# HEAT TRANSFER COEFFICIENTS IN A FULL SCALE ROOM WITH AND WITHOUT FURNITURE

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## ABSTRACT

The convective heat transfer coefficient at an outer ambient wall with a window exposed to natural climate was measured in a room with and without furniture. The method used was to estimate the heat flow from measured temperatures and solar radiation. The convective heat transfer was calculated as the difference between the heat flow through the building element and the calculated longwave radiation. The accuracy was at best ±15% for the window and ± 20% for the wall. Even so the effect of different heating and ventilation strategies on the inside could clearly be detected. Local coefficients may be more than 10 times the expected, due to ventilation or position of the radiator.

## INTRODUCTION

It is, with the thermal models used in today's building simulation programs, possible to calculate the major part of the heat transfer in a room with an ambient wall. The longwave radiation exchange between surfaces is complicated but possible to analyse. The heat transfer in ambient walls is dominated by conduction which is a well investigated phenomenon. Thus one can say that in general there is enough knowledge for rough estimates of hourly heat flows in a room. There are however some parameters which are more difficult to estimate: heat flows when walls and windows are poorly insulated and temperatures on the inside of ambient walls and windows.

The reason for this is the lack of data about the convective energy transport in a room exposed to ambient climate, including sun.

The convective heat transfer is related to the air movement in a room and is very difficult to calculate and then only for simplified models. Convective heat transfer coefficients from experiments in realistic situations are few. Most measurements have been made in special environments with, for example, metal coated walls.

## AIM OF STUDY

The aim of this study was to examine if it was possible to accurately measure the continuous heat flow through a window and a wall from temperature sensors and solar radiation measurements. If this was so could realistic values for the convective heat transfer coefficient between room and ambient wall be estimated ? These values were aimed for building simulation programs where the local air temperatures are not known. The study is described in detail in Wallentén (1998).

## METHOD

The method used was to estimate the heat flow through wall and window from measured solar radiation on the façade and temperatures. The temperatures are measured inside the wall, on the window panes, in the air, at inner surfaces etc. The longwave radiation is calculated from surface temperatures. The convective heat transfer is calculated as the difference between the heat flow through the building element and the longwave radiation.

$$q_c = q_{cond} - q_r \qquad (1)$$

For the window the absorbed solar energy has to be included:

$$q_c = q_{cond} - q_r - q_{abs} \qquad (2)$$

The convective heat transfer coefficient is then calculated as:

$$h_c = \frac{q_c}{T_{ref} - T_{surface}} = \frac{q_c}{\Delta T} \qquad (3)$$

Here $T_{ref}$ is a reference temperature that can be arbitrarily chosen and $T_{surface}$ is the surface temperature. The aim of this study was to estimate $h_c$ values for building simulation programs. Therefore $T_{ref}$ was chosen as the vertical average in the middle of the room, a value which should closely resemble the mean temperature used in these programs. The heat transfer coefficient will in general increase the further away from the wall the reference temperature is chosen. In case of ventilation inlet and outlet temperatures are other possible choices.

## SURVEY OF EXISTING FORMULAE

The formulae for the convective heat transfer coefficients can be determined theoretically, experimentally or by a mix of both. The theoretic formulae are derived from boundary layer theory from a vertical heated plane in an undisturbed surrounding. The reference temperature is taken as the undisturbed temperature. The formulae from experiments are derived from a wide range of situations where the reference temperature typically is chosen at a position close to the wall or in the middle of the test room. Here the laminar formulae from Bejan (1993) and the turbulent from Churchill and Chu (1975) are chosen as a reference. If these formulae are evaluated for $T$=15°C we get the following formula:

$$h_c = \begin{cases} 1.34\left(|\Delta T|/H\right)^{1/4} & |\Delta T|\,H^3 < 9.5\,\mathrm{m^3K} \\ 1.33|\Delta T|^{1/3} - 0.474/H & |\Delta T|\,H^3 > 9.5\,\mathrm{m^3K} \end{cases} \tag{4}$$

This formula is hereafter referred to as Churchill and Chu. Almadari and Hammond (1983) suggested a formula valid for both the laminar and turbulent region:

$$h_c = \left[\left(1.51\left(\frac{|\Delta T|}{H}\right)^{1/4}\right)^6 + \left(1.33\,(|\Delta T|)^{1/3}\right)^6\right]^{1/6} \tag{5}$$

Formulae based on full scale experiments give in general a slightly higher value than analytic or small scale experiments. This Min et al (1956) performed experiments in a test cell with floor areas 3.6 x 7.2 $m^2$ or 3.6 x 3.6 $m^2$ and the height 2.4 or 3m. For the floor heating case they got:

$$h_c = 2\,\frac{\Delta T^{0.32}}{H^{0.04}} \tag{6}$$

Hatton and Awbi (1995) made full scale experiments in a test cell with the floor area 2.78 x 2.78 $m^2$ and height 2.3 m. The walls were aluminium plated and longwave radiation was taken into account. The heat transfer coefficient was based on heated parts of the test cell. They found the mean convective heat transfer coefficient for the wall to be:

$$h_c = 1.57|\Delta T|^{0.31} \tag{7}$$

Khalifa and Marshall (1990) used a test cell with floor area 2.95 x 2.35 $m^2$ and height 2.05. The walls in the test cell were aluminium coated. The small longwave radiation exchange was not taken into account. With a radiator opposite to the cold wall they got for the wall:

$$h_c = 2.20|\Delta T|^{0.22} \tag{8}$$

With radiator under glazing, for the wall:

$$h_c = 2.35|\Delta T|^{0.21} \tag{9}$$

Khalifa and Marshall also calculated formula for other situations: at wall with floor heating, at window with radiator under window and at window with radiator opposite the cold wall. Delaforce et al (1993) made measurements in an outside test cell with the floor area 2.03 x 2.03 $m^2$ and the height 2.33 m. In Figure 1 the different heat transfer coefficients are shown. The results from Delaforce et al are indicated as regions surrounded by black squares. Churchill and Chu gives the lowest values and Min et al gives the highest or close to the highest. These two formulae will subsequently be used to compare with the measured values from this study.

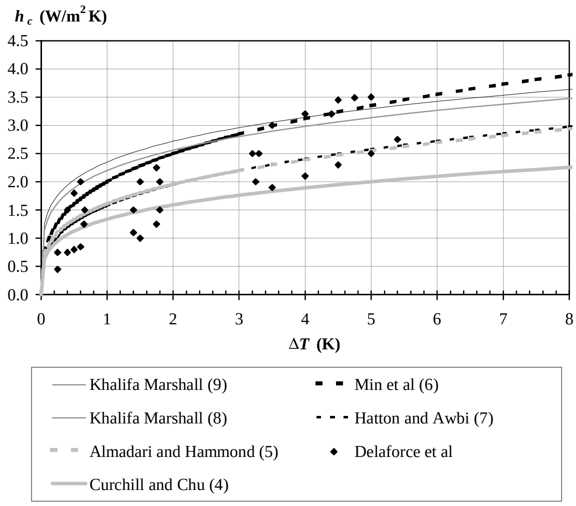


Figure 1: A selection of convective heat transfer coefficients found in the literature. $H$=2m.

Fisher and Pedersen (1997) performed full scale experiments in an indoor test room with ventilation rates between 3 to 12 air changes per hour. They found a strong coupling between $h_c$ and the ventilation rate when the inlet temperature was used as reference temperature. In the study shown here the ventilation rates never exceeded 1.0 ach and the reference temperature was chosen as an average room temperature.

## DESCRIPTION OF TEST ROOM

The test room was room had the dimensions 3 · 3.6 · 2.4 m and was located in a house in Lund, Sweden. Only the south wall was exposed to the ambient climate. A window of the dimensions 1m · 1.1 m was placed in this wall. The temperature in the room varied between 15 and 35 °C. The house was built with a sandwich construction of lightweight concrete and polystyrene. The interior surfaces were all painted with a mat white colour except for the floor which had a light brown linoleum flooring.

The ambient, south wall, was built with a normal non load bearing stud wall with an U-value of 0.27 W/m$^2$K. Four thermocouples were positioned in the wall at three heights on the wall as indicated in Figure 2. Two wooden frame windows were investigated: one superinsulated 3+1 window and one “normal” 2+1 window. The superinsulated window had a centre-of-glass U-value of 0.62 W/m$^2$K and a total U-value of 0.81 W/m$^2$K. The normal window had a centre-of-glass U-value of 1.75 W/m$^2$K and a total U-value of 1.80 W/m$^2$K.

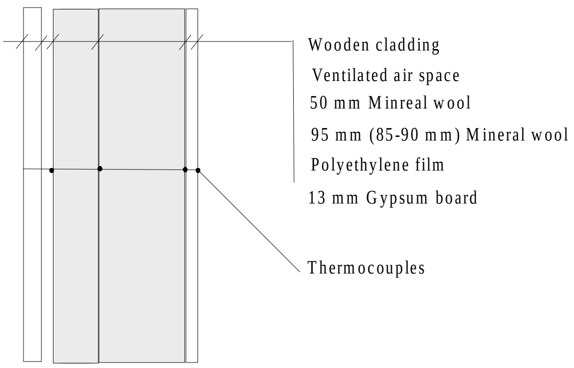


Figure 2: The ambient wall with thermocouples.

The room had either an electric radiator of maximum 500 W or a radiator with maximum 1000 W. The radiator was placed either 0.2 m from the north (back) wall in the centre of the wall or 0.12 m from south wall under the window. The ventilation system had an air inlet terminal in the ceiling 0.30 m from the south wall and an air outlet at the top east corner of the north wall 0.09 m from the ceiling and 0.40 m from the east wall. The inlet terminal blew air horizontally into the room parallel to the ceiling. The ventilation rate during the period was between 0 and 1.0 ach. No cooling system was installed.

The room was equipped with thermocouples of type T and a global solarimeter on the façade. Measurements were taken every minute and averaged to 4,10, 30 and 60 minutes mean values. The measurement system was four calibrated Acurex Netpak units.

The positions and names of the thermocouples on the south ambient wall are shown in Figure 3. There were 18 thermocouples in the air as shown in Figure 4. These thermocouples were made of 0.08 mm stripped wire and positioned in a section in the middle of the room, i.e. on the north-south symmetry axis.

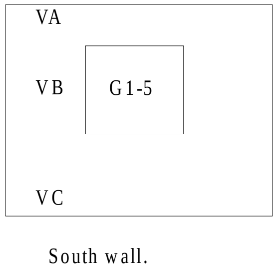


Figure 3: The positions and denotions of the thermocouples on the south wall.

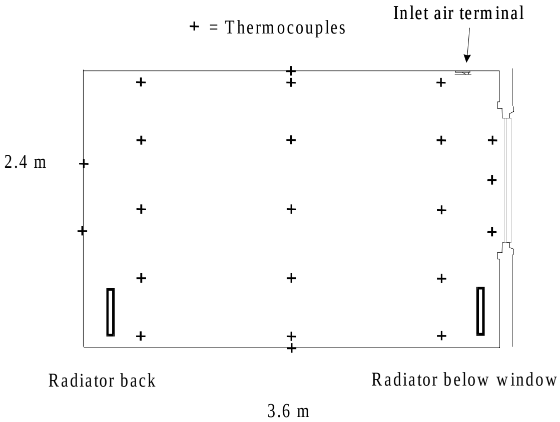


Figure 4: Air, back wall, floor and ceiling thermocouples in the middle section of the room.

There were 11 thermocouples in the window. They were made of 0.08 mm silver coated wire glued to the window pane under a microscope cover glass of 0.1 mm. This was made to maximise the thermal contact between the thermocouple and the window pane. The cover glass also ensured that the surface was plane thus not effecting the heat transfer coefficient and that the emissivity was that of glass.

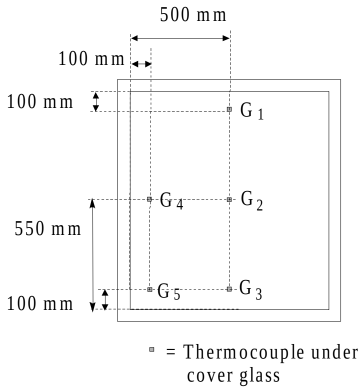


Figure 5: Thermocouples on window seen from the inside. There are thermocouples beneath these on the next pane.

## THERMAL MODELS

For the walls, an explicit finite difference method and a frequency method were used to calculate $q_c$ in equation (1). For the simulation of the wall the interior and third thermocouple (between the two mineral wool layers) were used as boundary conditions.

For the windows a lumped capacity model was used. For the inner pane this gives the following equation:

$$\frac{dT}{dt} C_{pane} = q_{2-1} + q_c + q_r + q_{abs} \qquad (10)$$

Here, $C_{pane}$ is the heat capacity of the window pane per unit area, $q_{2\text{-}1}$ is the total heat transfer from pane 2 to pane 1. Pane 1 is the inner pane with thermo-couples $G_1$-$G_5$. For the conductive/convective heat transfer between the panes the formulae from ElSherbiny et al (1982) were used. The longwave radiation between the window panes was modelled as grey diffuse longwave exchange between two infinite parallel planes. The shortwave radiation model was taken from Källblad (1998). Calculations were based on refraction indices and extinction coefficients and Fresnell's equations. In each pane parallel and perpendicular polarisation were treated separately. For coated panes fictive refraction index and extinction coefficients were used. The calculated reflectance was then adjusted by a factor to account for the fact that a coated pane has different reflectance on the coated and non coated side.

The longwave radiation in the room was calculated as radiation between diffuse grey surfaces. (Siegel and Howell, 1981). The view factors needed for the calculation were with formulae from Gross et al (1981).

## RESULTS

Experiments were performed during a period of four years. Typically an experimental lasted for a week. Three different representative experimental situations are shown in detail and 11 are summarized. Equation (3) was used to find the local convective heat transfer coefficient $h_c$. The reference temperature in all the calculations was the average of the middle vertical row of thermocouples. Choosing the mean value of all air thermocouples as reference temperature did not significally change the results. In all diagrams a positive temperature difference $\Delta T$ means that the reference temperature is warmer than the window pane or wall.

For the wall $h_c$ was calculated at positions VA and VB. For the window $h_c$ was calculated at positions $G_1$ to $G_4$ as shown in Figure 5. The temperature in a vertical section in the middle of the room where the air thermocouples were positioned are shown for some cases. The temperatures in the corners and between the walls and air thermocouples are interpolated and should therefore not be viewed as exact.

The results from calculations and measurements are shown without any filtering of the values. The exception is that for most values with $|\Delta T| < 0.4$ K and very large negative and positive values for $h_c$ are not shown. From error investigation is was found that for the window values for –4 K< $\Delta T$ < 2 K the estimated error was >40%. For the wall $h_c$ values for $\Delta T \approx 1$ K have an estimated error of around 35%. The reason for not filtering out these values was that the general trends can still be considered and that the increasing error is identifiable in most of the figures.

*Heat transfer coefficients with radiator at the back wall no ventilation*. The first case is with the radiator 0.2 m from the north (back) wall with no ventilation. The radiator had a bimetallic control and a maximal 10 min average effect of 180 W. The window was the 3-pane window. See Figure 6

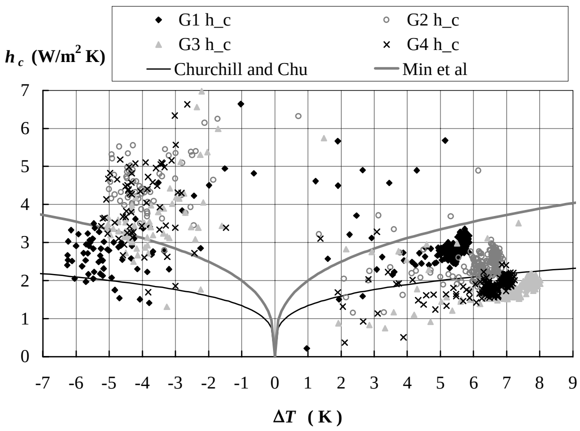


Figure 6: Convective heat transfer coefficient at 3-pane window with radiator at the back wall and no ventilation, 20min average, 3-pane window. (980128 -980130)

The order of the coefficients for positive $\Delta T$ are as expected since the natural convection is downwards the pane: $G_1$ $h_c$ >$G_2$ $h_c$ >$G_3$ $h_c$. The coefficient $G_4$ $h_c$ is lower than $G_2$ $h_c$ which might also be expected being close to the left edge of the window pane. The average values are slightly above Churchill and Chu. For negative $\Delta T$ the situation is not clear due to low accuracy. The lowest $h_c$ is from $G_1$ which it reasonably since the direction of the air flow is upwards here.

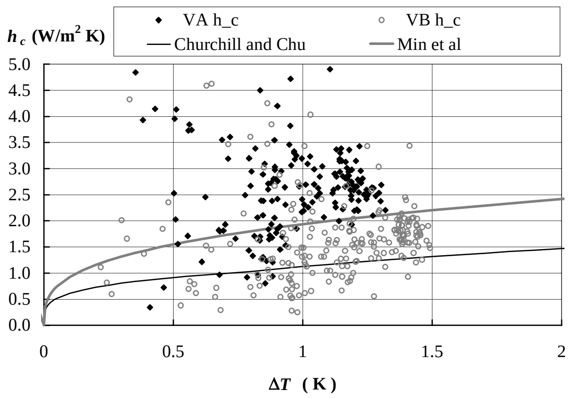


Figure 7: Convective heat transfer coefficient at wall with radiator at the back wall and no ventilation, 20 min average. (980128 -980130 )

Figure 7 shows $h_c$ for the wall. VA gives the highest value, whereas VB gives a value between Churchill and Chu and Min et al. If the air temperature close to the wall is different from $T_{ref}$ , the calculated $h_c$ will of course not match the formulae.

Figure 8 shows the air temperatures at night with the radiator effect 180 W. Figure 9 shows the temperatures with sun and no radiator effect. In Figure 8 the plume from the radiator is between the wall and the first row of thermocouples and can therefore not be seen. This row is located 0.5 m from the back wall.

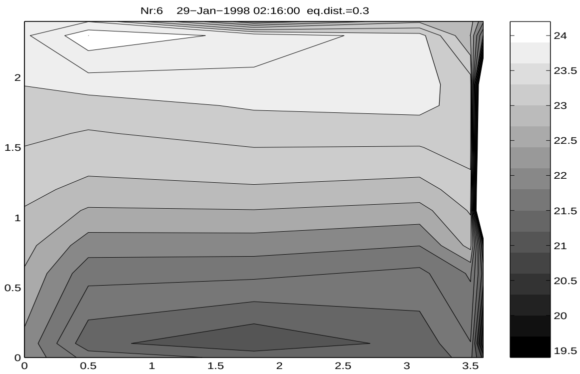

Figure 8: Air temperatures in the room 980129 2:16, radiator effect 180 W.

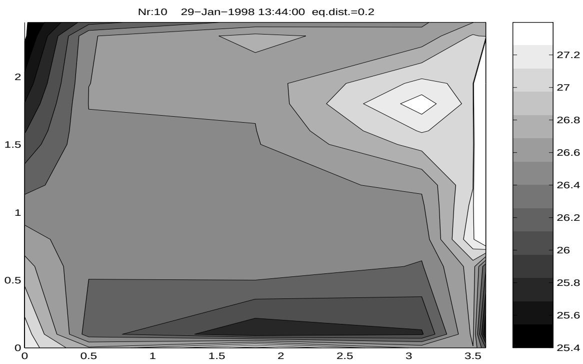

Figure 9: Air temperatures in the room 980129 13:44, radiator effect 0 W.

*Heat transfer coefficients with radiator at the back wall. Ventilation from window, 1.0 air changes per hour.* The radiator was at the back wall with PI-control, ventilation was 1 ach from the window and the window type 4-pane. The inlet temperature was about 2 K below the room temperature. Figure 10 shows $h_c$ for the window. All values for $\Delta T$>0 K are below Churchill and Chu which differs in sharp contrast to the previous case without ventilation. The direction of the ventilation is in opposite direction to the natural convection cell when $\Delta T$>0 K. When $\Delta T$<0 K the direction of the natural convection loop and the ventilation are the same.

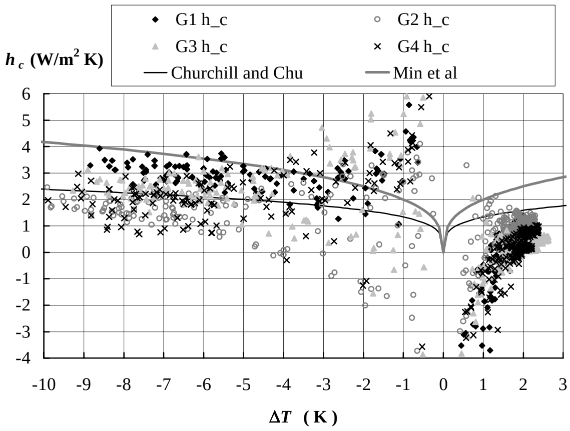

Figure 10: Convective heat transfer coefficient at the 4-pane window with radiator at the back wall and 1.0 ach. 940331 18.00-940409 16.00 , 30 min average.

Figure 11 show that the values from VA and VB have been reduced. Almost all the values are negative which means that the reference temperature taken is not representative close to the wall. The measurement error is high due to the low energy flow through the wall.

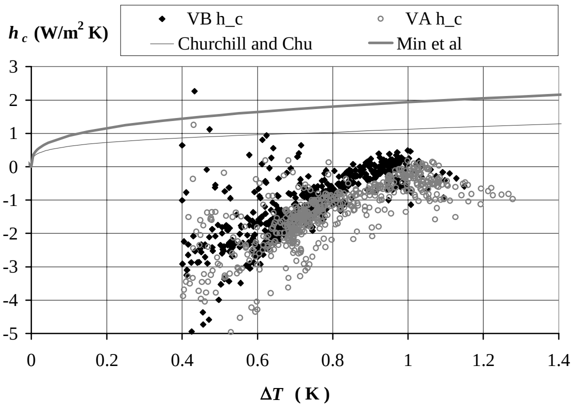

Figure 11: Convective heat transfer coefficient at wall with radiator at the back wall and 1.0 ach. (940331-940409 , 20 min average)

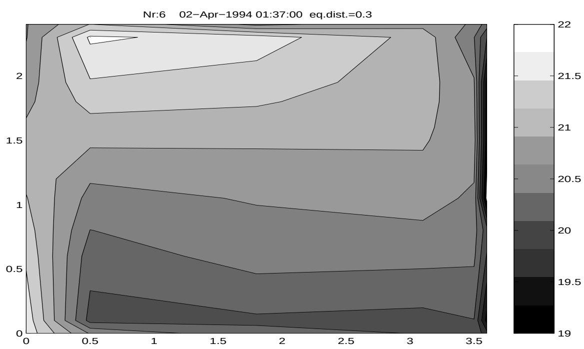

Figure 12: Air temperatures in the room 940402 1:37, radiator effect 120 W, inlet temperature 20.0 °C, outlet temperature 21.8 °C.

The air temperatures in Figure 12 differ from Figure 8 in that the horizontal temperature gradient is higher above 1.5 m. Close to the ceiling the warm air from the radiator is pushed back. The air seems to be more mixed in the middle of the room giving a lower temperature gradient in the vertical direction..

*Radiator below window. Ventilation from window, 0.6 air changes per hour. Room furnished..* The period (970217 – 970225) the room was furnished with a desk, two chairs and a small chest. There was also a window sill (depth 0.1m) positioned 0.02m from the window. During two of these days was the room equipped with a computer where one person worked during the day. The results from window and wall are shown in Figure 13 and Figure 14. In this case the longwave radiation exchange was not correctly calculated due to the furniture in the room. From a building simulation point of view this is a typical case however. The convective heat transfer coefficients calculated this way were actually quite close to those of a empty room with radiator below window and the same ventilation. The values at the window and wall are much affected by the radiator below. For the wall there is a zone of warm air close to the ceiling giving very large values from VA. The values from VB are much closer to Churchill and Chu.

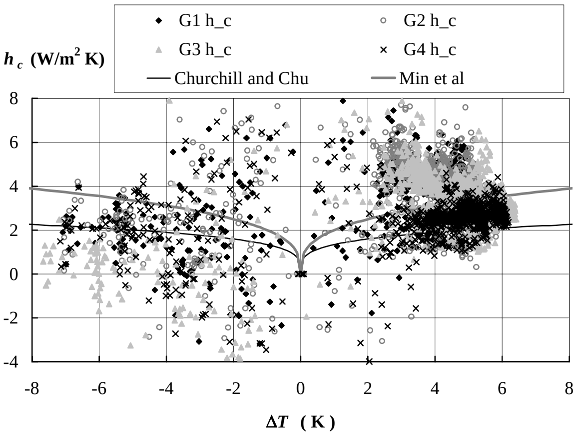


Figure 13: Convective heat transfer coefficient at 3-pane window with radiator below window, 0.6 ach, furnished room, 970217 -970225, 4 min average.

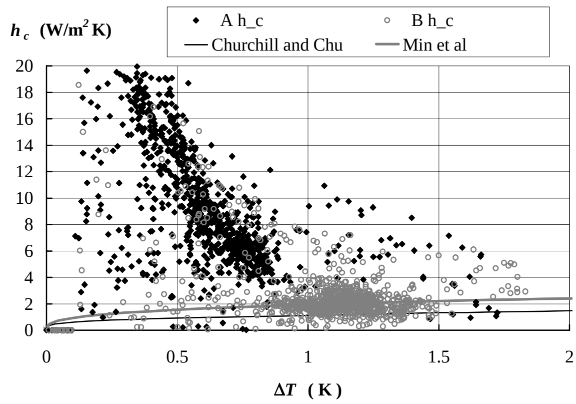


Figure 14: Convective heat transfer coefficient at wall with radiator below window, 0.6 ach, furnished room, 970217 -970225, 4 min average.

## GENERALISED VALUES

The results showed the importance of the ventilation design and position of radiator. The results from all the experiments are tentatively summarized in Table 1 and 2. The results are presented as a quotient of measured results to the function of Churchill and Chu.

$$f = \frac{\text{measured } h_c}{\text{Churchill and Chu } h_c} \qquad (11)$$

This gives at least a rough idea of how the convective heat transfer coefficients vary with temperature.

Table 1: The quotient, *f*, from (11) for the different tested situations, ↓ = ventilation from window, ↑ = ventilation towards window. Numbers in parenthesis mean that no temperature dependence could be established and that the absolute number could be used for $h_c$ (W/m$^2$K).

| Radiator position | Vent. | Nr.of panes | Window $\Delta T > 0$ | Window $\Delta T < 0$ |
|---|---|---|---|---|
| | ach (↓↑) | | *f radiator on/off* | *f* |
| back | off | 3 | 1 | 2 |
| back (big) | off | 3 | 1.5/0.9 | - |
| back | off | 4 | 1.4 | - |
| back | 0.5 ↓ | 3 | 0.8 | - |
| back | 1.0 ↓ | 4 | 0.5 | 1 |
| back | 1.0 ↑ | 4 | 2 | - |
| window | off | 3 | 2.5 / 0.7 | - |
| window | 0.5 ↓ | 3 | 3 /0.7 | 2 |
| window | 1.0 ↓ | 3 | 2.5/0.7 | 1.4 |
| window | 1.0 ↑ | 4 | (15) | 1 |
| furnished | 0.6 ↓ | 3 | 2/1 | 1 |

Table 2: The quotient, *f*, from (11) for the different tested situations.

| Radiator position | Vent. | Wall high | Wall middle |
|---|---|---|---|
| | ach (↓↑) | *f* | *f* |
| back | off | 2 | 1.3 |
| back (big) | off | (5) | (3) |
| back | off | 1.7 | 1.4 |
| back | 0.5 ↓ | 1.7 | 0.2 |
| back | 1.0 ↓ | (-1) | (-1) |
| back | 1.0 ↑ | 0.9 | 0.9 |
| window | off | (4) | 0.5 |
| window | 0.5 ↓ | (10) | 0.6 |
| window | 1.0 ↓ | (8) | 0.8 |
| window | 1.0 ↑ | (10) | 0.9 |
| furnished | 0.6 ↓ | (5-20) | 0.5 |

As stated before is the $h_C$ value ( and thus *f* ) sensitive to the choice of reference temperature, in this study the vertical mean temperature in the middle of the room. Since the temperature close to the wall and window is not accessible in the majority of the building simulation programs these temperatures were deliberately not used. An $h_c$ value based on local temperatures has no meaning in programs only dealing with mean air temperatures, other than producing smother curves and making the quotient *f* closer to 1.

It is not easy to recommend values to use based on this study. The accuracy was typically around 30%. Having said this, if the values *should* be generalised , a suggestion is that the results from the situations with 0.5 or 1.0 ach from the window should be used if nothing else is known. This means: Radiator at the back wall: *f*=0.7 for the window and *f*=1 for the wall. Radiator below window: *f*=2.5 for the window with the radiator power on and *f*=0.7 with radiator off and *f*=0.7 for the wall. For buildings with room height much above 2.5 m the results cannot directly be used.

## VERIFICATION AND ERRORS

The models were verified with heat flow meter test placed on wall and window, with convective heat transfer measurements based on temperature gradient measurements in the air close to the window (Mayer ladder), Mayer (1987). Figure 15 shows the heat flow through the inner window pane as measured by the Mayer ladder and calculated at the centre of the 4-pane window.

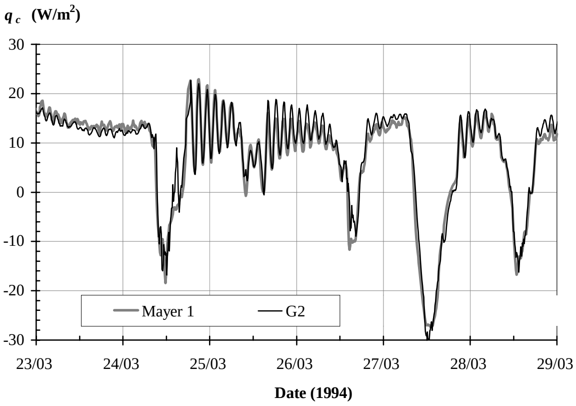


Figure 15: The heat flow through the inner window pane as measured by the Mayer ladder and calculated at $G_2$.

For the 3-pane window the heat transfer in the air between the panes was almost pure conduction. The coefficients $G_1$ $h_c$ –$G_4$ $h_c$ should therefore be good estimates for the local heat transfer. For the 4-pane window the heat transfer between the panes was up to 15% more heat transfer than by pure conduction so the values were in this case not so locally defined.

The absorption of solar radiation on the thin thermocouples (0.08 mm) glued to the window pane beneath a microscope cover glass was not a problem. To measure air temperatures in sunlit places thin (0.08 mm in this case) stripped thermocouples was a good alternative. With the thin thermocouples the measurement error was estimated to less than 0.5 K when exposed to 400 W/m$^2$ of solar radiation. Figure 16 shows the measured and calculated heat flow for the wall 1994-04-28 – 1994-05-08.

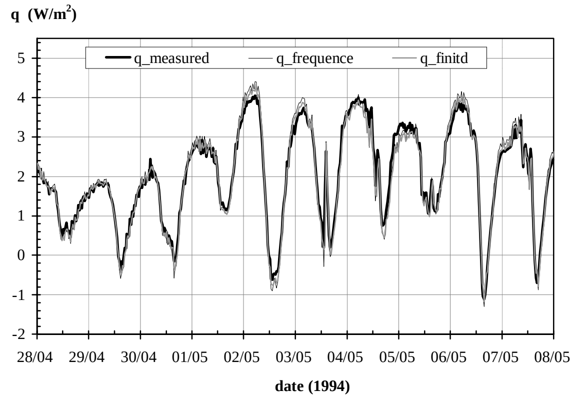


Figure 16: The measured, $q_{measured}$, and calculated heat flow through the heat flow meter from frequency analysis $q_{frequence}$ and finite difference $q_{FD}$.

The accuracy of the heat transfer coefficient calculation depends on the accuracy of the model and the measured parameters. If the errors are uncorrelated the sensitivity to error in the input parameters is:

$$\delta h_c = \frac{1}{\Delta T}\sqrt{\left(\frac{q_c}{\Delta T}\delta \Delta T\right)^2 + \delta q_r^{\,2} + \delta q_{cond}^{\,2}} \quad (12)$$

The estimated errors for the different parameters were:

| | |
|---|---|
| $\delta\Delta T$ | ±0.2 K |
| $\delta\, q_{cond,wall}$ | ± 0.3 W/m$^2$ |
| $\delta\, q_{cond,window}$ | ± 1 W/m$^2$ |
| $\delta\, q_r$ | ± 0.5 W/m$^2$ |

Based on the error estimation above, the absolute $\delta h_c$ and relative error $\delta h_c/h_c$ , when estimating the heat flow coefficient at the walls and at the window could be calculated. From these calculations it was clear that estimating the heat transfer coefficient by the method chosen in this study was not very accurate. For the window $q_c$ > 4 W/m$^2$ and $\Delta T$>2 K gave $\delta h_c/h_c$ < 0.3. This accuracy was also achieved for $\Delta T$ = 1 K and $q_c$ =5 W/m$^2$. For the wall the $\delta h_c/h_c$ <0.3 was achieved for $q_c$ = 4 W/m$^2$ and $\Delta T$=1 K. Unfortunately the heat flow through the wall was much lower than through the window due to the low U-value of the wall.

When comparing with the Mayer ladder with solar insolation the results were that the convective heat transfer coefficient on the window pane were reasonable if $\Delta T < -4$ K or $\Delta T > 1$ K.

## CONCLUSIONS

It was possible to measure the continuous heat flow through a window and wall from temperature sensors and solar radiation measurements. It was possible to continuously measure the convective heat transfer coefficient on the inner surface of a wall or a window with the suggested method. The accuracy was not very good: at best ±15% for the window and ± 20% for the wall. Even with this low accuracy the effect of different heating and ventilation strategies on the inside could clearly be detected. The presented results showed the importance of the ventilation design and the position of the radiator. Local convective heat transfer coefficients could be more than 10 times the expected, due to ventilation or position of the radiator. It is not obvious how the results of this study should be generalised. But a rough estimate was suggested. The effect of furniture and one person in the room was not obvious in a situation with radiator below window and ventilation 0.6 ach from window.

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## NOMENCLATURE

| | |
|---|---|
| $C$ | heat capacity per unit area(J/m$^2$K) |
| $f$ | factor used in comparison with equation (4)(-) |
| $H$ | height of surface (m) |
| $h_c$ | generally the mean convective heat transfer coefficient, except when referring to measured values when it is the local convective heat transfer coefficient(W/m$^2$K) |
| $q_r$ | longwave radiation heat flow (W/m$^2$) |
| $q_{cond}$ | conductive heat flow in material(W/m$^2$) |
| $q_c$ | convective heat flow(W/m$^2$) |
| $q_{abs}$ | absorbed shortwave radiation (W/m$^2$) |
| $q_{measured}$ | heat flow measured by the heat flow meter(W/m$^2$) |
| $q_{2\text{-}1}$ | heat flow from pane 2 to 1 (W/m$^2$) |
| $T$ | temperature (K) |
| $T_{ref}$ | reference (air) temperature (K) |
| $T_{surface}$ | surface temperature (K) |
| $\Delta T$ | $T_{ref}$-$T_{surface}$ (K) |
| $\delta$ X | estimated error in parameter X |

indices

| | |
|---|---|
| *frequency* | calculated with the frequency method |
| *FD* | calculated with the Finite Difference method |