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Calculating Energy Consumption and Thermal Dynamics of Buildings – Comparison of Some Calculation Models and the Effect of Input Data and Modelling Assumptions on Results

Introduction

The energy analysis of buildings includes many important issues e.g. the calculating of heating and cooling energy, the dimensioning of heating and cooling devices and the calculating of thermal indoor climate. The importance of all these issues is increasing. E.g. the need to reduce the CO_2-emissions means that also the energy consumption of buildings must be reduced because it makes up a considerable share of total energy consumption. Energy analysis programs are one method for selecting the most suitable and economic energy saving methods for buildings.

A good thermal indoor climate is an important factor especially in offices. Energy analysis programs are the only way to study the indoor climate in the building's design phase and to select proper building components and heating and ventilating systems.

There are two principally different methods for calculating the heating energy consumption. The former is *an energy balance method* and the latter *a dynamic simulation model.* In the energy balance method the heat losses of building components are calculated using a thermal transmittance for each component of the exterior envelope and the internal heat gains are taken into account using a utilisation factor. The interior is an temperature input value for calculations. Thus, with the energy balance method it is not possible to study the thermal indoor climate or the cooling energy. The calculation period is usually one month or the whole heating season. The national building regulations concerning the calculation of heating energy are usually based on the energy balance method. The new European standard EN832 *Thermal Performance of Buildings - Calculation of Energy Use for Heating* /1/ is an example of this.

Usually thermal analysis of buildings means using a dynamic model in which the calculation period is short (e.g. 1 hour) and the heat capacity of the building components is taken into account explicitly. A dynamic thermal simulation program calculates either the heating or cooling effect needed to achieve a certain interior temperature or the interior temperature when a certain heating or cooling effect is supplied into the building.

There are numerous dynamic energy analysis programs. There both widely used commercial and public domain programs (e.g TRNSYS, ESP and DOE) and national programs, such as our TASE /2/. The advantage of an own program is usually the fact that it can be easily modified when new systems and components are studied. As an example of the new systems studied with the TASE program is an electric window whose energy losses and effects on thermal indoor climate were analysed /3/.

Accuracy of energy analysis

The results of the energy analysis of buildings depend on three main points which all affect the accuracy:

1. The model itself, modelling principles, simplifications compared with reality and possible errors. There are e.g many methods for calculating the transient heat flux of walls and the thermal radiation between interior surfaces.
2. The input data used
3. The program user and how she/he models the physical reality.

It can be thought that an error less than 10 % in the annual energy or peak heating or cooling effect or an error less than 1 K in the interior temperature were good results in an energy analysis of buildings. However, the comparisons made on the accuracy of energy analysis show that the values of energy consumption can differ up to 40 % from each other when calculated by different persons with and different programs. In the calculated interior temperatures there can be differences up to 10 K.

International Energy Agency (IEA) has organised many interesting comparisons. One was the intermodel comparison of Task 12 of *Solar Heating and Cooling Programme* /4,5/. Figure 1 presents as an example the annual heating energy for 12 cases calculated with six programs. The ratio of the maximum and minimum energy consumption for a certain case is approximately 1.5-3. The maximum absolute difference is 200-300 kWh/a for all cases. The weather data of Denver (Colorado) was used in these calculations.

The intermodel comparisons tell only little from the 'truth'. Therefore it is interesting to compare calculation results with measurement results. This was done in the IEA Annex 21 of *Energy Conservation in Buildings and Community Systems Programme.* This kind of comparison should be performed 'blind' without knowing the measurement results in advance and accurate input data (e.g. thermal properties of building materials and weather data) should be obtainable. These points were taken into account in the comparison of Annex 21 /5,6/. Figure 2 shows the results, in which the energy consumption is presented for a one-week period in October. The average calculated energy consumption was less than the measured one. The ratio of the calculated maximum and minimum energy consumption was approximately 1.6. The test cell was situated in England.

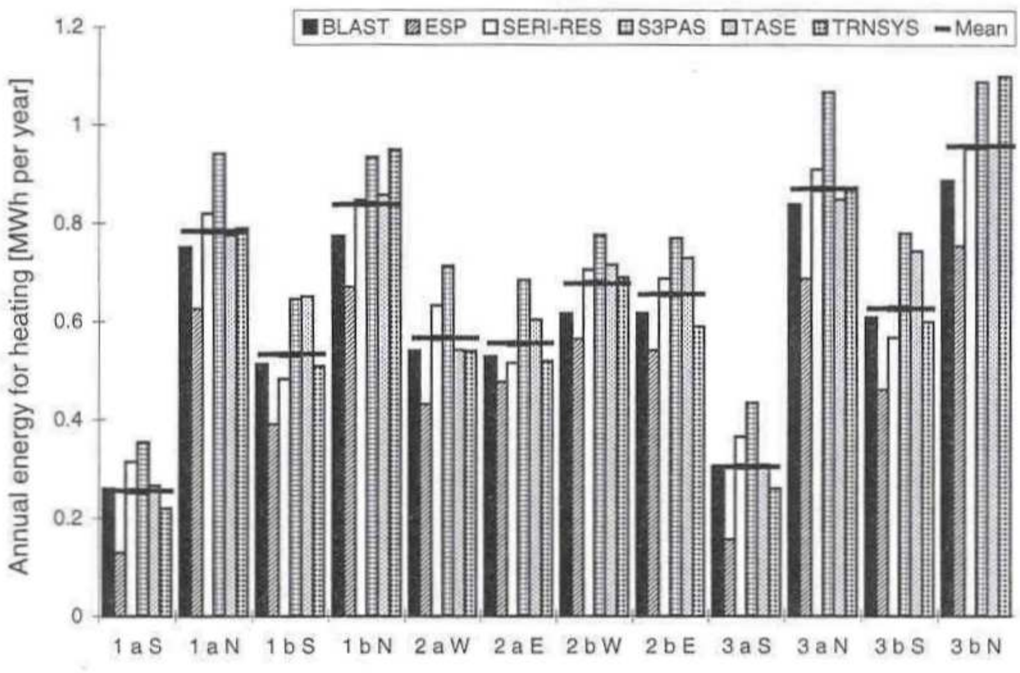

Figure 1. Calculated annual energy consumption with 6 programs for 12 cases. Denver climate. /4/.

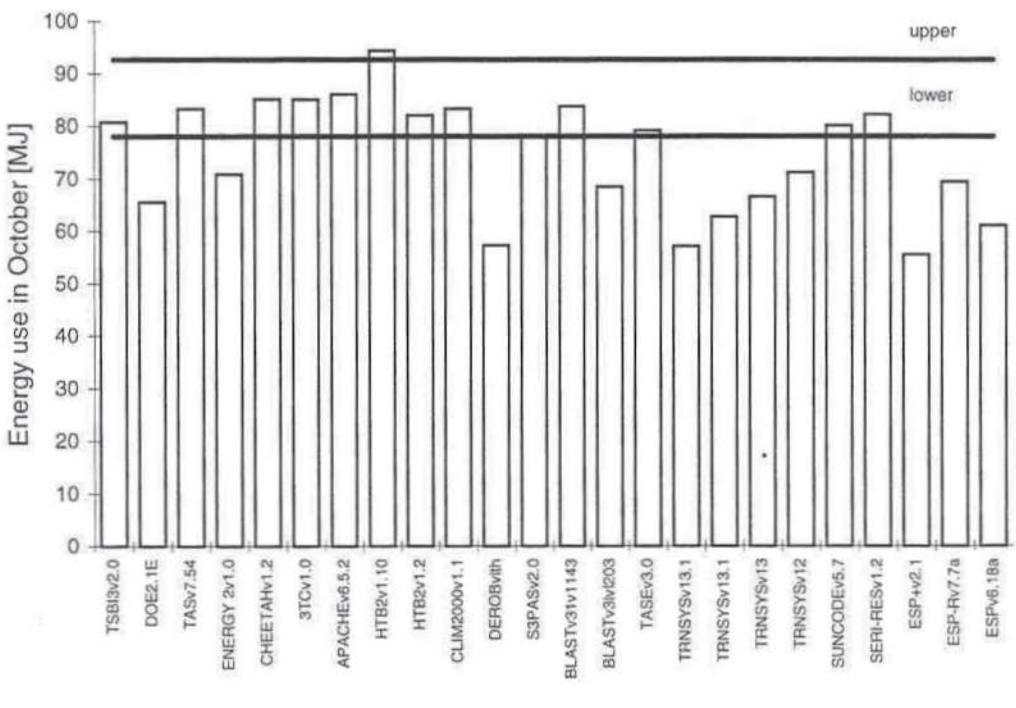

Figure 2. Comparison of measured (lower and upper estimation) heating energy with calculated ones. One-week period in October in England /6/.

Optimisation of thermal insulation of the envelope

It is possible to couple the energy analysis of buildings with an optimisation of the thermal insulation of the building's envelope. The natural objective function in the optimisation is the life-cycle cost which includes the investment costs in the thermal insulation of the envelope and those in the heating system as well as the annual energy costs. The design variables are the values of insulation and thickness of exterior walls, floor and ceiling the thermal transmittance (U-value) of windows. The heating system must be included in the model because the price of heat can be very different in various systems.

For the optimisation the unit costs of thermal insulation (FIM/m^3-insulation) and the unit costs of windows (FIM/m^2) as a function of the thermal transmittance are needed. The optimisation can include various constraints, e.g. a maximum allowable annual energy consumption, a maximum insulation thickness in a certain surface or a maximum investment cost.

We have made an optimisation model, called OPTIX, using Microsoft Excel worksheet. The calculation of annual energy consumption is made using the European standard EN 832 /1/ and the optimisation using the solver of Excel /7/. The solution algorithm in Excel is Generalised Reduced Gradient (GRG2).

Figure 3 presents an example of optimisation for a single family house (floor area 150 m^2) in Finnish climate . Optimal insulation thickness of exterior walls are compared for two heating systems (electric heating and heat-pump heating) and for two energy prices. It can clearly been noticed that with a heating system producing heat at low cost (heat pump) the optimal insulation thickness is clearly smaller than with electric heating producing expensive heat. *Optimised 1* presents a case in which present Finnish energy prices are used and *Optimised 2* a case in which the energy price is doubled compared with the first one. Case *Regulation* presents the insulation thickness according to Finnish building regulations. The economic advantage of optimisation in the life-cycle costs can be up 15-25 % compared with a case in which no optimisation is made and the envelope is insulated according to the minimum demand of building regulations. The smaller value concerns a case in which the heating systems is fixed and the latter a case in which the most economic heating system is selected.

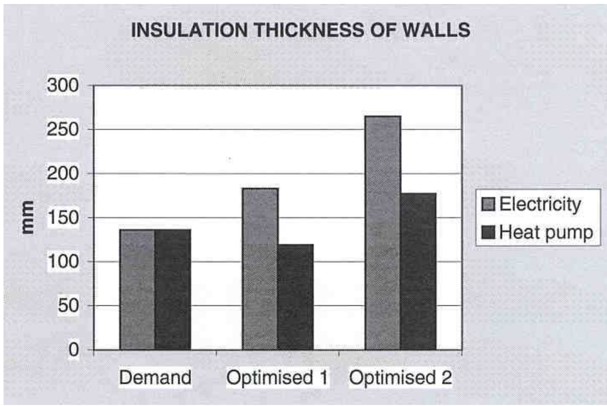

Figure 3. Optimal insulation thickness of exterior walls. *Optimum 1* calculated using present Finnish energy price and *Optimum 2* using a doubled energy price.

Cooling of buildings during interruption of heating

Winter 1998-99 was extremely cold in Lapland. The minimum exterior temperatures were – 40 - -50 ^{0}C. Most of Finnish single-family houses are light-weight structured whose time constant is approximately 50 hours. The most usual heating system is electric heating. The electric network is not dimensioned for such extreme low temperatures and therefore in many houses the heating was interrupted last winter due to lack of electricity.

The cooling of light-weighted buildings is rapid during extreme cold exterior temperatures. It takes approximately 30 hours when the interior temperature falls below 0 ^{0}C if the exterior temperature is -36 ^{0}C (Figure 4). TASE and IDA programs give almost the same cooling speed. The simple one time-constant model gives in the beginning of the cooling same results as the simulation programs, but later it gives clearly a too rapid cooling speed. The time-constant model can thus be used as a rough model in estimating the thermal dynamics /8/.

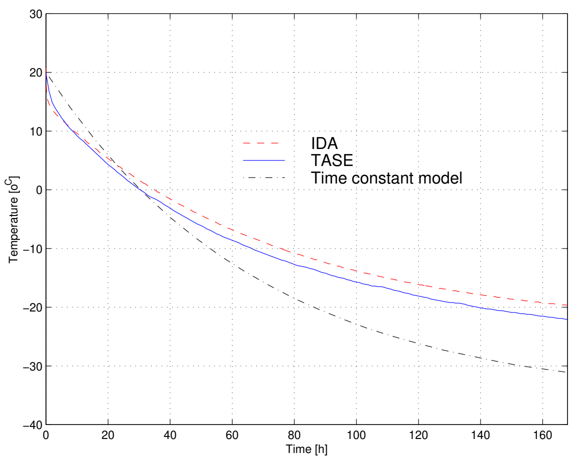

Figure 4. Cooling of a light-weighted single-family house during an interruption of heating. Exterior air temperature – 36 ^{0}C.

Energy analysis program TASE

The TASE program /2,9/ is a transient multiroom model for thermal analysis of buildings. The solution method is detailed in that thermal radiation between interior surfaces is taken into account and a heat balance for each interior surface and room air is calculated. The transient values of wall heat flux are calculated using transfer factors of Mitalas and Arsenault /10/.

IDA Indoor Climate and Energy

IDA Indoor Climate and Energy is a tool for simulation of thermal comfort, indoor air quality and energy consumption of buildings. IDA Simulation Environment consists of IDA NFM translator, IDA Modeller and IDA Solver. The mathematical models or components are described in terms of formal language, Neutral Model Format (NMF). To build a simulation model a number of components or modules are connected together. IDA Solver is then employed to solve the system of differential – algebraic equations that the equations of the modules will form /11/.

Interior heat transfer coefficients

In thermal analysis programs the convective heat flux from the room air to each surface is needed. Usually the room air and each surface are assumed to be isothermal. The convective heat transfer is calculated using appropriate correlation equation for each surface.

The two main methods for calculating convection are:

(1) Heat transfer coefficients are constant through the whole calculation
(2) Convective heat transfer coefficients are calculated iteratively from temperature differences between air and each surface. In addition, the direction of heat flux must be taken into account for horizontal surfaces

The convective heat transfer coefficients of walls are usually calculated from the equation

$$h_c = C\left(\frac{\Delta T}{L}\right)^n \tag{1}$$

where ΔT is the temperature difference between the air and surface and L is the characteristic length.

In reality the surfaces and the room air are not isothermal and the interior convection should be calculated for each time point using a program based on computational fluid dynamics (CFD).

In the earlier study /9/, in which various methods to calculate the interior convective and radiative heat transfer were compared, it was noticed that the heat transfer coefficients of Khalifa & Marshall /15/ for walls, ceilings and windows gave better results, compared with measurements, than the corresponding heat transfer coefficients of ASHRAE /16/ or Alamandri & Hammond /17/.
Table 1 presents coefficients C and n according to ASHRAE and according to Khalifa Marshall. The latter are suited for room having a radiator under a window.

This work compares interior temperatures of the TASE program obtained using constant values for interior heat transfer coefficients (Table 2) and temperature-dependent heat transfer coefficients calculated using the correlation of Khalifa and Marshall. Furthermore, heat transfer coefficients calculated using the commercial CFD program FLUENT were used in comparison

The CFD calculation of interior convective heat transfer coefficients was performed using a 100x120 two-dimensional grid and RNG k-ε model. The average heat transfer

Table 1. Coefficients C and n and characteristic length L of Eq. (1) according to ASHRAE /16/ and Khalifa Marshall /15/

Correlation	**Type of convection**	**C** *($W K^{-1} m^{-2}$)*	**n**	**L** *(m)*
ASHRAE	Vertical surface	1.31	0.33	1
	From horizontal surface	1.52	0.33	1
	Towards horizontal surface	0.59	0.25	L[a]
Khalifa Marshall	Walls	2.3	0.24	1
	Ceiling	3.1	0.17	1
	Window	8.1	0.11	1

[a]L is the length of the surface (in meters).

Table 2. Constant heat transfer coefficients for the test cell

Surface	**Heat transfer coefficient** *($W K^{-1} m^{-2}$)*	
	Convective	Radiative
Wall	1,9	6,0
Window	2,4	6,0
Floor	0,6	6,0
Ceiling	2,2	6,0
Radiator	9,8	7,4

coefficients were obtained assuming a steady-state air flow and constant temperature boundary conditions on the surfaces of the room. The temperatures of room surfaces were obtained from measurements and a steady-state calculation was performed every tenth minute.

Measurement arrangements

The capability of the TASE program and IDA Indoor Climate and Energy for calculating transient conditions was studied in a two-room test cell (Fig. 5). The purpose was to compare the calculated room air temperatures with corresponding measured values. Radiator heating was used and the radiators were placed under the windows. The thermal conditions outside the test cell were constant. The outside temperature was approximately 20 ^{0}C and there was no short-wave radiation. The thermal dynamics of the cell were studied by heating the cell from 20 ^{0}C to 40 ^{0}C and letting it cool freely. Only one of the two rooms was heated and the door between the rooms was closed.

The walls, the roof and the floor were light-structured (Table 3) and the windows were triple-glazed. The measured air temperature presented is the average value of seven measurement points in the middle of the room between heights of 0,1 and 2,4 m. A more exact description of the measurement arrangements is found of /9/.

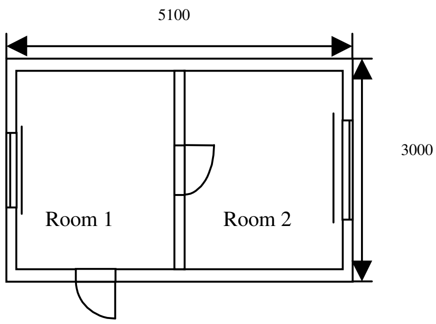

Fig. 5. Plan of the test cell used in the measurements.

Table 3. Structures information of the wall, the roof and the floor.

Surface	Structure and thickness
Wall	chipboard 10 mm + polyurethane insulation 120 mm + windshield 12 mm
Roof	chipboard 10 mm + air gap 50 mm + polyurethane insulation 120 mm
Floor	chipboard 25 mm + polyurethane insulation 120 mm

Table 4. The thermal properties of the wall layers.

Material	Density *($kg\ m^{-3}$)*		Thermal conductivity *($W\ K^{-1}\ m^{-1}$)*		Specific heat capacity *($J\ kg^{-1}\ K^{-1}$)*	
	Measured	Literature	Measured	Literature	Measured	Literature
Chipboard	630-720	600	0.107	0.13	1510	1380
Polyurethane	36	37	0.0264	0.030	1300	1050
Windshield	315	300	0.058	0.045	1150	1000

Dynamic situation in the test cell

The dynamic situation studied consists of three phases:

(1) First, one of the two rooms of the test cell was heated intensively (6.5 h) using an electric radiator. The power input into the room, in which the measurements were performed, was 770 W.

(2) In the second phase the power was reduced to 400 W for 4 h.

(3) In the third phase the power input was shut off and the room was allowed to cool freely 20 h.

Figs. 6 and 7 shows that the calculated changes in the air temperatures are more rapid than the measured ones, especially when the constant heat transfer coefficients are used in calculations. The temperature-dependent convective heat transfer coefficients of Khalifa and Marshall give clearly more accurate results for the air temperature during the rapid heating phase compared with the constant heat transfer coefficients. During the cooling phase the air temperatures, calculated with the basic thermal properties, decrease faster than the measured ones. One can see that not only the accurate calculation model but also the use of correct material properties has a noticeable effect on results.

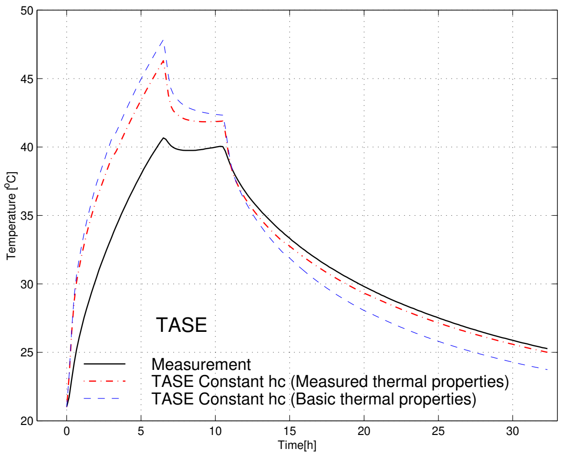

Fig. 6. Measured and calculated air temperatures, Constant convective heat transfer coefficients.

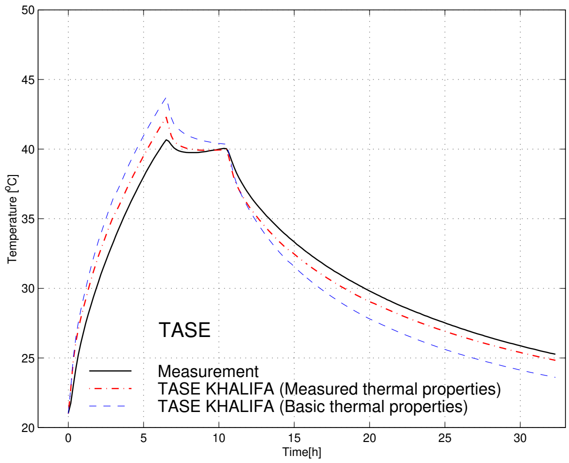

Fig. 7. Measured and calculated air temperatures. Temperature-dependent convective heat transfer coefficients (Khalifa and Marshall).

The convective heat transfer coefficients, calculated by FLUENT, give only slightly better results compared with the results obtained using the coefficients of Khalifa & Marshall.

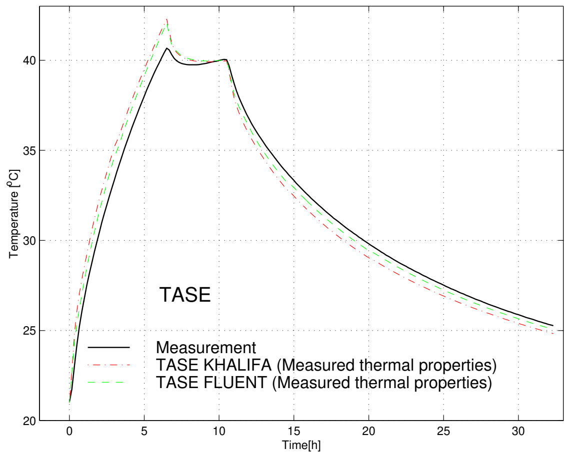

Fig. 8. Measured and calculated air temperatures. Temperature-dependent convective heat transfer coefficients calculated using FLUENT.

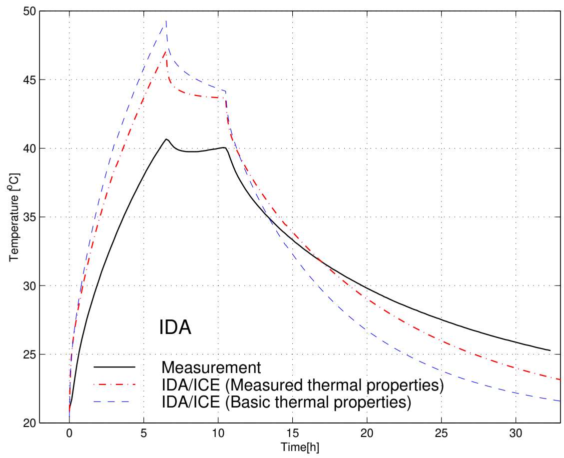

Fig. 9. Measured and calculated air temperatures. Temperature-dependent convective heat transfer coefficients.

Fig. 9 shows that IDA Indoor Climate and Energy gives too rapid changes in the air temperatures compared with the measurements. The difference compared with measurement results seems to be greater than that obtained with the TASE program even though IDA uses temperature-dependent heat transfer coefficients.

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