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EVALUATION OF THERMAL PERFORMANCES OF RESIDENTIAL VENTILATION SYSTEMS WITH HEAT RECOVERY

A-M Bernard[1], M-C Lemaire[2], B Spennato[3] and P Barles[4]

[1] CETIAT, BP 2042-F 69603 Villeurbanne Cedex, FRANCE
[2] ADEME-500, Rte des Lucioles-F 06560 Valbonne, FRANCE
[3] EDF DER ADEV-Route de Sens-Ecuelles-F 77250 Moret-sur-Loing, FRANCe
[4] 31 Bv Gambetta, 83460 Les Arcs, FRANCE

Evaluation of thermal performances of residential ventilation systems with heat recovery

Authors : A.-M. Bernard (CETIAT)-BP 2042 - F 69603 Villeurbanne Cedex
M.-C. Lemaire (ADEME)-500, Rte des Lucioles - F 06560 Valbonne
B. Spennato (EDF DER ADEB)-Route de Sens - Ecuelles - F 77250 Moret-s/Loing

Synopsis

Ventilation systems with heat recovery offer several advantages such as, of course, energy savings but also the possibility to add acoustic and filtration treatment. This study was to evaluate the thermal performances of such systems for residential ventilation in France.

These units usually combine exhaust and supply fans, filters and a heat recovery exchanger. To test them, a special draft is being written by the CEN experts of TC 156/WG2/AH7. The study included the test of several units according to this project to evaluate their performances : temperature ratios, airflow/pressure curves, electric power, internal and casing leakages... The influence of humidity, condensation, frost and several full duct-systems simulating a standard house equipment have also been tested.

List of symbols

θextr :	exhaust air dry bulb temperature
θwet, extr :	exhaust air wet bulb temperature
θdew, extr :	exhaust air temperature at dew point
θsupply :	supply air dry bulb temperature
$\Delta\theta$supply :	difference of temperature on supply air
$\Delta\theta$inlet :	difference of temperature between both supply and exhaust air entering the unit
η :	temperature ratio (defined in EN308) in dry conditions
ηsyst :	temperature ratio defined at the limits of the system (unit + ducts)
ηx :	humid temperature ratio (defined in EN308)
Δxsupply :	difference of water-contents on supply air (inlet-outlet)
Δxinlets :	difference of water-contents on unit inlets (supply-exhaust)

1. INTRODUCTION

This study aims the evaluation of the thermal performances of balanced ventilation systems with heat recovery, used in standard single houses in France. These systems also have other advantages like acoustic and filtration in relation with outside conditions. These advantages should be studied in the future for a global performance aspect.

The study focuses on :

- thermic and aerodynamic performances of the unit
- influence of humidity and frost on efficiency
- test and simulation of a full single, family-house system with ducts, in three standard climates in France.

2. PERFORMANCES ON THE UNITS

2.1 Description of the units

Five units combining exhaust and supply fans, filters and a heat recovery exchanger were tested. Four units had double speed fans to fulfill the conditions of French regulation which imposes a standard extract airflow and an increased one in kitchen when necessary.

Unit n°	1	2	3	4
Fan position				
Exhaust	downstream	upstream	upstream	downstream
Supply	downstream	downstream	upstream	downstream
Number of Heat exchanger tested				
number	2	1	1	2
material	plastic (PVC)	plastic (PVC)	plastic (PVC)	plastic (PVC) and aluminium

Figure 1: description of the units.

2.2 Temperature ratio

For each fan speed, temperature ratio were tested according to EN308 [1] and CEN TC156/WG2/AH7 [2] project : $\eta = \frac{\Delta\theta_{supply}}{\Delta\theta_{inlets}}$

During tests, exhaust and supply airflows were balanced and then slightly unbalanced. The test conditions, without condensation, were as followed :

- exhaust air dry bulb temperature : 25°C wet bulb temperature : < 14°C
- supply air dry bulb temperature : 5°C

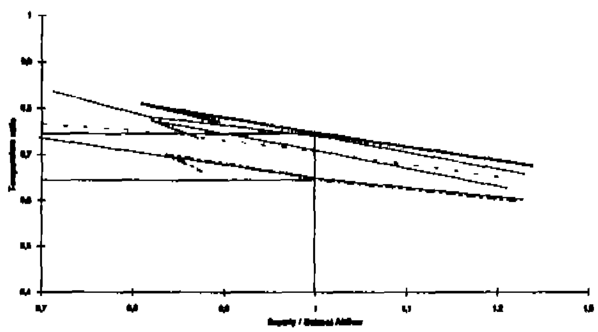

Figure 2: unit temperature ratio-standard speed-unbalanced airflows

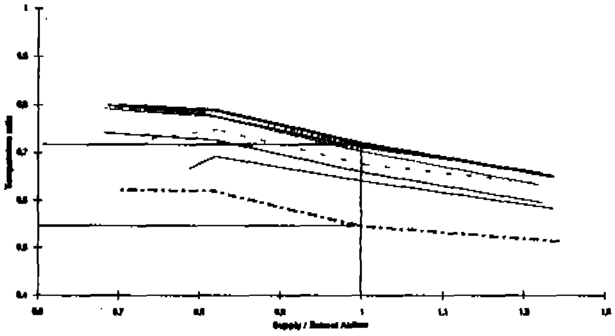

Figure 3: unit temperature ratio-maximum speed-unbalanced airflows

Results in figures 2 & 3 show that the temperature ratio of the units have varied from 64 to 74% in standard speed for balanced airflows. Unbalanced airflows modify these values.

2.3 Influence of humidity

On the same unit (# 1), tests were made for 4 test points with condensation of water vapor contained in exhaust air:

Test point	1	2	3	4
θsupply	7°C	2°C	7°C	2°C
θextr θwet, extr	20°C 12°C	20°C 12°C	20°C 17°C	20°C 17°C
Δθ = θdew, extr - θsupply	4.7 °C	9.6 °C	10.1 °C	15.1 °C
Relative Humidity	68.8%	81.3%	80.5%	84.3%

EN308 defines the humid temperature ratio as : $\eta_x = \frac{\Delta x_{supply}}{\Delta x_{inlets}}$

As supply air does not condensate, this ratio is equal to zero.

As condensation depends on :

- exhaust air moisture (θdew, extr)
- dry bulb temperature on supply air (θsupply).

Figure 4 shows the influence on the efficiency ratio depending on : $\Delta\theta = \theta_{dew, extr} - \theta_{supply}$

The increase of temperature ratio due to condensation is mainly linear (+ 1,5 %/°C) and was interpolated to estimate the seasonal performance (see § 3.3.4). This interpolation can be considered as valid as long as exhaust temperature is the same, which is more or less the case in the house we have considered.

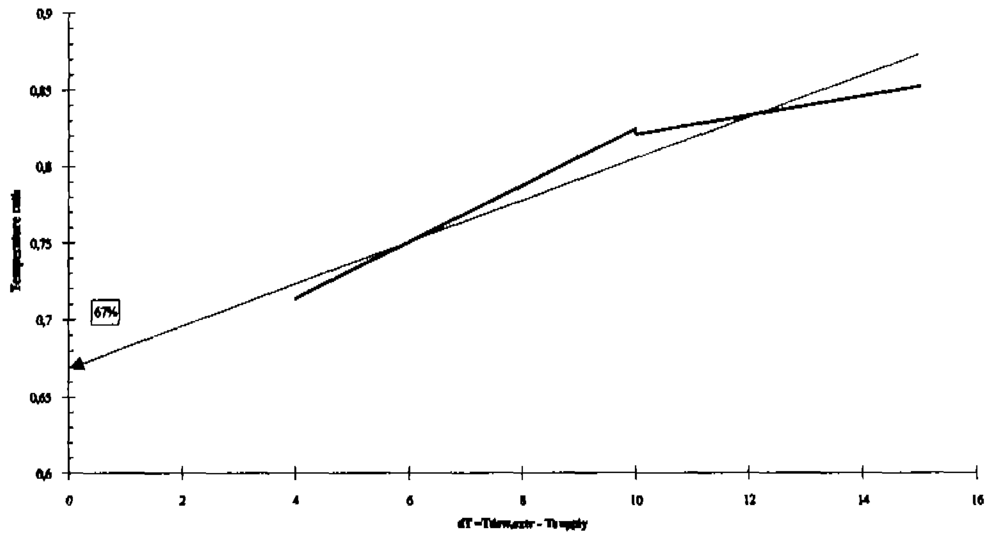

Figure 4: influence of condensation

2.4 Influence of frost

A test has been run during 4:30 hours with the following conditions :

θextr : 20°C (θdew, extr has varied between 16 and 17.4°C during the test)

θsupply : - 7°C

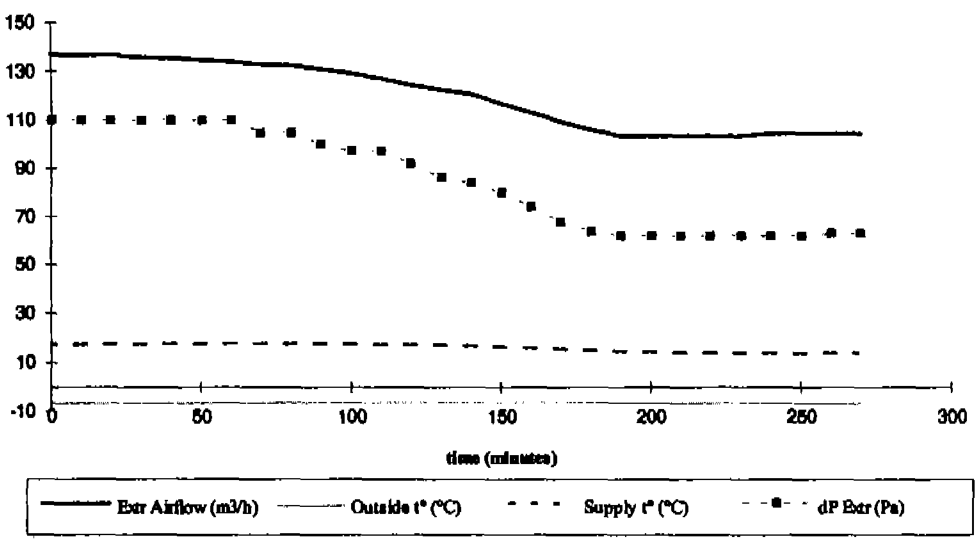

Figure 5: loss of performances due to frost

The loss of performance was stabilised after 3 hours and the airflow had decreased of 27%. The exchanger was not totally filled in by frost. We can consider that a lower supply temperature could induce a more important loss of airflow even down to zero. A variation of

the exhaust air moisture induces a variation of the time necessary to stabilise. For French climates, the risk of frost is mainly restricted to some continental areas like eastern part of France where outside temperatures are more extremes.
In this area, the use of a supplemental coil is necessary to prevent frost.

3. PERFORMANCES OF A SINGLE-HOUSE SYSTEM

3.1 Description of system

We have dimensioned two different distribution ducts for a standard house which are shown on figures 6 and 7.

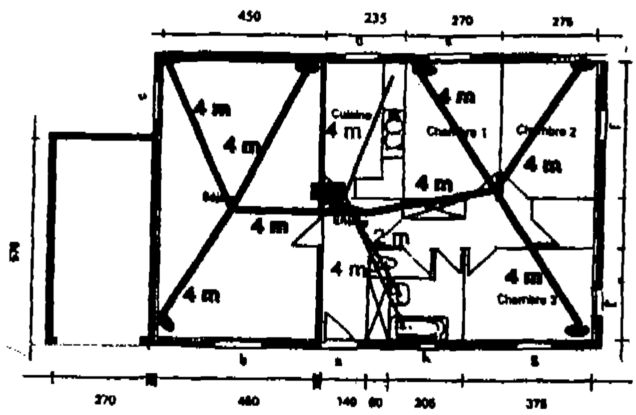

Figure 6: air distribution ducts serial configuration

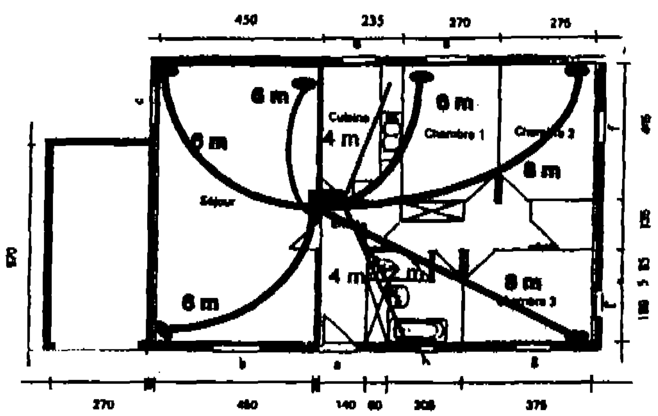

Figure 7: air distribution ducts parallel configuration

3.2 Test results

Tests have been made for two positions :

- unit and ducts situated in the attic (which was considered between fully-insulated and not-insulated). Ducts were insulated (25 mm or 50 mm) ;
- unit and ducts inside the heated volume. Ducts were not insulated.

The main results are indicated in appendix 1 and 2.

The "system temperature ratio" is defined as : $\eta_{syst} = \left(\frac{\Delta\theta_{supply\,air}}{\Delta\theta_{inlets}} \right)_{system}$

where temperatures were considered at the entrances or exits of the full system (duct + unit). When the system was inside the heated volume, we have excluded in temperature ratio calculations all exchanges with the volume itself through the unit casing or the ducts for example. When the system was in the attic, these exchanges were considered but limited to a maximum equal to heated volume results. As attics are indirectly heated by the house, when the construction techniques lead to a strong insulation directly in the attic and not between the attic and the heated volume, the system tends to act as a heated-volume system.

These results show that the system temperature ratio was between :
- 51 to 58% positionned in the attic
- 64 to 74% positionned in the heated volume.

3.3 Transient simulations

Simulations in dynamic state were run with TRNSYS/IISIBAT program. The house was totally described and 3 systems were modelised : mechanical exhaust (without heat recovery), supply and exhaust system in the attic, supply and exhaust system in the heated volume. Airflows were dimensionned according to French regulations and the system was slightly unbalanced (supply airflow ≈ 0.9 exhaust airflow).

3.3.1 Internal conditions

Room temperature was considered at 19 °C with 2 °C stratification that induced an exhaust temperature of 21 °C at minimum.
Scenarios of occupation were supposed corresponding to a family of 4 persons, 2 adults not present during the day and 2 children.
These elements induced humidity scenarios during the day as well as some heating gains.

3.3.2 Ventilation systems

The temperature ratio of the heat exchanger in "dry condition" was considered at 67 %. Influence of humidity was taken into account (according to § 2.3). A supplemental coil of 500 W to avoid frost was supposed to be installed and to start functionning when the outside temperature is below -5 °C.

3.3.3 Outside climate

Simulations have been run on three different site conditions in France :
- Nancy : continental climate with rough winter
- Trappes : near Paris, between Oceanic and continental
- Carpentras : South of France, almost mediterranean.

3.3.4 Results

The ventilation system with heat recorery compared to an exhaust ventilation system, without recovery, most common in France, leads to the following energy gains :
- system situated in the attic : 43 %
- system in the heated volume : 66 %.

Yet, when considering the infiltration airflow due to the fact that the house is at an internal pressure close to the external one, with the balanced ventilation system, and the fan absorbed power, these values decrease respectively to : 23 % and 44 %.
To obtain these values, infiltrations were calculated by a simplified method and these results should be checked with a more accurate modelisation, taking into account real wind.
Condensation increased in average the system temperature ratio of 1 to 3 %.
In France, the supplemental coil to avoid frost has an absorbed power which could have been neglected (32 kWh for one year, in Nancy).

4. CONCLUSION

Supply and exhaust systems with heat recovery allow important energy savings in residential ventilation systems :

- 43 % when the system is situated in the attic,
- 66 % when the system is in the heated volume.

Yet, these values are lower if you take into account infiltrations and fan asborbed power. A more precise study on infiltrations should be necessary.
The five heat recovery units tested have shown that temperature ratios :

- are between 61 and 74 % for dry air,
- increase proportionnally with (θdew, extr - θsupply).

The loss of performances due to frost can be important for a long period at low outdoor temperature and high indoor humidity, which is limited to the eastern part of France. A supplemental coil to avoid frost is necessary in these cases.

References

[1] EN 308 - "Heat Exchangers - Test procedures for establishing performance of air to air and flue gases heat recovery devices" - November 1997.

[2] CEN TC156/WG2/AH7 N3 - January 1998 "Components/Products for residential ventilation. Performance testing, Part 7 : Performance testing of a mechanical supply and exhaust ventilation unit".

APPENDIX 1: main test results on system situated in the attic

Duct insulation	mm	50	25	25	25	25	25
Configuration		serial	serial	serial	serial	serial	serial
SUPPLY AIR							
Outside temperature	°C	-0,1	0,0	0,0	-10,2	-5,1	10,0
Supply temperature	°C	12,3	11,2	11,2	5,9	8,8	16,9
Airflow	m3/h	119	119	175	128	120	119
Recovered power	W	497	453	666	706	571	275
Accuracy	%	+- 10 %	+- 10 %	+- 10 %	+- 10 %	+- 10 %	+- 10 %
EXHAUST AIR							
Room temperature	°C	21,1	21,1	21,2	20,8	21,1	20,9
Room temperature at dew point	°C	9,8	10,0	10,4	10,6	10,8	9,6
Inside Relative Humidity	%	48,5	49,2	50,4	51,9	51,5	48,6
Exit temperature	°C	9,4	9,7	10,7	4,4	6,6	15,1
Exit temperature at dew point	°C	7,5	7,7	8,8	2,9	4,8	9,5
Outside Relative Humidity	%	87,7	87,1	87,8	89,8	88,5	69,0
Airflow	m3/h	134	135	195	140	132	140
Power	W	659	653	832	1204	976	281
AMBIANCE		(attic temperature)					
Average temperature	°C	6,1	6,0	6,0	-7,1	0,0	15,0
Atm pressure	kPa	98,4	98,8	98,6	99	99,016	98,7
CALCULATION							
ηsyst	%	58,60%	53.2%	53.1%	51.7%	53.2%	53.2%

APPENDIX 2: main test results on system situated in the heated volume

SUPPLY AIR					
Outside temperature	°C	-4,4	-4,2	2,9	3,0
T° sèche sortie	°C	17,0	17,2	18,6	18,5
Airflow	m3/h	129	218	123	123
EXHAUST AIR					
Room temperature	°C	21,1	21,0	21,1	21,0
Room temperature at dew point	°C	10,8	10,6	2,9	10,4
HR entrée	%	51,7	51,5	30,2	50,9
Exit temperature	°C	10,2	10,7	13,8	13,8
Exit temperature at dew point	°C	2,3	9,3	2,9	7,5
HR sortie	%	57,8	91,6	47,5	65,7
Airflow	m3/h	134	155	136	136
AMBIANCE		(Room temperature)			
T ambiante Moy	°C	20,6	20,5	22,7	20,4
P Atm	kPa	99,3	98,02	98,9	99
UNIT					
θentrance, exhaust air	°C	20,8	20,8	21,3	20,7
θexit, exhaust air	°C	6,1	7,4	10,5	10,5
θambiance, supply air	°C	-1,0	-1,6	6,6	6,6
θexit, supply air	°C	13,2	13,9	15,8	15,4
Temperature ratio (corrected for heated volume)					
Equivalent outside temperature	*°C*	*-2*	*-2,1*	*5,6*	*5,6*
Recovered power	*W*	*818*	*1383*	*528*	*523*
ηsyst	%	74.4%	70.3%	64.4%	64.3%

APPENDIX 3: main results of dynamic simulations.

One year simulation - Quantity of Energy (kWh)

Mechanical exhaust

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	3981	208	3621	168	5532	198	3406	168
Trappes	6770	258	4396	188	9209	253	4248	188
Nancy	7200	270	4547	188	9806	265	4401	188

Exhaust and Supply system in the attic

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	2851	711	2058	336	4356	659	1459	336
Trappes	5290	865	2485	376	7643	832	1822	376
Nancy	5603	900	2552	376	8182	867	1919	376

Exhaust and Supply system in the heated volume

	Not insulated attic				Insulated attic			
Meteorology	Heating needs	Infiltration	Ventilation	Fan absorbed power	Heating needs	Infiltration	Ventilation	Fan absorbed power
Carpentras	2272	718	1290	336	4336	658	1449	336
Trappes	4497	871	1484	376	7612	832	1790	376
Nancy	4828	907	1549	376	8140	868	1859	376