ANALYTICAL AND NUMERICAL ANALYSIS OF DYNAMIC INSULATION

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ABSTRACT

Analytical solutions to the heat flow equation in a dynamic insulation is presented. A numerical least: squares method, called the gradient method, to estimate the steady state and transient air flow through a dynamic insulation is presented. The method was applied to measurements from a house with dynamic insulation in the ceiling. The analysis showed that 40% of the total inlet air passed through the insulation. The measured energy efficiency was very similar to the calculated one. The conclusion from the study was that dynamic insulation requires a house constructed to much more exacting standards than was the case, in order to work properly.

INTRODUCTION

In 'dynamic' insulation, part of the inlet or exhaust air or both passes through the insulation. If inlet air is used this is called counterflow insulation since the direction of air flow is opposite to that of heat flow. When exhaust air is used, this is called parallel flow insulation. There is also the possibility of using counterflow insulation in one part of the building envelope and parallel flow insulation in another part. There are three reasons for using dynamic insulation with inlet air in cold climates:

Energy loss is less than in a house without heat recovery in the ventilation system.
The insulation filters the air to a theoretically very high degree.
The inlet air is preheated, thus providing a high degree of comfort in the house.

Dynamic insulation is similar to a heat exchanger between the inlet and exhaust air. The energy efficiency of dynamic insulation depends on the amount of air that passes through the insulation. Typically, the air flow through the insulation is produced by keeping the house at a pressure lower than the pressure of the ambient air.

Only few laboratory and full scale experiments have been made with dynamic insulation (Anderlind 1983, Arquis 1986, Roots 1994). The experiments reported so far have been steady state ones, i.e. no variation with time has been studied. The most extensive studies have been performed in the laboratory. The main goal of the study presented here was to

- Analyse the steady state and transient behaviour of the dynamic insulation.
- Measure the performance of a dynamic insulation installed in a house exposed to realistic conditions.
- Measure the importance of wind speed, wind direction, ambient temperature and sunshine on the performance of the dynamic insulation.

A house with dynamic insulation in the ceiling was continuously measured for about one year (1993). The house, which was called the OPTIMA house, was a single storey one family house of 116 m^2 built at Dalby in the south of Sweden. The house was not occupied during the measurement period.

THE OPTIMA HOUSE

The OPTIMA house is described in detail in Elmroth and Fredlund (1993). The house was unoccupied during this period. The house was a single storey house with ambient ventilated attic. The ambient air passed from the outside into the attic, through 0.3 m of loose fill insulation and into a small air space over a gypsum board. The air passed into the living space through five inlet terminals in the gypsum board. The exhaust air passed to a central fan through five air outlets. The ventilation rate was set to 60 l/s or 0.75 ach. The important question was whether the air passed through the insulation or through leakage paths in the rest of the house. Three different methods of measuring the air flow through the insulation were used in the OPTIMA house:

1. Simulation and measurement of the temperature gradient inside the insulation.
2. Tracer gas technique.
3. Direct measurements of air flow in the inlet terminals.

In the study presented here the first method, called the *gradient method*, was used.

ANALYTICAL ANALYSIS

The air passes through a loose fill insulation at low velocity (~ 0.2 mm/s) which permits the use of a simple laminar model. For the macroscopic behaviour it is convenient to describe the loose fill insulation as homogeneous and isotropic. With these assumptions the steady state heat transfer equation in one dimension is (Anderlind 1983):

$$\frac{d^2T(x)}{dx^2} - \nu \frac{dT(x)}{dx} = 0, \qquad (1)$$

$$\text{with} \quad \begin{aligned} T(0) &= T_0 \\ T(L) &= T_L \end{aligned}$$

where T is the temperature in °C, $\nu = u\,\rho_a\,c_a/\lambda$ is the ratio of convection to conduction, λ is the heat transfer coefficient for the insulation, u is the air flow which is positive in the direction of increasing x, ρ_a is the density of air, c_a is the heat capacity of air and L is the height of the analysed part of the insulation. The upper part of the insulation is x=0 and the lower part is x=L. The air flow is downward. For the OPTIMA house was $u\approx0.2$ mm/s, and $\nu\approx6$ m^{-1}. The transient heat transfer equation in one dimension is:

$$a\frac{\partial^2 T(x,t)}{\partial x^2} - a\nu\frac{\partial T(x,t)}{\partial x} = \frac{\partial T(x,t)}{\partial t} \qquad (2)$$

$$\text{with} \begin{cases} T(0,t) = T_0(t), \quad T(L,t) = T_L(t) \\ T(x,t_0) = T^0(x) \end{cases}$$

where t is the time is seconds, ρ_i is the density of the insulation and c_i is the heat capacity of the insulation, $a=\lambda/\rho_i\,c_i$ is the diffusivity of the insulation. The product $a\nu$ is the *thermal velocity*. It describes at what rate a thermal front would move if conduction was neglected. For the OPTIMA house the thermal velocity was 0.01 mm/s with an air flow of 0.2 mm/s. This means that it would take roughly 8h for a thermal front to pass through an insulation of 0.3 m if the conduction was neglected.

The analytical solution to the steady state equation (1) is straightforward:

$$T(x) = T_0 + \frac{T_0 - T_L}{e^{\nu L} - 1}\left(1 - e^{\nu x}\right) \qquad (3)$$

The solution to the transient equation (2) is more complex. The standard approach is to use the Laplace transform in the time domain (Carslaw and Jaeger, 1959). The solution is shown in detail in Wallentén (1994) here is only the result presented:

$$T(x,t) = e^{x\nu/2}\,T_0\left(\frac{\sinh \frac{\nu}{2}(L-x)}{\sinh \frac{\nu}{2}L} + \sum_{n=1}^{\infty}(-1)^n e^{-\left(\frac{a\nu^2}{4} + \frac{a n^2\pi^2}{L^2}\right)t} \cdot \frac{\sin \frac{n\pi}{L}(L-x)}{\frac{\nu^2 L^2}{8n\pi} + \frac{n\pi}{2}} \right) \qquad (4)$$

The solution is thus the sum of an infinite series. The steady state solution is approached for large t. The terms in (4) decrease exponentially in time. The behaviour of the first term is:

$$e^{-\left(a\nu^2/4 + a\pi^2/L^2\right)t} = e^{-t/\tau}$$

$$\tau = \frac{}{a\nu^2/4 + a\pi^2/L^2}$$

$$\approx \frac{1}{20\cdot10^{-6} + 240\cdot10^{-6}} = 3846\text{ s} = 64\text{ min} \qquad (5)$$

where τ is a time constant for the insulation. Table 1 shows the value of τ for some different u for the OPTIMA house. The values for the parameters used in the calculations are: L=0.3 m, λ=0.042 W/mK, ρ_a=1.27 kg/m³, ρ_i =19 kg/m³, c_a=1005 J/kgK and c_i=1000 J/kgK.

Table Relationship between τ and u for the OPTIMA house

u (mm/s)	0.01	0.1	0.2	0.5	2.0
τ (min)	69	68	64	45	7

The ratio of the transient (4) to the steady state temperature (3) for different u and constant x for a step change in temperature is plotted in figure 1, and in figure 2 for constant u and different x, as a function of t/τ.

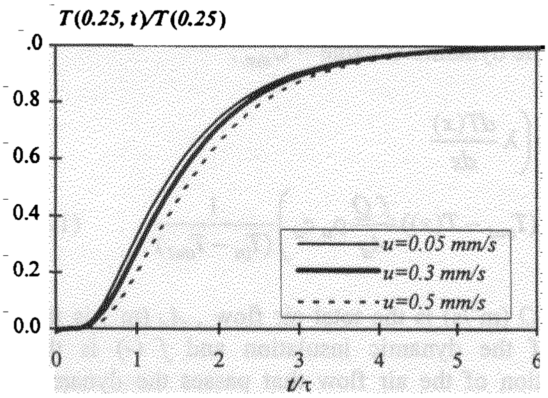

Figure 1: The ratio of the transient (4) to the steady state (3) temperature for u=0.05-0.5 mm/s, (x=0.25 m) as a function of t/τ.

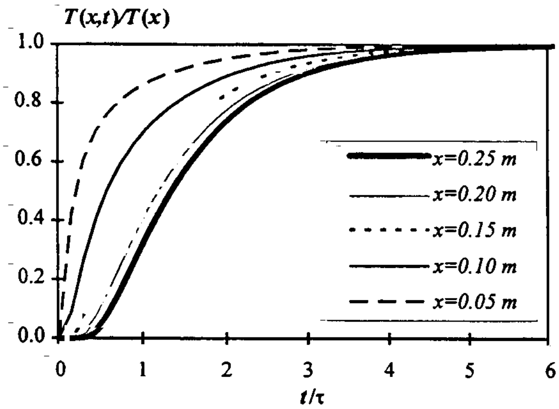

Figure 2: The ratio of the transient (4) to the steady state (3) temperature for x=0.05-0.25 m, (u=0.2 mm/s) as a function of t/τ. The full height if the insulation is 0.3 m.

From figure 1 it is clear that the relationship between the temperature and τ is fairly constant for different choices of the air flow u. It is therefore reasonable to use τ in describing how long the transient behaviour from a step change in the temperature lasts in the insulation. If the air flow is 0.2 mm/s about 30% of the effect of a step change will thus be reached in 1 hour (t/τ=1) at x=0.25 m and about 90% will be reached in 3.5 hours. This should be compared with the 8h needed for a thermal front to pass if conduction was neglected.

If the air flow is 0.2 mm/s the ratio of convection to conduction is $\nu \approx 6$ m^{-1}. This would indicate that the convective heat transfer dominates six times over the conduction. The more accurate analysis above shows that this is not true when the insulation has a finite height of 0.3m. The heat transfer by conduction is more important than heat transfer by convection in the OPTIMA house. The calculation above suggests that for time periods shorter than 5τ or approximately 6h it is necessary to use a transient calculation.

The solution (4) requires a smaller number of terms with increasing t. A solution with n=1 gives an error smaller than 10% for $t>\tau$ and a solution with n=2 gives an error smaller than 1%, for a step change in temperature. In order to get accurate results for times shorter than t=0.2τ a short term solution presented in Wallentén (1994) must be used.

THE GRADIENT METHOD

The goal of the gradient method is to find the air flow for which the theoretically calculated temperature distribution fits the measured temperature distribution as closely as possible. The theoretic distribution can result from either a steady state or transient calculation

The measured temperature distribution in the OPTIMA house was found by placing five thermocouples in a vertical column inside the insulation. The vertical distance between the thermocouples was 0.05 m. The temperatures were measured every third minute and averaged over 1 to 24 hours thus giving: $T_1^{\Delta t}$ .. $T_5^{\Delta t}$ with the averaging time Δt. The x coordinates for these measurements were $x_1, x_2 \,.\,.\, x_5$. The shortest averaging time was one hour. Four such columns were placed in the insulation. They were evenly distributed over the area and placed so as not to be disturbed by installations in the attic.

To calculate the steady state temperature distribution, the solution to the steady state heat transfer equation in one dimension (3) was used. The measured temperatures averaged over 24 hours were used. The temperatures T_1^{24h} and T_5^{24h} were used as boundary conditions in the solution (3). With $T(x,u)$=$T(x)$ for a given u, the nonlinear least squares error function $R^2(u)$ was constructed:

$$R^2(u) = \sum_{i=2}^{4} \left(T(x_i, u) - T_i^{24h} \right)^2 \qquad (6)$$

The problem was now to minimise $R^2(u)$ with respect to u. To find the minimum of $R^2(u)$ the Brent's method as described in Press et al (1986) was used. Brent's method is a mix between golden section search and parabolic interpolation. The value of u where this minimum occurs is called u_e and is thus a nonlinear least squares estimate of the true air flow. An approximate value of the standard deviation s of the temperature error $T(x_i, u_e)$-T_i^{24h} is:

$$s = \sqrt{R^2(u_e)/2} \qquad (7)$$

To calculate the transient temperature distribution a finite difference model with discretization both in time and space was used. The insulation was divided into 9 nodes. Each node was separated by the distance δx. The step between the discretized time levels was δt. The finite difference model for solving equation (2) was the MacCormack method, (Anderson 1984):

$$\text{Predictor: } \overline{T}_j^{n+1} = T_j^n - av\frac{\delta t}{\delta x}\left(T_{j+1}^n - T_j^n\right) + \frac{\delta t}{\delta x^2}\left(T_{j+1}^n - 2T_j^n + T_{j-1}^n\right) \quad (8)$$

$$\text{Corrector: } T_j^{n+1} = \frac{1}{2}\left[T_j^n + \overline{T}_j^{n+1} - av\frac{\delta t}{\delta x}\left(\overline{T}_j^{n+1} - \overline{T}_{j-1}^{n+1}\right) + a\frac{\delta t}{\delta x^2}\left(\overline{T}_{j+1}^{n+1} - 2\overline{T}_j^{n+1} + \overline{T}_{j-1}^{n+1}\right)\right] \quad (9)$$

The initial condition $T^0(x)$ and boundary conditions $T_0(t)$ and $T_L(t)$ were not known at specific times. The method used here was to guess an initial temperature distribution $T^0(x)$, use the hourly integrated temperatures T_1^{1h} and T_5^{1h} as boundary conditions over one hour and then calculate temperatures with (8, 9) up to the next averaging time step Δt which would typically be 2h to 6h. The calculated temperature at the nodes where the temperature was measured was integrated over Δt:

$$\tilde{T}^{\Delta t}(x_j) = \frac{1}{\Delta t}\sum_{t_n = t_0}^{t_0 + \Delta t} T_j^n \delta t \quad (10)$$

with $\tilde{T}^{\Delta t}(x_j, u) = \tilde{T}^{\Delta t}(x_j)$ for a given u. The least squares error function $R^2(u)$ was constructed:

$$R^2(u) = \sum_{i=2}^{4}\left(\tilde{T}^{\Delta t}(x_i, u) - T_i^{\Delta t}\right)^2 \quad (11)$$

The function is minimised as described above with respect to u to find the least squares estimate u_e of the air flow. The temperature distribution for u_e is used as initial condition for the next calculation.

DYNAMIC U VALUE: U_{dyn}

With the use of the 'dynamic U value' a house with dynamic insulation is described as a normal house with a better (lower) U value. To calculate the dynamic U value two houses are compared: one house with dynamic insulation and one house with the same air flow but without any heat recovery system for the ventilation. The total energy consumption of these two houses is equal for a certain U value of the normal house. This U value is called the dynamic U value = U_{dyn} :

$$U_{dyn} = \left(\lambda\frac{dT(x)}{dx} + (T_{out} - T(x))\frac{fQ}{A_d}\rho_a c_a\right)\frac{1}{(T_{in} - T_{out})} \quad (12)$$

where Q (m^3/s) is the total air flow , A_d (m^2) is the area of the dynamic insulation and f (-) is the proportion of the air flow that passes the dynamic insulation. The choice of T_{out} and T_{in} defines what part of the construction U_{dyn} represents. Here is T_{in} the temperature inside the house. If T_{out} is the ambient temperature T_{amb}, the ceiling, insulation and roof are included in U_{dyn}. If T_{out} is chosen as the attic temperature T_{attic} only the ceiling and insulation are included. If steady state conditions are assumed (12) becomes, with the use of the air velocity $u = fQ/A_d$:

$$U_{dyn} = \frac{u\rho_a c_a}{T_{in} - T_{out}}\left(\frac{T_L - T_0}{e^{u\rho_a c_a L/\lambda} - 1} + T_{out} - T_0\right) \quad (13)$$

where L is the distance between the points where the temperatures T_L and T_0 are measured.

ENERGY EFFICIENCY: e_{dyn}

Jensen (1982) and Arquis (1986) shows that the dynamic ventilation energy efficiency is perhaps a more adequate parameter for a dynamic insulation than the dynamic U value. To calculate the dynamic ventilation energy efficiency two houses are compared: one house with dynamic insulation and one house with the same air flow but with a heat exchanger between inlet and outlet air. The total energy consumption of these two houses is equal for a certain value of the energy efficiency for the heat exchanger. This value is here called the dynamic ventilation energy efficiency or just the energy efficiency = e_{dyn}:

$$e_{dyn} = 1 - \frac{T_{in} - T(x)}{T_{in} - T_{out}} + \frac{A_d}{fQ\rho_a c_a}\left(U_{normal} - \frac{\lambda}{T_{in} - T_{out}}\frac{dT(x)}{dx}\right) \quad (14)$$

Note that only the air that should have passed through the dynamic insulation ($f \cdot Q$) is used in the heat exchanger. In (14) U_{normal} is the U value of the investigated building component with the air flow set to zero. The choice of T_{out} and T_{in} defines what part of the construction e_{dyn} represents, as above for U_{dyn}. If steady state conditions are assumed and with the use of the air flow u (14) becomes:

$$e_{dyn} = \frac{U_{normal}}{u\rho_a c_a} + \frac{e^{u\rho_a c_a L/\lambda}(T_0 - T_{out}) + T_{out} - T_L}{\left(e^{u\rho_a c_a L/\lambda} - 1\right)(T_{in} - T_{out})} \quad (15)$$

If no extra energy is added (e.g. the sun) the energy efficiency can never be more than 0.5 for a dynamic insulation and this value is reached for air flows close to zero.

If the house with dynamic insulation is compared with a house with a heat exchanger between *all* inlet and outlet air, the total dynamic ventilation energy efficiency: $e_{dyn,tot}$ is obtained. This is the total energy efficiency for the house seen as one system:

$$e_{dyn,tot} = f \cdot e_{dyn} \quad (16)$$

The total energy efficiency takes into account the fact that only a fraction, f, of the inlet air has actually passed through the dynamic insulation. In figure 3 the energy efficiency e_{dyn} and the total energy efficiency $e_{dyn,tot}$ in the OPTIMA house are shown as functions of f.

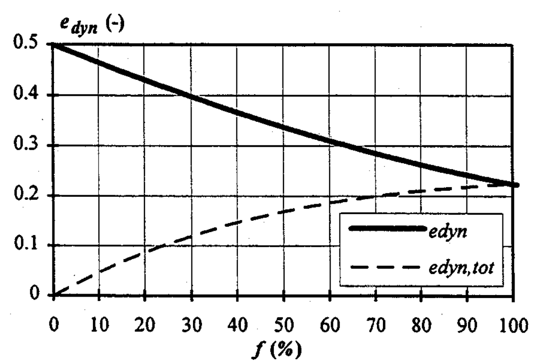

Figure 3: The energy efficiency e_{dyn} and the total energy efficiency $e_{dyn,tot}$ for the OPTIMA house. The calculation is based on 53 l/s exhaust air for f=100%.

Note that the total ventilation energy efficiency $e_{dyn,tot}$ for the OPTIMA house can never be more than 22% even if all air passes through the insulation. This means that a heat exchanger between all inlet and outlet air with energy efficiency >22% will always be better than the dynamic insulation in the OPTIMA house. A general conclusion is that a heat exchanger with an energy efficiency >50% is better than every possible type of dynamic counterflow insulation.

RESULTS FROM GRADIENT METHOD

The exhaust ventilation during the measurement period was approximately 53 l/s. If all air passed through the insulation the air flow would be $0.45 \cdot 10^{-3}$ m^3/m^2 s = 0.45 mm/s. Figure 4 shows the average air flow u_e and the ambient temperature over the period 930224-930421. The air flow was calculated with the steady state method using the 24 hour average. Figure 5 shows the average air flow u_e and the ambient temperature over the period 930829-931107. The average standard deviation in the temperature fit s was in all cases less than 0.3 °C.

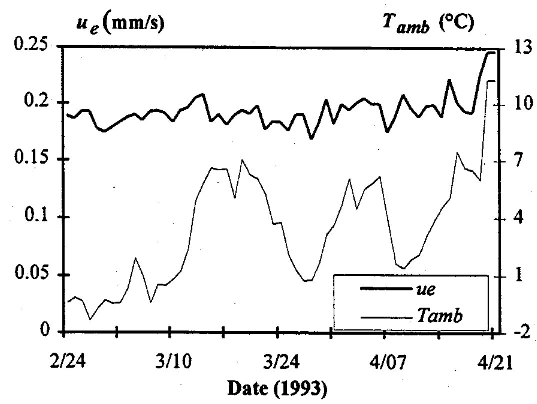

Figure 4: The 24h average u_e and the ambient temperature for the period 930224-930421.

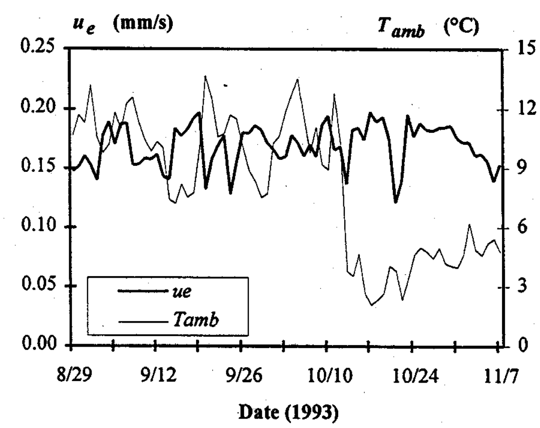

Figure 5: The 24h average u_e and the ambient temperature for the period 930829-931107.

The average u_e was 0.19 mm/s for the period 930224-930420 and 0.17 mm/s for the period 930829-931107. This represents a total airflow of 22 l/s and 19.7 l/s respectively or about 40% of the total inlet air.

The U value for the whole roof was 0.17 W/m^2K. The 24h average dynamic U values U_{dyn} for T_{out} in (13) taken as the attic and ambient temperature respectively are shown in figure 6. The 24h average dynamic energy efficiencies e_{dyn} are shown in figure 7. For a dynamic insulation without sunshine exposure the energy efficiency can never be above 0.5. The value 0.5 is only reached for air flows close to zero. The roof acted as a solar collector when $e_{dyn}(T_{out}=T_{amb})$ > $e_{dyn}(T_{out}=T_{attic})$. Note that this efficiency only affects the air that actually passes the insulation. If the previous calculation that about 40% of the total inlet air passes the insulation is true, the total energy efficiency should be $e_{dyn,tot}=e_{dyn}\cdot f$ or $e_{dyn,tot}\approx 16\%$.

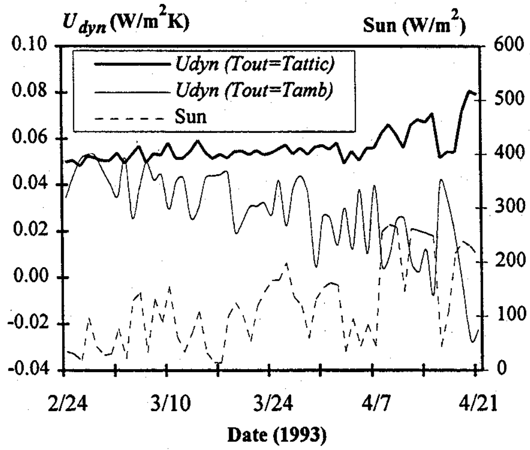

Figure 6: The 24h average $U_{dyn,}$ for $T_{out}=T_{amb}$ and $T_{out}=T_{attic}$ in equation (13) and the solar insolation for the period 930224-930421.

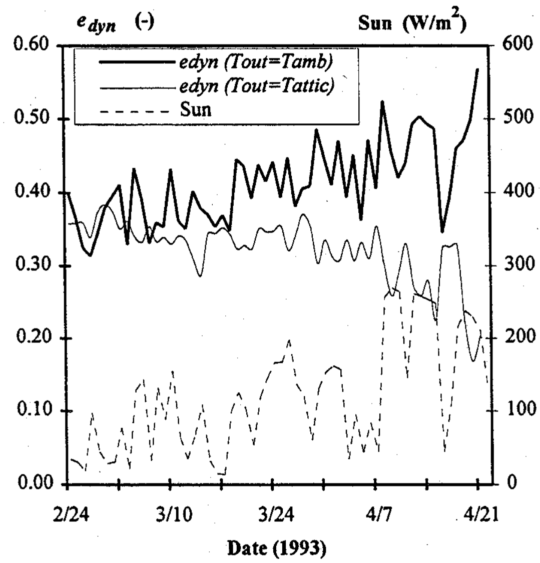

Figure 7: The 24h average $e_{dyn,}$ for $T_{out}=T_{amb}$ and $T_{out}=T_{attic}$ in equation (15) and the solar insolation for the period 930224-930421.

Figure 8 shows the 2h average air flow as measured by the four columns. From figure 8 it is clear that the measurements showed a daily variation of the air flow. This variation increased with increasing sunshine and was typically +/- 30% of the average flow with maximum at midnight and minimum at noon. The only parameter that was in phase with this variation was the global sun. No other measurement has so far confirmed that this variation actually occurred. There is no obvious reason why column 1 gave a higher air flow than the other columns. The calculated 24h average air flow increased slightly with increasing difference between inside and outside temperature. Both the 24h and 2h average air flow decreased with increasing wind speed. No obvious effect of wind direction was found

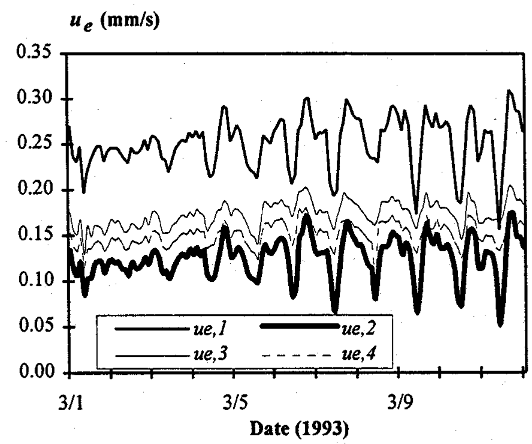

Figure 8: The 2h average at the four measuring columns ($u_{e,1}$ -$u_{e,4}$) for the period 930301-930311.

COMPARISON WITH OTHER MEASUREMENTS

Direct measurements showed that the air flow to the air outlets was about 58 l/s. Measurement over an orifice plate in the duct showed that the exhaust air should be 63 l/s. Direct measurements showed that 27 l/s came in through the inlet terminals. The airtightness of the house was 1.7 ach at 50 Pa negative pressure. During the period 930123-930206 tracer gas experiments were performed by Björn Hedin (1994). The inlet terminals were either modified to be as open as possible or in normal position. Figure 9 shows u_e calculated for the period when the tracer gas experiments were performed. One can clearly see when the terminals were changed from normal (N) to open (O) position. The tracer gas experiments gave that the total inlet air was 53 l/s. The results from the tracer gas, direct and gradient measurements are summarised in figure 10.

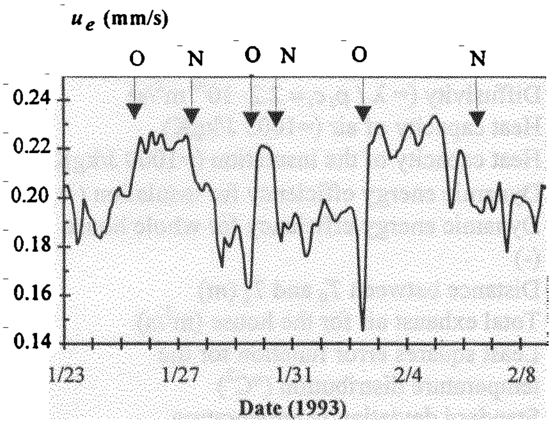

Figure 9: The 6h average u_e during the tracer gas experiment. The times when the terminals were open (O) or normal (N) are marked in the figure.

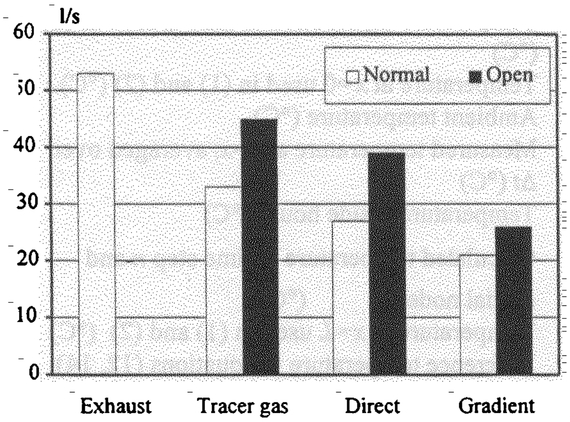

Figure 10: The air flow through the insulation as measured by the three different methods for the case with normal and open inlet terminals.

SENSITIVITY ANALYSIS

The least squares estimation model has been validated against the analytical solution and an accurate numerical simulation. Possible errors are to be found in the estimation of the physical parameters or in the physical model itself. The most sensitive parameters are λ, ρ_a and L and of course the temperature measurements. With the assumption that the errors are uncorrelated the accuracy of the steady state air flow calculation will be 8% or +/- 0.015 mm/s. The transient calculation will have a slightly higher error. The calculation of successive air flow variations will however be much more accurate, a relative error <4%. There is some indication that the density of the insulation was less than 19 kg/m^3. This would give an underestimation of the air flow of the order of 5%.

CONCLUSION

A nonlinear least squares method for continuous measurement of the air flow in 'dynamic insulation' has been presented. The method is called the *gradient method* to indicate that the temperature gradients in the insulation are used. The method includes one dimensional transient and steady state calculations of the heat transfer in the insulation. The model was verified by analytical and numerical simulations. With the use of this method the air flow through an insulation was continually measured in a single storey one family house during one year. The air flow through the dynamic insulation as measured by the gradient method was 40% of the total inlet air. Results from a few tracer gas and direct measurements indicated a higher flow, but this could be expected since the gradient method was the only method that measured the flow in the insulation itself. The major reason for the low percentage was that other parts of the house leaked too much air.

The dynamic ventilation energy efficiency for the insulation was about e_{dyn}=35%. This was very close to the calculated value of e_{dyn}=37% for a leakage of 40%. This should be compared with the maximal theoretic energy efficiency for any counterflow insulation of 50%. The total energy efficiency, i.e. with regard to the air that did not pass the insulation, was $e_{dyn,tot}$=14%. This was also very close to the calculated value of $e_{dyn,tot}$=14%. Even when heating of the attic by the sun was included this factor only increased to 16%.

The air flow was not very sensitive to external parameters e.g.: temperature difference, wind speed or wind direction. The transient calculation showed a so far not verified daily variation of the air flow.

The measurements in the OPTIMA house showed that the dynamic insulation of counterflow type could not be justified by the energy savings alone. However, the inlet air was preheated and filtered and there were fewer ventilation ducts for transporting the inlet air than would be needed in a house with a heat exchanger system. This taken together, makes the dynamic insulation an attractive choice, at least in theory. The general conclusion from the measurements was that the dynamic insulation needs a house constructed to much more exacting standards to work properly.

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NOMENCLATURE

A_d	Area of the dynamic insulation (=116 m^2)
a	Diffusivity (= $\lambda / \rho_i c_i \approx 2.2 \cdot 10^{-6}$ m^2/s)
c_a	Heat capacity of air (=1005 J/kgK)
c_i	Heat capacity of the insulation (=1000 J/kgK)
e_{dyn}	Dynamic energy efficiency for insulation (-)
$e_{dyn,tot}$	Dynamic energy efficiency for whole house (-)
L	Distance between T_0 and T_L (m)
Q	Total exhaust air for the house (m^3/s)
$R^2()$	Least squares error function for the temperature distribution (°C^2)
s	Standard deviation in temperature distribution (°C)
t	Time variable (s)
t_n	Time at time step n (s)
t_0	Initial time for boundary condition (s)
T	Temperature (°C)
$T^0(x)$	Temperature distribution at time t_0 used in (2) (°C)
T_0	Temperature at x=0 used in (1) and (2) (°C)
T_{amb}	Ambient temperature (°C)
$T_i^{\Delta t}$	Measured temperature at x=x_i averaged over Δt (°C)
T_{in}	Temperature inside house (°C)
T_j^n	Calculated temperature at time step n and spatial node j (°C)
T_L	Temperature at x=L used in (1) and (2) (°C)
T_{out}	Reference temperature in equations (12, 14) (°C)
$\tilde{T}^{\Delta t}$	Calculated temperature averaged over Δt (°C)
$\bar{T}_i$	Temporary temperature used in finite difference calculation (°C)
u	Air flow through the insulation (m^3/m^2 s)
u_e	Least squares estimate of the air flow (m^3/m^2 s)
U_{normal}	U value with zero air flow (W/m^2 K)
U_{dyn}	Dynamic U value for a specific u (W/m^2 K)
x	Length coordinate (m)
x_j	Coordinate for node j (m)
Δt	Time used in integration of temperature (s)
δt	Time step used in finite difference calculation (s)
δx	Distance between nodes in finite difference calculation (m)
λ	Heat transfer coefficient for the insulation (~ 0.042 W/mK)
ρ_a	Density of air (1.27 kg/m^3)
ρ_i	Density of the insulation (19 kg/m^3)
ν	Ratio of convective to conductive heat transfer (= $u\, \rho_a c_a / \lambda$ m^{-1})