Experimental tests on different systems for heat recovery on exhausted air

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ABSTRACT
This paper describes the realization of a test device which aim is to estimate the energy efficiency of the heat recovery systems on the exhausted air. The systems which will be tested are: a cross-flow heat exchanger, an enthalpy device and a reversible heat pump with evaporator and condenser (in cooling mode, viceversa in heating mode), respectively located on the supply air flow and on the exhausted one. These systems, characterized by low air flow rates and designed for residential use, can also work in stand-alone mode. The apparatus used for the tests had been built in order to make an energy efficiency analysis on the above mentioned three different systems, with controlled and variable conditions typical of the Italian climate.
The experimental activity goals are: the evaluation of the energy efficiency variations as a function of the temperature and the relative humidity boundary conditions of the two air-flows (supply and exhausted); the evaluation of the differences in the energy savings calculations performed using the experimental results; and finally, the evaluation of the actual seasonal averaged efficiency of the three compared systems.

1. INTRODUCTION
In Italy an Act (Act 10/91) obliges the heat recovery in ventilation systems (with a minimum sensible effectiveness of 50 %) whenever ventilation rates and annual working hours are higher than values determined by the climatic zones. So this prescription is only limited to the heating period. Although other laws have been recently issued (i.e Act 192/05, the Italian receipt of the European Directive 2002/91/EC) this prescription is still valid without any modification, despite energy costs are always increasing, and so energy saving is always more profitable. In any case it is suitable to consider the utilization of heat recovery devices, evaluating systems with an effectiveness higher than the mandatory one, and the possibility of latent heat recovery (using rotary enthalpy systems or reversible heat pumps), even during the cooling period.
In the last years, the heat recovery devices market in the household ventilating systems has increased in Italy as well in overall Europe. In literature there are several qualified studies based on theoretical energy efficiency of the recovery systems (Lazzarin et al. 1998, Noro et al. 2006). Moreover there are real operating systems that are monitored, so it will be possible to know the real seasonal performances of a active energy recovery ventilating system (Grisot, et al. 2006, Ranzato et al. 2006). However the recorded data from a real operating system, even if it is a precious data source, do not concur to a full

evaluation of seasonal performances as function of very different boundary conditions (locations, schedulings, etc) in which the ventilating systems operate.
So, as natural consequence, it has grown the need of implementing a device which can easily and accurately measure the performance data of the energy recovery systems. This device should be able to measure the household and the light-commercial heat recovery devices with very variable outside air conditions typical of Italian climate (hence outside air dry bulb temperatures ranging from -10 °C to +35 °C and relative humidity ranging from 40 % to 90 %); it will be possible to reduce the test time thanks to its compactness and the easy set-up and the easily simulate internal air conditions (20 °C in heating mode, 26 °C in cooling mode and 50 % relative humidity).
The first three heat recovery machines that will be tested by this new device, are:

1. a cross flow heat recovery with 54 % nominal efficiency (at -5 °C outside air and 20 °C internal air), 2.0 m/s frontal air speed, 570 m^3/h airflow and 2.8 kW heat recovery capacity;
2. enthalpic rotative heat recovery system with 72 % sensible efficiency and 63 % latent efficiency in heating working mode (at -5 °C and 80 % relative humidity outside air and 20 °C air temperature and 50 % relative humidity internal air) and 6.3 kW heat recovery capacity. In summer time, the sensible efficiency reaches 80 % while latent efficiency is 63% and recovers 2.5 kW (at 32 °C and 50 % relative humidity outdoor air and 26 °C and 50 % outdoor air) with 1.9 m/s frontal air speed and 650 m^3/h airflow;
3. an active heat recovery consisting of a reversible heat pump with 500 m^3/h air flow; at 30°C dry bulb temperature and 50 % relative humidity outdoor air and 25 °C dry bulb temperature and 50 % relative humidity indoor air, it gives a cooling power of 3.05 kW; at 7°C dry bulb temperature and 80 % relative humidity outdoor air and 20 °C dry bulb temperature and 50 % relative humidity indoor air, it gives a heating power of 3.58 kW.

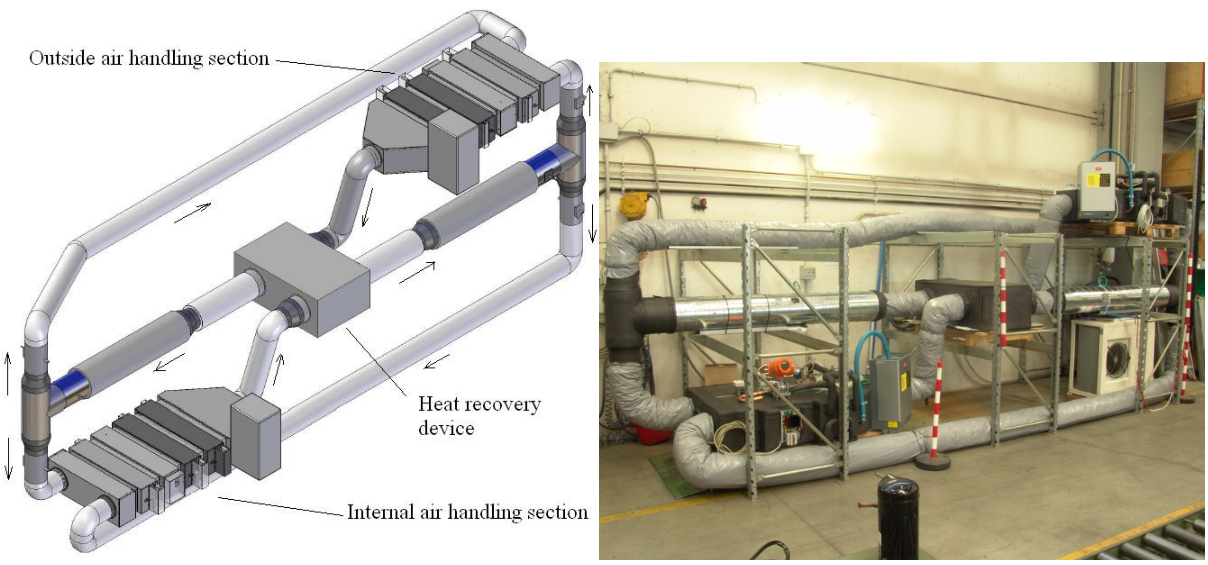

Figure 1: Three-dimensional model and picture of test device

2. TEMPERATURE AND HUMIDITY AIR CONDITIONS IN COOLING TEST

Beyond the standard conditions provided by UNI EN 14511-2 Norm, it will be possible to create all summer conditions: outdoor air flow temperature will vary from 20 °C to 35 °C dry bulb while the relative humidity will change from 50 % to 90 %. The indoor condition will be kept at 26 °C dry bulb temperature and 50 % relative humidity.

2.1. The active recovery

During the cooling period, the condensing coil is passed by a mixed air flow with the indoor air at 26 °C and 50 % relative humidity and the outdoor air coming from a bypass in the machine; the total air flow is 670 m^3/h (485 m^3/h indoor air and 185 m^3/h outdoor air).
In addition to the variation of the temperature and the humidity air conditions, the following trials will be done:

1. a working mode with a reduced outdoor air flow (375 m^3/h instead of 485 m^3/h) with 30 °C and 35 °C dry bulb temperature; (so it will be evaluated the supply air temperature shifting, compared with the full air flow operation);
2. the presence of a 18 °C / 23 °C water temperature cooling coil installed upstream the direct expansion coil;
3. the shutting down of the bypass device for 30 °C and 35 °C outdoor air tests; in fact in this way the air passing through the condensing coil will be at lower temperature but it will be counterbalanced by less air flow (from 670 m^3/h to 485 m^3/h). So it will be possible to determine if the bypass is really useful.

Sixteen cooling tests will be performed for this kind of system.

2.2. Passive heat recovery

For the passive heat recovery only four cooling tests will be performed for each system (cross flow and rotary heat exchangers) with the same temperature and humidity air conditions previously described and the internal air temperature ranging from 23 °C to 26 °C.

3. TEMPERATURE AND HUMIDITY AIR CONDITIONS IN HEATING TESTS

In this case too, the tests will be performed according to standard conditions provided by UNI EN 14511-2 Norm; moreover there will be other tests in which the outdoor air flow temperature will range from -10 °C to 10 °C with 5 K steps and 90 % relative humidity while indoor air will be kept at 20 °C dry bulb temperature and 50 % relative humidity constant.

3.1. Active recovery

Beyond the tests previously described, the following trials, with external air temperature lower than 0 °C, will be performed:

1. reducing inlet airflow;
2. using a 30 °C / 35 °C water coil before the condensing coil in order to avoid low air temperatures supply and too low condensing temperatures;
3. using an electrical resistance installed before the condensing coil for the same reasons seen before.

The total heating tests number is 23.

3.2. Passive recoveries

In this case the are only temperature and relative humidity variations of the external air, and five trials for each system will be performed.

4. TEST SYSTEM DESCRIPTION

The air handling sections which simulate indoor and outdoor air conditions are symmetrical and formed by (see *FIGURE 2*):

1. an intake and mixing section;
2. an inertial water and glycol coil able to stabilize the aeraulic circuit;
3. a 2 kW electrical resistance;
4. a first glycol-water heat exchanger for heating (45°C/40°C) and cooling purpose (7°C/12°C for standard tests or -15°C/-10°C for very low temperatures tests), with 0.6 litre/s water flow;
5. a second glycol-water heat exchanger with the same operating conditions seen before;
6. an inlet section with immersed electrodes humidifier producing a variable vapour flow till 8 kg/h.

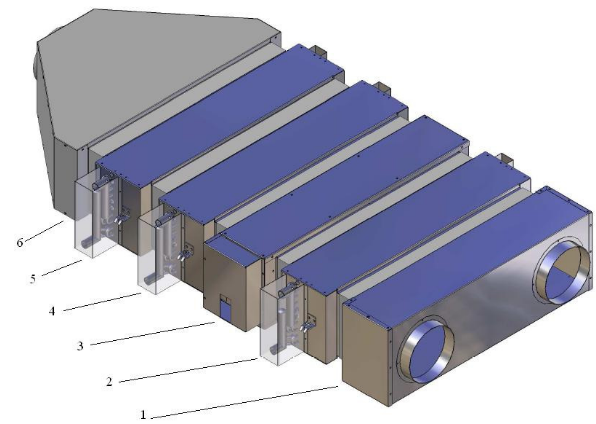

Figure 2: Air handling sections

All the test device sections are fully insulated for guaranteeing minimum thermal losses.

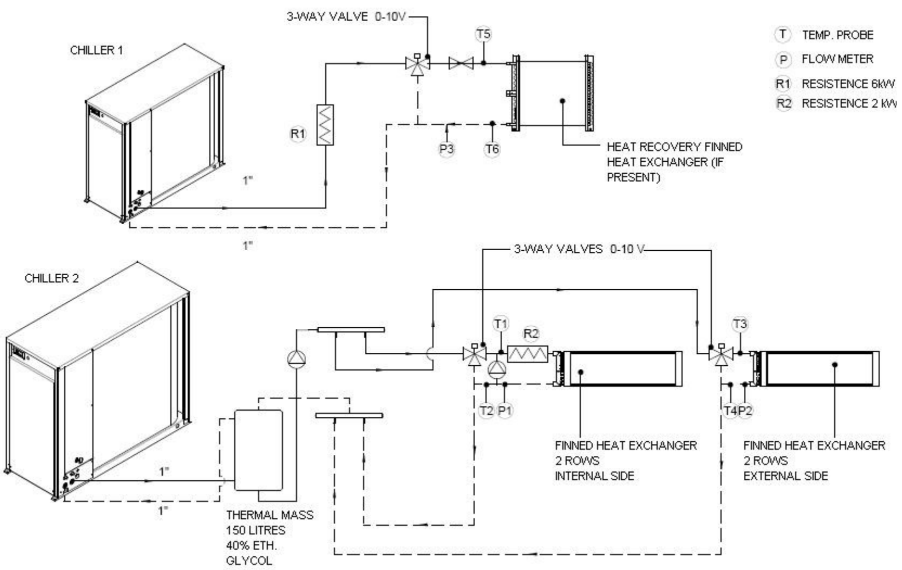

Figure 3: Hydraulic scheme of the test device

The two air flows, after the heat exchange occurred in the recovery device, are recirculated using the manual shutoff device in order to optimize the overall system efficiency (the cooling and thermal energy supplied by the heat recovery system is completely re-used in order to decrease the thermal load of the air handling sections). Another advantage of this system regards the absence of other fans, except the ones of the recovery machine.
It will be installed:

1. eight duct temperature probes (type PT100, 1/5 class DIN thermo-resistance) and duct relative humidity probes (ranging from -40 °C to 70 °C of air temperature and from 0 % to 100 % of air relative humidity with 2% accuracy), installed near the intakes and the supplies of the heat recoveries;
2. two duct air flowmeters, formed by a series of wings perpendicular to the air flow that measure the air speed in each subsection of the air duct; these instruments are characterized by a 2% accuracy for air speed ranging from 2 m/s to 100 m/s after the air handling sections.

The collected data will be recorded by a data logger and it will be possible to see them on a dedicated PC.

5. EXPECTED RESULTS

The first analysis target is the evaluation of the heat recovery system efficiency with variable boundary conditions. Moreover, when considering active recovery devices, the goal is the evaluation of the system behaviour as function of the different control strategies (by-pass, water coils adjustment, choice of fan speed, etc.), obtaining good hints to optimize the machine.
The different energy recovery systems test results will allow to:

1. fix efficiency figures shifting at several different boundary conditions compared to nominal values;
2. verify the humidity condensing and frosting risks of passive heat recoveries, when outdoor air reaches low temperatures; in the same way, verify the frosting risk of evaporating coil in the active energy recovery;
3. deduct optimization strategies in active energy recovery case.

Through efficiency figures obtained by the tests, some simulations will be carried out for different locations in order to calculate the possible energy savings as function of energy recovery kind of system.

6. CONCLUSIONS

The three tested systems are very different from the technological, energetic, economical point of view, although each of them has the same purpose: the extract air energy recovery on mechanical controlled ventilating systems. From experimental results, it could not be possible determine the best system but such an information could be carried out using calculation analysis starting from the results obtained by the test device. When the calculations will be done, it could be possible determine the best system as function of boundary conditions like climatic conditions, working period, main plant efficiency and energy costs.

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