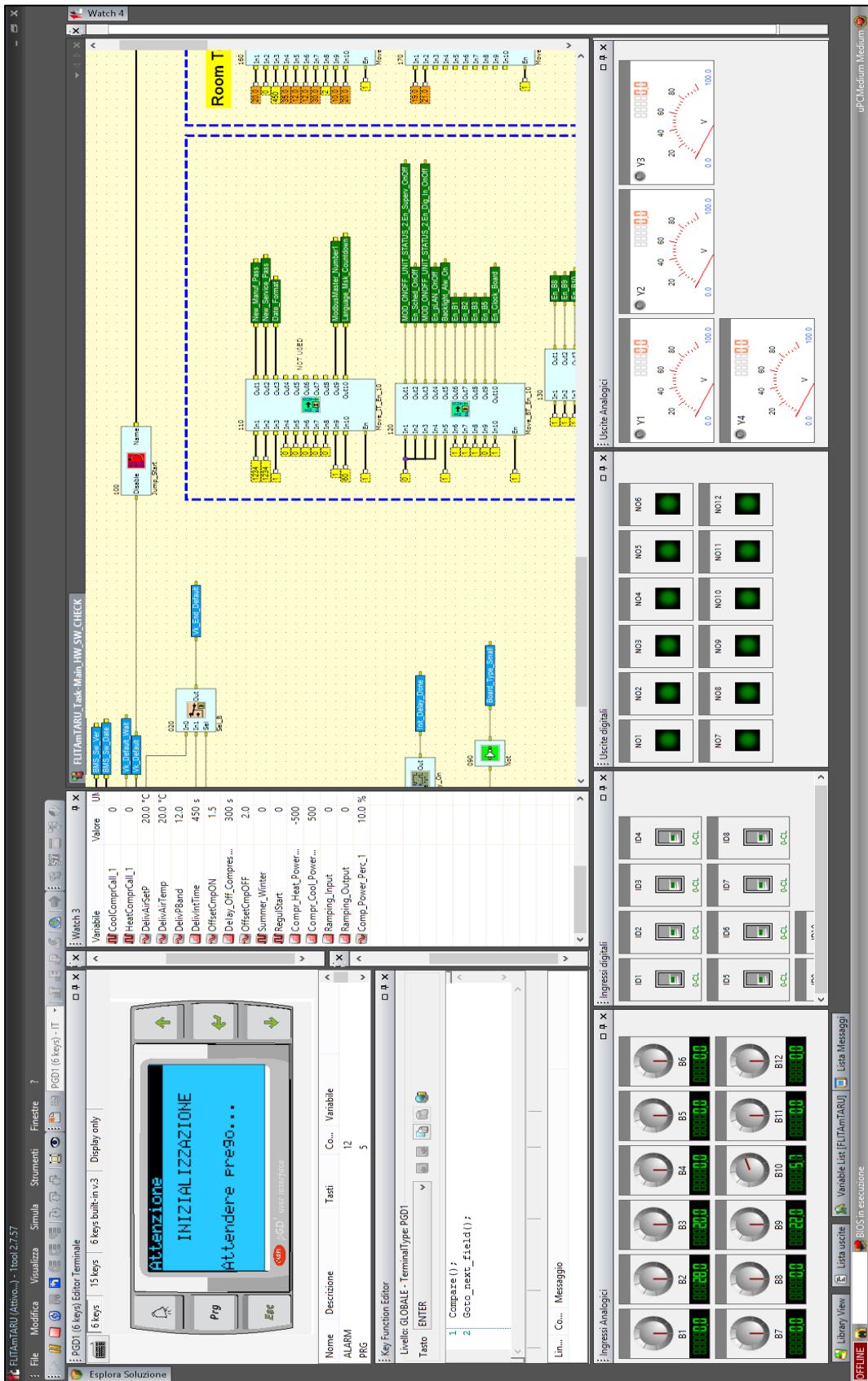


Figure 34: integrated simulator 1Tool®

A fundamental feature of the programming software is that it allows the working simulation of the application before installing it on the controller.

In Figure 34 this potential is highlighted. The analog and digital inputs are at the bottom of the page. By acting on these buttons it is possible simulating the variations of the quantities that, under normal operating conditions, would be detected from field. It is also possible to scroll through the PGD1 masks, modify and insert all the editable parameters. Given the complexity of the application, it is also possible to simulate the single page of the solution.

All SIVeMeC operating logics have been developed in the present PhD thesis with 1Tool® software. The result is a flexible software application, which allows, by editing and setting the desired parameters and the operation configuration that is more suitable for the application of interest.

Figure 35 reports a screen showing the pages nested in the project that constitute the strategy and a part of the block programming for the acquisition of the analog inputs.

The design of the application is complex and consists of numerous programming pages, some of which have macroblocks and modules inside them, which, as above mentioned, constitute additional programming pages. Hereinafter, a synthetic description will be reported to demonstrate the SIVeMeC working potential.

Four NTC probes were connected to the µPC (see Figure 21) to measure the temperatures of the supply, return, suction and expulsion airflows. Moreover, probes were installed for measuring the compressor suction and discharge temperatures (see Figure 24), the condensation and evaporation pressure, parameters on the basis of which the compressor work zones are defined. To ensure compressor operation within the permissible limits, a continuous comparison was made during the tests between the measured values and the compressor envelope curves. The optimal conditions regulation and the operating regime are managed by the inverter, which communicates through the Modbus serial communication protocol with the board. This connection allows the modulation of the load according to the climatic conditions detected from field and the maintenance of the operating parameters within the acceptable pressure and temperature limits.

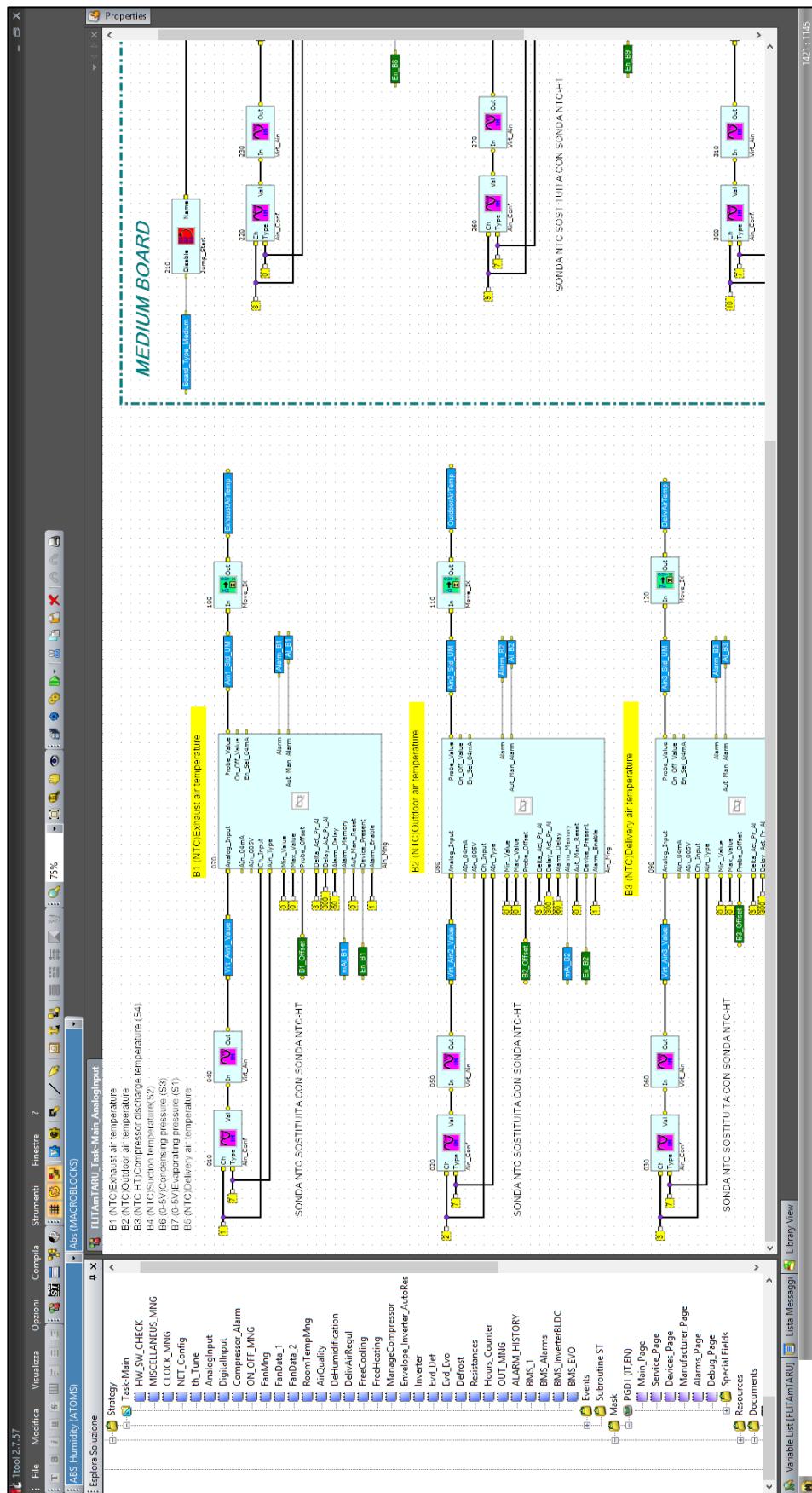


Figure 35: 1Tool® screen. The sidebar contains the pages that make up the strategy. The "AnalogInput" page is open in the main window.

A fundamental part of the application, as important as that of the compressor modulation, is the management of the electronic fans, driven by two analog outputs 0-10 Vdc. Flow adjustment can be controlled and managed in different ways. This solution has been designed to allow a wide margin of decision to the designer setting the active reference probe according to the type of application.

2.4.1 Temperature control

The inlet air temperature (T_s) adjustment occurs through two PID controllers (Figure 36). However, the time derivative coefficient is set to zero because it could make the controller too sensitive and lead to undesired fluctuations around the set point. So both controllers become PI ones. The first PI uses as set point the desired internal air temperature (T_{SET}) and as controlled input the value read by the exhaust temperature probe T_E . In this way, a calculated set point (T_{SET_calc}) is obtained. It deviates from the true set point the more the value of the exhaust temperature T_E deviates from the set point itself. This is powerful because it represents an in advance prediction, through a revision model (the first PI controller), capable to control the main components of the unit. In fact, the calculated set point is itself the input of a second PI controller which manages, through suitable conversions, the setting of the compressor ramp and of the rotation speed of both supply and exhaust air fans having as controlled input the inlet air temperature T_s .

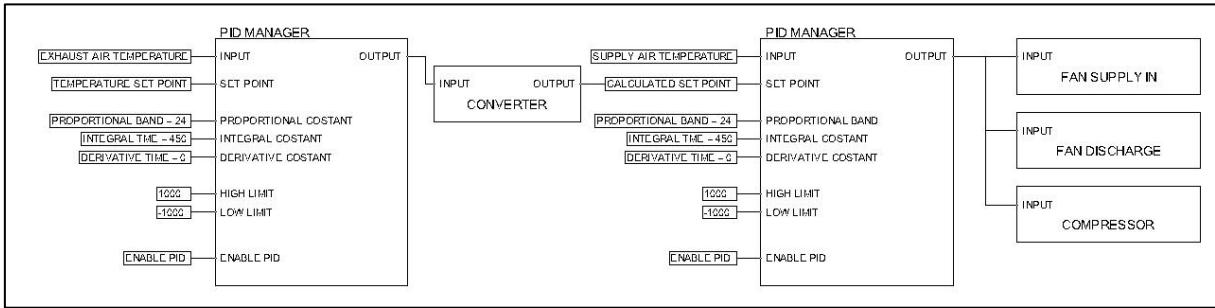


Figure 36: two PID controllers in series to manage compressor, delivery air fan and exhaust air fan.

2.4.2 Heat pump control

Heat pump control happens mainly through the compressor call and regulation. As above described, the compressor ramp is managed thanks to the second PID of Figure 37. Instead, Figure 37 shows the 1Tool® part of program related to the compressor turning-on. The compressor is powered on when the supply air temperature is below the calculated set point, diminished of an offset (1.5 °C hysteresis), for a time longer than 30 seconds. On the other hand

the compressor is disabled when the supply air temperature exceeds – for a time greater than 5 minutes – the set point plus an appropriate offset (sets to 2.0 °C and editable by user):

compressor on when $T_S < T_{(SET_calc)} - 1.5$

compressor off when $T_S > T_{(SET_calc)} + 2.0$

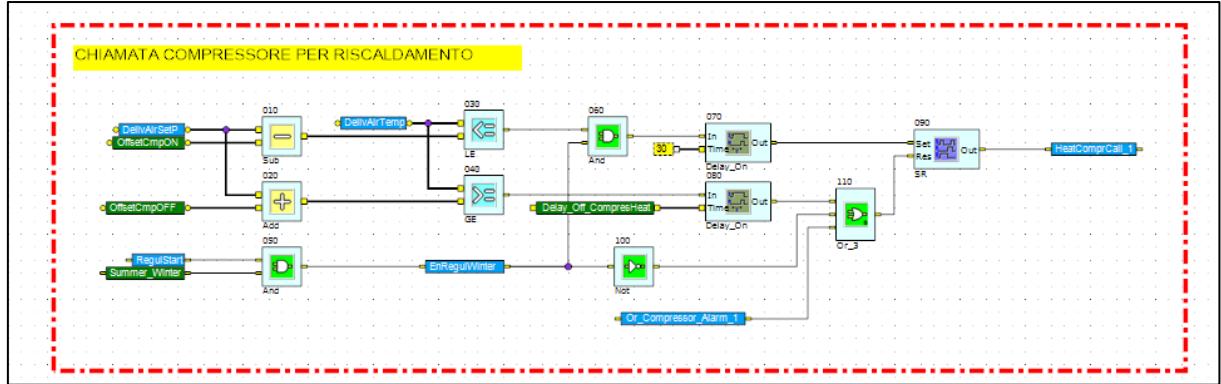


Figure 37: 1Tool® part of program for compressor calling.

2.4.3 Ventilation control

It is now clear that the regulation of the main SIVeMeC components is due to the PI controllers of Figure 36. Thus, the ventilation rate is chosen based on the set-point of indoor air temperature T_{SET} . Generally, this regulation allows ensuring also an appropriate replacement of carbon dioxide level. However, also a regulation depending on carbon dioxide level is possible.

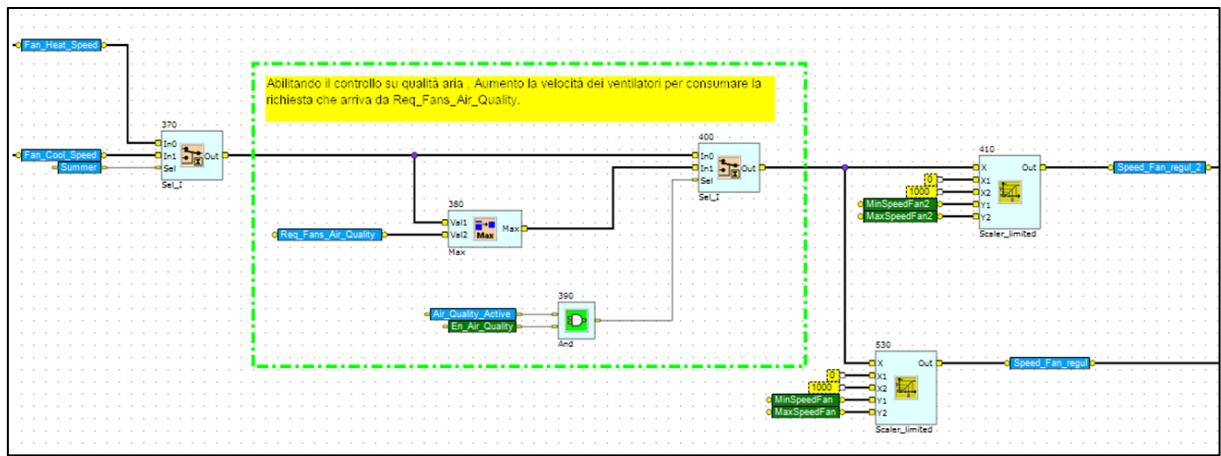


Figure 38: 1Tool® program part for fans speed regulation by using CO₂ probe as reference.

Figure 38 shows how it is possible to enable the regulation of fans speed by using the CO₂ probe as reference. In the green rectangle, the green variable (called En_Air_Quality) allows the managing of fans by using the carbon dioxide level of the indoor air rather than the exhaust air

temperature. In this way, when the CO₂ level falls below the minimum acceptable value the supply airflow rate is increased.

When the CO₂ content to be supplied is low it is possible to modulate the system by using as reference the temperature measured on the supply side or exhaust side (respectively T_S and T_E in Figure 21) depending on the main goal to be achieved. When fine-tuned neutral air is the main requirement the supply air temperature T_S is chosen as reference. Otherwise, when the indoor air temperature has to be controlled, it is possible to choose the exhaust air temperature T_E as reference. In the latter case, the system also provides to reduce a part of the heating load, because a higher supply air temperature (with respect to the neutral point) allows to partially covering the heat losses for transmission, in addition to the ventilation load. It is evident that in both cases the controlled temperature is the supply one.

3 LABORATORY EXPERIMENTAL TESTS

3.1 Experimental procedures

In order to test the SIVeMeC system, first, the two climatic rooms were stabilized at the different fixed temperatures (depending on trial conditions, see below), by setting the values of set points. Once reached the desired condition in each room, SIVeMeC is activated in fan only mode (compressor off) until the desired condition on the suction side is reached. Measurement variables acquisition begins when the boundary conditions were stable for at least 5 minutes. The data were acquired during the following 40 minutes with a sampling time (of each variable) of 5 seconds. Tests were carried out in heating mode by reproducing the following E.E. air temperature on the suction side -5 °C; 0 °C; 5 °C and 10 °C; and setting to 20 °C the C.E. air temperature controlled by SIVeMeC.

This value T_E is measured on the exhaust air side (T_E), while the renewal air temperature (T_0) was measured on the suction side of the recovery unit.

3.2 Energy parameters

As one of the goals of this work was to compare SIVeMeC with other existing commercial systems and with other experimental units found in literature, it is particularly interesting to carry out the trials in winter season simulated conditions, when the other systems offer the best performance. To this aim, a number of physical and performance parameter was measured. Mardiana and Riffat [27] presented a review on physical and performance parameters. Roulet et al. [28] defined a Coefficient Of Performance (COP) of heat recovery systems as the ratio of recovered heat and used electrical power. Also Han et al. [29] identified the COP for heat recovery systems in the same way. Riffat and Gillot [21], in the study of their novel mechanical ventilation heat recovery heat pump system, considered two kinds of COP: heating/cooling COP as the ratio of heating/cooling output to the electrical power; system COP as the ratio of heating/cooling output to the total electrical power used by the system (not only compressor). In the same way, we have considered the coefficient of performance of the overall system (COPs), which is the most suitable and important parameter, including both fans and compressor electrical demand:

$$COP_s = \frac{P_R}{P_e} \quad (3)$$

where:

P_R is the heat output of the system;

P_e is the electric power absorbed by the system.

Regarding to P_e , an energy meter is installed to measure the global electric demand of the whole recovery system. It is important to focus on this aspect because many studies and many data sheets given by constructors do not take into account the electric absorption due to ventilation (fans).

The heat P_R required by the renewal air, even in absence of any recovery system, can be evaluated as:

$$P_R = \dot{V}_S \rho_a c_{p_a} (T_S - T_O) \quad (4)$$

where

\dot{V}_S is the flow rate of the renewal air;

ρ_a is the density of dry air;

c_{p_a} is the air specific heat at constant pressure;

T_S is the supply air temperature;

T_O is the outdoor air temperature (measured on the suction side of the unit).

P_R includes only sensible heat because of the presence of an aluminium fixed-plate heat exchanger, so no moisture transfer is permitted. However, when the temperature of the exhaust air falls below its dew point also latent heat of condensation is transferred to the cold supply airstream. Another monitored parameter, frequently given by constructors too, is the temperature ratio η according to EN 308:1997. Almost always this parameter is known in literature as efficiency or as effectiveness, when referring to the ratio of supply air mass flow rate and the minimum mass flow rate (between the supply air and exhaust air) [30]. A set of NTC temperature probes are installed around the heat exchanger (T_E , T_O , T_{DIS_ST} and T_{S_ST} in Figure 21) to evaluate the heat transfer between supply and exhaust airstream. Particularly, we refer to the supply airstream to calculate η , measuring the temperature on the supply side (T_{S_ST}) with the NTC probe placed between the heat exchanger and the direct expansion coils.

$$\eta = \frac{T_O - T_{S_ST}}{T_O - T_E} \quad (5)$$

where:

T_{S_ST} is the supply air temperature downstream the static recovery;

T_E is the exhausted air temperature measured on the recovery side.

Another important topic is to determine the effects of the presence (by-pass dumpers closed) or absence (by-pass dumpers open) of the heat exchanger. D'Este et al. [23] already confirmed the effectiveness of the heat exchanger and proved that the heat exchanger should be always present, except only for some warm climates in which the heat needs are very low. Gustafsson et al. [31] compared three innovative HVAC systems (A, B and C) to a reference case (D), for renovation through dynamic simulation.

Case A considers a mechanical ventilation system with heat recovery and micro heat pump; case B consists of an exhaust ventilation system with exhaust air-to-water heat pump and ventilation radiators; case C configuration is an exhaust ventilation system with outdoor air-to-water heat pump and ventilation radiators; case D contemplates an exhaust ventilation system with outdoor air-to-water heat pump and panel radiator. They found that system A allows the largest reduction in energy consumption, by reducing the heating demand significantly compared to the other systems.

Anyway, to the same purpose, a comparison with an exhaust air heat pump (EAHP), studied by Fracastoro and Serraino [20], was made. It is well known that the presence of a static heat exchanger worsens the COP_{HP} (heat pump COP) compared to the case in which this component is absent [22]. In fact, downstream the static recovery, the temperature of the air entering the evaporator is lower than that which would occur when this component is not present. In the same way, temperature of air entering the condenser is higher for systems with a heat exchanger before the direct expansion coils. Indeed, thermodynamic recovery unit uses the air drawn from the internal environment to evaporate the refrigerant fluid and outdoor air to condense it. Higher evaporation and lower condensation temperatures lead to higher COP_{HP} . However, it is important to refer to the airside and not to the refrigerant side to correctly evaluate the COPs. Thus, a higher temperature of the air that enters the condenser implies a higher supply air temperature and therefore a greater heat output in heating mode [22].

Another parameter that can be used to determine the efficiency of this unit is the energy efficiency index i_{eR} . This energy parameter allows establishing if it is necessary to modulate or even by-pass the static recovery. The index refers to the only passive recovery since the heat pump is seen as an additional generator. Practically, the supply air temperature downstream the static heat exchanger must never be greater than the temperature useful to the purpose. In this

application this “useful temperature” is just T_{SET} , therefore the energy efficiency index can be so defined:

$$i_{\varepsilon R} = \frac{\eta}{\eta_{Umax}} = \frac{T_0 - T_{S_ST}}{T_0 - T_{SET}} \quad (6)$$

where η_{Umax} is the maximum useful temperature ratio:

$$\eta_{Umax} = \frac{T_0 - T_{SET}}{T_0 - T_E} \quad (7)$$

In fact, if $T_{S_ST} > T_{SET}$ it is possible to have, in a shorter or longer time, a loss in the control of the confined environment (C.E.) temperature and the heat pump should provide a thermal output of opposite sign (in opposition to the heat recovery). This is due to a supply air temperature greater in heating mode (or lesser in cooling mode) than the design supply temperature. The C.E. temperature becomes higher (or lower) than the set point, making the system inefficient. In fact, even if a temperature slightly higher than the set point is acceptable for the comfort of the occupants, it still represents an energy expenditure. Furthermore a supply air temperature higher than the design temperature leads, in a certain time, the air temperature of C.E. far from the set point. This effect is most visible in a highly insulated building, where the internal and solar loads (e.g. through windows) have an important weight. Therefore, it is necessary to operate the system in cooling mode during a heating period. For all these reasons the energy efficiency index is very important. Its value must be between 0 and 1:

- $i_{\varepsilon R} = 0$ when the unit is turned off or the static recovery is by-passed;
- $0 < i_{\varepsilon R} < 1$ when the static recovery acts at full load;
- $i_{\varepsilon R} = 1$ when the static recovery is modulated by regulating the speed of fans.

The heat pump downstream the heat exchanger must always provide the percentage difference of power from 1 to the actual value of the efficiency index. In cases that the index takes values higher than the unit, the power required by the generator would be of opposite sign, resulting in a waste of energy. This concept aforementioned is more immediate to understand if we consider equation 8:

$$P_{HP} = P_{SET} - P_{R_ST} = P_{SET} - (i_{\varepsilon R} \cdot P_{SET}) = P_{SET} \cdot (1 - i_{\varepsilon R}) \quad (8)$$

where

P_{HP} is the thermal power that heat pump must supply downstream passive recovery;

P_{SET} is the thermal power necessary to bring the supply air from the outdoor condition to T_{SET} ;

P_{R_ST} is the thermal power recovered by the static heat exchanger.

Juodis [32] studied the influence of heat gains on heat recovery and identified two values of the external air temperature that influence the efficiency of heat recovery. The first one at which the demand of heat generation is compensated by heat gain and the remainder by heat recovery. The second one at which heat gains compensate all heat loss and heat recovery system is switched off. Between these two temperatures the heat recovery must be modulated.

Commercial devices including heat exchanger retrofitted by a heat pump have high performance when the external conditions are optimal (mild days both in winter and summer). But these increases of COP are due to a supply air temperature far from the useful one (a higher temperature in winter and a lower temperature in summer than neutral point), so occupants could have local discomfort in addition to the loss of the above mentioned temperature control. Other studies highlight similar results. In fact, Nguyen et al. [22] in their study, analyse a unit which consists of compressor, outdoor fan and indoor fan, all with on/off control, indoor heat exchanger, outdoor heat exchanger, and sensible heat exchanger. It was realized and tested by using two “climatic chambers”. The evaluated COP, although acceptable, was obtained with a supply air temperature in the range 37.6 – 38.4 °C (function of ventilation flow rate), and considering only the compressor electrical absorption. Fracastoro and Serriano [20] evaluated the energy performances of an exhaust air heat pump (compressor speed is not modulated, but only on/off control was implemented). The COP was evaluated by using estimate mathematical model of the dynamic thermal behaviour of the system validated through field and laboratory experiments. Quite high COP values were obtained, in this case, but again the temperature of the supply air, when the outdoor air temperature increases, was higher than 35°C.

In these last cases acceptable COP values were achieved, but the temperature of the supply air was always over the desired set point making it possible to have some discomfort.

3.3 Results and discussion

In Table 7 and Table 8 the average values and the standard deviation of the main physical and energetic parameters are shown.

For each trial all temperatures drawn in Figure 21 were collected, but only temperatures useful to evaluate energy parameters of Table 8 are shown in Table 7. Each parameter is evaluated at different outdoor air temperature and in each compressor-operating mode. The outdoor air temperatures reproduced on the suction side of the recovery system were -5 °C, 0 °C, 5 °C and 10 °C, while the indoor set-point was 20 °C. Tests were performed in static (passive) plus

thermodynamic (active) arrangement and by using the exhaust temperature as a reference to control and regulate the heat recovery. In this way, SIVeMeC operates to balance the heating load, thus the supply airflow rate \dot{V}_S was almost always constant (on average 535 m³/h at $T_E = 0^\circ\text{C}$, 5°C and 10°C), while the supply air temperature T_S is continuously adjusted. The average values and standard deviations of each test are:

- 537.35 m³/h with standard deviation of 11.83 m³/h at 0°C ;
- 536.35 m³/h with standard deviation of 16.62 m³/h at 5°C ;
- 533.96 m³/h with standard deviation of 25.48 m³/h at 10°C .

Table 7: average values and standard deviation of the main physical measured parameters.

T_o	-5 °C		0 °C				5 °C				10 °C				
	COMPRESSOR OPERATING MODE														
	ON 60 rps	OFF 0 rps	ON 60 rps	MOD (1) rps	ATT 30 rps	OFF 0 rps	ON 60 rps	MOD (2) rps	ATT 30 rps	OFF 0 rps	ON 60 rps	MOD (3) rps	ATT 30 rps	OFF 0 rps	
T_s [°C]	\bar{x}	17.14	16.03	20.13	21.15	21.26	-	20.73	21.48	22.18	18.26	21.73	-	22.98	19.23
	σ	1.06	1.77	1.12	0.17	0.14	-	1.32	0.17	0.24	1.42	1.59	-	0.45	1.38
T_E [°C]	\bar{x}	18.40	18.55	19.04	19.75	20.27	-	19.25	19.54	20.15	19.99	19.01	-	19.84	19.83
	σ	0.16	0.18	0.11	0.22	0.20	-	0.27	0.19	0.19	0.34	0.36	-	0.37	0.38
$T_{s,ST}$ [°C]	\bar{x}	17.06	17.01	16.28	16.93	17.48	-	17.16	17.46	17.96	17.70	17.71	-	18.50	18.35
	σ	0.32	0.42	0.13	0.20	0.19	-	0.27	0.18	0.16	0.33	0.32	-	0.37	0.39
\dot{V}_s [m ³ /h]	\bar{x}	334.44	317.85	509.12	537.88	542.05	-	509.65	532.23	537.47	555.59	505.26	-	533.51	556.54
	σ	140.75	164.67	9.40	7.65	6.34	-	12.01	9.39	5.82	9.93	11.59	-	7.83	10.91

(1) Before attenuation (ATT) it is observed a period of about 10 minute of modulation at avg 33.94 rps and dev.std. 4.84rps
(2) Before attenuation (ATT) it is observed a period of about 8 minute of modulation at avg 34.33 rps and dev.std. 4.78rps
(3) Attenuation is not preceded by modulation

On the contrary, during the test at $T_o = -5^\circ\text{C}$, \dot{V}_s was lower (on average 326 m³/h) and greatly variable (standard deviation was very high: 152.71 m³/h) because the static and thermodynamic operating mode caused an on/off behavior of the heat pump instead of the modulation mode achieved in the other cases.

As said, SIVeMeC behavior has small changes at different outdoor temperature of the last three tests. As shown in Table 7 and Table 8 there are four compressor states:

1. ON, when the compressor speed ramps up to 60 rps and operates at nominal load;
2. MOD (modulation), when the compressor modulates the supplied power before attenuation;

3. ATT (attenuation), when the compressor works at the minimum allowed speed (30 rps);
4. OFF, when the compressor is disabled to avoid undesirable out-of-range temperatures (i.e. when the condition of equation 5 is verified and when T_{DIS} falls below a safety value to prevent the frost on the evaporator).

Table 8: average values and standard deviation of the main energetic calculated parameters.

T_o		-5 °C		0 °C				5 °C				10 °C			
		COMPRESSOR OPERATING MODE													
		ON	OFF	ON	MOD	ATT	OFF	ON	MOD	ATT	OFF	ON	MOD	ATT	OFF
		60 rps	0 rps	60 rps	(1) rps	30 rps	0 rps	60 rps	(2) rps	30 rps	0 rps	60 rps	(3) rps	30 rps	0 rps
P_R [kW]	\bar{x}	2.48	2.18	3.56	3.96	3.95	-	2.76	3.06	3.25	2.57	2.07	-	2.41	1.80
	σ	1.01	1.04	0.23	0.05	0.05	-	0.24	0.07	0.07	0.30	0.20	-	0.08	0.29
η [%]	\bar{x}	87.76	88.74	85.81	86.09	86.34	-	85.57	86.17	86.11	85.11	86.15	-	86.85	85.24
	σ	1.74	2.67	0.34	0.27	0.32	-	0.58	0.40	0.29	0.48	0.64	-	0.64	0.84
i_{eR} [-]	\bar{x}	0.82	0.94	0.82	0.85	0.89	-	0.81	0.84	0.87	0.85	0.78	-	0.85	0.83
	σ	0.02	0.15	0.01	0.01	0.05	-	0.02	0.01	0.01	0.02	0.03	-	0.03	0.04
COP_s [W _{th} /W _e]	\bar{x}	6.25	46.95	5.61	9.24	10.17	-	4.43	7.03	7.99	15.05	3.19	-	5.84	10.43
	σ	6.24	32.33	0.25	1.4	1.95	-	0.21	0.55	0.18	1.81	0.23	-	0.16	1.69
COP_s (4) [W _{th} /W _e]	\bar{x}	-		9.50				8.86				6.62			
	σ	-		2.20				3.72				2.70			

(1) Before attenuation (ATT) it is observed a period of about 10 minute of modulation at avg 33.94 rps and dev.std. 4.84rps
(2) Before attenuation (ATT) it is observed a period of about 8 minute of modulation at avg 34.33 rps and dev.std. 4.78rps
(3) Attenuation is not preceded by modulation
(4) Average coefficient of energy performance over the whole test

At outdoor temperature air $T_o = 5$ °C all four states occur according to the ordered list above. The compressor ramps up to nominal value (60 rps) and, once the regime was reached, the state was changed to MOD to keep the indoor temperature at set-point value. At $T_o = 10$ °C, modulation is not necessary and after the regime attenuation state immediately followed ON state. The OFF state occurs when the supply air temperature would lead to exceed indoor air temperature set point. At 0 °C the OFF state is absent, as during test time (40 minutes) heat load was too high to reach regime condition. In fact, the heat transferred to the supply air was 3.95 kW, 3.25 kW and 2.41 kW at $T_o = 0$ °C, 5 °C and 10 °C respectively. Moreover, even if T_E is 20.27 °C, at $T_o = 0$ °C the compressor is still active. Indeed, the reached T_S (21.26 °C) at $T_o = 0$ °C, switching through different compressor states, is smaller than that at $T_o = 5$ °C ($T_S = 22.18$ °C) and $T_o = 10$ °C ($T_S = 22.98$ °C) test, thus the second PI response is slower.

The energy efficiency index i_{eR} never exceeds 1 due to the fact that the only static exchanger is insufficient to guarantee the reaching of T_{SET} (20°C). The temperature T_{S_ST} measured

downstream the static recovery was $17.48\text{ }^{\circ}\text{C}$, $17.96\text{ }^{\circ}\text{C}$ e $18.35\text{ }^{\circ}\text{C}$ at $T_0 = 0\text{ }^{\circ}\text{C}$, $5\text{ }^{\circ}\text{C}$ and $10\text{ }^{\circ}\text{C}$ respectively. The slightly differences of T_{S_ST} at different compressor states are due to the influence of condensation direct expansion coil on measurement, because of the positioning of probe in a very compact system (see Figure 21). In fact, the probe is installed very close to the condenser. Also leakage through the heat recovery could affect the measured values [28]. It is important to emphasize that i_{eR} ranges between 0.78 and 0.89, with the exception of test at $T_0 = -5\text{ }^{\circ}\text{C}$ in which $i_{eR} = 0.94$, even if the static efficiency (temperature ratio η) is very high ($85 \leq \eta \leq 89\%$). This means that static heat recovery is never sufficient to balance the heating load of the indoor climatic room or, at least, to reach the neutral condition at C.E. inlet. Since that in reproducing indoor conditions no heat gains is simulated, the inlet air temperature useful to maintain the indoor set point matches just with this value ($20\text{ }^{\circ}\text{C}$). However, in real application T_{SET} (in equation 6) is a function of thermal properties of the building [33] and it is lower than the indoor air set point.

At $T_0 = -5\text{ }^{\circ}\text{C}$, SIVeMeC worked as an on/off device. Paying special attention to this case in Figure 39 and Figure 40 the trend of the compressor adjustment (Figure 39) and of the flow rate (Figure 40) are shown. This is confirmed by global values reported in Table 7 and Table 8, where the standard deviation of both renewal flow rate (\dot{V}_S) and overall coefficient of performance (COPs) is of the same order of magnitude of measured values. It may be pointed out the on/off behavior of the compressor although the heating load increases. Indeed, the T_{SET_calc} keeps growing inasmuch the T_E measured is far from the set point. The marked difference between T_{SET_calc} and T_S involves a reduction of the renewal flow rate in an attempt to lead T_S to the set point (Figure 40). The compressor is activated for the same purpose, but the low temperature values on the discharge side cause the switch-off of the compressor to prevent the frost on evaporator. After the first turn on of the compressor, the discharge air temperature is always below $1\text{ }^{\circ}\text{C}$ (Figure 41). The safety value of temperature under which the defrost system on the evaporator is activated is just $1\text{ }^{\circ}\text{C}$. Therefore, when the compressor turns on, the discharge air temperature is below $1\text{ }^{\circ}\text{C}$ and after a delay time of 30 seconds the compressor is turned off again to prevent the freezing on the direct expansion coil. In fact, the compressor could start to modulate after the start ramp and the following period of 120 seconds (at most) at 60 rps (nominal value).

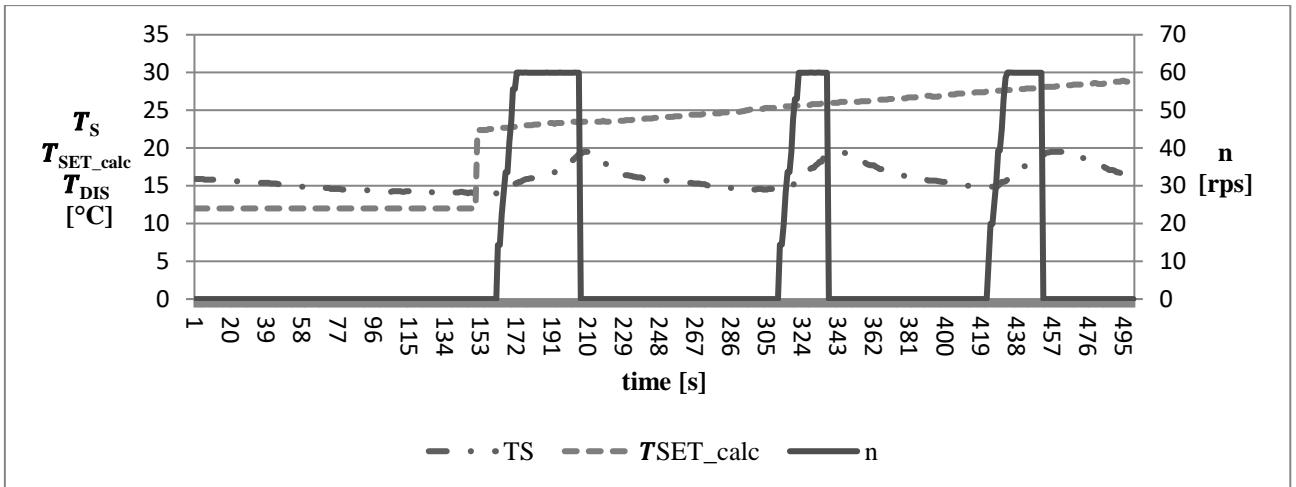


Figure 39: on-off behaviour of the compressor (test with $T_0 = -5^\circ\text{C}$).

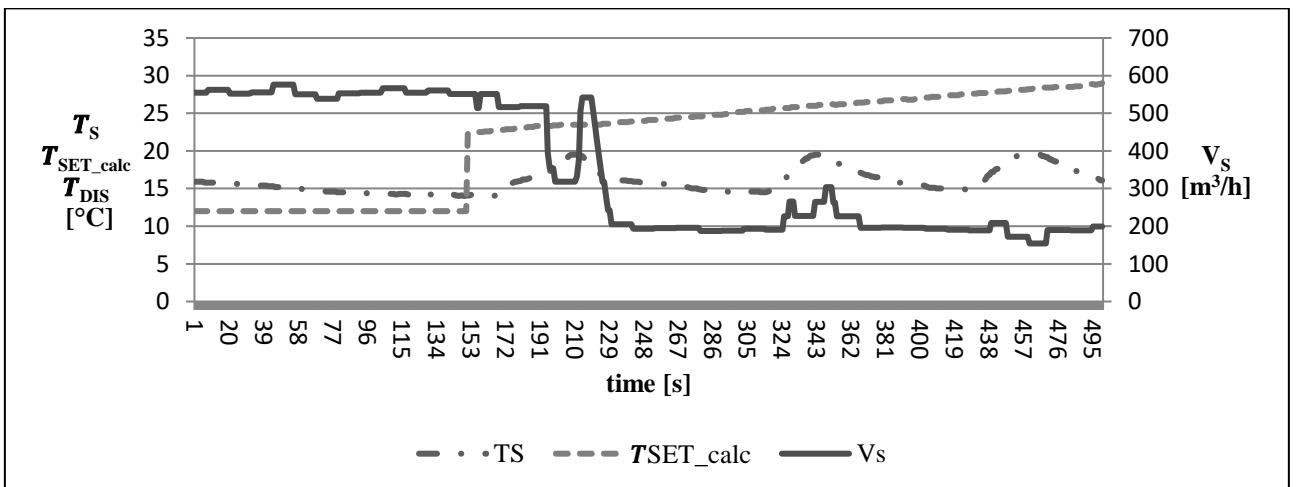


Figure 40: airflow rate reduction due to the high difference between T_s e T_{SET_calc} (test with $T_0 = -5^\circ\text{C}$).

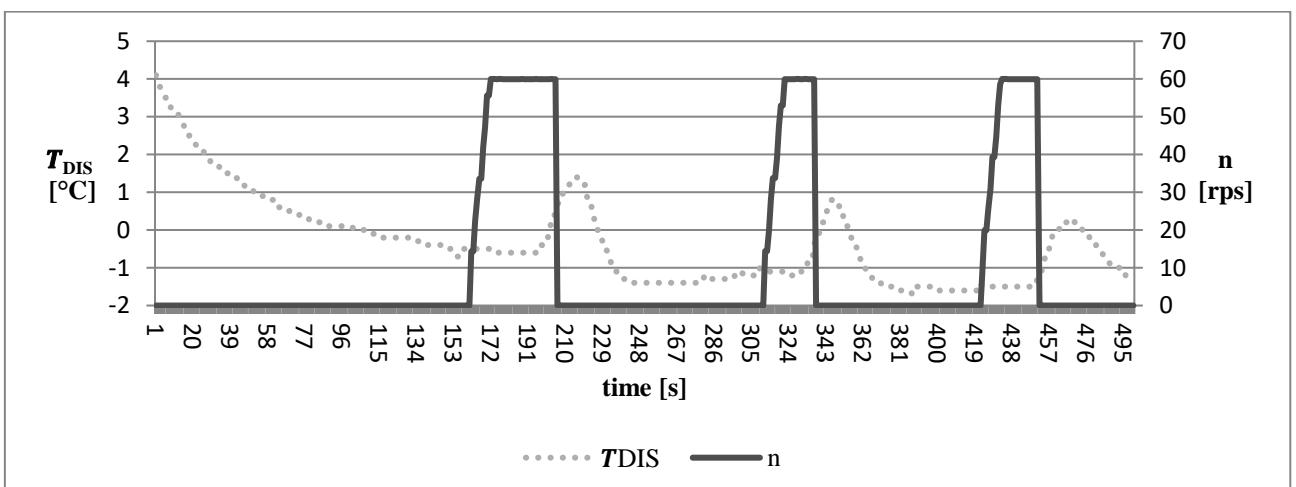


Figure 41: temperature trend at the evaporator inlet (test with $T_0 = -5^\circ\text{C}$).

Once again, at $T_0 = 5^\circ\text{C}$, the prevalent operating mode during the whole test is the attenuation one. During this state the average COPs is 8.0, while the average COPs along the whole test is

equal to 8.9 which is higher than that in attenuation mode. This is because when the compressor is off the COPs reaches its maximum value of 15.0. However, when the unit works with the only passive recovery, the supply air temperature decreases causing also a recovery temperature decreases. In this way, the compressor turned on again.

The same behavior occurs in the fourth test (suction air temperature $T_0 = 10^\circ\text{C}$). The COPs during attenuation mode is 5.8, while the average COPs is 6.6. In addition, in this case, the reason is that when the compressor is off the COPs reached its maximum value of 10.4. In this case, the loss of the recovery temperature control is more evident. With the only static recovery T_E drops to 19.8°C , so the T_{SET_calc} keeps growing until the compressor starts again.

This high sensitivity of the temperature control is due to a “dead band” equal to 0°C ($\Delta T_{DB} = 0^\circ\text{C}$). In an ordinary civil application ΔT_{DB} is set at least at 2°C : within it, control mode does not change. A ΔT_{DB} greater than 0°C allows to stabilize the control system inside the dead band (across the set point value). In this way, frequent fluctuations of temperature across the set point are prevented improving the energy performance. Therefore, ΔT_{DB} is set to 0°C to evaluate the performance in the worst case. Furthermore, setting $\Delta T_{DB} = 0^\circ\text{C}$ allows the unit to follow the heat load of the C.E. with high sensitivity.

In 2005, a similar system was tested [22]. It consists of a heat exchanger which acts before an heat pump equipped with an on/off compressor instead of modulated one. The obtained results with a ventilation flow rate of $300 \text{ m}^3/\text{h}$ are 4.13 for the refrigerant based COP (COP_{ref}) and 3.18 for the COP of the system ($\text{COP}_{net} \neq \text{COP}_s$ as in both cases, only electrical power of the compressor was considered. These results were obtained with an external air temperature of 7°C and inside air temperature of 21°C . The system studied by Fracastoro and Serraino [20] (EAHP), at the same condition of the outdoor air temperature, showed a COP_{EAHP} above 5.00 with an air flow rate of $490 \text{ m}^3/\text{h}$ and without considering the electrical power of the fans. Table 9 provides a comparison between the COPs of SIVeMeC and that of some prototype found in literature. Also a commercial system was taken into account. The collected information from the data sheets of constructors and from literature show that SIVeMeC has, for a wide range of external temperature, an excellent COPs when total electrical power (compressor and fans) is considered.

In Table 9 COPs⁽¹⁾ is the COP of the system calculated without taking into account fans electrical power (averaged along the ON mode), COPs⁽²⁾ is the COP of the whole system (both compressor and fans power and averaged along the whole test period). It is important to note

that the COP_S⁽²⁾ of SIVeMeC is higher than the COP_S⁽¹⁾. When the outdoor air temperature is T_O = -5 °C the COP_S⁽¹⁾ of SYSTEM C (commercial system) is greater than SIVeMeC one. This is due to the safety mechanism (frost protection) of SIVeMeC, which induces the on/off functioning of the compressor and reduces the renewal flow rate. In this way a lower heat P_R occurs. Also at 10 °C the COP_S⁽¹⁾ of the commercial system is greater than SIVeMeC one. In this case, the higher heat P_R of SYSTEM C than SIVeMeC and subsequently an increased COP are due to a supply air temperature far from the useful one. In fact, the aim of such a system is both energy saving and to avoid discomfort. As already said, a supply air temperature higher than the design one (as in some cases found in literature [20], [22] and above described) could lead to the provision of a thermal energy of opposite sign (e.g. cooling during winter season) by the plant system at service of the building.

Table 9: Comparison of the COP of different systems.

T _E [°C]	SIVeMeC		SYSTEM A (Nguyen et al.)		SYSTEM B (Fracastoro and Serraino)		SYSTEM C ^(*)	
	COP _S ⁽¹⁾	COP _S ⁽²⁾	COP _S ⁽¹⁾	COP _S ⁽²⁾	COP _S ⁽¹⁾	COP _S ⁽²⁾	COP _S ⁽¹⁾	COP _S ⁽²⁾
	[W _{th} /W _e]							
-5	8.3	-	-	-	5.3	6.8	9.6	-
0	9.0	9.5	-	-	4.8	6.5	7.8	-
5	7.5	8.9	-	-	4.3	5.6	6.6	-
7 ⁽³⁾	6.5	7.9	-	3.2	4	5.3	6.2	-
10	5.2	6.6	-	-	3.8	4.8	5.6	-

(1) COP of the overall system without electrical absorption of fans (for SIVeMeC at 60 rps of compressor speed)
(2) COP of the overall system
(3) COP of SIVeMeC and SYSTEM C obtained by linear interpolation of the previous and next values
(*) Commercial system with on/off compressor and fixed speed of fans: SYSTEM C with R410a; SYSTEM C with R407c

At T_O = -5 °C the COPs of SIVeMeC is very high. The reason is that in this condition the unit usually works with the only passive recovery. Nevertheless, as can be seen analyzing the values reported in Table 7, in this operating mode (heat pump off) T_E never reaches the set point as the only static recovery, albeit efficient, is not enough to enable a T_S useful for the purpose. Moreover, the heat pump, which should ensure the additional heating to lead the supply air at the desired conditions, is continuously stopped by the safety control to avoid frosting of the evaporator. This behavior suggests the existence of a temperature value at which is preferable to switch from the static plus thermodynamic operation mode to another arrangement and vice-versa.

For example, if the only thermodynamic recovery is working (static exchanger by pass open), the efficiency of the system is higher than that of conventional plant systems generally installed in buildings. It is possible to switch the operating mode in this arrangement or to reduce the ventilation flow rate to the minimum value necessary to ensure the indoor air quality and use the only passive recovery, leaving to the downstream plant systems the load coverage (if these are more efficient of the heat pump on the installed ventilation system). Moreover, when T_0 increases, it is easy to think that a value of the outdoor temperature exists above which it is correct to exclude the static heat exchanger and employ only the heat pump. This is true until the external air conditions allow the freeheating or freecooling.

4 GREENHOUSE EXPERIMENTAL APPARATUS

The mechanical ventilation system previously studied in laboratory, was then transferred in a real case by installing SIVeMeC at Vivaio Verde Molise, Termoli – Italy. To do that it was necessary to determine the size of the greenhouse that could be heated by SIVeMeC. In order to evaluate the performance of the system in this real case, a compact and smart supervisory system, for the monitoring of both field and prototype unit, was designed and realized.

All these elements, plus some minor modifications to existing installations, constitute the *experimental apparatus*.

In this section, firstly we focus on the energy and mass balance of the greenhouse of interest in order to understand the size compatibility with SIVeMeC. Subsequently, the installation of the experimental apparatus and its set up will be described in detail.

4.1 Greenhouse energy and mass balance

Greenhouse energy balance is obtained by evaluating each term that appears in the law of energy conservation. To do that the air within the greenhouse is considered as a control volume. While the energy conservation is related to sensible heat, the mass conservation is related to latent heat (humidity) and gaseous (or other) contaminants [1].

A schematic representation of each term appearing in the sensible energy transfer of an agricultural building (single airspace) is reported in Figure 42.

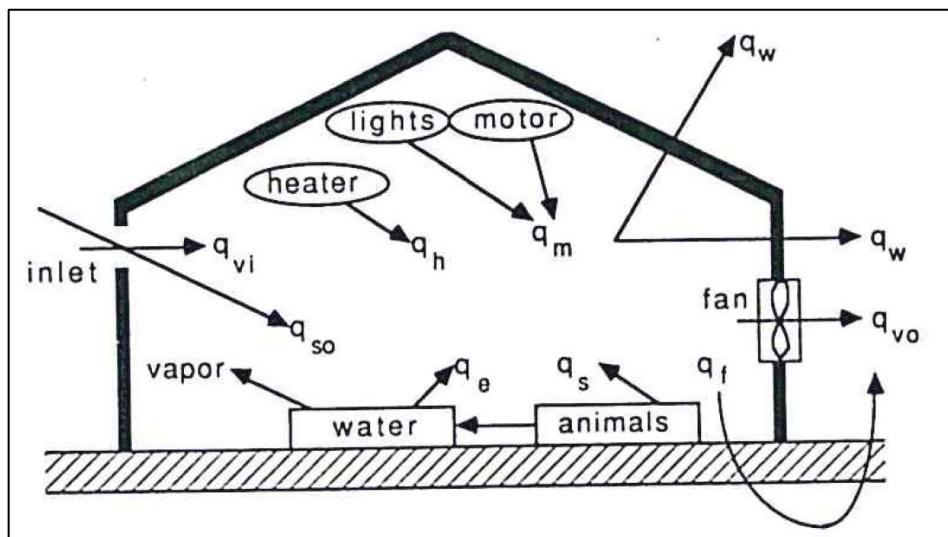


Figure 42: terms appearing in sensible energy balance of a typical agricultural building [1].

The energy flow terms are:

q_s : sensible heat gain from animals or people;

q_h : sensible heat gain from heating system;

q_m : sensible heat gain from motors and light (electrical energy converted to sensible heat);

q_{so} : sensible heat gain from the sun;

q_{vi} : sensible heat contained in the ventilation air entering in the airspace;

q_w : the transfer of sensible heat through the structural cover of the building (walls, ceiling, windows, etc.);

q_f : sensible heat transfer to the floor of the building;

q_e : the rate of conversion of sensible heat to latent heat within the airspace;

q_{vo} : sensible heat contained in the ventilation air exiting from the airspace.

The general form of the energy balance for the control volume is:

Gain – Losses = Change of Storage.

Generally, in agricultural buildings a steady state model is used rather than time-dependent models. In this case, there is no change of storage and the form of energy balance is simplified becoming:

Gain = Losses

that is:

$$q_s + q_m + q_{so} + q_h + q_{vi} = q_w + q_f + q_e + q_{vo} \quad (9)$$

Regarding mass balance it is referred to the same airspace (Figure 43).

The mass flow terms are:

m_p : the rate at which the material of interest (water vapor, carbon dioxide, etc.) is produced inside the airspace;

m_{vi} : the rate at which the material of interest is carried into the airspace by ventilation air;

m_{vo} : the rate at which the material of interest is carried out of the airspace by ventilation air.

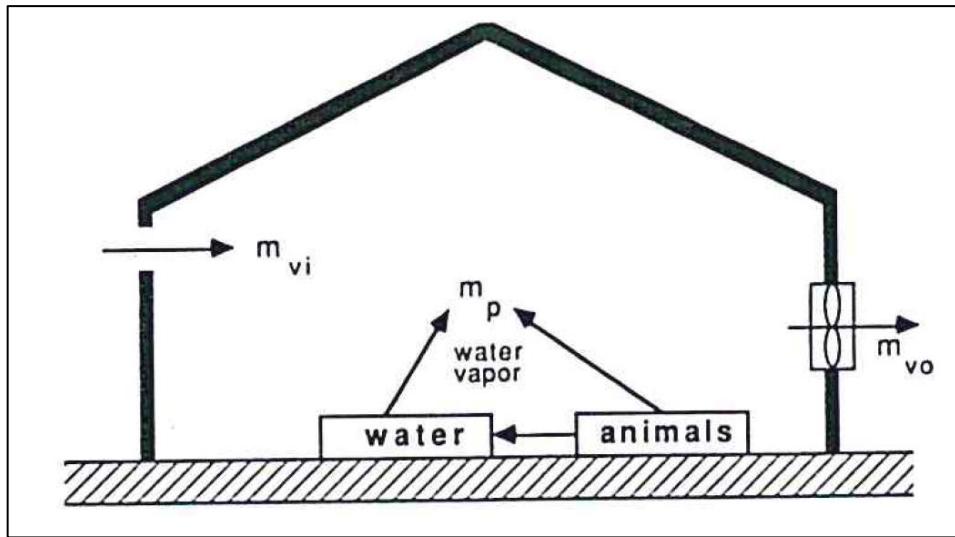


Figure 43: mass balance for a typical agriculture building [1].

By referring to Figure 43, the mass balance is:

$$m_p + m_{vi} = m_{vo} \quad (10)$$

The humidity inside a greenhouse is generally a function of plant population, light intensity and water management and has not been quantified in a way useful for environmental design. As a consequence, mass balance applied to greenhouse is seldom used to design its environmental control system [1].

To establish the heat necessary to maintain the desired temperature inside the greenhouse the sensible energy balance is used. In common practice, equation 9 is generally simplified depending on the case of study. Definitely, it is possible to do a consideration for each term of equation 9. The first term q_s is referred to the presence of animals and/or people, thus we can delete it. Also the q_f term could be deleted, because the heat exchange with the floor, in these buildings, is generally negligible. Moreover, during the production period the greenhouse ground was completely covered with plants. Thus, equation 9 can be rewritten:

$$q_h = q_w + q_f + q_e + q_{vo} - (q_m + q_{so} + q_{vi}) \quad (11)$$

which means that the heat provided by the heating system should match the heat losses diminished of the heat gains.

However, to select an appropriate heating system for a greenhouse we must determine the peak heating requirement for the structure [33]. In this case we consider only the following two terms: the transmission losses through the structural cover (q_w); infiltration and ventilation

losses caused by the heating of cold outside air ($q_{vo} - q_{vi}$). For this reason, to obtain the maximum heat that the heating system should provide, we must explicit these two terms.

The heat transfer through the structural cover could be calculated as:

$$q_w = \sum_i U_i A_i (T_i - T_o) \quad (12)$$

where

U_i is the thermal transmittance of the structural element considered [W/m²K];

A_i is the area of the i-th structural element considered [A];

$(T_i - T_o)$ is the temperature difference between indoor and outdoor air.

Some typical values of the thermal transmittance, for values of wind speed below 4 m/s, is reported in Table 10 [34]. If the wind speed exceeds this value, a correction factor must to be used.

Table 10: values of thermal transmittance for values of wind speed below 4 m/s.

Materials	Thermal transmittance [W/m ² K]
Single plastic film	6 ÷ 8
Double plastic film	4 ÷ 5.5
Glass	5
Double glass	2.5

Regarding the heat transfer due to the ventilation, even if not strictly true, we consider that the mass flow rate of air entering into the airspace is equal to the mass flow rate of the exiting one. In fact the moisture produced inside the airspace should be added to the exiting air flow rate. However, the difference is generally small, so we can assume a constant mass flow rate [1]. This consideration allows to explicit the heat due to ventilation as:

$$q_{vo} - q_{vi} = 1006 \rho \dot{V} (T_i - T_o) \quad (13)$$

where

1006 is approximately the specific heat of air [J/kgK];

ρ is the air density (which can be considered about 1.2 kg/m^3);

\dot{V} is the volumetric airflow rate [m^3/h].

the contribution airflow rate due to infiltration must be added also to the volumetric airflow rate:

$$\dot{V}_{\text{inf}} = nV \quad (14)$$

where

n is the number of air changes per hour [$1/\text{h}$];

V is the volume of the airspace [m^3].

By introducing the infiltration air contribution in equation 13 we obtain:

$$q_{vo} - q_{vi} = 1006\rho(\dot{V} + nV)(T_i - T_o) \quad (13)$$

Figure 44 shows a scheme of the greenahuse at service of which was installed SIVeMeC. It consists of a mini-tunnel greenhouse type of $1.8 \times 10 \text{ m}$ and 1.3 m high (to the ridge). The high of the “side wall” is 0.4 m . The material used for the covering structure is a single plastic film ($U = 6 \text{ W/m}^2\text{K}$).

To evaluate the heating requirements of the greenhouse object of study we must calucalate the total covering area A_T and the volume of the inner airspace V :

$$A_T = (2 \cdot 0.4 + 0.9 \cdot \pi) \cdot 10 = 36.26 \text{ m}^2$$

$$V = (0.4 * 1.8 + 0.5 \cdot \pi \cdot 0.9^2) \cdot 10 = 19.92 \text{ m}^3$$

The design outdoor temperature are generally reported in national standards (e.g. for Italian regulation UNI/TR 10349-2:2016) or provided by state energy office and organiztion. The design outdoor temperature of Termoli (Italy), where the grenhouse of interest is located, is of 0°C . As indoor air temperature we consider, as general value, 20°C (this value is function of the cultivated crop). Temperature is the most important variable for the microclimate of a greenhouses. Table 11 reports the optimum temperature for different crops cultivated in greenhouse [35]. Finally we assumes as number of air change per hour 0.5 (other values could find in literature as function of wind, greenhouse materials, greenhouse shape, etc.).

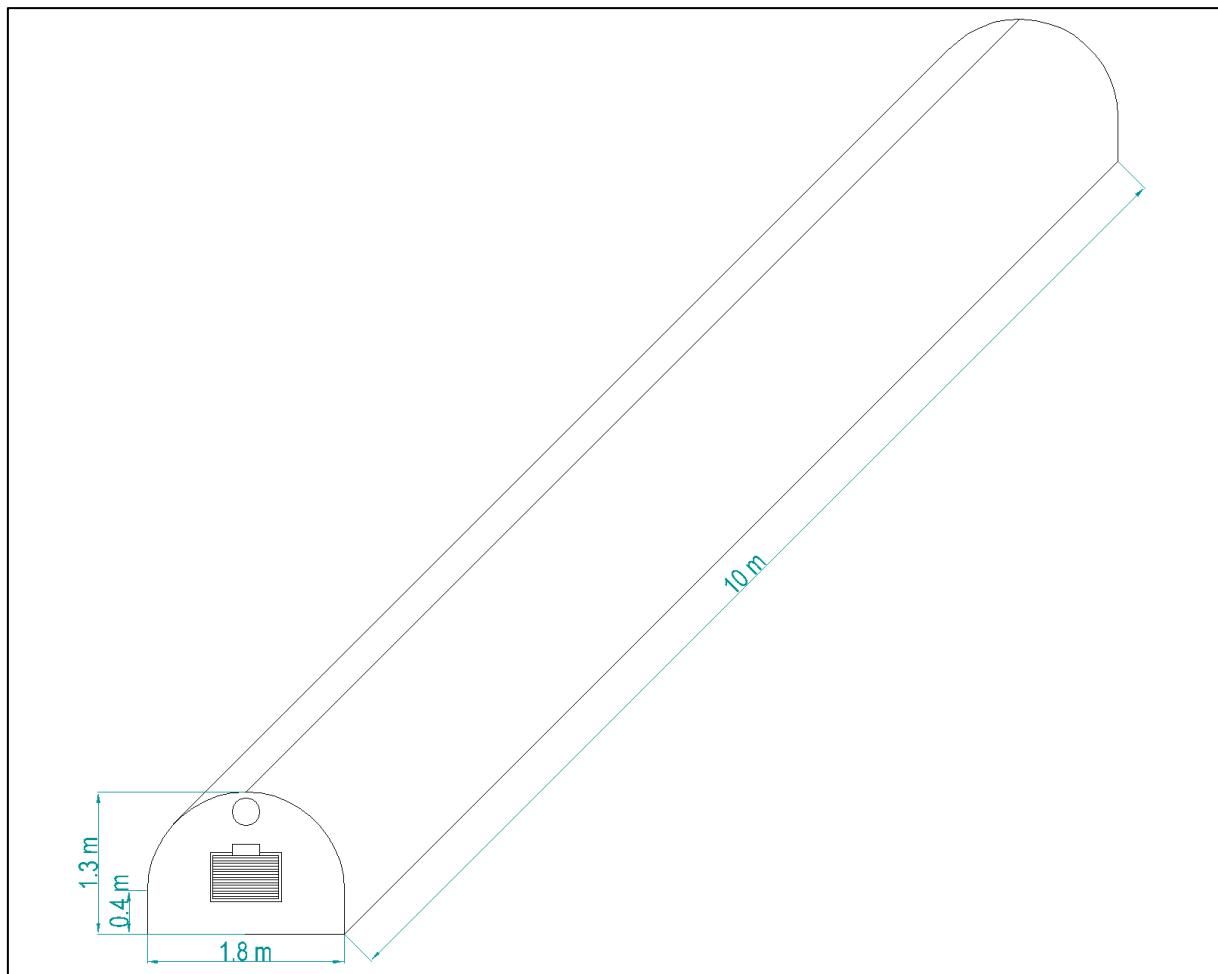


Figure 44: dimensions of the greenhouse served by SIVeMeC.

Table 11: Optimum temperature for different crops [35].

Thermal levels of indoor air for the main greenhouse crops		
Crop	Optimum temperature [°C]	
	Day	Night
Horticultural	20 – 24	12 – 16
Tomato	21 – 26	13 – 18
Lettuce	17 – 22	10 – 13
Melon	22 – 28	15 – 18
Floricultural	19 – 25	14 – 16
Rose	20 – 24	14 – 16
Chrysanthemum	17 – 21	16 – 17
Gerbera	21 – 27	12 – 16

It is now possible to evaluate the heating need of the greenhouse. For now it is advisable not to take into account the ventilation flow rate provided by SIVeMeC. In fact, SIVeMeC should be

seen as an additional heating system even if its principal means is ventilation. Table summarizes the main results.

Table 12: Heat losses by considering only transmission through structure and air infiltration.

HEAT LOSSES		
Description	Calculation	Heat [W]
Losses Through structure	$q_w = 6 \cdot 36.26 \cdot 20$	4351.20
Losses by infiltration	$q_{vo_{inf}} - q_{vi_{inf}} = 0.34 \cdot 0.5 \cdot 19.92 \cdot 20$	67.72
Total heat losses		4418.92
0.34 is the specific heat of air referred to the m^3/h : $c_{pa} \cdot \rho / 3600 = 1006 \cdot 1.2 / 3600$ [Wh/m^3K]		

By considering that SIVeMeC could supply an airflow rate of about $500 m^3/h$ at a supply air temperature of $35 ^\circ C$, this means an heating load of:

$$q_{vo_{SIVeMeC}} - q_{vi_{SIVeMeC}} = 0.34 \cdot \dot{V}_s \cdot (T_s - T_o) = 0.34 \cdot 500 \cdot (35 - 0) = 5950 W > 4419 W$$

It easy to note that SIVeMeC is suitable to balance the heating requirements of the greenhouse of interest. Another very important consideration springs from Table 4. The row relative to $T_o = 0 ^\circ C$ of outdoor temperature shows that the heat recovered by the passive heat exchanger is 2680 W. This imply that the heat pump must supply only 1738.92 W. Moreover, Table 4 also shows that the discharge air temperature downstream the passive recovery is $T_{DIS_ST} = 4 ^\circ C$. By matching this information with the performance curves (Figure 25) we can easily deduce that the heat pump should works in modulation mode.

4.2 Experimental apparatus installation

The experimental apparatus is composed of the mechanical ventilation system SIVeMeC, a distribution system and two Building Monitoring Systems (BMS), one to acquire data from field, another to manage and supervise SIVeMeC.

Before installing SIVeMeC at service of the greenhouse, some optimization was made. To add the possibility of recirculating the internal air it is necessary to realize a duct which connects the recovery side with the supply one. However, a strong recirculation of indoor air means an additional load for greenhouse dehumidification and consequently an increase in energy

consumption [36]. Hence the decision to delay the installation of recirculating duct. Nevertheless, a future installation of this system is possible since the application software for damper regulation is already developed. A regulation damper will be managed by SIVeMeC to adjust the airflow rates as a function of the damper degree of opening.

Another optimization of mechanical ventilation system for greenhouse application involves the approach of the unit about the management of the level of carbon dioxide. As opposed to the civil sector or agricultural buildings with animals, where the supply airflow rate increases linearly with the CO₂ concentration, in greenhouse application there is a CO₂ depletion. So a variation in operating logics related to the indoor air level of carbon dioxide was made. The control of the airflow rates was done primarily as a function of temperature and relative humidity. Thus, in contrast with previous application the airflow rate never decreases when CO₂ level increases.



Figure 45: experimental apparatus installed at Vivaio Verde Molise, Termoli - Italy.

Figure 45 shows the experimental apparatus. It consists of three mini-tunnel greenhouses type (A, B, C) of 1.8 x 10 m and 1.3 m high: A is the reference one, B is heated by a flat plate collector placed on the bench and C is air-conditioned with SIVeMeC.

SIVeMeC allows balancing the thermal load of the greenhouse C by adjusting the supply air temperature. In this way the internal temperature should be kept constant. Instead in the

greenhouse B temperature was not controlled because the collector on the bench are always supplied with water at 22 ± 2 °C. Thus, the internal air temperature was influenced by the external conditions (increases in mild days and decreases in cold days).

In the reference greenhouse A, the flat plate collector is closed with a gate valve. Only relative humidity is controlled by introducing water through misting nozzles. The relative humidity set point was set at RU 80 ± 5 %. The same system and regulation of humidity was used in the greenhouse B. In greenhouse C the RU has the same set point, but its regulation happens through a separate water line (see section 4.2.1).

4.2.1 Air and water distribution systems

For a suitable distribution of the supply air a perforated duct was designed and realized (Figure 46). A stainless steel tube of diameter 250 mm, with 5 holes each side of 10 mm and with a distance of 40 mm, was installed in the ridge of the greenhouse. It was connected with the supply side of the unit through an insulate hose. The perforations were designed to guarantee homogeneity of distributions and to avoid high values of air speed near the plants. The recovery side was connected to the mechanical ventilation system through a hose and a plenum. The aspiration and discharge duct are galvanized steel tubes of diameter 250 mm.



Figure 46: perforated duct for supply air distribution.

To regulate relative humidity in greenhouse object of study (C) a separate line of water supply and a solenoid valve were installed (Figure 47). The field supervisory system (see 4.2.3) reads

the internal value of relative humidity and manages the solenoid valve opening. This new line is necessary since the presence of mechanical ventilation system produces a higher level of dehumidification than reference case.



Figure 47: solenoid valve.

4.2.2 Field probes

The data collected from field are the outputs of a set of probes. The main parameters acquired are temperature, relative humidity and carbon dioxide level values inside the greenhouse C and air speed inside both supply and exhaust duct.

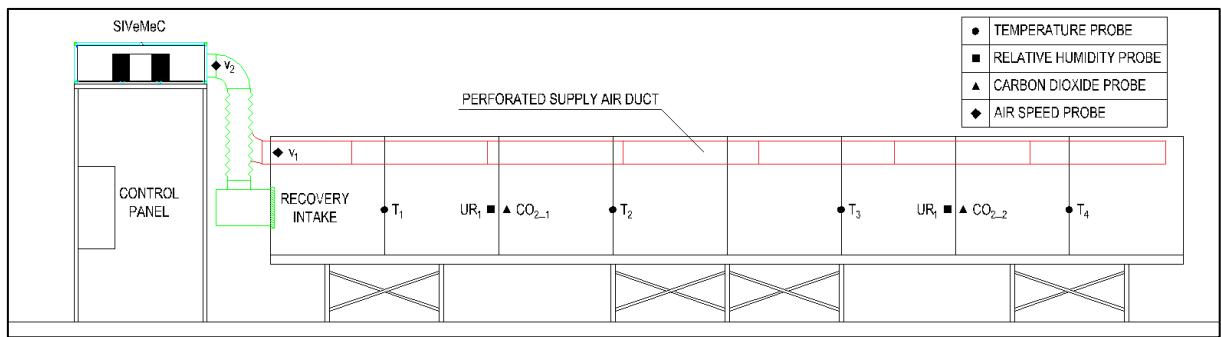


Figure 48: experimental apparatus layout and probes location.

Figure 48 shows the experimental apparatus layout and probes location inside the mini-tunnel greenhouse of interest C. The temperature probes are installed along the greenhouse at 2.5 m distance each other, while relative humidity and carbon dioxide level probes at 5 m. The probes are numbered in ascending order moving away from the recovery intake. Regarding to air speed probes, v_1 is the one on the supply side and v_2 on the exhaust side.

The probes were installed spaced uniformly in order to evaluate the uniformity of the indoor air parameters. The most important parameter, as previously said, is undoubtedly the temperature. For this reason, we tried to acquire its value in representative points of the greenhouse (first zone T₁, mean zone T₂ and T₃ and end zone T₄).

The air speed probe were useful to evaluate the volumetric airflow rate of both supply and exhaust air streams.

Table 13 shows the characteristics of the probes used for data acquiring from field.

Table 13: probe used in field.

<i>Probe</i>	<i>Accuracy</i>
NTC Temperature probe, -50-105°C	± 1% at 25°C
Duct humidity-temperature sensor, range 5-95% - -30-70°C, output 0-10Vcc	± 0,3K at 25°C ± 2,5% at 20°C 10-95% rh
CO ₂ transmitter, output. 0-10 Vcc	+ 2%
Duct speed sensor; range 0-20 m/s, output 0-10 Vcc	< 10%

4.2.3 Field supervisory system

The BMS system used for field data monitoring, the PlantWatch PRO® (Figure 49) of Carel S.p.A. is a supervisor able to communicate with field devices through Modbus protocol.



Figure 49: PlantWatch PRO®.

PlantWatch PRO® is a solution for monitoring and supervision of small and medium-sized systems. It has a 65 K colour - touch screen display for user interface. It allows controlling and

optimising refrigeration and air-conditioning systems. Among its main futures, it is worth remembering:

- possibility to connect and control CAREL or other Modbus devices over RS485 bus;
- log up to 500 variables;
- log data for up to one year (with sampling every 15 min);
- 2 relay outputs, for alarm signals or managing lights;
- extraction of data such as temperature, energy consumption, alarms, events, configurations and device models, using a USB memory key or SD Card;
- display graphs;
- complete alarm configuration;
- access to the system via login credentials and profiling to allow different levels of control;
- possibility of remote access from PC, smartphone or tablet (suitable for future extension of the system to Industry 4.0 applications).

An I/O (Input/output) module and three parametric controllers (DN33) was configured on a bus line of PlantWatch PRO®. I/O module allows to collect temperature values along the greenhouse (4 NTC probes), while the DN33 controllers allow to acquire values of relative humidity (2 RH probes), CO₂ concentration (2 CO₂ probes) and air speed (2 ILV probes).

Particularly, the DN33 at which was connected relative humidity probes permits also the activation of a solenoid valve to regulate the activations of a fog system. The DN33 platform is a product made up of integrated electronic microprocessor controllers with LED display, designed especially for the control of stand-alone units.

Also an energy meter was installed on a second Modbus line (because of a different speed of communication: 9600 baud/s). In this way it is possible to evaluate the coefficient of performance of the SIVeMeC because the energy meter was inserted on the power line of the ventilation unit.

All these components are meld together in a compact control panel showed in Figure 50.

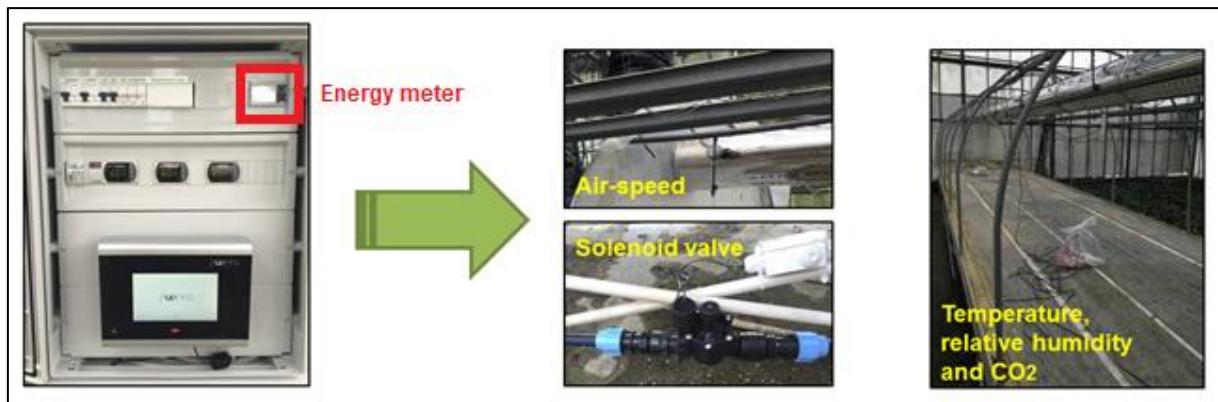


Figure 50: control panel equipped with energy meter.

4.2.4 SIVeMeC supervisory system

SIVeMeC supervisory system consists of software developed by Carel S.p.A. installed on a PC. This software is called pCOManager® and is a program that lets to manage all the configuration, debugging and maintenance operations on CAREL pCO system devices (including). Other futures are:

- upload and download to Flash and NAND;
- configure log files;
- fully debug the application, display alarms, trace variables, force inputs and outputs, set configuration parameters

In addition, it allows to create different debug environments to either show or hide variables and parameters, making maintenance easier. All variable lists can be printed.

There are three modules in pCOManager® which can be opened: Commissioning, pCOLoad and LogEditor.

Commissioning is a real-time configuration and monitoring software that allows to monitor the operations of the application installed on μ PC for its start-up, debugging and maintenance. Thanks to this tool we can set configuration parameters, edit volatile and permanent variable values, save the trends of main device values to a file (.csv), manually manage machine I/O in a simulation file and monitor/reset alarms on the machine where the device is installed. By using commissioning configuration functions we decided which variables are subject to monitoring/log/trend/event monitoring, and organized variables in categories by setting parameters configuration.

Figure 51 shows a user interface of the BMS used for SIVeMeC data acquisition, and a set of variables chosen for commissioning.

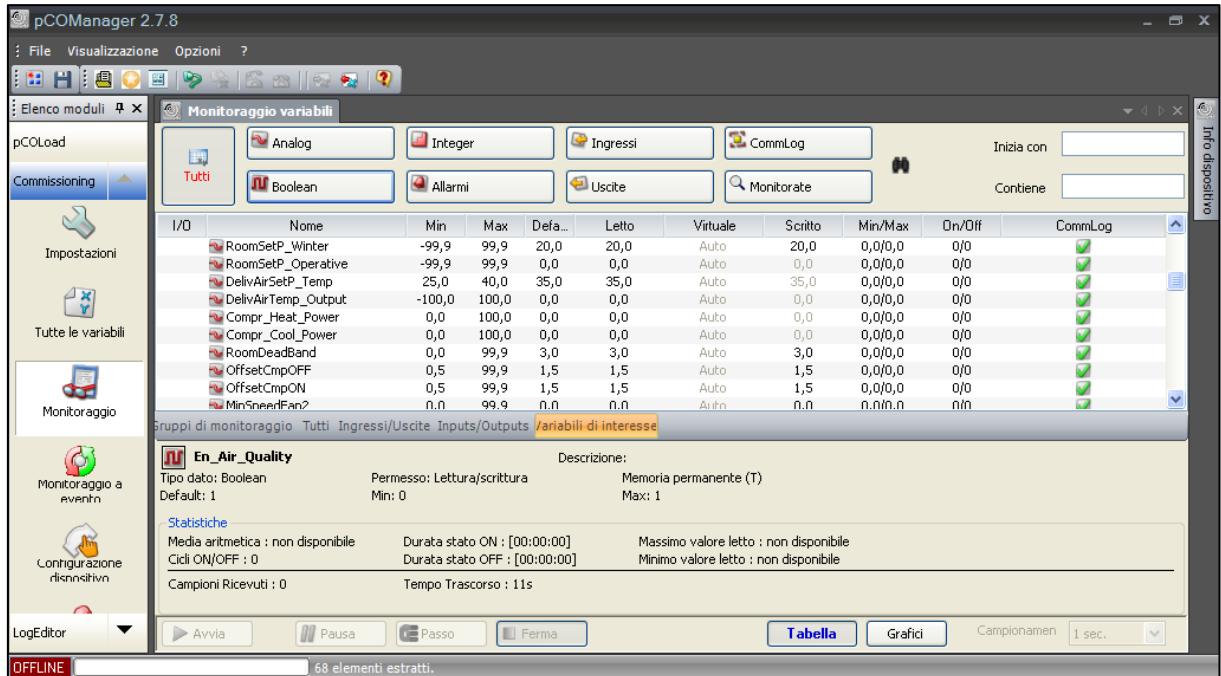


Figure 51: user interface of pCOManager® for SIVeMeC monitoring.

Variable monitoring can be manually run. The system reads variables according to the set sample interval (minimum 1 s), and simultaneously writes the variables to record to a file. Up to four variables can be monitored on a graph simultaneously. Curves can be saved to file or printed.

pCOLoad is the module that allows to upload/download the application (containing the working logics) on the µPC. We used it after the design of the application software with 1Tool®.

LogEditor is the module used for pCO® device log configuration (pCO log). pCO log configuration consists in defining a certain number of variable sets in which the variables to record, the saving procedure (frequency or event) and minimum number of guaranteed records are specified. Once log configuration files are loaded on the pCO®, the pCO® saves log data in the following files:

- .bin file with all data in binary format;
- .csv file that contains the same data but in a generic format with values separated by commas;

- *_graph.csv file with the same data to be used to display graphs.

Files can be downloaded using the download function.

5 GREENHOUSE EXPERIMENTAL TESTS

5.1 Experimental procedures

To test SIVeMeC system in the real application, it is firstly to understand the temperature difference established between the outdoor environment and the mini-greenhouse C. In fact, even if the system could switch automatically from “winter” to “summer” mode, currently we set this switch manually. Therefore, the first task to be performed is to monitor the temperatures using pGD1 or pCOManager®. This is essential, because if there is a high outdoor temperature T_o (greater than the desired set point) the machine will have to work according to the summer mode instead of winter mode.

Considering that there are two supervision systems, one for SIVeMeC (pCOManager®), the other one for field (PlantWatch PRO®), before starting tests we must synchronize the two BMS clocks. This procedure simplifies the following data elaboration.

The temperature chosen as set point for trials is 27 °C in order to understand the behavior of the mechanical ventilation systems when the heating requirement is related to a temperature difference $T_{SET} - T_o$ in the range 10 – 20 °C.

Measurement variables acquisition begins after the clocks synchronization. The data were acquired with a sampling time of 1 second on pCOManager® and 30 seconds on PlantWatch PRO® (minimum setting). Tests were carried out in heating mode by choosing as reference probe the exhaust air temperature T_E and working, as well as for laboratory tests, in passive plus active recovery mode.

5.2 Results and discussion

The main aim of these first set of tests is to understand whether SIVeMeC is suitable for temperature control inside greenhouse. Another significant target is to evaluate the energy efficiency of the overall system. As explained above (see section 3.2), the most important energetic parameter is the coefficient of performance of the overall system COPs. This parameter is not directly measured, but is calculated as the ratio of the heat supplied by the system (P_R) and its energy absorption (P_e).

In this work one of two main and representative tests carried out last winter (2016/2017) are reported and discussed.

The first test (Test 1) starts with temperature values of 17.4 °C and 20.3 °C for outdoor and exhaust air respectively. Moreover, the mean value of outdoor air temperature T_O was 18.0 °C. The whole Test 1 has had a duration of 44 minutes.

Figure 52 shows the exhaust air temperature T_E , supply air temperature T_S and the coefficient of performance of the overall system COPs as the outdoor air temperature T_O changes. The test is carried out with a set point of internal air temperature of 27 °C and a mean airflow rate of 410 m³/h. The probe for the regulation of SIVeMeC is installed on the recovery side of the unit (on board).

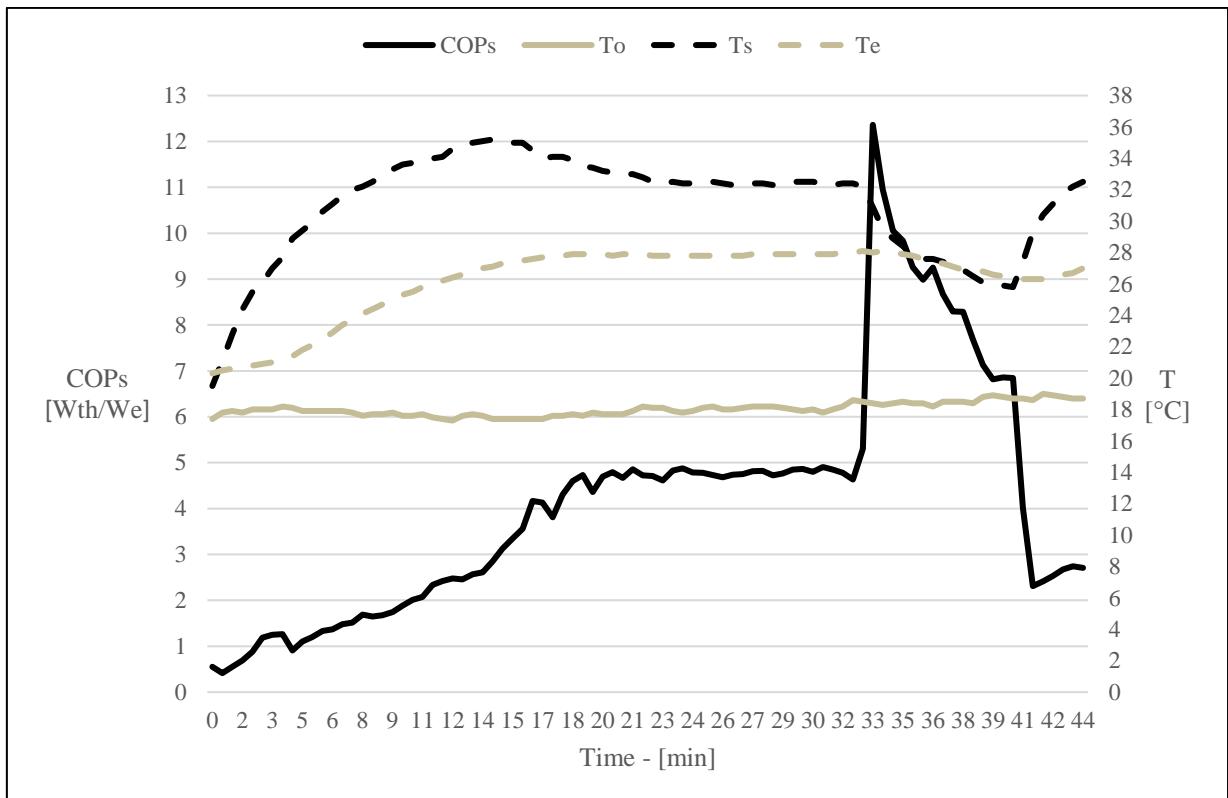


Figure 52: Test 1 – exhaust air temperature (T_E), supply air temperature (T_S) and coefficient of performance (COPs) as a function of outdoor air temperature (T_O).

At the beginning of the test the T_S grew up quickly to balance the thermal load and to bring as soon as possible T_E to the set-point. In fact, after a first step in which SIVeMeC worked with only passive recovery, the compressor turned on. This is well shown in Figure 53 where the COPs is reported together with the compressor speed in rps (revolution per second). When T_E reached the set point the heat pump modulated and then deactivated. However, when only passive recovery occurred, T_E slightly decreased so the compressor turned on again.

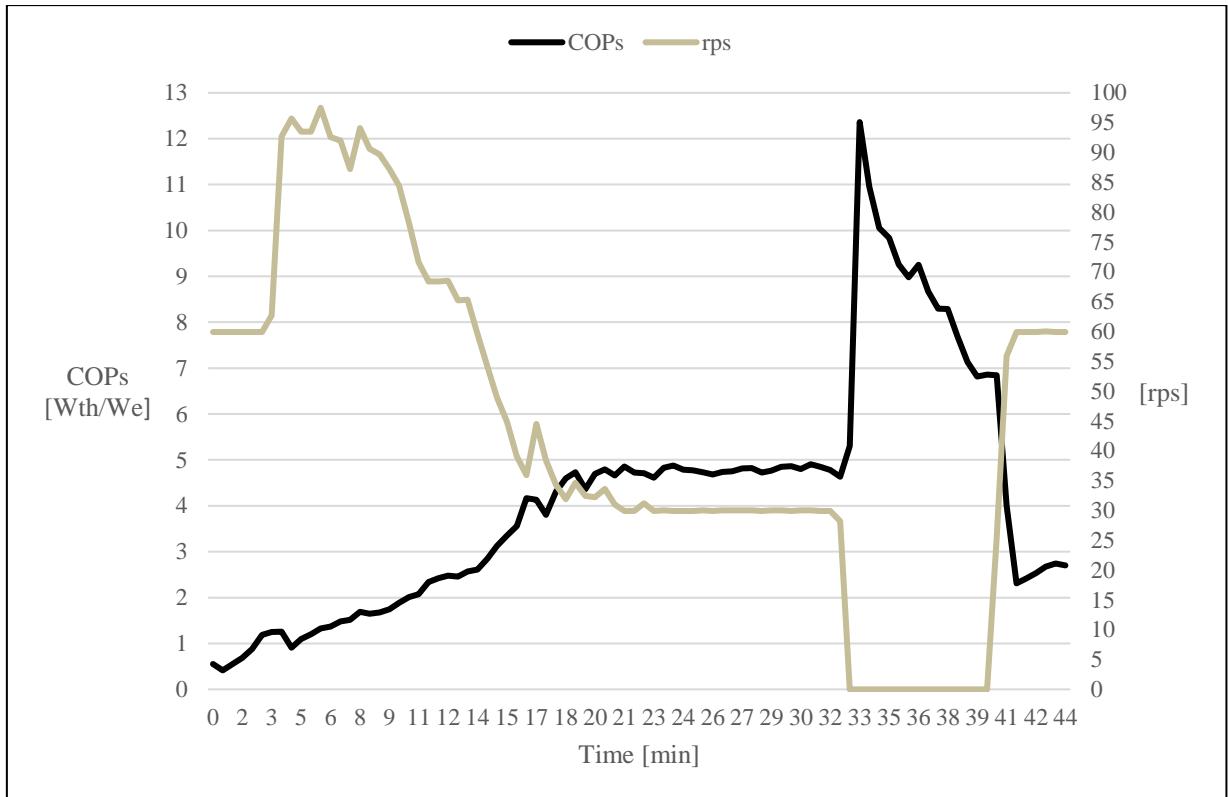


Figure 53: Test 1 – coefficient of performance (COPs) as a function of the compressor speed.

As shown in Figure 53 the COPs, after the initial transitory (about 15 minutes), stabilized at about 4.6, with a compressor speed of 30 rps (that means attenuation: minimum allowable speed). This is a very high energy performance by considering that the temperature difference between supply and outdoor side ($T_s - T_o$) is of about 15 °C and hence the heat supplied by the system is of about 2100 W. When the compressor shuts down COPs experiences a very high and fast increase and its mean value is of about 8.6. By considering this operating cycle the mean COPs is of about 5.4.

Figure 54 shows the air temperatures inside the greenhouse. A heat loss along the recovery duct explains a slightly higher temperature in the greenhouse respect to T_e . However, it could be seen that the internal distribution of temperature is almost homogeneous. A slight gradient of temperature towards the final part of the compartment is revealed. This is mainly due to the return air system configuration. A single intake grid is installed at the beginning of the greenhouse (Figure 48). Thus, the air introduced in the first part of the compartment is also the first to exit. Dividing the return air system with a grid on the beginning side and a second grid on the end side could completely equalize the indoor air temperature, but this was not in the aims of this PhD thesis.

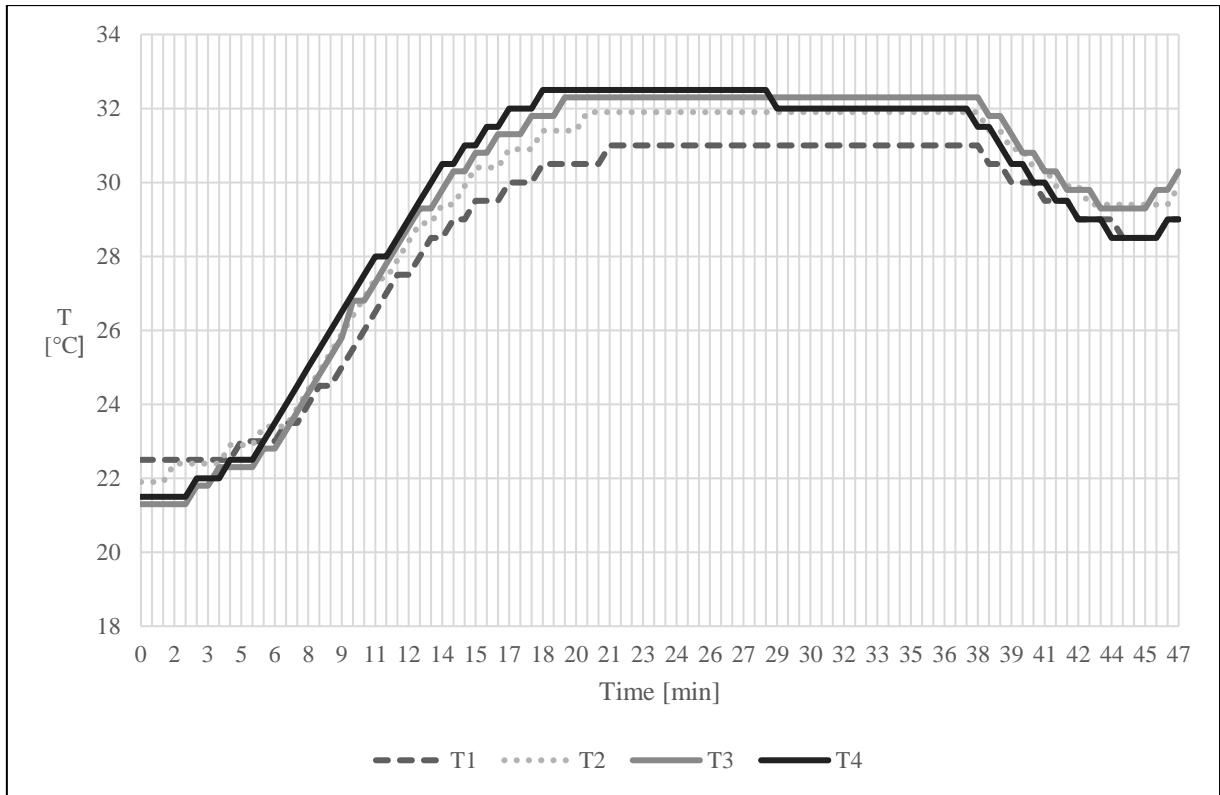


Figure 54: Test 1 – temperature distribution inside greenhouse.

To date the results are encouraging and it confirm that the greenhouse conditioning is possible with this system.

The second test (Test 2) here reported started with an outdoor air temperature T_o of $16.4\text{ }^{\circ}\text{C}$ and had a mean value of $15.7\text{ }^{\circ}\text{C}$ during the whole test. Figure 55 shows a sudden increase of T_o at the beginning of the test, reaching about $20\text{ }^{\circ}\text{C}$, then return to lower values. This fact could be explained by considering that when the fans start, the first amount of suctioned air is the stagnant air inside duct. Particularly, only a portion of duct in charge of suctioning the outdoor air was hit by sun. Regarding to the air inside the greenhouse, the starting value of the exhaust air T_E was $18.4\text{ }^{\circ}\text{C}$. The average value of the supply airflow rate, similarly to the previous one, was $418\text{ m}^3/\text{h}$.

At first sight it is easy to understand that the boundary conditions of this second test are much severe. In fact, remembering that the set point temperature T_{SET} is $27\text{ }^{\circ}\text{C}$, the initial and mean temperature differences are of $10.6\text{ }^{\circ}\text{C}$ and $11.3\text{ }^{\circ}\text{C}$ respectively. For this reason, it can be seen that the compressor was activated from the beginning of the test (Figure 56).

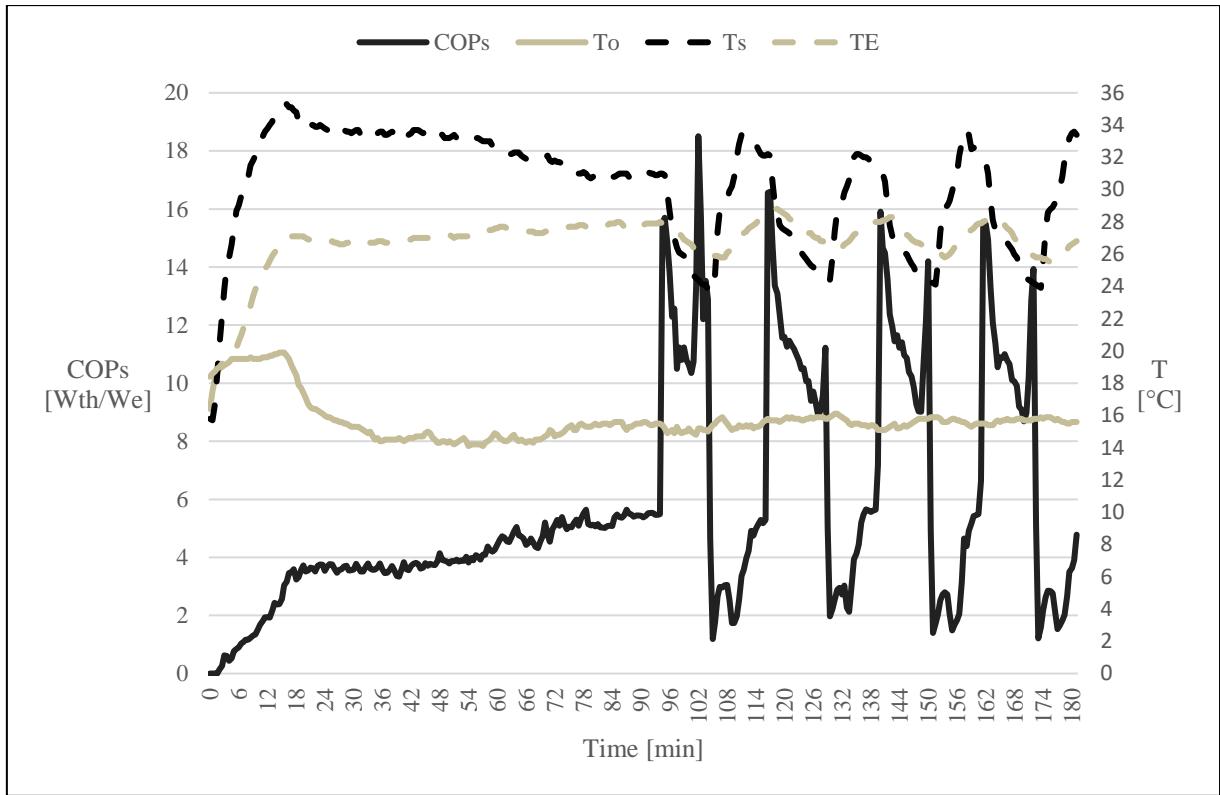


Figure 55: Test 2 – exhaust air temperature (TE), supply air temperature (TS) and coefficient of performance (COPs) as a function of outdoor air temperature (TO).

Figure 55 shows, according to Figure 56, a very fast increase of the supply air temperature T_s . In fact, after few minutes of compressor regime (after the initial ramp) at 60 rps, the compressor experiences a second ramp until the full load (100 rps). Figure 57 displays the heat supplied by SIVeMeC during the test. It can be noted that after the initial transitory, the unit is able to provide a heating load of about 3200 W. In this way the T_E quickly reached the set point of 27 °C. Due to the thermal inertia of the system, even if the compressor was modulated and then attenuated, the exhaust air temperature exceeded the set point. As discussed for experimental tests in laboratory, the high sensitivity of the temperature control is due to a dead band equal to 0 °C ($\Delta T_{DB} = 0$ °C). The control mode does not change within this band. This is why Figure 56 shows frequent fluctuations of temperature across the set point, representing a worst operating condition. A ΔT_{DB} greater than 0 °C could allow to stabilize the control system inside the dead band (across the set point value).

In this study ΔT_{DB} was set to 0 °C to evaluate the performance in the same arrangement of laboratory test. Furthermore, we remember that setting $\Delta T_{DB} = 0$ °C allows the unit to follow the heat load of the greenhouse with high sensitivity.

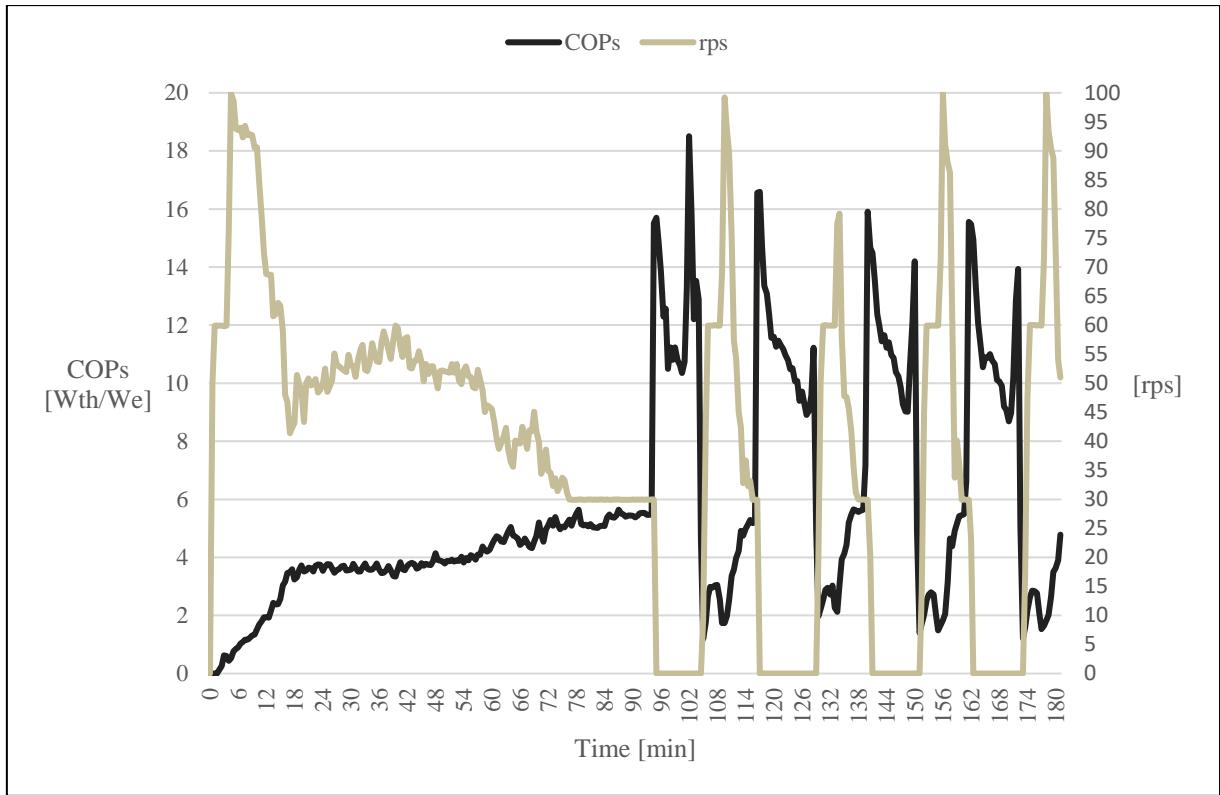


Figure 56: Test 2 – coefficient of performance (COPs) as a function of the compressor speed.

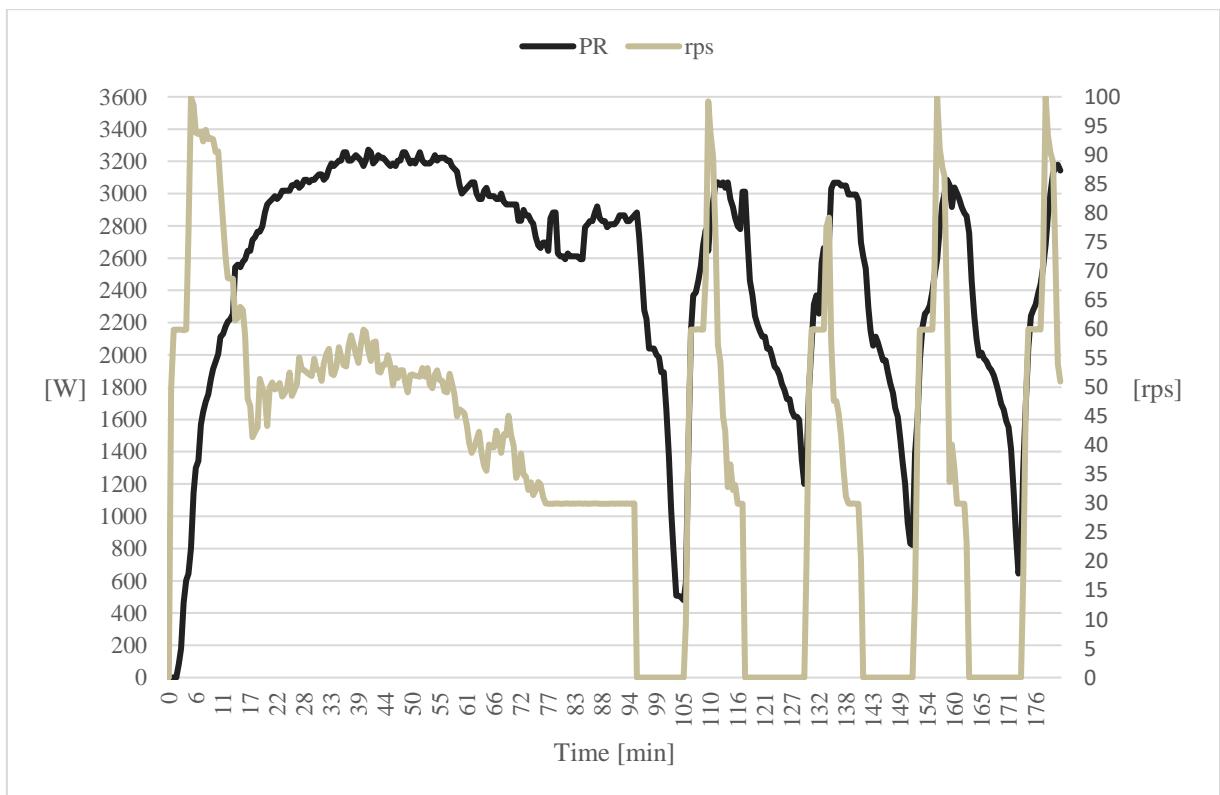


Figure 57: Test 2 – heat supplied by SIVeMeC (P_R) as a function of compressor speed.

The high sensitivity of the system resulted in an on/off behaviour with the compressor ran up to maximum load after each turning on. In fact when T_E reached or over 28 °C, the compressor was turned off and when fell to about 26 °C the compressor started again (Figure 58).

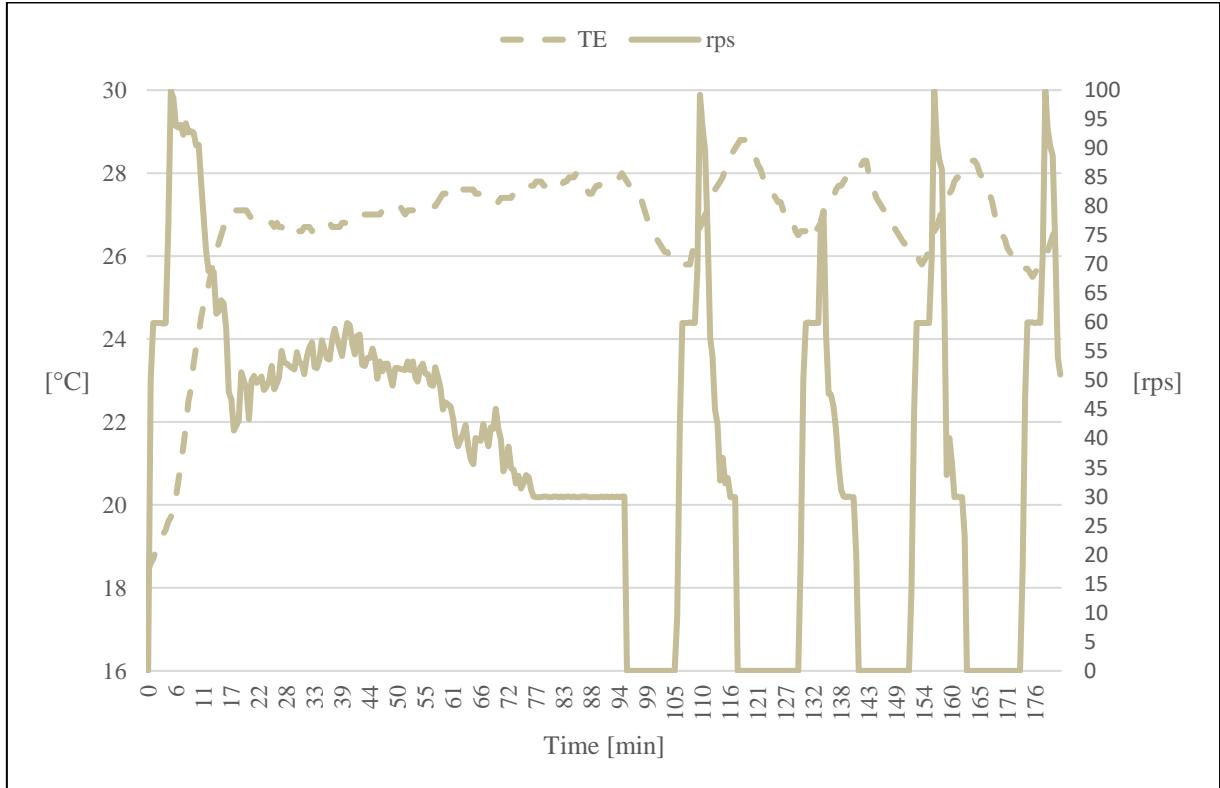


Figure 58: Test 2 – compressor speed as a function of exhaust air temperature T_E .

However the energy performance evaluation of the system during the whole test (about 3 hours) showed a mean COPs of 5.7. This is a very remarkable value by considering that we have no loss of the indoor air temperature control (excessive overheating).

Finally, Figure 59 shows the air temperature inside the greenhouse. As for Test 1, heat loss along the recovery duct explains a higher temperature in the greenhouse respect to T_E . Also in Test 2 it could be seen a gradient of temperature towards the final part of the compartment. This is mainly due as aforementioned to a single intake grid installed at the beginning of the greenhouse (Figure 48). However, by considering T_1 and T_2 it is possible to note a very similar trend. Maybe this fact is related to the duration of test. After 3 hours of heat supplied by the system, the temperature values inside the greenhouse start to conform each other.

In order to facilitate a fast uniformity of indoor temperature, the distribution system, particularly the return side, should be re-designed, but as said above this is not in the aims of the study.

Additional investigation could lead to a further adjustment of the regulation of the unit. For example, it is possible to set an appropriate offset on the exhaust air temperature to avoid a higher value of the internal air temperature than desired one, or by using an indoor air temperature probe to regulate the supply air temperature. Moreover, a $\Delta T_{DB} = 2$ °C, as for the civil sector, can be set to prevent unwanted fluctuations.

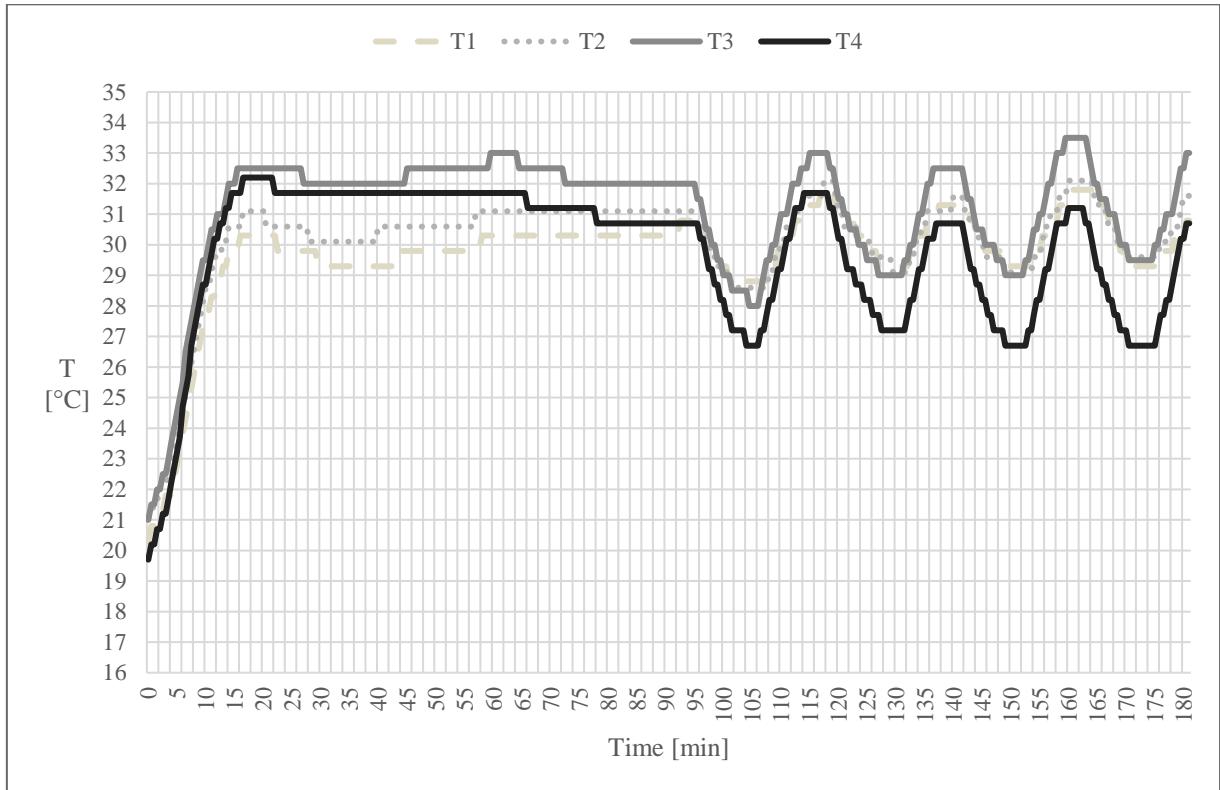


Figure 59: Test 2 – temperature distribution inside greenhouse.

6 CONCLUSIONS

In this PhD thesis, an integrated mechanical ventilation system with heat pump was designed, realised and studied. The system can work according to different operating modes using or not either the static heat exchanger or the heat pump. In the present study, system performance was evaluated in static plus thermodynamic recovery arrangement. In this way, a comparison with other prototypes found in literature and actual commercial systems was possible, considering worst conditions for tested prototype and best ones for the other [15].

The mechanical ventilation prototype was firstly studied in laboratory, by reproducing the worst-case boundary conditions in two climatic rooms, and then in real case, by installing the unit at service of a mini-tunnel greenhouse type.

The system test in laboratory showed an average overall coefficient of performance COPs of 9.5, 8.9 and 6.6 at $T_o = 0^\circ\text{C}$, 5°C and 10°C (outdoor air temperature) respectively. Particularly, at $T_o = 0^\circ\text{C}$, the static recovery worked efficiently (η equal to 86.2 %), making it possible to reduce the use of the heat pump in attenuation mode (i.e. when the compressor runs at its minimum allowable speed) for almost the entire cycle duration. On the contrary, when the outdoor air temperature increases ($T_o = 5^\circ\text{C}$ and $T_o = 10^\circ\text{C}$) the compressor was turned off during the last period of these two trials. This is due to a close control of the supply air temperature caused by the “dead band” setting equal to zero ($\Delta T_{DB} = 0^\circ\text{C}$). At $T_o = -5^\circ\text{C}$ the system behaves as an on/off type due to a safety mechanism to prevent frosting on the evaporator. This means that an upper and a lower limit value of external temperature exist, making it necessary to change the operating setting: this is not at all a common practice both in existing prototypes and in commercial systems. For values below the lower limit, it is possible to operate with the only passive recovery and a heating system at service of the confined environment. On the other hand, for temperature values higher than the upper limit, it is possible to operate with the only active recovery, until the outdoor conditions will allow freeheating or freecooling.

Ultimately, the global system energy performance (COPs) appears to be excellent if compared with that of typical units equipped with an on/off compressor and a fixed speed fans both commercial and prototypes found in literature. This becomes more evident when considering that many data sheets provided by manufacturers qualify the COPs without the electric consumption due to the ventilation. Furthermore, with increasing of the outdoor temperature (in winter conditions), the data are calculated with high values of supply air temperature, so as to have high heat supplied P_R which is hardly useful to the purpose. In fact, a supply air

temperature higher than the designed one causes an increase of indoor air temperature above the set point. This is neither useful for the comfort nor for energy saving.

The possible future developments involves the investigation of the optimal values of temperature at which switching between different operating modes of the unit is preferable. Most likely, these temperature values are not constant but they are a function of outdoor air temperature, airflow rate (both supply and discharge ones), indoor temperature set point and other plant systems installed at service of the building. So, a new concept in the operating logic is required to make the system fully adaptable to the intended use.

Energy savings to manage the internal microclimate conditions of greenhouses is becoming increasingly important. Thus, this new concept of mechanical ventilation with heat recovery and heat pump was installed at service of a mini-tunnel greenhouse. The experimental apparatus involves also two supervisor systems, one for the monitoring of the field (PlantWatch PRO®), the other for managing the mechanical ventilation unit (pCOManger®). The ventilation unit should allow a proper control of indoor climate condition (temperature, relative humidity and CO₂).

A first set of tests on the regulation of the indoor air temperature were carried out with a set point of 27 °C. The results show a suitable regulation when the reference probe is that installed on the recovery side, while an offset of few Celsius degree is observed with the probes inside the greenhouse. This is due to the heat loss along the recovery duct and the configuration of the recovery system (only one intake grid installed at the beginning of the greenhouse).

It can be concluded that the greenhouse conditioning is possible with this system which had also shown remarkable energy performance: COPs of 5.4 and 5.7 at outdoor air temperature of 18.0 °C and 15.7 °C respectively.

Future tests will allow to better set up the system for a real application and make an energy comparison with other air-conditioning systems leading to an economical evaluation in terms of energy saving and increasing of crop yield.

REFERENCES

- [1] L. Albright, Environment Control for Animals and Plants, 2950 Niles Road, St, Joseph, Michigan 49085-9659 USA, 1990.
- [2] V. Sethi e S. Sharma, «Survey and evaluation of heating technologies for worldwide agricultural greenhouse applications,» *Solar Energy*, n. 82, pp. 832-859, 2008.
- [3] G. Rai, Solar Energy Utilization, vol. III, New Delhi: Khanna Publisher, 1997, pp. 418-441.
- [4] M. Gupta e P. Chandra, «Effect of greenhouse design parameters on conservation of energy for greenhouse environmental control,» *Energy*, n. 8, pp. 777-797, 2002.
- [5] M. Santamouris, C. Balaras, E. Deskalaki e M. Vallindras, «Passive solar agricultural greenhouses: a worldwide classification and evaluation of technologies and systems used for heating purpose,» *Solar Energy*, n. 53, pp. 411-426, 1994.
- [6] M. Santamouris, «Active solar agricultural greenhouse. The state of art,» *International Journal of Solar Energy*, n. 14, pp. 19-32, 1993.
- [7] D. Jain e G. Tiwari, «Modelling and optimal design of ground air collector for heating in controlled environment greenhouse,» *Energy Conversion and Management*, n. 44, pp. 1357-1372, 2003.
- [8] R. Singh e G. Tiwari, «Thermal heating of controlled environment greenhouse: a transient analysis,» *Energy*, n. 41, pp. 505-507, 2000.
- [9] J. White e R. Aldrich, «Progress report on energy conservation for greenhouses research,» *Floriculture Review*, n. 41, pp. 505-507, 1975.
- [10] D. Willis, «Cooling fan ventilated greenhouse: a modeling study,» *Biosystems Engineering*, n. 84, pp. 315-329, 2003.
- [11] V. Sethi e S. Sharma, «Survey of cooling technologies for worldwide agricultural greenhouse applications,» *Solar Energy*, n. 81, pp. 1447-1459, 2007.

- [12] G. Giacomelli, M. Ginger, A. Krass e D. Mears, «Improved methods of greenhouse evaporative cooling,» *Acta Horticulturae*, n. 257, pp. 29-41, 1985.
- [13] M. Coomans, K. Allaerts, L. Wittemans e D. Pinxteren, «Monitoring and energetic performance of two similar semi-closed greenhouse ventilation system,» *Energy Conversion and Management*, n. 76, pp. 128-136, 2013.
- [14] N. V. d. Bulk, M. Coomans, L. Wittemans, J. Hanssens e K. Steppe, «Monitoring and energetic analysis of an innovative ventilation concept in a Belgian greenhouse,» *Energy and Buildings*, n. 57, pp. 51-57, 2013.
- [15] F. Fucci, C. Perone, G. L. Fianza, L. Brunetti, F. Giometta e P. Catalano, «Study of a prototype of an advanced mechanical ventilation system with heat recovery integrated by heat pump,» *Energy and Buildings*, n. 133, pp. 111-121, 2016.
- [16] J. Dieckmann, K. Roth e J. Brodrik, «Air-To-Air Energy Recovery Heat Exchanger,» *ASHRAE Journal*, n. 45, pp. 57-58, August 2003.
- [17] A. Mardiana-indayu e S. Riffat, «Review on heat recovery technologies for building applications,» *Renewable and Sustainable Energy Review*, n. 16, pp. 1241-1255, 2012.
- [18] ASHRAE, «Chap. 44, Air-to-Air Energy.,» in *ASHRAE Handbook of HVAC Systems and Equipment*.
- [19] J. Wallin, H. Madani e J. Claesson, «Run-around coil ventilation heat recovery system: A comparative study between different system configurations,» *Applied Energy*, n. 90, pp. 258-265, 2012.
- [20] G. Fracastoro e M. Serraino, «Energy analyses of buildings equipped with exhaust air heat pumps (EAHP),» *Energy and Buildings*, n. 42, pp. 1283-1289, 2010.
- [21] S. Riffat e M. Gillott, «Performance of a novel mechanical ventilation heat recovery heat pump system,» *Applied Thermal Engineering*, n. 22, pp. 839-845, 2002.
- [22] A. Nguyen, Y. Kim e Y. Shin, «Experimental study of sensible heat recovery of heat pump during heating and ventilation,» *International Journal of Refrigeration*, n. 28, pp. 242-252, 2005.

- [23] A. D'Este, A. Gastaldello e L. Schibuola, «Energy saving in building ventilation,» *Ecosystems and Sustainable Development V*, n. 81, pp. 335-344, 2005.
- [24] COACH AX – CentraLine AX – Engineering Tool Set, Product Data. *CentraLine by Honeywell*.
- [25] ARENA AX – BACnet Supervisor, Product Data. *CentraLine by Honeywell*.
- [26] HAWK 600E, Product Data. *CentraLine by Honeywell*.
- [27] A. Mardiana e S. Riffat, «Review on physical and performance parameters of heat recovery systems for building applications.,» *Renewable and Sustainable Energy Reviews*, n. 28, pp. 174-190, 2013.
- [28] C.-A. Roulet, F. Heidt, F. Foradini e M.-C. Pibiri, «Real heat recovery with air handling units,» *Energy and Buildings*, n. 33, pp. 495-502, 2001.
- [29] H. Han, Y. Ghoo e Y. Kwon, «An Experimental Study on the Effect of Outdoor Temperature and Humidity Conditions on the Performance of a Heat Recovery Ventilator,» in *Proceedings of Clima 2007 WellBeing Indoors..*
- [30] R. Besant e C. Simonson, «Air-To-Air Energy Recovery,» *ASHRAE Journal*, n. 42.5, pp. 31-38, 2000.
- [31] M. Gustafsson, G. Dermentiz, J. Myhren e C. Bales, «Energy performance comparison of three innovative HVAC systems for renovation through dynamic simulation,» *Energy and Buildings*, n. 82, pp. 512-519, 2014.
- [32] E. Juodis, «Extracted ventilation air heat recovery efficiency as a function of a building's thermal properties,» *Energy and Buildings*, n. 38, pp. 568-573, 2006.
- [33] Kevin D. Rafferty, P.E., *Geothermal Direct Use Engineering and Design Guidebook Chapter 14* (1998), OIT Geo-Heat Center Klamath Falls, OR 97601, p. Guidebook Chapter 14.

- [34] Campiotti C.A., 2013. *La risorsa rinnovabile per la sostenibilità ambientale ed energetica della filiera dei sistemi serra in Italia. Tesi di dottorato in Tecnologie per la sostenibilità ed il risanamento ambientale*, Palermo..
- [35] ENEA, 2014. *CERTIFICATI BIANCHI. Guida Operativa per la scheda 40E – Gli incentivi del DM 28 dicembre 2012 sui sistemi serra..*
- [36] C. Perone, F. Fucci, G. L. Fianza, L. Brunetti, F. Giometta, P. Catalano e B. Bianchi, «Experimental Study of a Mechanical Ventilation System in a Greenhouse,» *Chemical Engineering Transactions*, n. 58, pp. 811-816, 2017.

STANDARDS

EN 308:1997 – Heat exchangers – test procedures for establishing performance of air to air and flue gases

UNI/TR 10349-2:2016 – Riscaldamento e raffrescamento degli edifici - Dati climatici - Parte 2: Dati di progetto