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The Efficiencies of the IACC! Ventilation Module.

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July 1998

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SAMMANFATTNING.

En laboratorieundersökning och analys av mätdata har utförts på ventilationsmodulen IACC!. Modulen, som är ett kompakt decentraliserat ventilationssystem (måtten 34x19.5x22.3 cm^3) med två fläktar och en värmeväxlare, fungerar regeneratrivt eftersom till- och frånluft passerar apparaten växelvis med 27.5 sekunders intervall. Av denna anledning är det nödvändigt att bestämma systemets termiska prestanda. På grund av att systemet är kompakt och de snabba luftväxlingarna, är det experimentellt svårt att bestämma själva värmeväxlarens verkningsgrad.

Ett antal verkningsgrader har definierats och studerats i rapporten. Temperaturverkningsgraden, som ofta används i Sverige, visar sig inte vara tillämpbar på IACC!-modulen eftersom definitionen inte omfattar fläktarnas energi. En ny verkningsgrad, energiverkningsgraden, har formulerats på ett liknande sätt som temperaturverkningsgraden men denna tar hänsyn till fläktarnas energi. Därutöver har en modell och entalpiverkningsgraden uttryckts, som tar hänsyn till fukttransporter, värmeförluster och luftläckage i systemet. Entalpiverkningsgraden har inte bestämts numeriskt i rapporten.

Temperaturverkningsgraden för hela systemet varierade mellan 89-116% beroende på luftflöde och utomhustemperatur. Temperaturverkningsgraden för värmeväxlaren varierade mellan 64-104%. Siffrorna visar att fläktarnas energi påverkar de uppmätta temperaturerena kring värmeväxlaren. Energiverkningsgraden erhöll värden mellan 80-91% för olika driftsförhållanden.

En så-kallad förlustfaktor har definierats. Faktorn indikerar hur mycket ventilationsförlusterna minskar om ett regeneratrivt ventilationsystem installeras gentemot självdragsventilation. Här antas det att luftväxlingen är densamma för bägge ventilationstyper.

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SUMMARY.

A laboratory test and an analysis of measured data have been performed on the ventilation module called the IACC!. The ventilation module, which is a compact decentralized ventilation system (measuring 34x19.5x22.3 cm^3) composed of two fans and a heat exchanger, is regenerative since the supply and exhaust air flows are alternated through the module with 27.5 second intervals. For this reason, it is necessary to determine the system's thermal performance. Due to the compactness and the fast switching of the air flow direction, it is difficult to experimentally determine the efficiency of the heat exchanger alone.

A number of various efficiencies are defined and studied in this report. The use of the traditional efficiency used in Sweden, the temperature efficiency, proves invalid when applied to the IACC!-module since the definition does not include energy from fans in the system. A new efficiency, the so-called energetic efficiency, where the energy supplied to the fans are accounted for by definition, is formulated in a similar fashion as the temperature efficiency. Also, a detailed model and a so-called enthalpy efficiency is formulated that takes into consideration moisture transfer, heat losses, and air leakage in the system as a whole. The enthalpy efficiency is not numerically determined in this report.

The temperature efficiency of the whole system varies between 89-116% for different air flows and outdoor temperatures. The temperature efficiency for the heat exchanger varies between 68-104%, indicating that fan energy interferes with the temperatures of the heat exchanger. The energetic efficiency has values ranging between 80-91% for different running conditions.

A so-called loss ratio is defined. This ratio gives an indication on how much ventilation losses are reduced by insertion of a regenerative ventilation system as opposed to natural ventilation. This ratio assumes that the rate of ventilation are the same for both ventilation types.

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1. Background.

The annual energy demand caused by ventilation of dwellings in Sweden is of the order of magnitude of 22 TWh (ELAB 1994). This quantity is dependent on the types of ventilation systems that are installed in the houses. In 1994, there existed 1.353 million single-family houses of which 80% had ventilation due to stack effect and 12% ventilation with exhaust fans. The rest, 8%, had balanced ventilation with or without heat exchange. For multi-family apartments in 126000 buildings, the respective numbers are 58%, 35% and 7%. On an environmental and economical basis, it is wished that this energy demand is reduced. Also in terms of interior environment, many buildings have insufficient ventilation resulting in poor indoor quality. If the ventilation in these buildings would be increased, the energy demand would rise.

The most beneficial way of reducing the energy demand is by means of recuperation (heat exchange) or regeneration (heat and possibly mass exchange). This would mean that natural ventilation would have to be substituted by mechanical ventilation. For the dwelling owners, it would mean not only an investment cost but also run costs, primarily that of fan energy. This must however be put in relation to how much energy is saved through these measures.

The conventional way of establishing mechanical ventilation is by installing a central unit with ducts to every building zone. An alternative method is to use decentralized units, such as the ventilation module called IACC! produced by TAB System AB in Sweden. The IACC!-system is quite unique with respect to that each module is a small entire compact ventilation system that can fulfill the need of ventilation in individual zones as well as in multiple zones. The system is regenerative: during the supply mode, air is preheated with recovered heat from the previous exhaust mode.

For a small house, a couple or three IACC! modules fulfill the ventilation demand at a relatively low investment cost. The run cost is dependent on two factors. The first is the energy demand of the fans. This cost can be brought down depending on fan power; small ventilation units have small fans. The run cost is also dependent on the outdoor climate, primarily the outdoor temperature. This cost will be reduced proportional to the regenerated energy. With this in mind, this study was brought on to assess the thermal performance of the IACC!

2. Scope.

The aim of this project was to investigate and acquire experience from a measurement on the IACC!-system. The module is complex and differs in several ways in comparison to traditional central ventilation systems.

- The flow direction is reversed every 27.5th seconds.
- The short periods require that the gauges have extremely short responses (time constants).
- The system is compact, meaning that is it difficult to measure on individual components of the system.
- The system involves both mass and heat transfer in normal climatic conditions.

The aim of this project was also to establish an efficiency of the module that includes fan energy but is comparative to the so-called temperature efficiency commonly used in Sweden. Different theoretical models have been set up to establish the temperature and energetic efficiencies. Validity of the models and what order of magnitude different types of efficiencies give was studied. In terms of measurements, this project will give experience on how and what to measure in future elaborate tests.

3. The IACC!-Module: A brief description.

The IACC!-module is a small compact ventilation system that can be installed in or adjacent to external building constructions, for example walls. The system has an inner casing of steel. It encloses two sections: the fan motor compartment and the air flow section. The air flow section contains two radial fans, an electrical heating coil and a regenerator (heat and moisture exchanger). The regenerator is composed of a sandwich-like construction containing strips of cardboard between multiple layer aluminum grids. The upper aluminum layer contains a filter.

The inner casing is circumscribed by an aesthetic outer casing having the height of 34 cm, a width of 19.5 cm and a total depth of 22.3 cm. On opening the hatch, the operator can reach two knobs to make service adjustments. One is to set the air flow. The other is used for regulating the temperature of the incoming supply air; in other words the control of heating coil if necessary.

The module is mounted inside or in connection to a wall. An opening in the exterior wall has to be made for the duct. The duct has an adjustable length and a rectangular cross-section area measuring 19x4 cm^2.

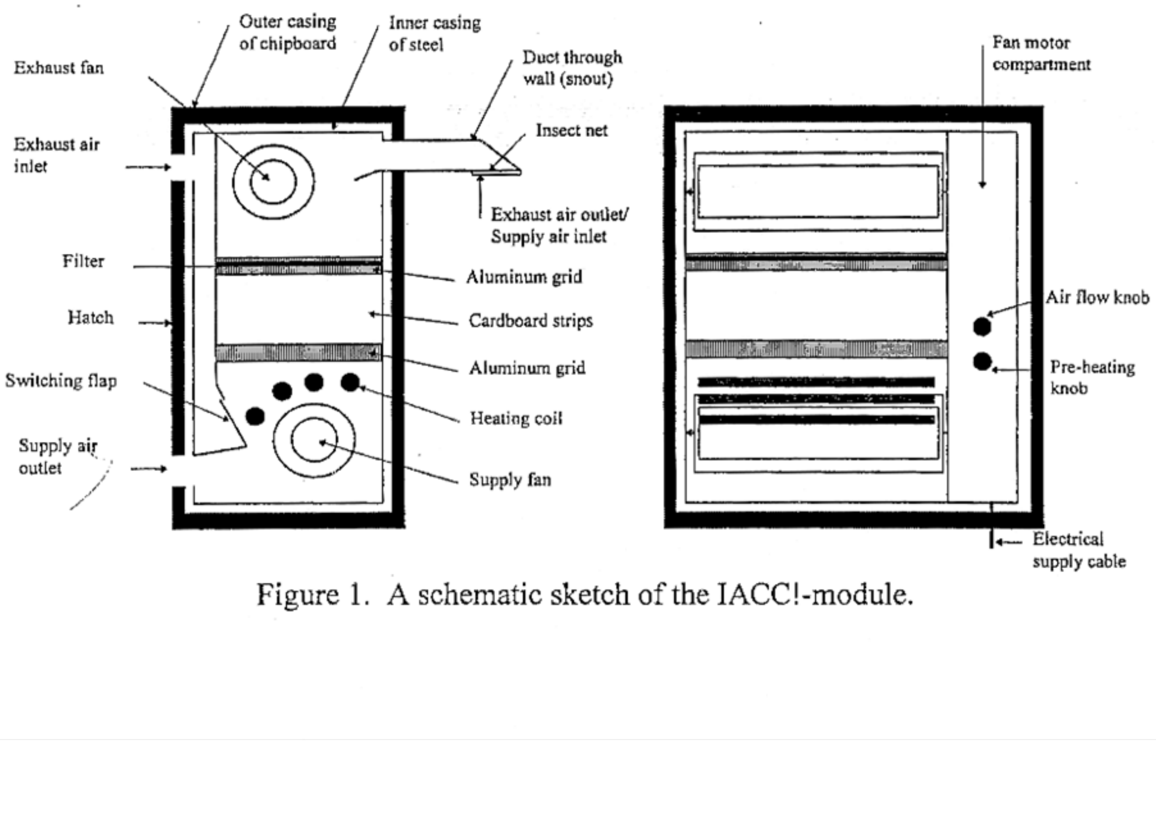

Figure 1. A schematic sketch of the IACC!-module.

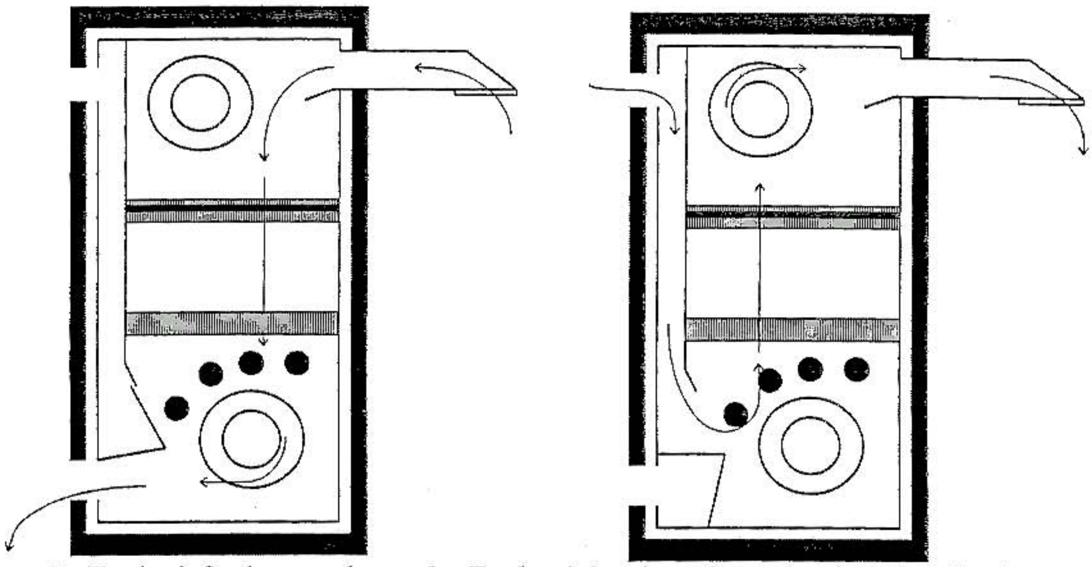

Figure 2. To the left: the supply mode. To the right: the exhaust mode. Note the function of the switching flap.

The module is regenerative, meaning that it has two working modes as illustrated in figure 2. In the supply mode, fresh uncontaminated air as taken from the external environment by means of the lower supply fan. The air passes the regenerator, resulting in a temperature (and probably a moisture) rise. A small hole between the two compartments located next to the supply fan allows warm air from the motor compartment to be mixed with the supply air. This warm air is transferred by means of the ejector effect.

During the exhaust mode, exhaust air passes through the module as the exhaust fan reverses the flow whereas the supply fan is shut off. The switching flap closes the supply outlet. The exhaust air is led through the regenerator to discharge heat (and moisture), to finally be rejected to the exterior environment. Each mode lasts for 27.5 seconds.

The air flow in each mode is usually the same. However, the switching flap can be adjusted to give a non-balanced net flow.

4. Efficiencies.

Within the field of ventilation and also thermodynamics in general, there are many types of efficiencies and special efficiency definitions. Therefore, some will first be listed, and some will be defined.

At this point, it will be mentioned that energy, work, heat and power are terms which will continuously be used in the following text. Q [J] depicts energy. The rate of change of energy is energy flow, such that the time derivative of energy is symbolized by a letter with a dot, for example that $dQ/dt = \dot{Q}$. $\dot{Q}$ is the symbol for heat flow [W] whereas $\dot{W}$ denotes the rate of change of work, power [W].

4.1 The Temperature Efficiency of a Heat Exchanger.

The most commonly used efficiency which characterizes thermal performance of a recuperator (heat exchanger) is the "temperature" efficiency η_t (in Swedish *temperaturverkningsgraden*). It is defined as the quotient between temperature rise of the supply air ($T_{so} - T_{si}$) and the temperature difference between the entering exhaust air temperature and the entering supply air temperature (T_{ei} and T_{si}), such that

$$\eta_t = \frac{T_{so} - T_{si}}{T_{ei} - T_{si}} \tag{1}$$

Equation 1 assumes that the mass flow of the medium is the same in both directions. The largest number this efficiency can have is 1, which indicates that all heat is recovered.

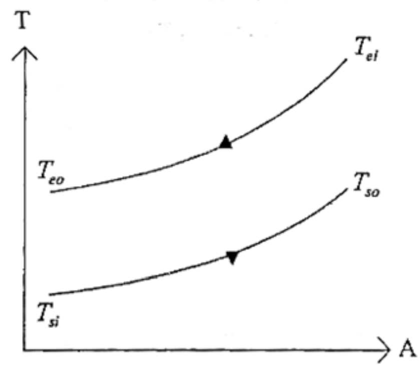

Figure 3. The temperature profiles for exhaust and supply air in a heat exchanger.

4.2 The Temperature Efficiency of a System.

The temperature efficiency of a heat exchanger is however often used as the efficiency of a ventilation system with a heat exchanger. By definition, it will be the quotient between temperature rise of the supply air and the temperature difference between the interior and the exterior environments ambient temperatures (T_i and T_a), such that

$$\eta_{t\ sys} = \frac{T_s - T_a}{T_i - T_a} \tag{2}$$

This equation is derived from several assumed conditions. Figure 4 illustrates a cycle of an ideal ventilation system containing a heat exchanger. The mass flow is $\dot{m}$ with the initial ambient temperature T_a. The air passes a heat exchanger, receiving a heat flow corresponding to $\dot{Q}_c$ which results in a temperature rise to the supply air temperature T_s. By means of an external heating device, such as a radiator, the air is heated to the internal air temperature T_i. During the exhaust mode, air with the initial temperature T_i enters the heat exchanger, dissipates heat to the supply air $\dot{Q}_c$, and exits the system with a temperature of T_e . Note that no energy is supplied to or from fans since these are neglected.

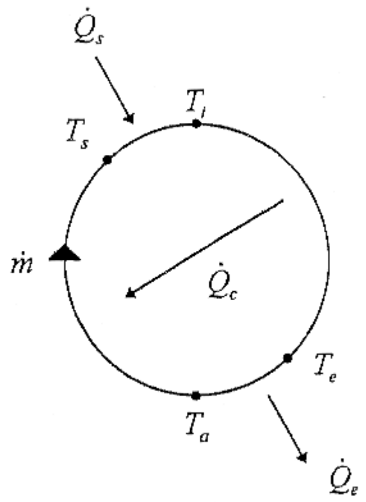

Figure 4. A cycle of an ideal ventilation system containing a heat exchanger.

The total heat delivered to the system is $\dot{Q}_c$ and $\dot{Q}_s$, whereof $\dot{Q}_c$ is recovered. This gives an efficiency of

$$\eta_{t\ sys} = \frac{\dot{Q}_c}{\dot{Q}_c + \dot{Q}_s} = \frac{\dot{m} \cdot c_p (T_s - T_a)}{\dot{m} \cdot c_p (T_i - T_a)} = \frac{T_s - T_a}{T_i - T_a} \tag{3}$$

The definition is unfortunate in this application since the equation does not contain any terms representing fan work delivered to the system, even though this work may be represented in terms of temperature in measurements. It is therefore convenient to set up an "energetic efficiency" (in Swedish *energetiska verkningsgraden*), which preferably can be comparable with the so-called temperature efficiency.

4.3 The Energetic Efficiency.

It is desirable that the definition of the energetic efficiency contains work delivered to the system, and yet gives a representation on how much heat is recovered. The maximum value should be 1.

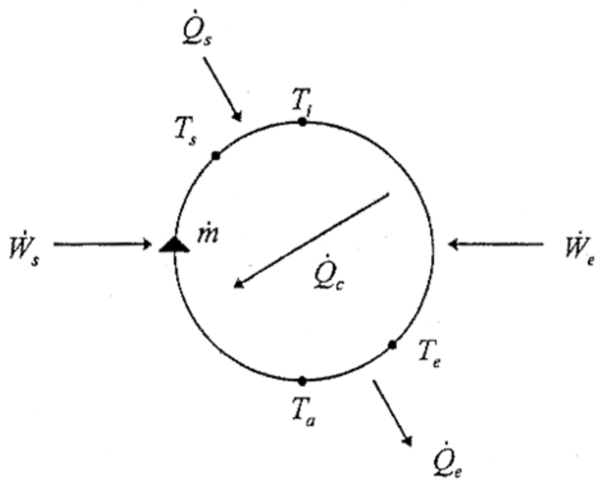

Figure 5. A cycle of an ideal ventilation system containing a heat exchanger and two fans.

Figure 5 displays a cycle as in figure 4 with the introduction of fan work. $\dot{W}_s$ is work delivered to the supply fan whereas $\dot{W}_e$ is delivered to the exhaust fan. At this point, it is assumed that all energy delivered to the fans is dissipated into the flowing air.

The energetic efficiency of the heat exchanger will be defined as the heat that is recovered divided by the total energy that is supplied to the system, such that

$$\eta_e = \frac{\dot{Q}_c}{\dot{Q}_c + \dot{Q}_s + \dot{W}_s + \dot{W}_e} \tag{4}$$

Since the supplied heat of the supply mode, $\dot{Q}_c$, $\dot{W}_s$ and the external heat $\dot{Q}_s$, results in the temperature increase that can be measured,

$$\dot{Q}_c + \dot{Q}_s + \dot{W}_s = \dot{m} \cdot c_p \left(T_i - T_a\right) \tag{5}$$

Insertion of equation 5 into the denominator of equation 4 gives that

$$\eta_e = \frac{\dot{m} \cdot c_p \left(T_s - T_a\right) - \dot{W}_s}{\dot{m} \cdot c_p \left(T_i - T_a\right) + \dot{W}_e} \tag{6}$$

However, not all fan work may be dissipated into the flowing air. If it is assumed that only a part b_s of $\dot{W}_s$ is delivered to the flowing air, equation 5 can be written such that

$$\eta_e = \frac{\dot{m} \cdot c_p \left(T_s - T_a\right) - b_s \cdot \dot{W}_s}{\dot{m} \cdot c_p \left(T_i - T_a\right) + \left(1 - b_s\right) \cdot \dot{W}_s + \dot{W}_e} \tag{7}$$

4.4 Discussion on Efficiencies.

The thermal efficiency η_T (in Swedish *termiska verkningsgraden*) is within thermodynamics is defined as the useful energy Q_{gains} divided by the energy that is totally supplied to the system in question, namely Q_{tot}.

$$\eta_T = \frac{Q_{gains}}{Q_{tot}} \tag{8}$$

Since the totally supplied energy to a system is conserved during a process, this quantity can be separated into a useful part (gains) and the rest, which usually accounts for losses and supplied work to keep the process running.

$$Q_{tot} = Q_{gains} + Q_{losses} \tag{9}$$

The efficiency of the system can therefore be expressed by means of the losses, giving an equation such that

$$\eta_T = 1 - \frac{Q_{losses}}{Q_{tot}} \tag{10}$$

These equations are commonly used within the field of power cycles, where the gains Q_{gains} are actually work delivered from a system whereas Q_{tot} is energy put in the system, for example, the energetic content of the combustion fuel.

In the case of ventilation, there is clearly a thermal cycle present, but no power is generated. There is energy being put into the system, and there are losses associated with the process. Therefore, what is meant by useful energy within this context, and thus the meaning of ***energetic efficiency***, *is the part of the totally supplied energy to the system that is regenerated from the exhaust air and delivered to the supply air.*

On recalling equation 7, the terms can be identified as follows:

$$\eta_e = \frac{\dot{m} \cdot c_p\left(T_s - T_a\right) - b_s \cdot \dot{W}_s}{\dot{m} \cdot c_p\left(T_i - T_a\right) + \left(1 - b_s\right) \cdot \dot{W}_s + \dot{W}_e} = \frac{\dot{Q}_{gains}}{\dot{Q}_{tot}} \tag{11}$$

Application of equation 9 gives that

$$\dot{m} \cdot c_p\left(T_i - T_a\right) + \left(1 - b_s\right) \cdot \dot{W}_s + \dot{W}_e = \dot{m} \cdot c_p\left(T_s - T_a\right) - b_s \cdot \dot{W}_s + \dot{Q}_{losses} \tag{12}$$

so that

$$\dot{Q}_{losses} = \dot{m} \cdot c_p\left(T_i - T_s\right) + \dot{W}_s + \dot{W}_e \tag{13}$$

The energy that has to be supplied during one cycle constitutes heat to raise the supply air temperature from the supply outlet temperature to the interior environments temperature (corresponding to $\dot{Q}_s$) and energy to the fans. A conclusion that can be made is that the energetic efficiency is a type of thermal efficiency.

The use of an energetic efficiency involves several benefits for the purchaser of such a system. The following points should appear in the definition:

- The system user can predict the ventilation system's total energy demand.
- The utilization costs are accurately determined. The energetic efficiency of a system is less than the temperature efficiency of the heat exchangers which are given by manufacturers.
- The effect of fan power is present in this term.

The commonly used temperature efficiency on the supply side, expressed as

$$\eta_t = \frac{T_s - T_a}{T_i - T_a}$$

can only describe part of the thermal performance of a heat exchanger. It reveals nothing about system energy requirement nor run costs.

Efficiencies expressed as functions of temperatures are actually only valid as long as the moisture content of the indoor and outdoor air are the same, or that the difference can be neglected as well as phase changes. A more accurate and physical efficiency for a system can be formulated on basis of enthalpies instead of temperatures only. The following sections will deal with such a treatment as well as a module model that accounts for transmissions, leakage and moisture phase changes. Since the air flow in the IAAC!-module changes direction, not only heat is exchanged, but also moisture, which should be accounted for in the modules efficiency.

4.5 The Loss Ratio.

An interesting efficiency, or factor to be more precise, is a type of temperature efficiency of the exhaust mode. The exhaust air that leaves the ventilation system has an energy content that is higher than that of the ambient outdoor air. This energy corresponds to the heat flow

$$\dot{Q}_e = \dot{m} \cdot c_p \left(T_e - T_a\right) \tag{14}$$

and is in essence the heat that is removed from the buildings zone by means of ventilation. Equation 14 can be related to the heat removed by natural ventilation, in other words if there were no mechanical ventilation. If the mass flow $\dot{m}_{nat}$ were generated by natural ventilation (due to stack effect), the losses would be

$$\dot{Q}_{nat} = \dot{m}_{nat} \cdot c_p \left(T_i - T_a\right) \tag{15}$$

The loss ratio γ can now be defined as

$$\gamma = \frac{\dot{Q}_e}{\dot{Q}_{nat}} \tag{16}$$

For the special case where $\dot{m}_{nat} \approx \dot{m}$, the loss ratio becomes

$$\gamma = \frac{T_e - T_a}{T_i - T_a} \tag{17}$$

This term is interesting from different points of view. The denominator expresses the initial energy content of the indoor air and therefore the potential of energy that can be regenerated. How the heat and moisture have been delivered to the indoor air in the first place is irrelevant. The nominator expresses the energy content of the rejected air, in other words the heat loss of the actual ventilation system.

The loss ratio also shows how much of the energy requirement is reduced if a mechanical ventilation system replaces natural ventilation. The loss ratio can be larger than 1, take for instance a ventilation system with an exhaust fan without heat recovery. The exhaust fan increases the temperature of the exhaust air to be above T_i; this temperature rise accounts for the fan power requirement. This also applies to balanced ventilation systems without heat recovery. Heat from the supply fan is implicitly represented in the denominator, whereas the energy of the exhaust fan increases the exhaust air temperature to above T_i. Systems without heat recovery will have a loss ratio larger than 1.

5. Setting up a Model for the IAAC!-Module.

5.1 The General Equation for an Energy Balance.

The general equation of an energy balance as the system changes from state 1 to state 2 can be formulated in terms of the rate of change of energy in a system dE/dt that is

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}(h_1 - h_2) + \frac{\dot{m}}{2}(c_1^2 - c_2^2) + \dot{m} \cdot g(z_1 - z_2) \tag{18}$$

where

$\dot{m}$ is the mass flow of the gas [kg/s]

$\dot{Q}$ is heat supplied to the system, positive value to the system [W]

$\dot{W}$ is work delivered from the system, positive value from the system [W]

h_1 and h_2 are gas enthalpies at states 1 and 2 [J/kg]

c_1 and c_2 are velocities at states 1 and 2 [m/s]

z_1 and z_2 are heights in relation to a reference level [m]

g is the gravitational acceleration

Several general assumptions are made in the setting up of the models and during the calculations:

- Air is treated as an ideal gas. Temperature differences are small, which allows the use of a constant value for the air's specific heat capacity $c_{pa}(T) = c_{pa}$ as well as a constant value for the vapors specific heat capacity $c_{pv}(T) = c_{pv}$
- The system is defined as the complete apparatus and air on both sides that have the temperatures of the interior and exterior air, respectively.
- Energy is conserved during the cycle. Kinetic and potential energies are neglected.
- The mass flow is constant in both working modes and independent of the switching of direction.
- Moisture phase change and transfer processes are not explicitly modelled. However, the effect of these processes is present if the processes influence monitored temperatures.

Enthalpy can for humid air be expressed as a function of temperature and moisture content, such that

$$h = c_{pa} \cdot t + x\left(r_o + c_{pv} \cdot t\right) \tag{19}$$

where

t is the air temperature in °C
x is the airs moisture content, kg water per kg dry air.
c_{pa} is the specific heat capacity of dry air = 1.01 kJ/(kg·K).
c_{pv} is the specific heat capacity of vapor = 1.84 kJ/(kg·K).
r_o is the specific heat of vapor = 2505 kJ/(kg·K) at 0 °C

5.2 Neglecting the Kinetic and the Potential Energies.

The latter terms of equation 14 containing velocities depicts the change of kinetic energy in the system whereas the last term containing heights represents change of potential energy. The change of potential energies can be neglected since inlets and outlets are situated at the same heights. In the assumptions, the kinetic energy is assumed to be very small, and the order of magnitude of this term is briefly shown below.

Values from an earlier test (Norberg 1991) are used to verify the assumption of the kinetic energy. Air velocities were not measured, but will in this context be calculated. The velocities can be calculated by means of the mass flow $\dot{m}$ and cross-section areas A of inlets and outlets.

$$c = \frac{\dot{m}}{\rho \cdot A} \tag{20}$$

The mass flow was determined to be $\dot{m} = 14.8 \cdot 10^{-3}$ kg/s and the air density estimated to be $\rho = 1.21$ kg/m^3. The areas and the calculated mean velocities are shown in table 1 below.

The change of kinetic energy is in the order of magnitude

$$\frac{\dot{m}}{2}\left(c_i^2 - 0\right) \approx 0.028\ W \tag{21}$$

in other words very small.

Location	Cross-section area [m^2]	Velocity [m/s]	Kinetic power [W]
Entering the indoor environment	0.009	1.37	0.028
Entering outdoor environment	0.0063	1.95	0.056

Table 1. Inlet- and outlet cross-section areas, calculated velocities and kinetic powers.

Also, work that is delivered from the fan to the air is a function of the pressure difference. In the test (Norberg 1991), pressure differences were monitored. The work delivered to the gas to generate the flow is basically

$$\dot{V}(p_i - p_e) = 12.3 \cdot 10^{-3}(5) = 0.062\ W \tag{22}$$

This indicates that work delivered to generate flow is very small.

5.3 A Model for the Module.

The setting up of a model for the IACC! has two objectives. The first is to define the enthalpy efficiency of the system with the model as base. The second is to identify and quantify the different energy flows so that measures can be taken to eventually improve the system.

Figure 5 illustrates a model of the IACC! during one cycle. The entities in the figure depict the cumulated energy during each mode.

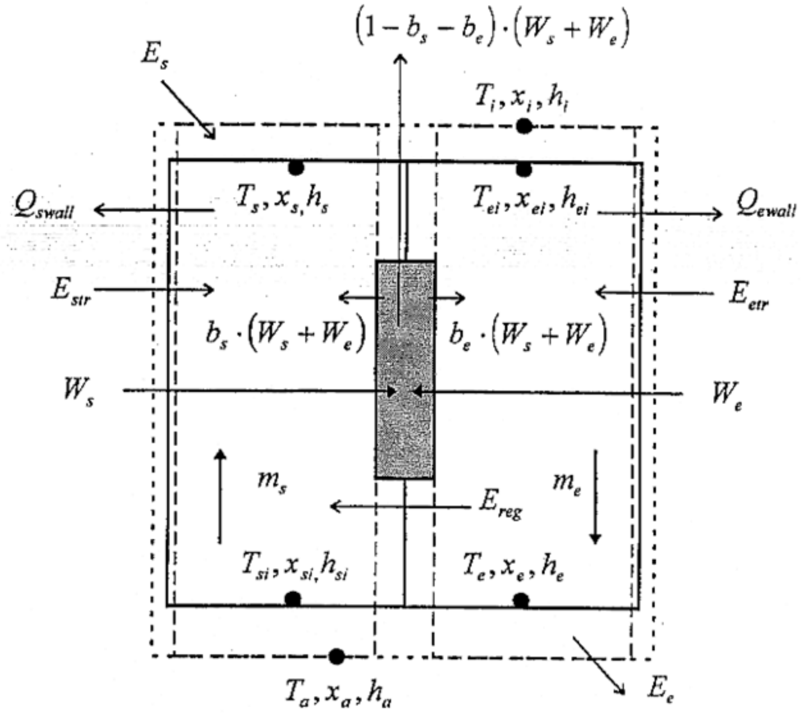

Figure 5. An illustration of the IACC! model.

The left side of figure 5 represents the supply mode of the cycle, through which the mass m_s of air has passed during the time t_s. Air having the temperature T_{si} and enthalpy h_{si} (*si* for *s*upply *i*nlet) enters the module. Energy is supplied to the air, here corresponding to the sum of $E_{reg}+E_{str}-Q_{swall}+b_s\cdot(W_s+W_e)$. E_{reg} is regenerated energy from the previous exhaust mode. b_s is a number between zero and one that represents the part of the fan work that has been converted to heat in the fan compartment and that is delivered to the supply air. Q_{swall} is heat transmitted from the module via the exterior wall, whereas E_{str} is energy transmitted to the module from the interior environment. E_{str} represents accumulated energy losses of any kind, such as heat transmission and air leakage. The supply air leaves the module having the temperature T_s and is supplied with heat and moisture from external devices E_s (not belonging to the ventilation system) as to reach the temperature of the interior air temperature T_i and enthalpy h_i.

On the right hand side, the exhaust mode allows the air quantity m_e to pass the module during the time t_e. Interior environment air enters the module having the temperature T_{ei} and enthalpy h_{ei} (*e*xhaust *i*nlet). The now cold module chills the warm air to the temperature T_e. This involves the net energy transfer from the air to the module that corresponds to the sum of $-E_{reg}+E_{etr}-Q_{ewall}+b_e\cdot(W_s+W_e)$. E_{reg} is energy to be regenerated to the next cycle, b_e is a number between zero and less than one (so that $b_s+b_e\le 1$) that represents the part of the fan heat that is delivered to the exhaust air and $\dot{Q}_{ewall}$ represents heat losses from the module to the exterior wall. Exhaust air is finally discharged with the energy content $\dot{E}_e$.

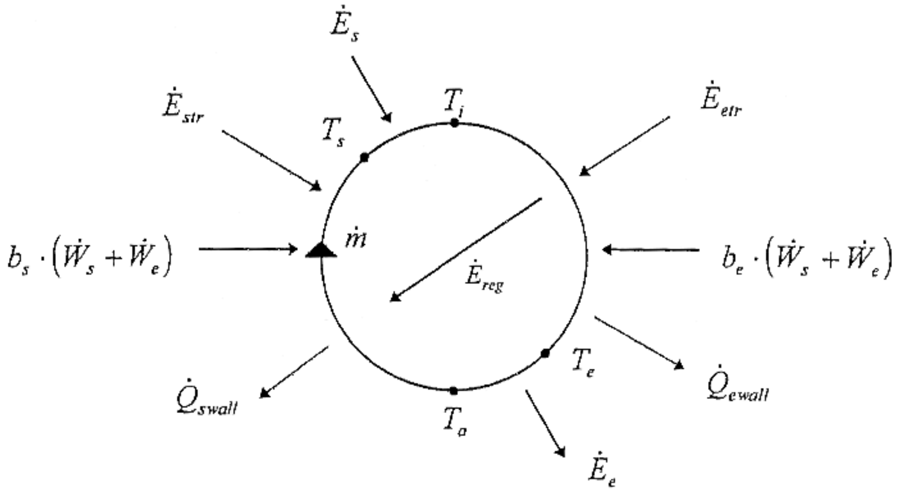

Figure 6. The process represented as being cyclic.

The change in total energy ΔE_s for the supply air within the dashed boundary, the supply sub-system, gives that

$$\Delta E_s = E_{reg} + E_{str} + E_s - Q_{swall} + b_s \cdot (W_s + W_e) + m_s \cdot (h_a - h_i) \tag{23}$$

Likewise, for the exhaust mode, the change of total energy in the sub-system is

$$\Delta E_e = -E_{reg} + E_{etr} - E_e - Q_{ewall} + b_e \cdot (W_s + W_e) + m_e \cdot (h_i - h_a) \tag{24}$$

On eliminating E_{reg} from both equations, the result becomes

$$\Delta E_s + \Delta E_e = E_{str} + E_{etr} - (Q_{swall} + Q_{ewall}) + (b_s + b_e) \cdot (W_s + W_e) + E_s - E_e \tag{25}$$

For the whole cycle, the change of energy of the enclosed system should be

$$\Delta E_s + \Delta E_e = 0 \tag{26}$$

if equilibrium in running conditions has been reached. This gives the relationship as follows

$$E_{str} + E_{etr} = (Q_{swall} + Q_{ewall}) - (b_s + b_e) \cdot (W_s + W_e) + E_e - E_s \tag{27}$$

What is interesting with this relationship is that $E_{str} + E_{etr}$ are measures to how close the actual process is to the ideal process. For the ideal process, $Q_{swall} + Q_{ewall} = 0$. This can be achieved with well-insulated external walls (this will be shown in a following section).

Also, if the system is balanced, the following equations are true.

$$W_s = W_e = W\,; \quad t_s = t_e = t\,; \quad m_s = m_e = m\,; \tag{28}$$

By dividing equation 27 by the time t, and substituting the terms of equation 28 into equation 27, the result becomes

$$\dot{E}_{str} + \dot{E}_{etr} = \left(\dot{Q}_{swall} + \dot{Q}_{ewall}\right) - \left(b_s + b_e\right) \cdot 2\dot{W} + \dot{m}\left[\left(h_e - h_a\right) - \left(h_i - h_s\right)\right] \tag{29}$$

where $\dot{E}_s$ and $\dot{E}_e$ have been substituted by $\dot{m}\left(h_i - h_s\right)$ and $\dot{m}\left(h_e - h_a\right)$, respectively.

The summing up of energy supplied to the system, divided by t, can be expressed as

$$\dot{E}_{tot} = \dot{E}_{reg} + \left(\dot{E}_{str} + \dot{E}_{etr}\right) + 2 \cdot \dot{W} + \dot{E}_s \tag{30}$$

Note that equation 30 has the setup according to equation 31 below, where the gain is the regenerated energy.

$$\dot{E}_{tot} = \dot{E}_{gain} + \dot{E}_{losses} \tag{31}$$

The energy supplied to compensate for the losses are thus

$$\dot{E}_{losses} = \left(\dot{E}_{str} + \dot{E}_{etr}\right) + 2 \cdot \dot{W} + \dot{E}_s \tag{32}$$

With equation 29 inserted and substitution with $\dot{E}_s = \dot{m}\left(h_i - h_s\right)$ gives the resultant

$$\dot{E}_{losses} = \left(\dot{Q}_{swall} + \dot{Q}_{ewall}\right) + \left(1 - b_s - b_e\right) \cdot 2\dot{W} + \dot{m}\left(h_e - h_a\right) \tag{33}$$

or in words: the sum of heat transmitted from the module walls and the energy content of the discharged exhaust air.

The quantity $\dot{E}_{reg}$ can be determined if equation 19 is divided by time t and $\Delta\dot{E}_s = 0$, giving

$$\dot{E}_{reg} = \dot{Q}_{swall} - \dot{E}_{str} - \dot{E}_s - b_s \cdot 2\dot{W} + m \cdot \left(h_i - h_e\right) \tag{34}$$

By inserting equations 32 into equation 30, the supplied energy corresponds to

$$\dot{E}_{tot} = \dot{m}\left(h_i - h_a\right) + \left(1 - b_s\right) \cdot 2\dot{W} + \dot{Q}_{swall} + \dot{E}_{etr} \tag{35}$$

5.4 The Enthalpy Efficiency.

The enthalpy efficiency of the system is on basis of the definition

$$\eta_e = \frac{E_{gains}}{E_{tot}} = \frac{\dot{E}_{gains} \cdot t}{\dot{E}_{tot} \cdot t} \tag{36}$$

Insertion of the entities from the previous section gives

$$\eta_e = \frac{\dot{m}\cdot\left(h_s - h_a\right) - b_s \cdot 2\dot{W} - \dot{E}_{str} + \dot{Q}_{swall}}{\dot{m}\cdot\left(h_i - h_a\right) + \left(1 - b_s\right)\cdot 2\dot{W} + \dot{Q}_{swall} + \dot{E}_{etr}} \tag{37}$$

or

$$\eta_e = \frac{\dot{m}\cdot\left(h_i - h_e\right) + b_e \cdot 2\dot{W} + \dot{E}_{etr} - \dot{Q}_{ewall}}{\dot{m}\cdot\left(h_i - h_a\right) + \left(1 - b_s\right)\cdot 2\dot{W} + \dot{Q}_{swall} + \dot{E}_{etr}} \tag{38}$$

There can arise difficulties in using equations 37 or 38. These come about in the terms b_s, b_e, $\dot{Q}_{swall}$, $\dot{Q}_{ewall}$, $\dot{E}_{str}$ and $\dot{E}_{etr}$ which cannot be directly measured (with exception to the wall heat transmission). The effect of b_s, b_e, $\dot{Q}_{swall}$ and $\dot{Q}_{ewall}$ is predicted to be small in association to the efficiency. The order of magnitude of $\dot{Q}_{swall}$ and $\dot{Q}_{ewall}$ are presumed to be small. Values for b_s and b_e can roughly be guessed whereas their sum can be calculated using material parameters of the modules inner and outer casings. The problem is quantifying the terms $\dot{E}_{str}$ and $\dot{E}_{etr}$. Their sum can be determined by the use of equation 29, but no relationships so far allow these to be determined individually.

5.5 Estimation of Unmeasured Entities.

The area of the motor compartment inner casing which allows one-dimensional heat transfer to the interior environment measures 19x(32+2*4) cm^2. The material is a chipboard with a thickness of 16 mm. The module itself is a steel casing having a thickness of 1 mm. The distance between the steel casing and the chipboard is a 3 mm wide air cavity. The task here is to evaluate the heat flow through this construction. There are several assumptions made here:

- A steady state approximation is used since the temperature inside the motor compartment was found to be relatively constant, as well as the temperature of the interior environment.
- The heat conductivity of each material is assumed as well as values on surface resistances.
- The heat transfer from the motor compartment to the interior environment is one dimensional and takes place only through the side wall.

The equation to be used is

$$\dot{Q}_{trm} = U_{trm} \cdot A_{trm}\left(T_m - T_i\right) \tag{39}$$

with

$$U_{trm} = \frac{1}{R_{si} + \sum \frac{d_j}{\lambda_j} + R_{sm}} \tag{40}$$

where

$\dot{Q}_{trm}$ is heat transmitted from the compartment to the interior environment.
A_{trm} is the area through which the heat is transmitted.
T_m is the temperature in the motor compartment.
T_i is the temperature of the interior environment.
R_{si} is the surface resistance between the modules wall and the interior environment.
R_{sm} is the surface resistance between the motor compartments mean temperature and the steel casing.
$\sum \frac{d_j}{\lambda_j}$ is the sum of the thermal resistances of each material layer.

R_{si} is assumed to have a value of around 0.13 m²·K/W. R_{sm} is more difficult to evaluate as the convection within the motor compartment is difficult to assess. The radiative heat transfer coefficient from each fan is in the order of magnitude of $4 \cdot \varepsilon \cdot \sigma \cdot T_m^3$ (where ε is the emissisivity and σ the Stefan-Boltzmanns constant) or with values inserted $4 \cdot 0.9 \cdot 5.67 \cdot 10^{-8}(273+40)^3 \approx 6.3$ W/(m²·K). If the convective heat transfer coefficient is of the same order of magnitude, the surface resistance will be around 0.08 m²·K/W.

The thermal resistance of the air cavity between the metal casing and the chip board is calculated in a similar way as R_{sm}. The air in the cavity will have a Nusselts number equal to one unless forced convection is present due to air leakage. Heat conduction through the air gives a heat transfer coefficient of $\frac{\lambda_{air}}{d_{air}} \approx \frac{0.026}{0.003} = 8.7$ W/(m²·K). Heat radiation gives a heat transfer coefficient of approximately $4 \cdot 0.8 \cdot 5.67 \cdot 10^{-8}(273+40)^3 \approx 5.6$ W/(m²·K), giving the air cavity a thermal resistance of totally 0.07 m²·K/W.

The thermal resistances of the steel and the chipboard layers are $\frac{d_{steel}}{\lambda_{steel}} = \frac{0.001}{40} \approx 0$ and $\frac{d_{chip}}{\lambda_{chip}} = \frac{0.016}{0.14} \approx 0.114$ m²·K/W.

The U-value of the construction is estimated to be 2.54 W/(m²·K). The area through which the heat is transferred is 0.076 m² or

$$\dot{Q}_{trm} = 0.193(T_m - T_i) \tag{41}$$

With values where the temperature difference is around 20 °C gives 3.9 W. The fan power ranges from 16 to 20 W. This gives a value for $(1-b)$ ranging from 0.25 down to 0.2. This calculation indicates that most of the heat generated by the fans is transferred to the ventilated air.

The quantities $\left(\dot{Q}_{swall} + \dot{Q}_{ewall}\right)$ have to be estimated. In a similar calculation as made above, it is assumed that the internal surface of the module that is adjacent to the "exterior" wall has a mean temperature. The mean heat flow from this surface and out to the external environment reads as follows:

$$\left(\dot{Q}_{swall} + \dot{Q}_{ewall}\right) = \frac{A_{wall}}{R_{wall}}\left(T_{wall} - T_e\right) \tag{42}$$

The area in contact with the exterior wall was measured to be 34x30 -(18.5x4)cm^2, where the latter term is the cross-section of the exhaust duct. The area A_{wall} is 0.095 m^2.

The thermal resistance is as before the sum of the resistance of each material layer. The thermal resistance of the exterior wall was $\frac{d_{cellpl}}{\lambda_{cellpl}} \approx \frac{0.10}{0.036} = 2.78$ m^2·K/W. The chipboard gives $\frac{d_{chip}}{\lambda_{chip}} = \frac{0.016}{0.14} \approx 0.114$ m^2·K/W and R_{se} can be expected to be in the regions of 0.09 m^2·K/W. Note that the latter two thermal resistances can be neglected in this calculation.

Equation 36 now takes on the appearance

$$\left(\dot{Q}_{swall} + \dot{Q}_{ewall}\right) = 31.8 \cdot 10^{-3}\left(T_{wall} - T_e\right) \tag{43}$$

A temperature difference of 30 °C gives the modest heat flow of 1 W. The term can in this particular experiment be neglected.

6. The Measurements.

6.1 Measurement Setup.

The measurements were performed in a laboratory where the internal and the external environments were simulated by climate chambers. The volume of each chamber is in the region of 35.5 m^3. The chamber are not air tight and therefore the flow, generated by the module, can be approximated to be the one under natural conditions. The largest volume of air transported by the module during one mode is around 0.4 m^3, and therefore a small part of the total volume. The pressure difference between the chambers was also read.

The "membrane" that separated the chambers was composed of, to a large extent, real external building constructions. However, the small part of the wall that the module was placed at was 10 cm cellular plastics.

Thermocouples of type T called B-10-T cc IEC/CEI were used. Figure 7 shows the set up. Some of these were screened from long wave radiation by means of a cylindrical glossy aluminum sheet, whereas the rest were exposed, either for the reason that screening was impossible or that the time constant was wished to be held at a minimum. Temperatures were gathered at a sampling rate of one second.

Other variables were momentarily read between the temperature measurements. The relative humidity was measured in the warm zone. Fan power and pressure differences between the zones were also checked and registered.

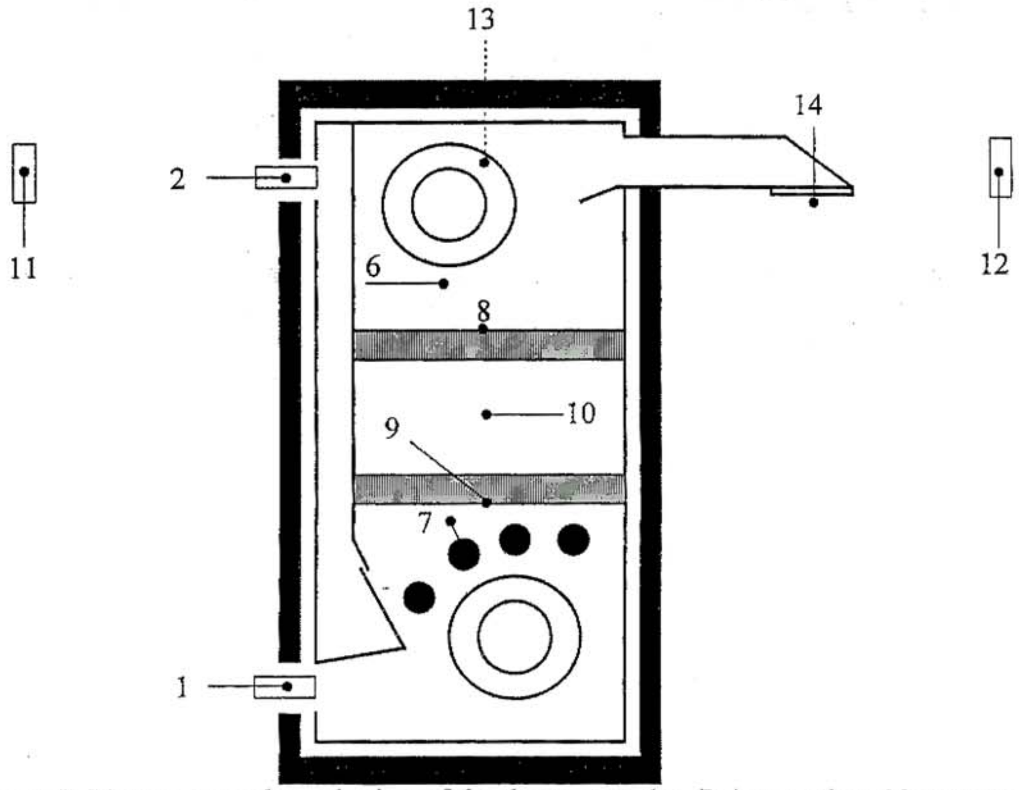

Figure 7. Placement and numbering of the thermocouples. Points enclosed by a rectangle indicate that the thermocouples were screened from long wave radiation.

Number		Number		Number	
1	Supply air outlet	7	Mounted on heating coil	10	Among the cardboard strips
2	Exhaust air inlet	9	Attached to aluminum grid	8	Attached to aluminum grid
6	Mounted next to the exhaust fan	11	Indoor air temperature	12	Outdoor air temperature
13	Fan compartment (upper region)	14	Exhaust outlet/ supply inlet		

List 1. Thermocouple location.

In general, the conditions in the climatic chambers were allowed to settle as the compressors and fans of the cool zone were on, as well as fans for the warm zone. The module was also allowed to function, even it was noticed that the fans of the chambers disturbed the flow through the module. The measurement system was turned on when the chambers heating/cooling and fan systems where turned off. In order to let the flow and temperatures of the module to reach a "stationary" cyclic state, the system was allowed to run at the expense of a slow continuous increase of temperature in the cold zone.

6.2 The Influence of the Thermocouples Mass.

The switching between modes occurs every 27.5th second, where the flowing air mass changes direction. Physically, this means that the measurement gauges that generate monitored values in form of air temperatures should be massless. However, the thermocouples do have a certain mass, meaning that the monitored value is under influence of the gauges response to a sudden temperature change. This matter is illustrated by Norberg, 1991.

The thermocouple used in this experiment was a thin kind of type T called B-10-T cc IEC/CEI (delivered by the company Pentronic AB, Sweden). The diameter of each metal leader was each 0.2 mm. Initially, the time constant of the thermocouple was said to be around 1 second. A later checkup of tables delivered from manufacturers indicated that the time constant was about 1.5 seconds. This is the value if the air velocity is around 20 m/s.

The time constant can be determined using measured data from the experiment. However, first the definition will be formulated in this section. A system (gauge and the medium) initially at equilibrium where both gauge and medium will have the same temperature, here called T_{init}. A sudden decrease of the medium temperature resulting from a step change, dropping ΔT units, will result in a temperature decay registered by the gauge. The temperature decay can be approximated to be exponential according to the equation below,

$$T_{gauge}(t) = T_{init} + \Delta T\left(1 - e^{-\frac{t}{t_o}}\right) \tag{44}$$

where

t is the time lapse from the temperature change of the medium.
t_o is the time constant.
$T_{gauge}(t)$ is the gauge temperature at time t.
T_{init} initial equilibrium temperature.
ΔT temperature change, negative value for temperature drop.

The time constant is determined when the term t/t_o equals one. This occurs when the term $1 - e^{-\frac{t}{t_o}} = 1 - e^{-1} \approx 0.632$ or in other words when 63.2% of the change has occurred.

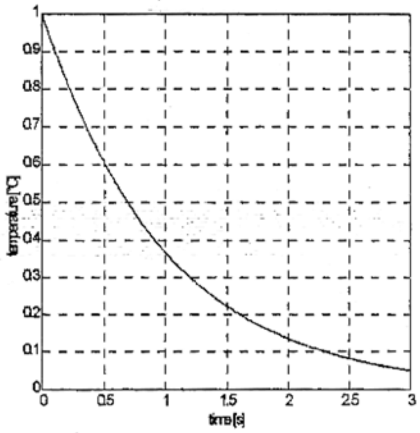

Figure 8. The time constant is in this illustration 1 second since 63% of the process has occurred at that point, where $T_{gauge}(t_o) \approx 0.37\,°C$.

In the experiment, the thermocouple placed under the exhaust air outlet is subject to large and fast temperature variations. In the shift between the exhaust and the supply mode, the thermocouple should immediately respond to the temperature drop from the last exhaust air temperature registration to the first supply air temperature. This temperature should reasonably be in the order of magnitude of the exterior environment air (unless long wave radiation from the snout should interfere with the values). The gauge is in this case subject to a step function as illustrated above.

A simple model can also be set up, by means of finite differences. The model is illustrated in figure 9 below.

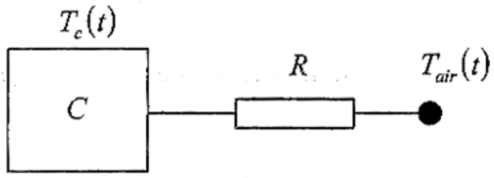

Figure 9. A model of the thermocouple in contact with air.

The figure shows a thermocouple junction with the thermal capacity C. The junctions temperature $T_c(t)$ is the temperature that is being measured at the time point t. The temperature which is wished to be measured is the air temperature $T_{air}(t)$, which for the moment can be set as constant from the time instance $t > 0$. The thermocouple and air node are connected by a thermal resistance. The thermal resistance is mainly the inverse of the convective heat transfer coefficient for two reasons: the convective heat transfer coefficient is larger than the radiative heat coefficient since the flow is forced, and the emissisivities are small since the surfaces are of non-corroded metal.

On applying the heat equation to this model, one finds that

$$C\frac{dT_c(t)}{dt}=\frac{(T_{air}-T_c(t))}{R} \tag{45}$$

with the condition that the thermocouples temperature was constant and equal to the air's initial temperature T_o before the temperature drop occurred, $T_c(0)=T_o$.

The solution of this problem is

$$T_c(t)=T_{air}+(T_o-T_{air})e^{-\frac{t}{R\cdot C}} \tag{46}$$

or in essence equation 44, where $t_o = R\cdot C$. This time constant should have a constant value during the process unless the convective heat transfer coefficient changes, for example due to air velocity or mass flow variations. However, the values for $R\cdot C$ is dependent on the mass flow and must be given different values depending on the flow set-point of the module.

Since the temperatures are not perfectly exponential, the discretized solution of equation 45 will be used, and this equation can be formulated such that the air temperature can be solved.

$$R\cdot C\frac{(T_c(t+\Delta t)-T_c(t))}{\Delta t}+T_c(t)=T_{air}(t) \tag{47}$$

T_c are the measured temperatures whereas $R\cdot C$ has to be determined experimentally. This has already been done in the supply mode of the cycle when the warm exhaust air, having heat the thermocouple, is substituted by cold supply air. The supply mode is illustrated in figure 10.

The appearance of the curve with compensation is "jumpy". It may seem that the mathematical model is under the influence of so-called numerical instability. This is however not the case: take the series close to sample number 250 in figure 10 as an example. The monitored values for samples 246 - 249 have pretty constant gradient, whereas sample 250 has a different gradient due to a disturbance of some kind, for example disturbances in flow due to some pressure build-up shortly after the reversing of the flow. The change in gradient of the recorded temperatures is a damped reflection of a larger change that occurred in the air. In other words, a small change in the recorded temperatures is the result of a larger change at the measurement point.

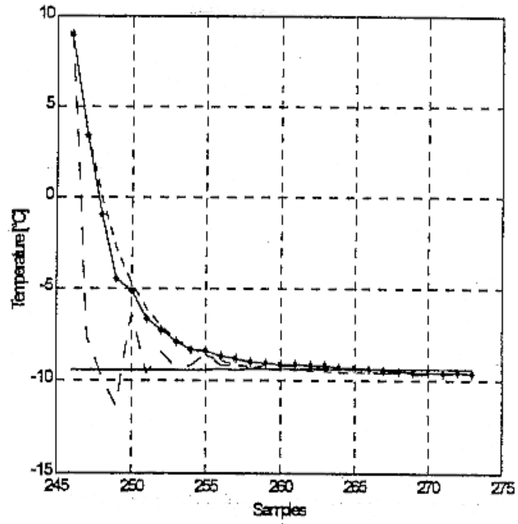

Figure 10. The measured air temperature at the snout is the filled curve with stars (*). The dotted curve is from the exponential equation with a time constant of 3 seconds. The dashed curve represents the actual air temperature based on the model. The somewhat horizontal curve shows the exterior temperature. The figure to the left depicts the case -10 °C outdoors and max. flow.

In the case where the outdoor air temperature was simulated to be -10 °C and the flow at a maximum, the original measured values (point 14) give a mean of -7.3 °C totally and -7.9 °C if the initial point is neglected. The corresponding means of the compensated values are -8.6 °C and -9.3 °C. The mean of the monitored "outdoor" air is -9.4 °C. The influence of the thermocouples mass corresponds to a mean of 1.3 to 1.4 °C in this supply mode. The time constant was 3 seconds.

The difficulty with the exhaust mode is that the exhaust air's temperature variation in time is not a step-like event. However, this can be tackled with the simple model with the assumption that the time constant is the same in exhaust mode as for the supply mode. The compensated values for the exhaust mode are illustrated in figure 11.

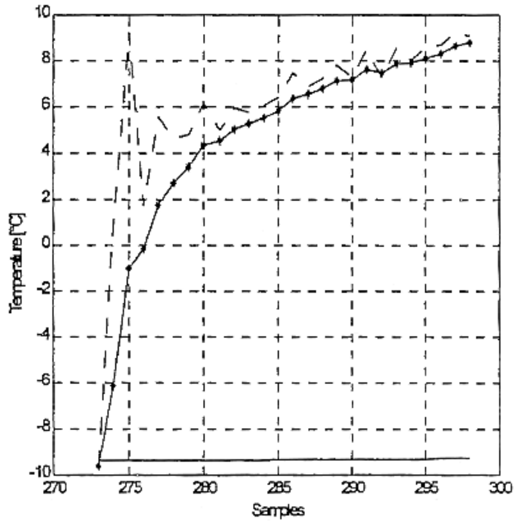

Figure 11. The original values of the exhaust air are plotted with the filled lines. The dashed curve represent compensated values with a time constant of 3 seconds. The horizontal line is the "outdoor" temperature. This test case is for -10 °C outdoors with max. flow.

The compensated values are somewhat larger than the measured values. Point 14 gives a mean of 4.6 °C for all values and 5.2 °C if the initial value is excluded. Corresponding values for the compensated series are 6.0 and 6.6 °C. Again, the difference is 1.4 °C between originally monitored and compensated values.

7. Results.

The mean values of the measured variables for each mode were used to calculate the different efficiencies. The results from measurements on the air flow and fan power are shown in table 2 below. Full details on that measurement procedure and results for the flow and fan power are documented in appendix 1.

Flow	Mean mass flow rate per mode [kg/s]	Mean power [W]
Min	$8.5 \cdot 10^{-3}$	16.4±0.3
Med	$12.6 \cdot 10^{-3}$	17.4±0.3
Max	$17.7 \cdot 10^{-3}$	19.3±0.3

Table 2. Mass flow and fan power.

Table 3 below shows the mean values at different measurement points. Variables on the left hand side of the column with the time constant τ are means of actually measured values, whereas the columns to the right hand side of τ list means that are based on compensation for the influence of the thermocouples thermal inertia. These are the values which will be used in the different definitions of efficiency. The notations for each variable is listed below table 3. The bar above each "T" denotes that it is a mean value.

Flow	$\overline{T}_{12}$	$\overline{T}_{s14}$	$\overline{T}_{e14}$	$\overline{T}_{1}$	τ	$\overline{T}_{s14C}$	$\overline{T}_{e14C}$	$\overline{T}_{1C}$	$\overline{T}_{2}$	$\overline{T}_{11}$
Max.	-15.47	-13.55	1.32	15.43	3.3	-15.37	3.32	15.08	18.90	18.90
Max.	-8.41	-6.94	5.22	15.93	3.3	-8.37	6.78	15.63	18.68	18.62
Max.	0.31	1.50	10.03	17.58	3.3	0.51	11.13	17.42	19.24	18.99
Max.	4.51	5.67	13.90	19.49	3.3	4.76	14.92	19.40	19.92	19.67
Max.	9.90	10.60	16.79	20.28	3.3	9.92	17.54	20.26	20.02	19.66
Med.	-15.62	-12.71	2.84	18.46	4.5	-15.54	5.95	18.32	19.18	19.14
Med.	-9.64	-7.05	6.81	19.22	4.5	-9.61	9.65	19.18	19.05	18.96
Med.	-0.48	1.41	10.96	19.71	4.5	-0.43	12.89	19.69	19.52	19.31
Med.	5.02	6.58	14.38	20.47	4.5	5.18	15.83	20.51	19.78	19.58
Med.	9.87	11.06	17.46	21.39	4.5	9.94	18.64	21.47	20.27	19.92
Min.	-16.01	-10.55	5.94	19.94	5.5	-14.55	10.33	19.99	19.20	19.01
Min.	-9.79	-5.55	8.54	20.23	5.5	-9.05	12.34	20.32	19.15	19.01
Min.	0.44	3.10	12.64	20.36	5.5	0.706	15.14	20.46	19.46	19.24
Min.	5.07	7.08	15.96	21.40	5.5	4.99	17.63	21.53	20.00	19.83
Min.	9.76	11.39	18.05	21.51	5.5	9.79	19.62	21.66	20.00	19.71

Table 3. Mean values from the last four modes of each measurement series.

$\overline{T}_{12}$ Mean exterior temperature.

$\overline{T}_{s14}$ Mean temperature at the supply inlet during the supply modes.

$\overline{T}_{e14}$	Mean temperature at the exhaust outlet the exhaust modes.
$\overline{T}_1$	Mean temperature of supply air.
τ	Assumed time constant.
$\overline{T}_{s14C}$	Compensated mean exterior temperature (should be of the same order of magnitude as $\overline{T}_{12}$).
$\overline{T}_{e14C}$	Compensated mean temperature at the snout during the exhaust modes.
$\overline{T}_{1C}$	Compensated mean temperature of the supply air.
$\overline{T}_2$	Mean temperature of the exhaust air entering the module.
$\overline{T}_{11}$	Mean interior temperature.

7.1 The Temperature Efficiency of the System.

This data of table 2 has been processed to determine the efficiency (the temperature efficiency) for the whole system, as defined in equation 2.

$$\eta_{t\,sys} = \frac{\overline{T}_{1C} - \overline{T}_{s14C}}{\overline{T}_2 - \overline{T}_{s14C}} \tag{48}$$

The reason for using $\overline{T}_2$ instead of $\overline{T}_{11}$ is that $\overline{T}_{11}$ is more exposed to long wave radiation than $\overline{T}_2$, see figure 7. However, the largest difference between these two variables is around 0.3 °C,

Flow	$\overline{T}_{s14C}$	η_{tsys}	$\overline{T}_2$	RH	$\overline{T}_{dew}$
Max.	-15.37	0.889	18.90	37	3.1
Max.	-8.37	0.889	18.68	38	3.3
Max.	0.51	0.903	19.24	39	4.1
Max.	4.76	0.966	19.92	44	6.6
Max.	9.92	1.024	20.02	47	7.7
Med.	-15.54	0.975	19.18	37	3.7
Med.	-9.61	1.005	19.05	38	3.6
Med.	-0.43	1.009	19.52	39	4.4
Med.	5.18	1.050	19.78	44	6.5
Med.	9.94	1.116	20.27	47	8.0
Min.	-14.55	1.023	19.20	37	3.3
Min.	-9.05	1.041	19.15	38	3.7
Min.	0.706	1.053	19.46	39	4.4
Min.	4.99	1.102	20.00	44	6.7
Min.	9.79	1.163	20.00	47	7.7

Table 4. The cold chambers air temperature, the temperature efficiency of the system and the temperature of the warm chamber as well as the relative humidity (RH) and dew point temperature are listed.

which is within the inaccuracy level 0.5 °C. The same applies to the variables $\overline{T}_{12}$ and $\overline{T}_{s14C}$ with the exception of the two min-flow series having the external temperatures -16 °C and -10 °C.

Results from the application of equation 48 are listed in table 4 and illustrated in figure 12. Note that the difference in $\eta_{t\ sys}$ between the minimum and maximum flow is fairly constant for all exterior temperatures. The change in slope on each side of the exterior temperature corresponding to 0 °C is also interesting, indicating that the phase changes of moisture affects the thermal performance of the module. The dew point temperature is called $\overline{T}_{dew}$ in table 4 and has been calculated on basis of the relative humidity and temperature of air in the warm chamber. $\overline{T}_{dew}$ indicates that only measurements where moisture phase change does not occur at any instance was when the air temperature in the cold chamber was around 10 °C.

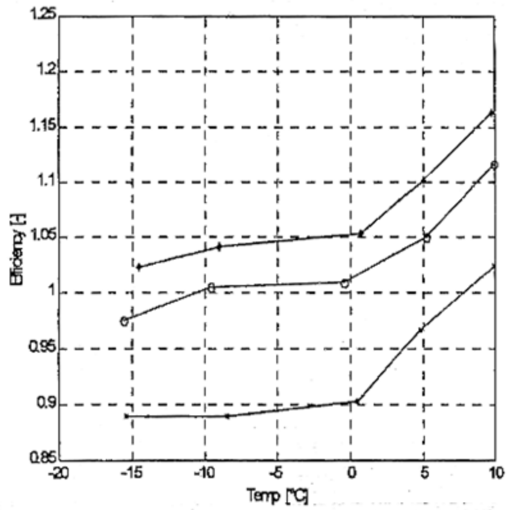

Figure12. The temperature efficiency of the system at maximum flow (x), at medium flow (o) and at minimum flow (*) as function of the "outdoor" temperature.

A problem with equation 48 is that the fans work, which is represented in the monitored term T_{1C}, does not appear in the definition. This can result in efficiencies larger than 1, which according to the definition is impossible for conventional heat exchangers without work or heat input. The application of this type of efficiency is inappropriate.

7.2 The Temperature Efficiency of the Heat Exchanger.

To study the temperature efficiency of the heat exchanger itself, the two measurement points T_6 and T_7 are processed with compensation for the thermocouples mass. The temperature that these two gauges show are air temperature and long wave radiation from the enclosing surfaces. What also must be kept in mind is that a small part of the heat that is generated by the fans motors will interfere with the performance of the heat exchanger itself. This heat, which is transmitted from the motor compartment to the measurement zone by means of conduction and convection, is assumed to be small in comparison to heat that is ejected from the motor compartment to the exiting supply air. The difference between $\overline{T}_{7C}$ and $\overline{T}_{1C}$ is explained by this ejector phenomenon, and also to the fact that the thermocouple that measures T_7 is exposed to cold surfaces in the module, whereas the thermocouple that measures T_1 is exposed to the warmer interior surfaces.

The definition of interest is equation 1, which formulated with the present variables gives

$$\eta_t = \frac{\overline{T}_{s7C} - \overline{T}_{s6C}}{\overline{T}_{e7C} - \overline{T}_{s6C}} \tag{49}$$

where the index e is for exhaust and s is for the supply cycle.

Flow	$\overline{T}_{s14C}$	$\overline{T}_{s7}$	$\overline{T}_{e7}$	$\overline{T}_{s6}$	$\overline{T}_{e6}$	$\overline{T}_{s7C}$	$\overline{T}_{e7C}$	$\overline{T}_{s6C}$	$\overline{T}_{e6C}$	η_t
Max.	-15.37	11.78	16.31	-5.39	3.53	10.83	17.35	-6.87	5.32	0.731
Max.	-8.37	12.16	16.72	-0.83	6.78	11.35	17.54	-2.05	8.09	0.684
Max.	0.51	14.86	18.02	5.83	11.07	14.38	18.56	5.04	12.00	0.690
Max.	4.76	17.25	19.54	9.62	14.57	16.88	19.91	8.89	15.36	0.725
Max.	9.92	18.44	19.98	13.62	17.15	18.19	20.24	13.10	17.74	0.713
Med.	-15.54	15.65	18.25	-4.72	5.38	15.01	18.99	-6.82	7.83	0.846
Med.	-9.61	17.59	18.82	0.70	9.28	17.20	19.24	-1.23	11.45	0.900
Med.	-0.43	18.10	19.33	6.87	12.71	17.72	19.73	5.52	14.18	0.859
Med.	5.18	19.15	19.89	10.84	15.44	18.95	20.11	9.79	16.53	0.888
Med.	9.94	20.28	20.70	14.42	18.21	20.14	20.86	13.60	19.08	0.901
Min.	-14.55	19.58	19.54	-3.47	10.10	19.55	19.56	-7.04	14.02	1.000
Min.	-9.05	19.63	19.76	1.28	11.88	19.59	19.80	-1.62	15.01	0.990
Min.	0.706	19.99	19.90	8.64	14.91	20.00	19.91	6.77	16.85	0.931
Min.	4.99	20.96	20.82	12.18	17.81	21.00	20.79	10.63	19.38	1.021
Min.	9.79	21.00	20.79	15.42	19.24	21.03	20.76	14.26	20.39	1.042

Table 5. Mean values from the last four modes of each measurement series as well as the temperature efficiency of the heat exchanger.

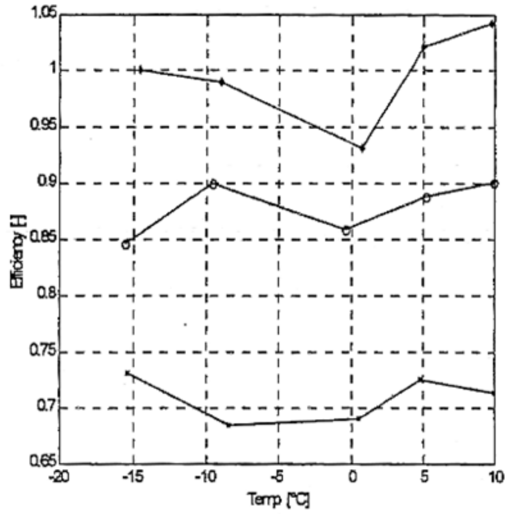

Figure13. The temperature efficiency of the heat exchanger at maximum flow (x), at medium flow (o) and at minimum flow (*) as function of the "outdoor" temperature.

The efficiency of the heat exchanger surpasses 1.00 for the low flow, indicating that heat from the fans is present in the temperature measurements. The most reliable results are those for the maximum flow since heat from the fans is relatively small in proportion to regenerated heat and due to the fact that the thermocouples mass have a relatively smaller influence on the measured values. It should be noted that it's not only the heat exchanger alone that regenerates the heat. The whole module itself acts as a heat exchanger and due to this reason, the temperatures at the inlets and outlets should be used in the efficiency definition. This can be done by using the energetic efficiency.

7.3 The Energetic Efficiency.

Equation 7 states the energetic efficiency, such that

$$\eta_e = \frac{\dot{m} \cdot c_p\left(T_s - T_a\right) - b_s \cdot \dot{W}_s}{\dot{m} \cdot c_p\left(T_i - T_a\right) + \left(1 - b_s\right) \cdot \dot{W}_s + \dot{W}_e} \tag{50}$$

Since heat generated by the fans motors is delivered to the fan motor compartment and from there to the air, this equation is more suitable if the work of the two fans are summed and b_s represents the part of the total fan energy that is delivered to the supply air. If the fan work of the two modes are the same, the energetic efficiency becomes

$$\eta_e = \frac{\dot{m} \cdot c_p\left(T_s - T_a\right) - b_s \cdot 2\dot{W}}{\dot{m} \cdot c_p\left(T_i - T_a\right) + \left(1 - b_s\right) \cdot 2\dot{W}} \tag{51}$$

This equation is directly comparable with equation 37, where the losses are set to zero and enthalpies are replaced with temperatures.

b_s cannot be measured. By means of calculations and assumptions, the order of magnitude can be estimated as follows: Let b_e be the part of the total fan energy that is supplied to the exhaust air. $(1-b_s-b_e)$ will therefore corresponds to the part that is transmitted from the motor compartment to the surroundings, or in absolute values Q_{trm} as described in section 5.5. This value is dependent on the difference in temperature of the motor compartment, $\overline{T}_{13}$, and the indoor air, $\overline{T}_2$ as shown in table 6.

Now, if $b_s \approx b_e$, the energetic efficiency using this assumption will be denoted by $\eta_{e\,50}$. A small speculation will however indicate that $b_s > b_e$ for two reasons. The first reason is that the temperature of the supply air is lower than that of the exhaust air, resulting in an increased heat transfer to the supply mode. The second reason is that the ejector effect is larger for the supply mode than that of the exhaust mode. If it is assumed that b_s accounts for two thirds of (b_s+b_e), the efficiency denoted by $\eta_{e\,67}$ can be determined. (Note that the difference between $\eta_{e\,50}$ and $\eta_{e\,67}$ is less that half a percent, see table 6 below.)

The equation used for determining the energetic efficiency is with the current notations

$$\eta_e = \frac{\overline{\dot{m}} \cdot c_p\left(\overline{T}_{1C} - \overline{T}_{s14C}\right) - b_s \cdot 2\overline{\dot{W}}}{\overline{\dot{m}} \cdot c_p\left(\overline{T}_2 - \overline{T}_{s14C}\right) + \left(1-b_s\right) \cdot 2\overline{\dot{W}}} \tag{52}$$

Flow	$\overline{T}_{s14C}$	$\overline{T}_2$	$\overline{T}_{13}$	Q_{trm}	b_s+b_e	η_{e50}	η_{e67}
Max.	-15.37	18.90	31.9	2.5	0.87	0.831	0.829
Max.	-8.37	18.68	33.2	2.8	0.85	0.815	0.813
Max.	0.51	19.24	34.7	3.0	0.84	0.800	0.797
Max.	4.76	19.92	37.5	3.4	0.82	0.836	0.833
Max.	9.92	20.02	39.0	3.7	0.81	0.830	0.825
Med.	-15.54	19.18	34.8	3.0	0.83	0.901	0.913
Med.	-9.61	19.05	34.9	3.1	0.82	0.913	0.910
Med.	-0.43	19.52	35.8	3.1	0.82	0.880	0.878
Med.	5.18	19.78	38.6	3.6	0.79	0.876	0.873
Med.	9.94	20.27	40.6	3.9	0.78	0.871	0.867
Min.	-14.55	19.20	34.8	3.0	0.82	0.915	0.913
Min.	-9.05	19.15	35.8	3.2	0.80	0.912	0.910
Min.	0.706	19.46	36.7	3.3	0.80	0.864	0.861
Min.	4.99	20.00	39.0	3.7	0.77	0.866	0.862
Min.	9.79	20.00	40.5	4.0	0.76	0.825	0.818

Table 6. List of the part of the fan energy that is supplied to the air. The two latter columns show the energetic efficiency with the assumption that half of fan energy that is delivered to the flowing air goes to the supply air, η_{e50}, and the energetic efficiency with the assumption that two thirds of fan energy that is delivered to the flowing air goes to the supply air, η_{e67}.

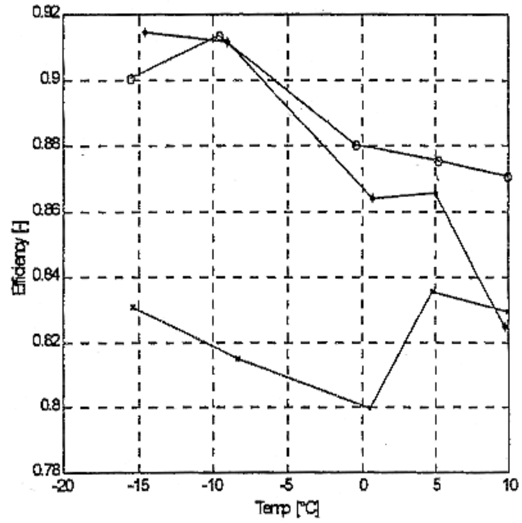

Figure 14. The energetic efficiency of the system η_{e50} at maximum flow (x), at medium flow (o) and at minimum flow (*) as function of the "outdoor" temperature.

What can be seen in figure 14 is that the efficiency does not exceed 1.00. The efficiency is in the region between 0.8 and 0.92 depending on the air flow and "outdoor" temperature. A rather surprising result is that most values of the medium flow curve are larger than those of the minimum flow. A reason for this may be that the adjustments of air flow during the temperature measurements were not the same as that for the air flow measurement. However, if this is untrue, the results become very interesting since this indicates that the system's efficiency, according to the definition, can be optimized in terms of air flow versus mode period (cycle time).

As stated in section 4.3, the equation applied within this context is the definition based on an ideal system and on measured temperatures. The results shown in table 6 and figure 14 can be compared with the so-called temperature efficiency of the supply air. However, if one wishes to determine the true amount of energy that is regenerated in relation to the total energy supplied, the enthalpy efficiency (see section 5.4) would have to be applied for two reasons. The first is that energy involved in moisture transfer and phase changes cannot be neglected in normal running condition in, for example, Scandinavian countries. The second reason is that effect of mass and heat transfer in form of air leakage and heat transmission have to be taken into account.

The use of the enthalpy efficiency is more complex than the other efficiencies listed in this report for several reasons. One reason is that the relative humidity has to be measured in all inlets and outlets of a system. The second is that air leakage and heat transmission have to be included; these are difficult to determine. This leads to problems in the calculation of the total energy involved in the process, as earlier described in section 5.4. With these points in mind, the energetic efficiency is a simpler entity to determine. It is also comparable with the temperature efficiency of the supply air. However, it cannot within this context be used to determine the total energy supplied nor the losses of a system. The losses of the system will be dealt with in the following section.

7.4 The Loss Ratio.

The loss ratio is an factor that can relatively easily be determined. The definition was proposed in section 4.5, equation 17 as the ratio between energy content of the rejected air and the energy content of the indoor air. With the monitored values, equation 17 takes on the appearance such that

$$\gamma = \frac{\overline{T}_{e14C} - \overline{T}_{s14C}}{\overline{T}_2 - \overline{T}_{s14C}} \tag{53}$$

The results are plotted in figure 15. These results show the reduction in energy losses by replacing natural ventilation with the IACC!, provided that the change of air rate in the ventilated zone is the same. The reduction is substantial for high air flows and low outdoor temperatures: for natural ventilation the air change rate increases as the outdoor temperature decreases!

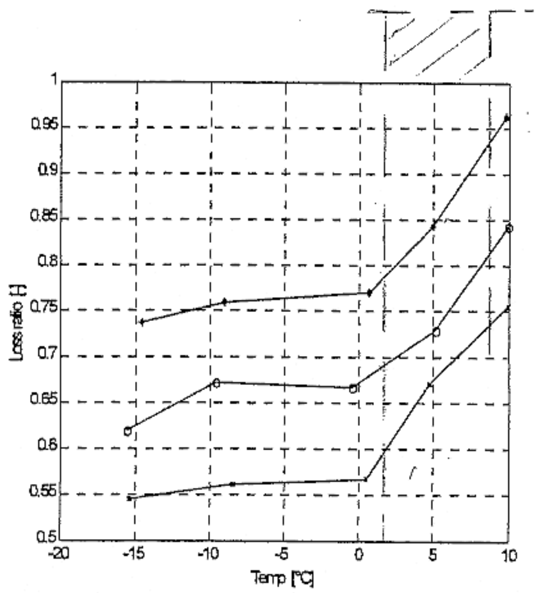

Figure 15. The loss ratio at maximum flow (x), at medium flow (o) and at minimum flow (*) as function of the "outdoor" temperature.

Equation 53 does not take into account differences in moisture content. Again, this can be taken into consideration by substituting the temperatures with enthalpies. Below, an example is shown on how this can be achieved by means of a Mollier diagram.

The indoor air condition is approximately 19 °C with a relative humidity of 37%. The moisture content is found to be 5.1 g/kg, see figure 16. The outdoor temperature is slightly below -15 °C with a relative humidity that can be assumed to approximately 100%. This gives a moisture content of 1 g/kg. On a line between the indoor and the outdoor air states, where that line crosses the exhaust air temperature of around 3 °C, the moisture content can be expected to be 3.1 g/kg. On applying these temperatures and corresponding moisture contents with equation 19, the enthalpy is for the indoor air 32 kJ/kg, -13 kJ/kg for the outdoor air and 11 kJ/kg for the rejected air. The enthalpy loss factor would be formulated such that

$$\gamma_h = \frac{\bar{h}_{e14C} - \bar{h}_{s14C}}{\bar{h}_2 - \bar{h}_{s14C}} \tag{54}$$

With the enthalpies insert, the value would give some 0.535 whereas the use of temperatures alone give a value corresponding to 0.543.

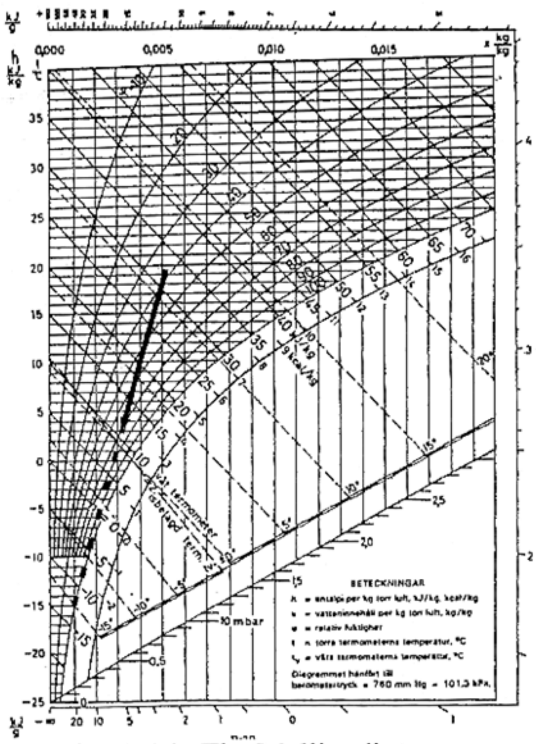

Figure 16. The Mollier diagram.

10. Literature.

ELAB 1994: Luftkvalitet, ventilation och energi. ISBN 91-972335-3-6. (In Swedish).

Norberg, C. 1991: PM Betavärmeväxlaren - Koncept. Vattenfall Utveckling AB.

APPENDIX 1.

DETERMINATION OF THE VOLUMETRIC FLOW.

A tracer gas technique was used to determine the volumetric flow of the module. The exhaust snout was placed in a sealed compartment which is usually used for determining the air tightness of building constructions (guarded pressure box). The compartments internal volume measured V_o = 4.426 m^3. Since this is a rather small volume in relation to the pulsing volumes of the module, the volume of the compartment was given capabilities of expanding by means of a bag, commonly used in "bag method". The bag has a volume of 530 liters, whereas the module can deliver around 400 liters ($\approx$ 15 l/s * 27.5 s) one way at maximum flow state.

Three separate measurement series were performed, where the air flow was varied. These are here called *min*imum, *med*ium and *max*imum. The pressure difference between the interior and the exterior of the compartment was registered in all series as well as the power supplied to the fans. The temperatures were the same internally and externally.

Prior to every series, tracer gas N_2O (laughing gas) was inserted in the compartment. The decaying concentration of the tracer gas was continuously registered by an IR analyzer of the type Binos, produced by Leybold-Heraeus. For the three series, concentration decay over time are shown in figures 1.1, 1.2 and 1.3.

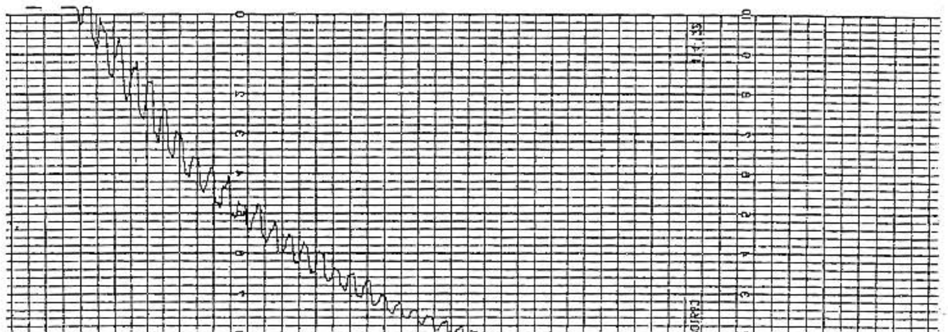

Figure 1.1 The module has been set to min flow. The y-axis represents concentration with the unit ppm (part per million) ranging from 0 - 1000 ppm. The x-axis represent time with 1 minute per tick.

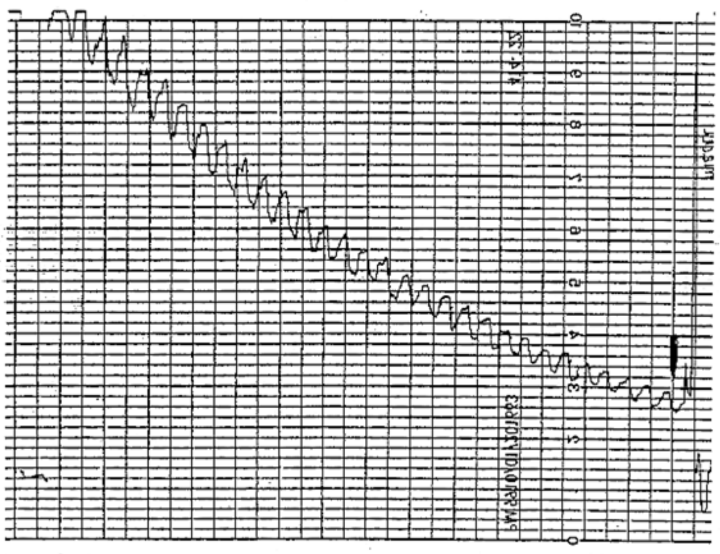

Figure 1.2 The module has been set to med flow. The y-axis represents concentration with the unit ppm (part per million) ranging from 0 - 1000 ppm. The x-axis represent time with 1 minute per tick.

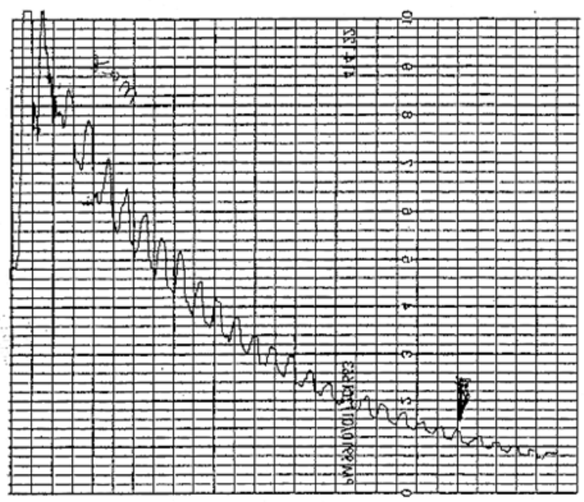

Figure 1.3 The module has been set to med flow. The y-axis represents concentration with the unit ppm (part per million) ranging from 0 - 1000 ppm. The x-axis represent time with 1 minute per tick.

The calculation procedure for interpreting monitored values assumes that the concentration of tracer gas decays exponentially in time and that the air change rate is constant. The rate of air change n in the measured enclosure can be determined using equation 1.1 below:

$$n = \frac{1}{t} \ln \frac{C_o}{C_t} \qquad (1.1)$$

where

t is the lapsed time during the measurement.
C_0 is the initial concentration of tracer gas at time = 0 s.
C_t is the concentration of tracer gas at time = t s.

Concentration has the unit *ppm* (part per million).

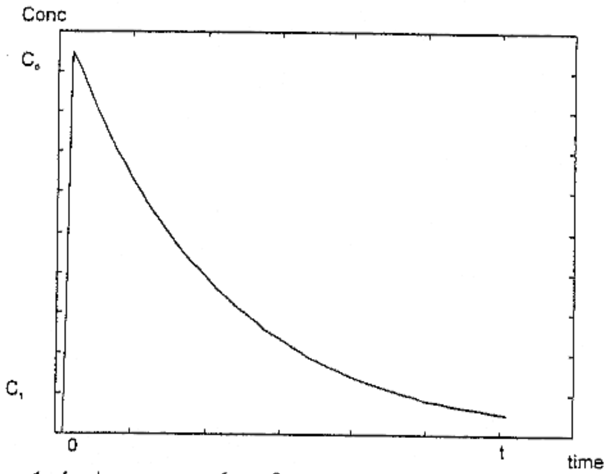

Figure 1.4 An example of tracer gas concentration decay.

Since the rate of air change is pulsed in this particular experiment, a slight modification has to be done when dealing with equation 1.1. The ordinary supply mode is in this experiment actually an exhaust mode, since air is taken from the compartment. The end of the supply mode will result in a local maximum concentration and a minimum volume in the compartment. Therefore, the air change rate n_s can be determined from the interpolation of maximums along the measured curves. The volumetric flow of the supply cycle $\dot{V}_s$ will be

$$\dot{V}_s = n_s \cdot V_o \qquad (1.2)$$

n_s is calculated from equation 1.1. However, the curves are seldomly perfectly exponential. Therefore, some points have been chosen from the curves, been plotted in a time-logarithm of concentration diagram, so that the air change rate has been found as the slope of a straight line "passing" all points by means of linear regression. See figure 1.5.

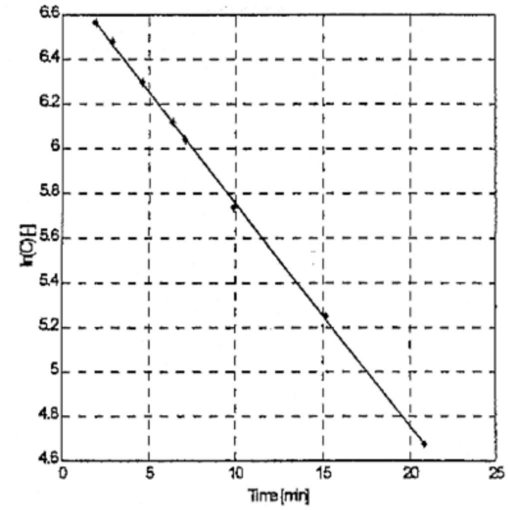

Figure 1.5 Plot of time versus the logarithm of concentration for the maximum flow at the supply mode.

The same can be done for the exhaust mode (in this experiment it supplies uncontaminated air into the compartment). The supplying of uncontaminated air and the increase of compartment volume result in local minimums along the decaying curve. The volumetric flow, based on the nominal volume, is

$$\dot{V}_e = n_e \cdot V_o \tag{1.3}$$

However, the compartment will have expanded and the following equation should be more accurate, giving the exhaust mode's volumetric flow $\dot{V}_e^*$

$$\dot{V}_e^* = n_e\left[V_o + n_e \cdot V_o \cdot t_c\right] \tag{1.4}$$

where t_c is the time lapse during the mode, here 27.5 seconds.

The measured and calculated results are presented in table 1.1 below.

Flow state	Mean power [W]	Total pressure difference [Pa]	$\dot{V}_s$ [l/s]	$\dot{V}_e$ [l/s]	$\dot{V}_e^*$ [l/s]	$\overline{\dot{V}}$ [l/s]
min	16.4±0.3	-	3.567	3.502	3.578	3.6
med	17.4±0.3	0.6	5.289	5.182	5.348	5.3
max	19.3±0.3	1.2	7.391	7.170	7.490	7.4

Table 1.1 Measured and calculated results. The volumetric flows contain 3 decimals for the reason of comparison. $\overline{\dot{V}}$ is the mean of $\dot{V}_s$ and $\dot{V}_e^*$. It should be noted that these values represent a *one-way flow during the cycle time* t_C. The actual flow during each mode is twice the number in the table.

It should be noted that the volumetric flows here depict the net rate of air exchange. The volumetric flow per mode of the cycle (the modes being exhaust or supply) is twice the numbers of table 1.1.

A well balanced module should give that $\dot{V}_s \approx \dot{V}_e^*$, which is the case above. However, it is from this measured data impossible to measure small unbalances. It is the first time this type of measurement procedure was used, and the equipment with regards to the compartment can be improved (such as the mix of air volumes within the compartment and location of the IR-analyzer gauge). At the time of the experiment, the advantages of this procedure were unknown. This method has several benefits. One is that it is possible from one set of measurement series to determine the volumetric flow in both directions. Another is that the module can operate in normal conditions with regard to pressure differences. The method can give good and accurate results which is representative for the cycle. A problem with the method is the relatively long time it takes compared to "hand" instruments and that the method is suited for laboratory applications only.

The mass flows at the given flow states are found by multiplying the volumetric flow by the air's density. In future calculations, the mean values will be used based on the assumption that the flow rates are balanced.

$$\overline{\dot{m}} = \rho \cdot \overline{\dot{V}} \qquad (\rho_{20^\circ C} = 1.188 \; kg/m^3) \qquad (1.5)$$

The equation gives results that are listed in table 1.2.

	min	med	max
Mass flow [kg/s]	$8.48 \cdot 10^{-3}$	$12.64 \cdot 10^{-3}$	$17.68 \cdot 10^{-3}$

Table 1.2 The mass flows per mode as function of the flow state.

Using this data, a binomial equation was fit, giving

$$\overline{\dot{V}} = -82.8021 + 8.6318 \cdot P - 0.2051 \cdot P^2 \qquad (1.6)$$

and a

$$\overline{\dot{m}} = 2 \cdot \left[-98.3689 + 10.2545 \cdot P - 0.2436 \cdot P^2 \right] \cdot 10^{-3} \qquad (1.7)$$

valid for active powers ranging between 16.4 and 19.3 W.