Designing for comfort and energy savings through the use of heat recovery

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SUMMARY

The air quality in a building is one of the main factors in assuring comfort in the interior. The air must be continuously exchanged to keep it clean. To change the air, a given amount must be exhausted from the conditioned space at the comfort temperature and be replaced by an equivalent amount of outside air.

To bring this fresh air to the comfort temperature, a certain amount of energy must be spent. The consumption of energy can be reduced by using a heat recovery system between the exhaust and supply air flows. The primary source for this energy is the heat in the exhaust air flow, which would otherwise be wasted. As VDI 3803 states: *"the use of waste heat can be equated, at least in terms of energy use, with renewable energy sources"*.

This paper defines the classification of heat recovery systems and determines that buildings must be equipped with a heat recovery system of at least class H3, as set out in UNI EN 13053.

In addition, as an aid to the designer, we provide a method for calculating the energy class of a heat recovery system. This paper concludes by illustrating the potential energy savings, net of energy spent in transporting the air through the heat recovery unit, for two climate zones in the North and South of Italy.

1. INTRODUCTION

Indoor air quality (IAQ) is a fundamental factor to ensure comfort in a building. The increasing attention to energy efficiency has led to new buildings that are more and more air tight. The integrated design of a controlled mechanical ventilation system has to minimize the level of contaminants, gas, and airborne particulate inside the building. UNI 10339 (UNI 1995) defines the minimum quantity of outdoor air required for proper air exchange. This means that a certain amount of air, at the comfort temperature, must be exhausted while an equal amount of air at the outdoor temperature, must be brought to the comfort temperature and supplied to the interior.

This process consumes a considerable amount of energy. However, the energy costs of the process can be reduced with a heat recovery unit. Such units recover the heat energy between the two air streams passing through it. The exhaust air gives up a large part of its thermal energy to the intake air, which returns it to the interior of the space.

In such systems, the exhaust air acts as a source of renewable energy. As stated in VDI 3803 (VDI 2012): *"the use of waste heat can be equated, at least in terms of energy use, with renewable energy sources"*.

2. INDOOR AIR QUALITY

UNI 10339 provides guidelines regarding the amount of outdoor air required to ensure a proper exchange of air and hence comfort indoors. For example, to renew the air in a supermarket with a floor area of 1200m², a supply airflow of 10000m³/h of outdoor air is required.

2.1. Air exchange

Air exchange consists of exhausting a certain quantity of air at the comfort temperature from the conditioned interior and replacing it with the same amount of air at the outdoor temperature.

This means that a certain amount of energy is consumed in bringing the outdoor air to the thermal and hygrometric conditions required to reduce the indoor thermal load.

However, this energy consumption can be reduced by installing a heat recovery unit between the two air streams.

2.2. Exhausted heat as a renewable source

In reality, the exhaust air is a renewable heat source, as stated in VDI 3803. By placing a heat recovery unit between the supply and exhaust air streams, we can recover energy which would otherwise be wasted, reducing the energy consumption of the air conditioning system by more than 50%.

2.3. Performance of heat recovery units

Heat recovery units are characterised by their performance ratings at certain operating conditions:

- Temperature effectiveness[1];
- Pressure drop.

The temperature effectiveness indicates the capacity of the exchanger to recover heat from the exhaust air, while the pressure drop is the loss of energy of the airflow caused by its passing through the exchanger itself.

UNI EN 308 (UNI 1998) and ASHRAE 84-1991 (ASHRAE 1992) define a method for measuring these indices and calculating them.

In parallel, UNI EN 13053 (UNI 2011) introduces a single coefficient for determining the exchanger's performance, called *energy efficiency*. This is equal to the temperature effectiveness, corrected by the ratio between the capacity of the heat recovery system and the electrical power consumed.

3. ENERGY EFFICIENCY

3.1. Calculation

The method to calculate the energy efficiency as given in UNI EN 13053 as shown below.

[1] In this work, and in the standards to which it refers, the terms “temperature ratio” and “temperature effectiveness” are synonyms; they coincide when the supply and extract flow are equal, as is assumed in this article.

Given:
- T_{11} [°C] exhaust temperature;
- T_{21} [°C] outside air temperature;
- T_{22} [°C] supply temperature ;
- ηt recovery unit temperature ratio.

Table I shows the reference values specified in the standard.

Table I: Reference values for calculating the energy efficiency according to UNI EN 13053.

T_{11} = 25°C	T_{21} = 5°C
Balanced flow between supply and exhaust (1:1)	
Dry air, without condensation	

We have:

$$\eta_t=((T_{22}-T_{21}))/((T_{11}-T_{21})) \quad (1)$$

The total pressure drop in the recovery unit Δp_{HRS} [Pa] is the sum of the two pressure drops Δp_{supply} [Pa] and $\Delta p_{\text{exhaust}}$ [Pa] of the supply and exhaust air streams respectively.

$$\Delta p_{HRS}= \Delta p_{\text{supply}}+ \Delta p_{\text{exhaust}} \quad (2)$$

The temperature ratio and pressure drops are characteristic indices of the heat exchanger. To determine these values in this article, we use a specific calculation program, available online free of charge.

These values have been calculated with the above referenced values. The resulting characteristic data are therefore not applicable to all operating conditions.

Given:
- q_v [m^3/s] air flow;
- η_D overall static efficiency of electrical system;
- P_{el} [W] electrical power input;
- $P_{el, aux}$ [W] electrical power input of auxiliary equipment (as applicable).

We calculate:

$$P_{el}=q_v*\Delta p_{HRS}*1/\eta_D +P_{el, aux} \quad (3)$$

The electrical power (P_{el}) thus depends on the flow (q_v), the total pressure losses (Δp_{HRS}) and the efficiency of the electrical system (η_D). If not otherwise specified, we can consider the latter to be 0.6. If the recovery system includes moving mechanical assemblies, one must also consider the consumption of such components: these will include the motor driving rotary recovery units, for instance.

Once the electrical power required to operate the energy recovery system has been evaluated, we can calculate the coefficient of performance (COP).

Given:

- Q_{HRS} [W] capacity of the heat recovery system;
- ρ_a [kg/m^3] air density;
- cp_a [J/(kgK)] specific heat of air.

$$Q_{HRS} = q_v * \rho_a * cp_a * (T_{22} - T_{21}) \quad (4)$$

We calculate:

- COP coefficient of performance.

$$COP = Q_{HRS} / P_{el} \quad (5)$$

The coefficient of performance represents the benefit obtained from using a heat recovery unit, per unit of power consumption.

We now have everything we need to calculate the energy efficiency:

$$\eta_e = \eta_t \, (1 - 1/COP) \quad (6)$$

Equation (6) thus allows us to calculate an index which gives global information about the performance of the heat recovery unit. Indeed, the temperature ratio, calculated for dry air without condensation, is reduced by consumption due to the loss of energy caused by the system's operation. The calculation of this index enables us to determine the energy class of the unit, using the following Table II.

Table II: Intervals for the determination of the energy class.

Class	$\eta_{e\ 1:1min}$ [%]
H1	≥ 71
H2	≥ 64
H3	≥ 55
H4	≥ 45
H5	≥ 36
H6	no requirement

To speed up the design process, we have generated the Figure 1 graph which shows, for three energy classes (H3, H2 and H1), and as a function of the total pressure drop in the recovery system, the temperature ratio and supply air temperature of the recovery unit, for the energy classes in question.

These graphs have been calculated in relation to the definition of energy efficiency set out by UNI EN 13053. We have derived a constant "*k*", equation (7), which is system dependent, and equation (8), which gives the limit curves for the energy classes.

$$k = cp_a * \rho_a * \eta_D \quad (7)$$

$$\eta_t = \eta_e + \Delta p_{HRS} / (k * (T_{11} - T_{21})) \quad (8)$$

In this article, the constant *k* has been calculated in relation to the following operating conditions:

- η_D=0.6;
- $P_{el,\ aux}$=0 W;
- cp_a =1004 J/(kg K);
- ρ_a=1.2 kg/m^3.

In particular, the diagram gives the values for energy class H3, which is the minimum acceptable class pursuant to VDI 3803, which states *"…all air-conditioning systems that are installed in a building or refurbished must be equipped with a heat recovery that at least corresponds to the classification H3…"*, while adding the two superior classes H2 and H1, to highlight how, for these classes, one must use recovery units with higher temperature ratios.

The adoption of these superior classes is in line with the trend, in European countries, towards greater energy savings.

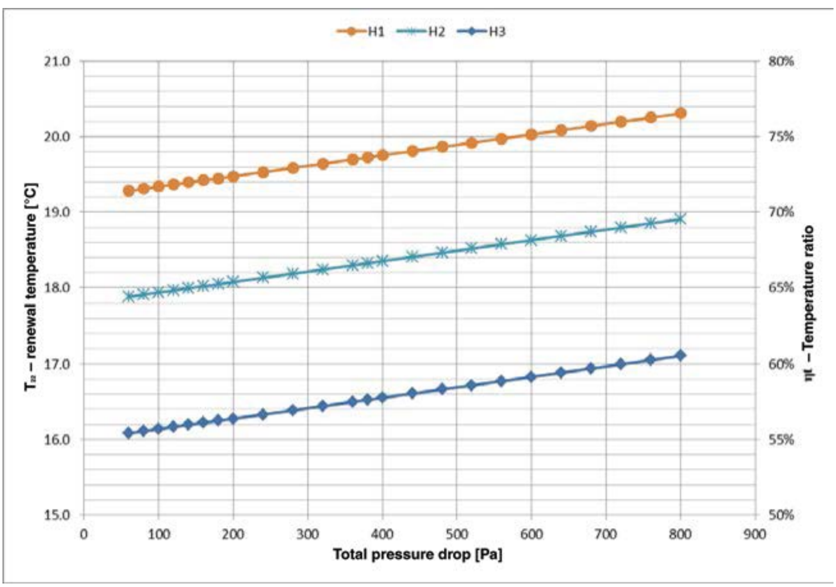


Figure 1: Energy class limit curves.

3.2. Example calculation

We give an example calculation below as an aid to understanding the method used in calculating the energy class.

The following case is based on the technical data of a commercial product whose specifications are given in Figure 3. This type of heat recovery unit is characterised by a plate geometry which ensures optimally turbulent flow, as can be seen in Figure 2.

Figure 2: Detail of the plate geometry of the heat recovery unit in question.

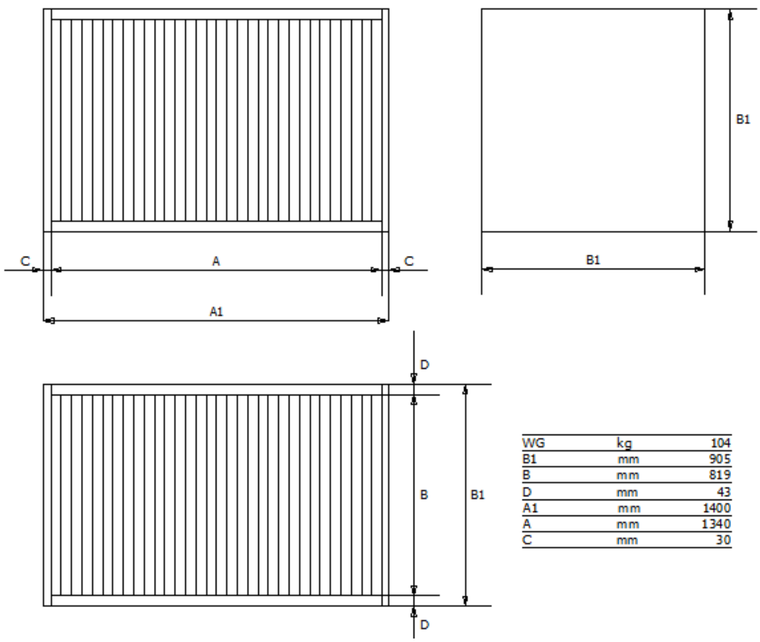


Figure 3: Geometric characteristics of the class H3 heat exchanger used in the analysis (§4.1, §4.2, §3.2).

Table III: Inputs to the example calculation.

Symbol	Value	Unit	Definition
T_{11}	25	°C	Exhaust air temperature
T_{21}	5	°C	Outside air temperature
q_v	2.78	m³/s	Air flow
η_D	0.75		Electrical energy efficiency
$P_{el,aux}$	0	W	Auxiliary equipment power

Since the plate recovery unit is a static heat exchanger, the system has no auxiliary equipment of interest, so we set $P_{el,\ aux}$ to zero. The air flow (**q**$_v$) is that required for a proper exchange of air in a supermarket with a floor space of 1200m² (10000m³/h). In the example, we consider the efficiency of the electrical system to be 0.75, as in the following calculations.

Using the performance calculation software, we obtain the following results, as shown in Figure 4 and Table IV.

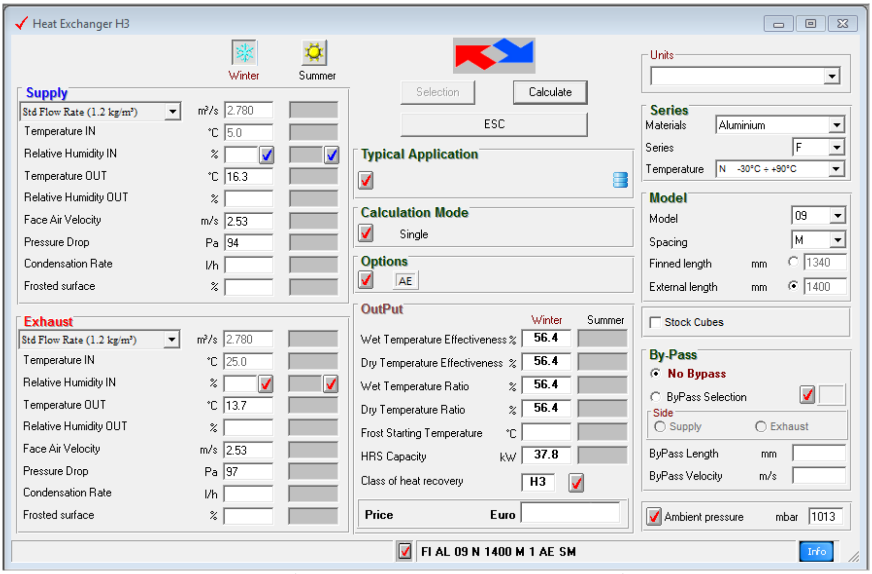


Figure 4: Screenshot of class H3 heat exchanger selection software (§4.1, §4.2).

Table IV: Output of calculation.

Symbol	Value	Unit	Definition
T_{22}	16.3	°C	Supply air temperature
η_t	56.4	%	Temperature ratio
Q_{HRS}	37800	W	Capacity of the heat recovery system
$\Delta p_{renewal}$	94	Pa	Supply pressure drop
$\Delta p_{expulsion}$	97	Pa	Exhaust pressure drop

The software calculates in and out air temperature, temperature ratio, pressure drop of both flows and the capacity; the results are given in Table IV. We conclude the example by calculating the energy efficiency below.

Using the above formulas, we obtain the following results, summarised in Table V.

Table V: Results calculated with equations (2), (3), (5). and (6).

Symbol	Value	Formula	Unit	Definition
Δp_{HRS}	191	=94+97	Pa	Sum of pressure losses
P_{el}	708	=2.78*191*1/0.75+0	W	Electric power consumption
COP	53.4	=37800/708		Coefficient of performance
η_e	55.3	=56.4*(1-1/53.4)	%	Energy efficiency

From the values obtained in Table V and with reference to Table II, we can conclude that the recovery unit is in class H3, as can be seen from Figure 5.

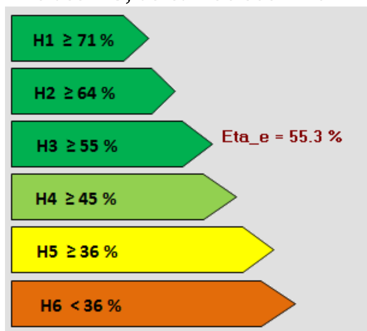


Figure 5: Energy class.

4. EVALUATING THE ENERGY SAVINGS

Once we have determined the recovery unit's energy class, it is interesting to calculate the energy savings net of the energy supplied to the air in driving it through the heat exchanger itself, on a variety of configurations in different climatic zones of Northern and Southern Italy.

The analysis is divided into three sections:

1. Comparison of the value of the investment at Milan and Palermo for the installation of a class H3 recovery unit;
2. Calculation of the benefit of installing an evaporative cooling system on a class H3 recovery system in Palermo;
3. The increase in energy savings in Milan achieved by replacing the class H3 recovery unit with a class H1 unit.

The supply and exhaust air flow passing through the heat recovery unit has been calculated using the terms of UNI 10339 for a commercial facility with floor space 1235m².

The above-mentioned standard defines an occupancy factor for the facility of 0.25pp/m². Each occupant requires an air exchange of $9*10^{-3}$m³/s. This yields a supply and exhaust air flow equal to 10000m³/h (2.78m³/s).

The analysis has been run in two different areas of Italy to highlight the effect of the outdoor temperature on energy savings and, hence, on return on investment.

To analyse the energy savings, we must define the type of air heating and cooling system. For the heating part, we have considered a natural gas boiler with efficiency of 85% and fuel cost of 0.50€/Nm³. In the warm season, the air conditioning plant is characterised by a chiller with cooling performance coefficient of 3 and electrical power cost of 0.18€/kWh. The fan required to compensate for the pressure drop in the recovery unit has an efficiency of 0.75. The system runs Monday to Saturday, 11 hours per day.

The cost of the investment is not simply equal to the cost of the heat recovery unit, but this option increases costs as a result of increased fan power and the number of profiles required to construct the air handling unit itself. To proceed with the analysis, we have assumed the increase of cost due to the installation of the recovery unit to be equal to the value of the heat exchanger itself: 973€.

Figure 6: Map of Italy. The cities covered by the analysis have been circled: Milan and Palermo.

Milan and Palermo temperature profiles in 2013 have been taken from a database[2]; Figure 7 shows the results in graphic form.

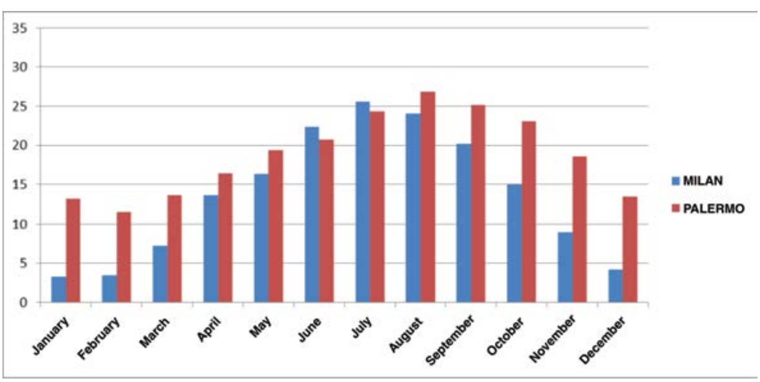


Figure 7: Temperature profile for Milan and Palermo, year 2013.

As can be seen from Figure 7, Milan has a wider temperature range, with winter temperatures lower by around 10°C than Palermo.

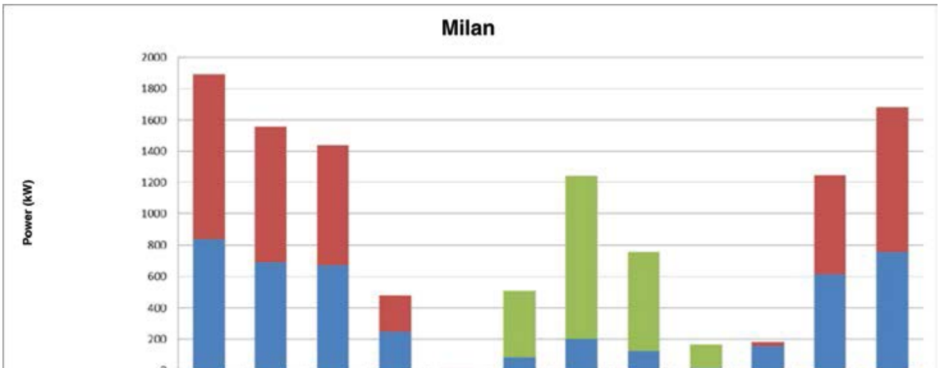
We have assumed indoor temperatures of 20°C and 24°C in the winter and summer respectively.

[2] archivio-meteo.distile.it/.

4.1. Milan and Palermo, class H3 recovery unit

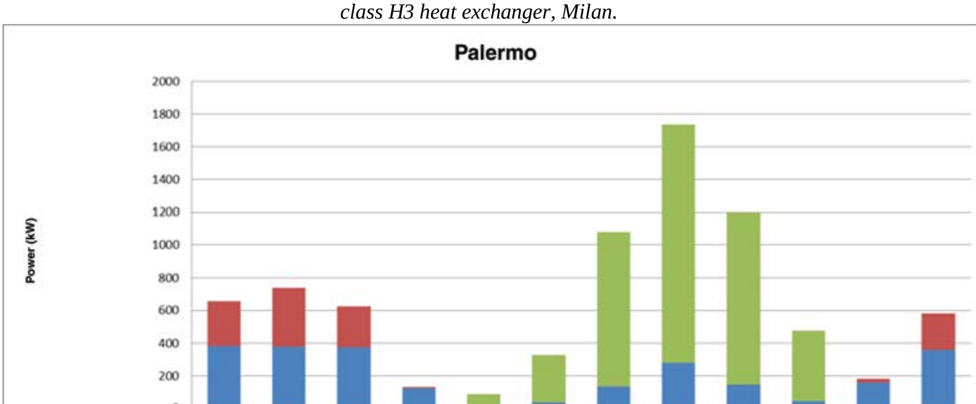
As defined in the introduction, the analysis considers a heat recovery unit with the geometric configuration given in Figure 3.

We used recursively the dynamic link library of the selection program to calculate all the operating conditions of the recovery unit for the specified temperatures and humidities, day by day, thus giving the values of temperature effectiveness and pressure drop for every day of the year in question. We then calculated the energy savings, obtaining the values given in Figure 8 and Figure 9 for Milan and Palermo respectively.



	January	February	March	April	May	June	July	August	September	October	November	December
Chiller (kW)	0	0	0	0	0	426	1036	632	153	0	0	0
Boiler (kW)	1054	861	762	232	0	0	0	0	0	27	631	925
Power recovery H3 (kW)	838	692	674	246	0	80	203	126	9	153	614	755

Figure 8: Monthly power draw of powered mechanical ventilation system, and power recovered with class H3 heat exchanger, Milan.



	January	February	March	April	May	June	July	August	September	October	November	December
Chiller (kW)	0	0	0	0	78	291	941	1460	1048	432	0	0
Boiler (kW)	273	360	247	9	0	0	0	0	0	0	25	223
Power recovery H3 (kW)	383	377	375	122	9	36	136	278	146	45	159	356

Figure 9: Monthly power draw of powered mechanical ventilation system, and power recovered with class H3 heat exchanger, Palermo.

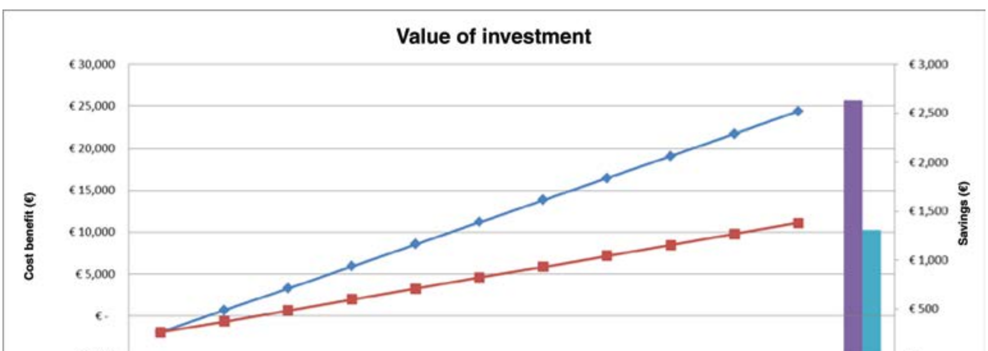
It is evident from the above figures that the total power draw is partly supplied by the heat exchanger, supplemented by the air conditioning system.

Note also that the energy recovery in Milan is greater during the winter and lesser during the summer than in Palermo.

During the intermediate seasons, the heat recovery is negligible or zero. Thanks to the mild outdoor temperature, the interior can be air conditioned with free cooling, by fitting a ByPass in the exchanger, driven by the controller (Figure 10).

Figure 10: Heat recovery unit equipped with ByPass.

To evaluate the cost savings, we have shown the value of the investment over several years in Figure 11.



	0	1	2	3	4	5	6	7	8	9	10	Annual savings
MILAN H3	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ 2,632
PALERMO H3	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ 1,304
MILAN H3	-€ 1,946	€ 686	€ 3,318	€ 5,950	€ 8,583	€ 11,215	€ 13,847	€ 16,479	€ 19,111	€ 21,743	€ 24,376	
PALERMO H3	-€ 1,946	-€ 642	€ 662	€ 1,967	€ 3,271	€ 4,575	€ 5,879	€ 7,183	€ 8,487	€ 9,792	€ 11,096	

Figure 11: Graph of value of investment over years of operation and annual savings, Milan and Palermo, class H3 recovery unit.

Lower energy recovery means lower annual savings, resulting in a longer time required to pay back the investment. In particular, the investment is paid back in around 0.7 years in Milan, while in Palermo it takes 1.5 years to start turning a profit. If we exploit the heat in the exhaust air flow, we save 2632€ and 1304€ a year in Milan and Palermo respectively.

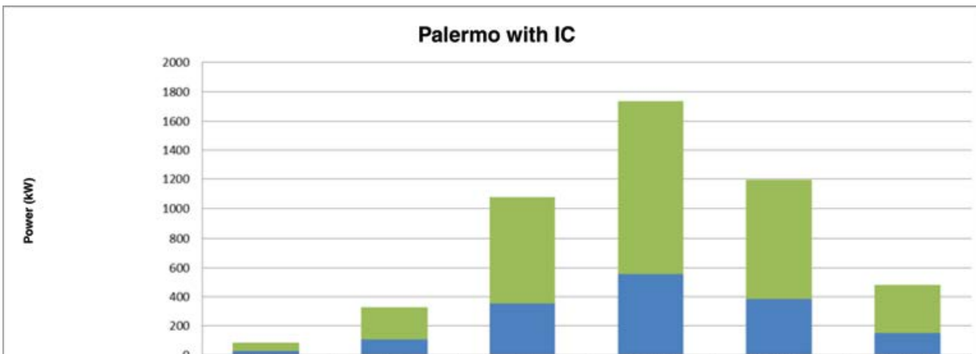
4.2. Palermo, class H3 recovery unit and evaporative cooling

The energy recovery during the summer can be increased by installing an adiabatic humidification system in the exhaust air stream, with a saturation efficiency of 0.90 (equation (9)) for instance.

$$\eta_{sat}=((x_2-x_1))/((x_S-x_1)) \quad (9)$$

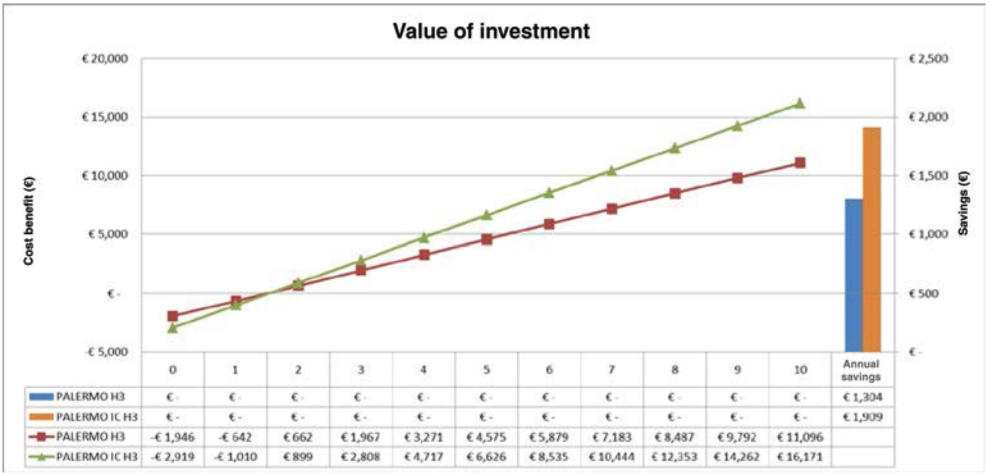
Indirect evaporative cooling can be used to increase the thermal differential between the supply air and the exhaust air. If the exhaust air is humidified, the nebulised water evaporates, thus absorbing the required latent heat and cooling the air.

In particular, by using a proper exhaust side humidification system the recovery air temperature decreases from 24°C to 19.1°C. The result is a greater temperature difference between the two air streams, which yields higher energy savings throughout the summer, as can be seen from Figure 12.



	May	June	July	August	September	October
Chiller (kW)	59	216	725	1179	809	323
Power recovery H3 IC (kW)	28	110	352	558	385	154

Figure 12: Monthly power draw of powered mechanical ventilation system, and power recovered with class H3heat exchanger equipped with evaporative cooling, Palermo.



	0	1	2	3	4	5	6	7	8	9	10	Annual savings
PALERMO H3	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ 1,304
PALERMO IC H3	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ -	€ 1,909
PALERMO H3	-€ 1,946	-€ 642	€ 662	€ 1,967	€ 3,271	€ 4,575	€ 5,879	€ 7,183	€ 8,487	€ 9,792	€ 11,096	
PALERMO IC H3	-€ 2,919	-€ 1,010	€ 899	€ 2,808	€ 4,717	€ 6,626	€ 8,535	€ 10,444	€ 12,353	€ 14,262	€ 16,171	

Figure 13: Cost benefits of class H3 recovery unit, with and without evaporative cooling (IC), Palermo.

We have assumed the cost of installing an evaporative cooling system to be equal to that of the heat exchanger itself.

To evaluate the cost savings, we can see from Figure 13 that the energy savings increase and the investment is paid back in around 1.5 years, notwithstanding the higher cost of the installation.

4.3. Milan, class H1 recovery unit

Milan is much colder in the winter than Palermo. Since the temperature differential is much greater during the winter, it is of interest to analyse how the situation is changed by installing a class H1 heat exchanger.

A more efficient recovery unit will be bigger and with more finely spaced fins (larger number of plates). This means that the exchanger and the system as a whole will be more costly.

In the following case, we have chosen a recovery unit with the geometric configuration given in Figure 15. This recovery unit costs 2149€.

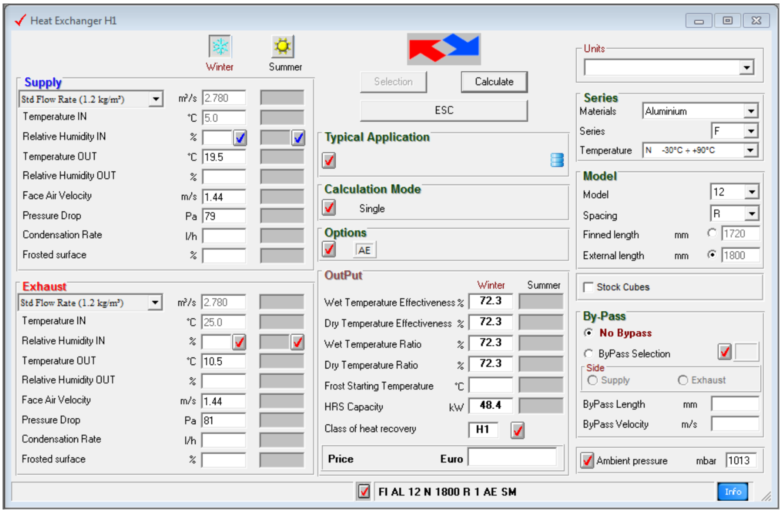
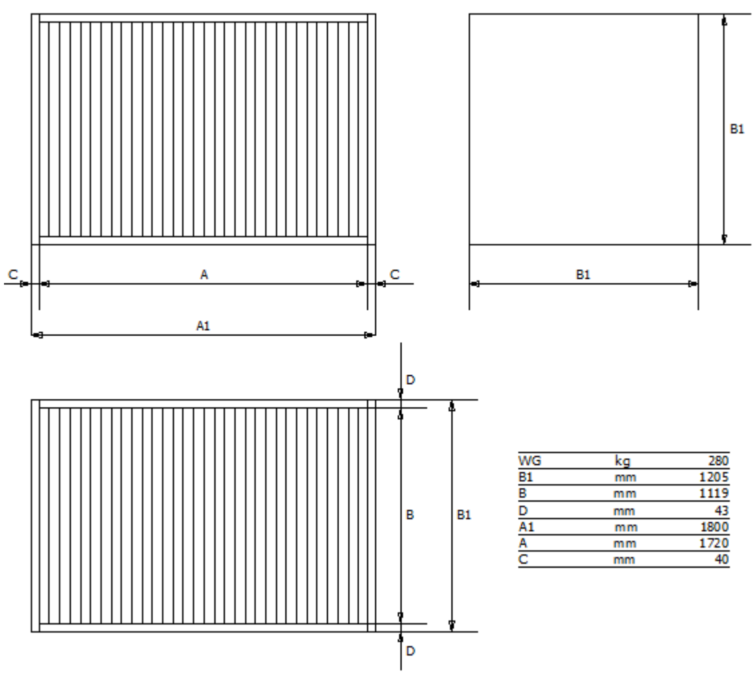


Figure 14: Screenshot of class H1 heat exchanger selection software (§4.3).



WG	kg	280
B1	mm	1205
B	mm	1119
D	mm	43
A1	mm	1800
A	mm	1720
C	mm	40

Figure 15: Geometrical characteristics of the class H1 heat recovery unit used in the analysis (§4.3).

One can see from Figure 16 that choosing a class H1 heat exchanger in Milan, with much colder winter temperatures than in Palermo, results in higher heat recovery.

This is more evident when the temperature differential is high. Indeed, as can be seen from Figure 16, the benefits of a class H1 over a class H3 unit are greater in the summer than in the winter.

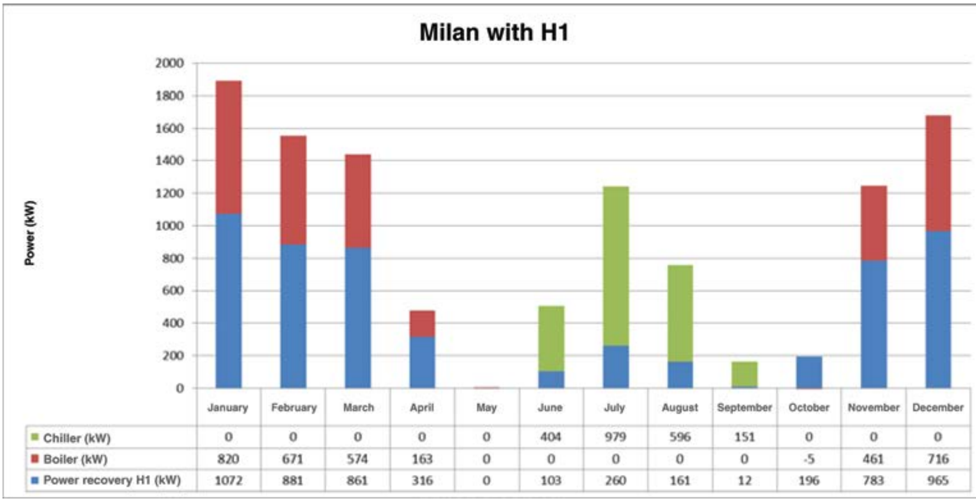


Figure 16: Monthly power draw of powered mechanical ventilation system, and power recovered with class H1heat exchanger, Milan.

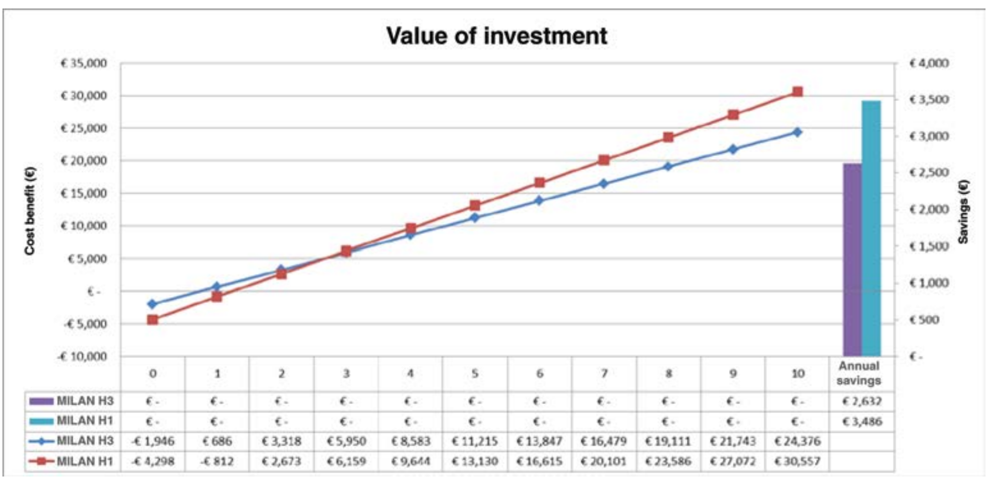


Figure 17: Value of investment for class H1 and class H3 recovery units, Milan.

To assess the cost savings, we have compared the value of the investment over time for the class H1 and class H3 heat exchanger in Figure 17.

Even though the H1 unit requires a longer time to pay back the investment (1.2 instead of 0.7 years), annual savings have increased by around 30% and the long term benefit is greater.

5. CONCLUSIONS

Air exchange is essential to comfort inside a building. The exhaust air flow contains a large amount of energy, which can be recovered before it is exhausted from the building. It is renewable energy source, which can be used by a heat recovery unit.

Such units are characterised by a certain number of indices, which include the temperature ratio and pressure drop. These two values are summarised in the energy classes introduced by UNI EN 13053.

This article illustrates the method of calculation employed to assess the energy class. We have also presented a sample calculation as an aid to comprehension.

The recovery unit used in the example, which is a class H3 unit under the terms of VDI 3803, has also been used for an economic analysis.

We have presented a graph, as an aid to designers, which shows the temperature ratio and exchanger supply temperature (as functions of the pressure drop) which the exchanger must provide to be rated as belonging to a certain energy class.

Heat recovery units have considerable benefits in relation to energy recovery. The time required to pay back the initial investment depends on the outdoor climatic conditions, but is nonetheless always very short. Indeed, in Palermo, where winter temperatures are much milder than they are in Northern Italy, the payback time is less than two years, while in Milan it is under a year.

In a hot climate like that of Southern Italy, installing an adiabatic cooling system increases the benefits of using a heat recovery system, and results in energy savings of a further 50% over installations not having evaporative cooling.

Annual savings can also be increased in Milan. Given the considerable temperature differential during the winter, increasing the efficiency of the recovery system by using a class H1 heat exchanger instead of a class H3 device will increase the annual savings by 30% with a payback time of around 1.2 years. However, using a higher efficiency recovery unit regardless of the local climatic conditions does not always yield benefits capable of justifying the increased initial investment. One must therefore evaluate the potential benefits on a case by case basis.

GLOSSARY

T_{11}	Extract air temperature, °C.
T_{21}	Outside air temperature, °C.
T_{22}	Supply air temperature, °C.
η_t	Temperature ratio.
Δp_{supply}	Pressure drop of the supply air, Pa.
$\Delta p_{exhaust}$	Pressure drop of the exhaust air, Pa.
Δp_{HRS}	Sum of supply and exhaust pressure drops, Pa.
q_v	Air flow, m³/s.
η_D	Overall static efficiency of power consumption.
P_{el}	Electrical power input, W.
$P_{el,\ aux}$	Electrical power input of auxiliary, W.
Q_{HRS}	Heat recovery capacity, W.
ρ_a	Air density, kg/m³.
cp_a	Specific heat of air, J/(kg K).
COP	Coefficient of performance.
η_e	Energy efficiency.
k	Constant, J/(m³K).
H1, .., H6	Energy class.
IC	Evaporative cooling.
η_{sat}	Humidifier saturation efficiency.
x_1	Absolute humidity of air at humidifier intake, g/kg.
x_2	Absolute humidity of air at humidifier outlet, g/kg.
x_S	Absolute humidity of saturated air at wet bulb temperature, g/kg.

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