HEAT LOSSES IN DETACHED HOUSES WITH GIVEN THERMAL COMFORT CRITERIA

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ABSTRACT

This paper describes a methodology for the evaluation of the thermal indoor climate and energy losses in detached houses.

The paper suggests a method for determination of energy losses that takes into account the actual distribution of temperatures caused by the heating system. Determination of energy losses also takes into account the necessity for maintenance of thermal comfort. The use of the method is demonstrated by applying it to different heating systems. Results are expressed as the boundary thermal resistance of internal surfaces caused by the particular heating system or as the efficiency of heat emission of the heating system. This efficiency could be used in calculations to determine the heat balance of the building.

The results show that real energy losses could be more than 10% higher than those indicated by theoretical calculations using fixed design values for internal air temperature and internal boundary thermal resistance. This discrepancy decreases in buildings with higher levels of insulation.

A study of three types of electric panel radiators in a moderately insulated building shows a difference in energy losses of 6%. The best results were obtained in the case of an enclosed electric heater having a high proportion of heat transferred by radiation. A comparison of three methods of heating—heated ceiling, warm-air heating, and an enclosed heating panel—in a highly insulated building showed no significant differences among the three systems.

INTRODUCTION

For a given building, the choice of systems for heating and ventilating will affect internal air currents and temperature distributions. Different types of heating systems distribute the heat differently within a given room. This has a direct effect on both the building's heat losses and on the indoor thermal climate. The building's ventilation losses depend on the exhaust air temperature, while the transmission losses depend on the temperature distribution over the inside of the building envelope. The occupants' subjective experience of the thermal climate depends on air temperature, surface temperatures, and air velocity.

Calculations of building heat losses (annual or at steady-state conditions), e.g., for the purposes of building design, normally describe the thermal insulation characteristics (from air to air) of the various parts of the building envelope by means of either their U-factors or total thermal resistance. The concept requires some simplifications:

1. The internal conditions in a room are expressed by a single internal temperature.
2. The internal surface resistances are given constant nominal values by a standard, by the building code, or by the test equipment used for determination of U-factors.

Although the thermal resistances (surface to surface) of the various parts of the building envelope are accurately determined by calculations or by measurements, errors are introduced in the calculations by these simplifications. Such calculations also disregard the differences caused by the type of heating system.

This paper describes a method of investigating the effects of temperature distribution by means of comparative testing of different heating systems in the same building but not at the same time. The purpose is to estimate if the errors introduced in calculations by the simplifications mentioned are significant. The method is based on the use of the building envelope as a set of heat flowmeters to obtain any discrepancies in surface resistance for different heating systems. In addition, the paper also describes a method of allowing for the effect of the type of heating system on thermal comfort.

METHOD OF EVALUATING ENERGY LOSSES

A Strategy for Evaluating Energy Losses

The following strategy can be applied to assess how energy losses in a detached house are affected by different heating system principles.

For a given building, the choice of heating and ventilating systems will affect temperature distributions in the air and at various points on the interior of the building en-

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velope. Normally, energy losses are calculated on the basis of the temperature difference between the indoor and outdoor air, using the U-factors of the various elements of the building envelope or the total thermal resistance between the two air sides. If, instead, we employ the total thermal resistance from the inside of the building to the outside (surface-surface), or from its inside to the outdoor air (surface-air), we can determine the transmission losses in relation to the actual temperatures on the insides of the various parts of the building. In the case of ventilation losses, we use the actual temperature of the exhaust air. If the effects of different heating systems are compared in the same building, observed differences in temperature distribution can be directly translated into differences in the transmission and/or ventilation losses. The method is based on the facts that the thermal resistances of the various parts of the building (surface-surface) can be assumed to be constant and are determined with sufficient accuracy and that comparison tests are performed with approximately the same outdoor climate. The observed difference in temperatures is then regarded as being due solely to the effects of the heating systems. It can be said, in other words, that the parts of the building are being used as heat flowmeters.

Instead of making a direct comparison between two different heating systems, the actual energy losses can be related to a theoretical reference system. In the case of Swedish conditions, it is natural to use as a reference a heating system that corresponds to the assumptions made for traditional calculations of heat losses in accordance with the Swedish building regulations. This implies the use of a heating system that produces the same "homogeneous" air temperature at all points throughout a room, which results in an internal surface resistance that is in accordance with the nominal values set out in the building regulations. In Sweden, these calculations are normally made (BBR 1993) on the basis of an indoor temperature of 20°C and internal and external surface resistances of 0.13 and 0.04 $m^2 \cdot K \cdot W^{-1}$, respectively.

A similar technique is used in CEN (1993), which assumes a reference system that produces the same operative temperature at all points throughout a room. The operative temperature is determined as the mean value of the air temperature and the mean radiant temperature.

Allowance for Actual Temperature Distribution

The actual temperature distribution for the particular building is determined on the basis of measurements or calculations. As the absolute temperature level in a room is determined by the thermostat setting, it is difficult to make use of absolute temperatures directly. It is often appropriate to describe the temperatures in a room relative to a common reference, and it is usually the air temperature at the center of the room that is used as such a reference. The temperature at a given point is then expressed as the difference between it and the air temperature at the center of the room. The absolute surface temperatures can then be recalculated with sufficient accuracy for slightly different air temperatures at the center of the room.

When the actual temperature distribution is known, it can be used in steady-state calculations in a number of different ways:

1. Actual surface temperatures can be employed directly when calculating the density of the heat flow rate for the various parts of the building envelope by basing the calculations on the thermal resistance from the internal surface to the external air. Similarly, the actual exhaust air temperature can be used when calculating ventilation losses.
2. As a further step, the actual thermal resistances can be calculated on the basis of the obtained density of heat flow rate and the temperature difference between the air at the center of the room and on the surfaces of the various parts of the building envelope. The U-factors can then be modified to allow for these new internal thermal resistances.
3. When the actual heat losses influenced by the heating system have been calculated using one of the methods specified above, they are summarized over the whole building envelope. The actual heat losses are compared to the total values of nominal energy losses, as calculated by use of the nominal values of surface resistance but apart from that at similar conditions. This can be made through the application of an efficiency factor for heat release from the heating system. The efficiency is calculated as the quotient of the nominal energy losses and the actual energy losses.

Design Indoor Temperature

Calculation of the energy losses of a building is normally based on a constant air temperature in the building. However, in practice, the indoor temperature is determined by the fact that the occupants require a certain degree of satisfaction with the indoor thermal climate. When the outdoor temperature is decreasing, the temperature of the internal surfaces is also decreasing. This results in a demand for increased air temperature, which means that the requirements relating to air temperature depend on the outdoor climate.

The thermal indoor climate can be suitably assessed as described in ISO (1984/1990). Assessment of energy losses should be based on conditions in an empty room, so that it is immediately comfortable as soon as anyone enters it. If there is a risk of excessive temperatures, this is a problem that should be considered, although it does not affect energy use under winter conditions.

The most usable parameter to describe the temperature level in a room is the operative temperature, which is

determined as the mean value of the air temperature and the mean radiant temperature. Determination of the operative temperature is not affected by the room's use. In addition, air currents are generally so slight that use of the physiologically more correct equivalent temperature concept does not require any additional information or degree of accuracy. However, the most practical method is to allow for the operative temperature in a number of steps.

- The variation of the operative temperature within each individual room is determined by measurements or by calculation. The results should at least indicate the value at the center of the room, together with the highest and lowest values encountered in the room. In addition, they should indicate which points in the room are critical with regard to local discomfort. Poor comfort factors, such as temperature gradients, radiation asymmetry, or drafts, cannot be compensated for by increasing the temperature level in the room.
- All temperatures are expressed in relation to a defined reference temperature in the room; the ventilation exhaust air temperature is often used for this purpose. A practical way is to express the temperatures as the difference between them and the air temperature at the center of the room. This method has been employed consistently in the experiments described later in this paper (Johansson 1990, 1993; Johansson and Ekstrand-Tobin 1989; Johansson and Pettersson 1984). The reason for using the air temperature at the center of the room is because it is representative of the temperature in the central parts of the room and because it is a suitable reference temperature for use when determining energy losses.
- The air temperature is determined as required at the center of the room in order to meet the comfort limits for the operative temperature at all points in the occupation zone for different outdoor temperatures. Use this design air temperature for subsequent calculations of energy losses.
- With only modest correction of the air temperature at the center of the room, the surface temperatures of the interior surfaces of the room can be used directly when expressed as differences related to the temperature at the center of the room. Where the required adjustments are greater, it is more accurate to correct the temperature differences in relation to the change in the temperature difference between interior and exterior. Surface temperatures on the exterior of the building could preferably be related to the outdoor temperature.

The Effect of Outdoor Temperature

Both the design air temperature (as described above) and the actual temperature distribution are affected by the ambient temperature and should therefore be expressed as functions of the ambient temperature. This dependence can be determined through repeated measurements made under varying outdoor climatic conditions. These measurements can suitably be combined with a theoretical model for determining the effect of the ambient temperature; a simplified model for describing certain heating systems is described in Johansson (1993).

When calculating the energy losses, dependence on the outdoor temperature can be allowed for when using duration diagrams or when making computer calculations. When using specific heating requirements (degree-hours) for different parts of the country as starting points for calculating the energy losses, the data that should be used are those corresponding to the mean temperature during the heating season.

RESULTS

Surface Temperatures—Surface Resistance

The measured temperatures on the surfaces and those of the room air are interesting in themselves, as they can be used for calculation of such parameters as the operative temperature at various points. An alternative to this is to present true thermal resistances, calculated from the air temperature at the center of the room, to the various surfaces. If, for various reasons, there is interest in estimating the surface temperatures in a room, they can be calculated using the true values of surface resistance.

Efficiency in the Release of Heat

When determining the true energy losses for fulfillment of comfort criteria, they can be compared with the theoretical calculated losses. This can be done through introduction of an efficiency factor, η_ε, indicating the ability of the heating system to warm the room effectively. This efficiency can be calculated from the following relationship:

$$\eta_\varepsilon = \frac{Q_{nominal}}{Q_{comfort}}$$

where $Q_{nominal}$ is calculated by nominal calculation of the losses, and $Q_{comfort}$ is calculated from the actual conditions when the comfort criteria are fulfilled.

Error Estimation

The proposed method is based on the assumption that the thermal resistance of the parts of the building envelope are known and can be regarded as constant, so that the parts of the building can be used as heat flowmeters. In addition, it is assumed that the external surface resistance is constant if the transmission losses are calculated relative to the outdoor temperature. These assumptions are not always true, and it may be necessary to look more closely at the errors that can arise.

Table 1

Part of the Building	U-Factor $W \cdot m^{-2}K^{-1}$	R $m^2 \cdot KW^{-1}$	Rsi $m^2 \cdot KW^{-1}$	Rss $m^2 \cdot KW^{-1}$	Rse $m^2 \cdot KW^{-1}$	q $W \cdot m^{-2}$
Wall	0.2	5.00	0.13	4.83	0.04	4
Window	1.9	0.52	0.13	0.35	0.04	38

Determination of the Thermal Resistance of Individual Parts of the Building Envelope

Different parts of a building envelope have different thermal insulation performances. Consider, for example, an exterior wall and a triple-glazed window. With an indoor temperature of +20°C and an outdoor temperature of 0°C, the mean temperature in these elements of the building is +10°C, with the characteristics listed in Table 1, assumed to be representative when used in the nominal calculations.

The expected surface temperature on the interior surfaces of these two parts of the building are then +19.48°C for the wall and +15.0°C for the window. If the surface temperature actually measured shows a departure of 1.0°C from these values, it is then interpreted as a departure in the value of the heat flow density of 0.205 $W \cdot m^{-2}$ for the wall and 2.56 $W \cdot m^{-2}$ for the window caused by the heating system. Let us assume that there has been some error in determining the insulation performance, so that the actual performance has been overestimated by 10% in making the nominal calculations. This will mean that the actual departures in the heat flow density for a difference of 1°C in the surface temperature are 0.228 and 2.82 $W \cdot m^{-2}$, respectively.

The error in determining the thermal resistance of the parts of the building is constant, which means that it will have the same effect on the various heating systems when making comparative measurements. This means that if the thermal resistance is overestimated by 10%, an observed difference between the performance of the different heating systems will also be overestimated by 10%. *But the important point is that even small observed differences are significant. Errors in determination of the thermal resistance of the parts of the building envelope used as heat flowmeters are not very serious for the accuracy of the suggested method if they are constant.*

The method assumes that the thermal insulation performance and draftproofing of the various parts of the building are up to standard. If not, the thermal insulation performance can vary substantially with different wind and pressure conditions. If in doubt, checks should be made using a thermal camera.

Inaccuracy of Measurement

Assume that the surface temperatures can be measured with an accuracy of ±0.2°C. For the wall, this means that there is a potential inaccuracy when determining the heat flow density of ±0.04 $W \cdot m^{-2}$ or ±1.0%. For the window, the corresponding potential errors are ±0.51 $W \cdot m^{-2}$ or ±1.3%. When using calibrated temperature sensors, the error can be halved by direct measurement of the temperature difference across the various parts of the building.

Dependence of Insulation Performance on Temperature

The insulation performance of porous insulation materials decreases as the mean temperature of the materials increases. For the majority of porous materials, this decrease is about 0.5% per °C. Allowance can be made for this when making measurements. If we ignore the temperature dependence and assume that the thermal resistance is constant, this will result in an error contribution when making comparison measurements of two different heating systems if the outdoor temperature differs between the two measurement occasions. If the outdoor temperature differs by 2°C between the two occasions, this will mean that there is a change of 1°C in the mean temperature. The insulation performance changes by 0.5%, resulting in a corresponding error when determining the differences in transmission losses between different heating systems. When calculating the expected surface temperature on the inside of the wall, this gives an error of 0.1°C.

In the case of an ordinary triple-glazed window, insulation performance drops by about 0.8% per °C. For a 2°C difference in outdoor temperature, this is equivalent to an error of 0.8% when determining the differences in performance between different systems. The expected surface temperature changes by about 0.1°C.

A double-glazed window with a low-emission coating has approximately the same insulation performance as a triple-glazed window, except that its insulation performance increases by about 0.3% per °C as the mean temperature of the window rises.

Changes in the External Surface Resistance

The outdoor climate can be controlled well when performing tests under laboratory conditions, but the external surface resistance can vary substantially when performing full-scale experiments in real buildings. The effect of this can be avoided by measuring the surface temperatures on both the interior and exterior surfaces of the building and calculating the heat flow density on the thermal resistance between the two surfaces. This applies particularly for windows.

As comparison tests are based on measurement of the outdoor temperature, measurements should be made on cloudy days, when the external surface resistance varies

more modestly. The effect of the variations is also limited by the fact that the external surface resistance gives rise to only a smaller part of the total thermal resistance of the particular part of the building.

Assume that the external surface resistance varies within the range $0.04 \pm 0.04\ m^2 \cdot K \cdot W^{-1}$ between the various measurement occasions. For the wall, as described above, this is equivalent to an error in determination of the heat flow density of ±0.8%, or $\pm 0.03\ W \cdot m^{-2}$, for a temperature difference of 20°C between the inside and outside. The corresponding error in calculating the surface temperature on the inside of the wall is ±0.16°C.

In the case of the window, the effect is considerably larger at ±7%, or $\pm 2.6\ W \cdot m^{-2}$. The corresponding error in determining the surface temperature of the inner pane is ±1.2°C.

Maximum Error

The maximum error in determining the transmission losses can be illustrated by a calculation. Assume that the indoor temperature is about +20°C and that the outdoor temperature is 0°C. Assume also that the thermal resistance of the window has been estimated at $0.52 \pm 0.05\ m^2 \cdot K \cdot W^{-1}$ and that the thermal resistance of other parts of the building has been estimated as $5.0 \pm 0.5\ m^2 \cdot K \cdot W^{-1}$. Assume, too, that the surface temperature of the window has been found to be 15°C for system A and 13°C for system B, with the surface temperatures of the other parts of the building being 19.5°C and 19.0°C, respectively.

When performing the tests in a laboratory, the outdoor climatic conditions can be controlled and both the external surface resistance and the thermal resistance of the building envelope can be assumed to be constant. This leaves only the effect of uncertainty in the determination of the thermal resistance of the parts of the building envelope used as heat flowmeters together with the effect of uncertainty of measurement of temperatures. Heat flow density can then be determined as follows: (Calculated flux + Contribution from constant error in determination of the thermal resistance of the particular part of the building + Contribution from random error in temperature measurement).

The heat flow density through the window is:

For system A	38.5 + 3.8 ± 0.5	$W \cdot m^{-2}$
For system B	33.3 + 3.3 ± 0.5	$W \cdot m^{-2}$
Difference	5.2 + 0.5 ± 1.0	$W \cdot m^{-2}$

The heat flow density through the walls, etc., is:

For system A	4.00 + 0.40 ± 0.04	$W \cdot m^{-2}$
For system B	3.90 + 0.39 ± 0.04	$W \cdot m^{-2}$
Difference	0.10 + 0.01 ± 0.08	$W \cdot m^{-2}$

The method provides good results under laboratory conditions, allowing differences in heat losses greater than about 2% to be noted. The window will cause greater errors then other parts of the building with higher thermal resistance. But when the total heat losses for the building are summarized, the area of the windows is small in comparison to the total area of the building envelope.

A similar calculation in the case of field measurements shows increasing errors, especially for the heat transmission through windows. If the temperatures of the exterior surfaces of the various parts of the building are measured, then differences in total heat losses greater than about 3% can be detected. If the outdoor temperature is used, the difference in the total heat losses caused by the heating systems must be greater than about 6%.

The conclusion that can be drawn from this is that the temperature of the exterior surfaces of the various parts of the building should be measured directly. This applies particularly for windows.

APPLICATION OF THE METHOD TO DIFFERENT HEATING SYSTEMS

This suggested method of working has been applied to measure data from earlier tests on heating systems reported in Johansson (1990, 1993), Johansson and Ekstrand-Tobin (1989), and Johansson and Pettersson (1984). These systems have included electric ceiling heating, airborne heating with ceiling-fitted inlet distributors, and various types of electric radiators. Most of the measured data have been obtained from tests performed in test rooms in climate chambers, although the results of field measurements are also included. The test conditions have varied from one set of trials to another in that the proportion of cooled surfaces has varied, as has the amount of insulation. The outdoor temperature has also varied. However, all tests have included measurements on some type of enclosed electric radiator, the results of which can be compared with other heating systems.

The following briefly describes the results of calculations of energy losses based on the various series of tests. A more detailed presentation of the calculations, and of the input data from the various underlying tests, is given in Johansson (1994).

Ceiling Heating and Electric Radiator in a Climate Chamber

Testing was performed in a test room (3.3 m long, 3.0 m wide, and 2.1 m high) constructed inside a climate chamber so that its ceiling, floor, and two end walls were cooled. One of the end walls contained a 1.2-m^2 double-glazed window with a wooden frame. The amount of insulation in the structure was quite low, with 5 cm of foamed plastic in the floor and 10 cm in the walls. The measurements here are those for a system with electric ceiling heating and an enclosed electric radiator. The ceiling heating system provided 125 $W \cdot m^{-2}$, with a total nominal power of 800 W. The ceiling heating foil covered about 65%

of the area of the ceiling, with the highest power near the window wall. The indoor air temperature was about 22°C and the outdoor temperature was about −15°C. The cold air circulated past the ceiling, the rear wall, and the floor, with the temperature rising progressively to about −9°C. All the results used as-measured surface temperatures from the outside of the building envelope.

Table 2 shows the observed temperature distributions on the internal room surfaces for both heating systems, expressed relative to the air temperature at the center of the room. Table 3 shows the corresponding internal surface resistance, while Table 4 summarizes the efficiencies.

Different Electric Radiators in Climate Chamber Tests

The following example describes laboratory testing of different electric radiators (Johansson and Ekstrand-Tobin 1989; Johansson 1990). These tests were performed in a test room that measured 3.5 m long, 3.0 m wide, and 2.5 m high. One wall of the room, with 10 cm of thermal insulation and incorporating a double-glazed window, was cooled. The room was ventilated with supply air through an air inlet above the window. From the total pool of results available, those for three different types of radiator have been selected for consideration here:

A. Convection radiator, discharging warmed air in an upward direction.
B. Convection radiator, discharging the air inward to the room.
C. An enclosed radiator with a relatively large surface area, emitting much of its warmth as radiation.

Tables 5 through 7 present the most important results from this comparison.

Table 2 **Air/Surface Temperature Differences for Ceiling Heating and Radiators**

Part of Room	Ceiling	Floor	Exterior Wall		Window
Nominal	−1.7	−2.4	−1.6	−1.8	−13.6
Actual, ceiling heating	+10.8	−0.6	−0.4	−0.5	−9.2
Actual, radiator	−1.4	−3.6	−1.9	−0.4	−2.5

Table 3 **Internal Surface Resistance for Ceiling Heating and Electric Radiator ($m^2 \cdot KW^{-1}$)**

Part of Room	Ceiling	Floor	Exterior Wall		Window
Nominal	0.13	0.13	0.13		0.13
Actual, ceiling heating	−0.62	0.03	0.03	0.03	0.07
Actual, radiator	0.10	0.20	0.16	0.03	0.02

Table 4 **Efficiency for Ceiling Heating and Electrical Radiator**

Assessment Basis	Efficiency, η_ε	
	Ceiling Heating	Electric Radiator
For the same air temperature at the center of the room	0.86	0.91
For the same comfort	0.86	0.85

Table 5 **Air/Surface Temperature Differences for Different Types of Radiators**

Part of Building	Outer Wall	Window
Nominal	−1.4	−11.3
Radiator A	+0.1	−2.5
Radiator B	−1.2	−6.1
Radiator C	+2.0	−6.4

Table 6 **Internal Surface Resistance for Different Radiators**

Part of Building	Outer Wall	Window
Nominal	0.13	0.13
Radiator A	−0.01	0.02
Radiator B	0.11	0.06
Radiator C	−0.17	0.06

Table 7 **Efficiency, η_ε, for Different Radiators**

Assessment Basis	Efficiency, η_ε		
	Radiator A	Radiator B	Radiator C
For the same air temperature at the center of the room	0.82	0.88	0.88
For the same comfort	0.80	0.85	0.87

Airborne Heating and Electric Radiator in a Well-Insulated Detached House

The results from full-scale tests (Johansson 1993) of two heating systems have been used in the following example. The tests were performed in the bedroom of a well-insulated detached house having an airborne heating system with ceiling-mounted air supply devices. The same room also contained an electric radiator, which could be used as an alternative source of heating. The supply air temperature with radiator heating was about +15°C. The radiator was of the enclosed type and identical to radiator C in the previous example.

Tables 8 through 10 show the main results of these tests.

Summary of the Various Systems' Efficiencies

The greatest difference between the different types of heating compared here was obtained in a comparison of the different types of electric radiators.

Table 11 shows the efficiency, η_ε, related to thermal comfort in the coldest part of the room. The difference in energy losses between the poorest and the best radiator systems is 6%. The best results were obtained with the enclosed radiator with a large surface area, which delivered much of its heat by radiation.

When ceiling heating and airborne heating are compared with an enclosed radiator, the differences in efficiency are slight, as shown in Table 12. The amount of insulation in the building structure, on the other hand, has a substantial effect. For the best-insulated building, the true energy loss was about 2% to 3% higher than cal-

Table 8 **Air/Surface Temperature Differences for Airborne Heating and Radiators (°C)**

Part of Room	Ceiling	Floor	Exterior Wall	Window
Nominal	−0.2	−0.3	−0.4	−2.9
Actual, ceiling heating	0.6	−0.7	−1.0	−2.6
Actual, radiator	−0.3	−0.4	(−0.7)	−1.2

Table 9 **Internal Surface Resistance for Airborne Heating and Electric Radiator ($m^2 \cdot K \cdot W^{-1}$)**

Part of Room	Ceiling	Floor	Exterior Wall	Window
Nominal	0.13	0.13	0.13	0.13
Actual, ceiling heating	−0.32	0.27	0.34	0.11
Actual, radiator	−0.20	0.15	0.22	0.05

Table 10 **Efficiency, η_ε, for Airborne Heating and Electric Radiator**

Assessment Basis	Efficiency, η_ε Ceiling Heating	Efficiency, η_ε Electric Radiator
For the same air temperature at the center of the room	1.02	1.00
For the same comfort	0.98	0.97

Table 11 **Efficiency, η_ε, for Different Radiators**

Type of Radiator	Efficiency, η_ε
A. Through-flow with upward air discharge	0.80
B. Through-flow with front air discharge	0.85
C. Enclosed	0.87

Table 12 **Efficiency, η_ε, for Ceiling Heating and Airborne Heating in Comparison with Electric Radiator**

Building Type	Type of Heating	Efficiency, η_ε
Poorly insulated	Ceiling heating	0.86
Poorly insulated	Enclosed radiator	0.85
Well insulated	Airborne heating	0.98
Well insulated	Enclosed radiator	0.97

culated, while for the less insulated building, the difference was more than 10%.

Comment

Surface temperatures and internal surface resistance vary considerably between the different types of heating systems. Ceiling heating and airborne heating both result in high surface temperatures of the ceiling, with negative values of the surface resistance as a result. This is to be expected; ceiling heating supplies all the thermal power to the surface of the ceiling, from where it heats the rest of the room by radiation. The airborne heating system also tends to heat the surface of the ceiling, from which the heat is again emitted by radiation.

The main effect of radiator heating systems is that the surface temperature of the window is higher. When the surface temperature of a poorly insulated part of the building envelope is raised, transmission losses increase accordingly, although this is compensated for in several of the systems by a raised operative temperature.

CONCLUSIONS

The proposed method of working can be employed with relatively good accuracy if the measurements are planned to suit its needs. Attempting to analyze the results afterward can be difficult if important data are missing.

From the examples studied here, it can be seen that when allowance is made for the true thermal climate produced by the heating systems, the corresponding true energy use is higher than the calculated value. This applies particularly in the case of the poorly insulated buildings, for which the difference exceeds 10%.

The difference is less in the case of well-insulated buildings, estimated to be on the order of 2% to 5%. When comparing different heating systems, the interesting result obtained is that there is a significant difference in efficiency between different types of electric radiators. The energy losses differ by 6% between the best and poorest radiators; the best results were obtained from an enclosed radiator releasing much of its heat as radiation.

The differences are small when comparing results between ceiling heating and airborne heating with an enclosed-type radiator.

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