eenvoudig EXCEL progr. : goed om voeling te krijgen met dynamiek
↳ niet juist genoeg voor conditionering!

VENTILATION AND COOLING
18TH ANNUAL AIVC CONFERENCE
ATHENS GREECE, 23 - 26 SEPTEMBER, 1997

Title

Thermal analysis of rooms with diurnal periodic heat gain, ThermSim.
Part 2 : Practical use and comparison

Part 1 = ? → tafel achteraan..

Authors

M.Sc. Tor Helge Dokka[1,2] and B.Sc. Hanne Dokka[1]

Affilation

1 ProgramByggerne ANS, Pirebergv. 36, 7045 Trondheim, Norway
2 Norwegian University of Science and Technology, 7034 Trondheim, Norway

Software firm!

doel v. cooling load is swallowed by thermal mass

goed om dyn aspect te verstaan!

eenvoudig model in EXCEL

doel beter verstaan van bouwfysica

nt mogelijk om ventilatie te wijz in de tijd

Thermal analysis of rooms with diurnal periodic heat gain, ThermSim. Part 2 : Practical use and comparison

Synopsis

Temperature and cooling demand in a room summertime are influenced by numerous factors, like internal gains, ventilation, solar gain, behaviour of occupants, thermal inertia of the room and outdoor conditions (climate).

The thermal environment and cooling demand summertime are often analysed using detailed computer programs, which take into account the factors mentioned above (among others). Often the overview, transparency and some of the physical insight is lost using these advanced computer programs.

In a predesign phase of a project it is preferable to do simple calculations of the thermal behaviour of a room. These simple calculations often gives more physical insight and overview than using computer programs. Simple calculations also gives a quality assurance of later computer analysis of the room.

This is part 2 of two related papers concerning a simplified method for thermal design of rooms, called *ThermSim*. Part 1 (the accompanying paper) is concerned with derivation and interpretation of the model.

This paper is concerned with practical guidance in choosing appropriate input to the model. Comparison to the advanced simulation program BRIS is also presented.

The model shows good agreement with computer analysis when the model assumptions is fulfilled.

List of symbols

Symbol	Description	Unit
A_{fac}	Facade area	m^2
A_{floor}	Floor area	m^2
A_{win}	Area for whole window construction (including frame)	m^2
C_{air}	Heat capacity of air (can be set to 0.34 Wh/m^3K)	Wh/m^3K
F_{sh}	Effective total shading factor	-
L''	Normalized mechanical or natural air flow rate	m^3/hm^2
n	Air infiltration in ACH	1/h
n_{per}	Occupation time for persons	h
$n_{l\&a}$	Operation time for lighting and appliances	h
q_{per}	Heat gain from persons	W
$q_{l\&a}$	Heat gain from lighting and appliances	W
q''_{sol}	Solar intensity through a vertical pane	W/m^2
Q''_{sol}	Daily sum of solar gain through a vertical pane	Wh/m^2
$\overline{T}_e$	Mean daily external temperature	°C

$\hat{T}_e$	Daily amplitude external temperature	°C
ΔT_{fan}	Temperature rise over the supply fan	K
U_{fac}	U-value facade construction	W/m²K
U_{win}	U-value window construction	W/m²K
V	Air volume	m³

1 Introduction

This is part 2 of two related papers concerning a simplified method for thermal design of rooms. Part 1 (the accompanying paper) presents a method/model, called *ThermSim*, that can be used in the thermal design of rooms.

This paper is concerned with practical guidance in choosing appropriate input to the model. In addition this paper gives examples in the use of the model, and comparison to advanced computer simulation.

2 Guidance in choosing appropriate input data

This sections gives guidance in selecting appropriate input to the model. Most of the input can be "normalized" by dividing the value with the floor area. E.g. internal loads in Watt can be normalized into Watt per m² floor area (W/m²), which gives a much smaller range for the value. With this approach it is possible to make tables where normalized input values can be picked from, making the calculation process quick, and reducing the chance for calculation errors.

Values presented in the tables below are related to "Scandinavian" building standards, and may not be representative for other countries and other climates. It should however be easy to modify the tables to other standards and climates.

To exemplify how values in the tables have been determined, an office room is used as a case study throughout the section.

2.1 Normalized specific external loss

The specific loss to the external is comprised of window losses, facade losses and loss due to infiltration. In modern (Scandinavian) buildings the external walls are well insulated, and heat loss through windows are dominating. In less airtight buildings (older buildings) might infiltration have some impact on the external loss.

Given an office 3 x 4 m (12 m²) large, with a facade area of 9 m² (incl. window) and a window area of 2 m². U-value for the facade construction is 0.25 W/m²K (15 cm mineral wool) and the U-value for the window is 2 W/m²K. Infiltration is estimated to 0.3 ACH (Ceiling height is 3 m). Normalized specific external loss is then given by :

$$H''_{ext} = \frac{U_{fac}A_{fac} + U_{win}A_{win} + C_{air}nV}{A_{floor}} = \frac{(9-2)\cdot 0.25 + 2\cdot 2 + 0.34\cdot 0.3\cdot 36}{12} = 0.785\ W/m^2K$$

This figure is quite typical in office rooms, it normally lies between 0.4 W/m²K and 1.2 W/m²K. Table 1 presents typical values of normalized specific external loss, as a function of normalized windows- and facade loss, and infiltration (in ACH).

Table 1 : Specific external loss

Wind&Facad/ Infiltration	None	0.2W/m²K Low	0.5 W/m²K Medium	1.0 W/m²K High	3.0 W/m²K Very high
n = 0.1 ACH (Low)	0.1	0.3	0.6	1.1	3.1
n = 0.2 ACH	0.2	0.4	0.7	1.2	3.2
n = 0.3 ACH (Med)	0.3	0.5	0.8	1.3	3.3
n = 0.5 ACH	0.5	0.7	1.0	1.5	3.5
n = 0.7 ACH	0.7	0.9	1.2	1.7	3.7
n = 1.0 ACH (High)	1.0	1.2	1.5	3	4.0
n = 1.3 ACH	1.3	1.5	1.8	2.3	4.3

2.2 Normalized total specific loss

The total specific loss is the sum of the specific external loss and the ventilation loss. Air flow rate is often given in m³/h per m² floor area (normalized air flow), which is convenient here.

The room in subsection 2.1 is ventilated (balanced mechanical vent.) with 10 m³/hm² (120 m³/h). The total specific loss is the given by :

$$H''_{tot} = H''_{ext} + C_{air} L''_{vent} = 0.785 + 0.34 \cdot 10 = 4.185\, W / m^2 K$$

Table 2 gives normalized total specific loss as a function of ventilation rate and normalized specific external loss.

Table 2 : Normalized total specific loss

External loss/ Ventilation rate	None	0.5W/m²K Low	1.0 W/m²K Medium	2.0 W/m²K High	4.0 W/m²K Very high
0 m³/hm²	0	0.5	1.0	2.0	4.0
3 m³/hm² (Low)	1.0	1.5	2.0	3.0	5.0
5 m³/hm²	1.7	2.2	2.7	3.7	5.7
8 m³/hm² (Medium)	2.7	3.2	3.7	4.7	6.7
11 m³/hm²	3.7	4.2	4.7	5.7	7.7
15 m³/hm² (High)	5.0	5.5	6.0	7.0	9.0

2.3 Normalized specific heat capacity, timeconstant and time-lag

The effective heat capacity of the room can be treated in the same manner as the specific losses. The effective heat capacity of a building construction exposed to a 24 hours cycle temperature variation, can be limited to the inside 10 cm of the construction, or inside the insulating layer. If heavy material as concrete or brick is covered with insulating materials (i.e. carpet or lowered ceiling), the accumulating layer is reduced considerably. These "rules" gives specific (per m²) heat capacity of : ~ 50 Wh/m²K for a massive concrete wall, ~ 35 Wh/m²K for a massive brick wall, ~ 4 Wh/m²K for a insulated composite wall with gypsum board or wood panelling, ~15- 25 Wh/m²K for a concrete slab covered with carpet or lowered ceiling.

Given the room in section 2.1 with concrete floors covered with carpet, mineral wool lowered ceiling (beneath concrete construction) and brick walls in facade and partition walls. The normalized heat capacity of the room can be calculated to :

$$C_a'' = \frac{C_{air}V + \sum C_a''A}{A_{floor}} = \frac{0.34 \cdot 36 + 12 \cdot 20 + 12 \cdot 20 + (9-2) \cdot 35 + (3+4+4) \cdot 3 \cdot 35}{12} = 158\ W/m^2K$$

With the normalized heat capacity and normalized total specific loss, the timeconstant and time-lag can be readily calculated :

$$\tau = \frac{C_a''}{H_{tot}''} \qquad \tau_{lag} = \frac{\arctan[\tau\omega]}{\omega}$$

Table 3 gives timeconstant and time-lag values as a function of normalized heat capacity and total specific loss.

Table 3 :Timeconstant/time-lag

Total specific loss/ Normalized heat capacity	1.0 W/m²K Low	3.0 W/m²K	5.0 W/m²K Medium	7.0 W/m²K	9.0 W/m²K High
20 Wh/m²K (Very light)	20/5.3	7/4.1	4/3.1	3/2.5	2/1.8
40 Wh/m²K (Light)	40/5.6	13/4.9	8/4.3	6/3.8	4/3.1
80 Wh/m²K (medium)	80/5.8	27/5.5	16/5.1	11/4.7	9/4.5
140 Wh/m²K (Heavy)	140/5.9	47/5.7	28/5.5	20/5.3	16/5.1
260 Wh/m²K (Very heavy)	260/5.9	87	52/5.7	37/5.6	29/5.5

Example : Total specific loss : 3.0 W/m²K and specific heat capacity : 80, gives a timeconstant of 27 hours and a time-lag equal to 5 hours and 30 minutes.

2.4 Normalized internal load and solar gain

Heat gain from persons, light and appliances is often normalized with the floor area. In addition to the maximum instantaneous heat gain (to determine the amplitude heat gain), we have to estimate the diurnal mean heat gain. If balanced mechanical ventilation is used, we also have to estimate the heat gain from the supply fans.

The room in subsection 2.1 is occupied by one person (gain : 100 W) 8 hours a day. Lighting (120 W) and a computer (50 W) gives a mean heat gain of 170 W, and both are operated 8 hours a day. The supply fan rise the supply air flow 1 Kelvin (the fans are operated 24 hours a day). The normalized heat gain amplitude related to the internal load is then given by :

$$\hat{q}_{int} = \frac{q_{per} + q_{l\&a}}{2 \cdot A_{floor}} = \frac{100 + 170}{2 \cdot 12} = 11.25\ W/m^2$$

The normalized mean heat gain related to the internal load becomes :

$$\bar{q}_{int} = \frac{q_{per}n_{per} + q_{l\&a}n_{l\&a}}{24 \cdot A_{floor}} + C_{air}L_{vent}''\Delta T_{fan} = \frac{100 \cdot 8 + 170 \cdot 8}{24 \cdot 12} + 0.34 \cdot 10 \cdot 1 = 10.9\ W/m^2$$

Table 4 gives normalized amplitude heat gain and mean heat gain related to internal loads. It is given as a function of persons per 10 m² floor area (a normal office room) and the normalized gain from lighting and appliances. Heat gain from supply fan is included in the figures (air flow 10 m³/hm² and temperature rise 1 K). Operation of lighting and appliances is assumed to be 10 hours, and effective occupation time is set to 6 hours.

Table 4 : Normalized heat gain amplitude/mean heat gain

Lighting&Applianc./ Person density	5 W/m² Low	7 W/m²	10 W/m² Normal	15 W/m²	25 W/m² High
0.5 pers/10 m² (low)	5/7	6/8	8/9	10/11	15/15
1 pers/10 m² (office)	8/8	9/9	10/10	13/12	18/16
1.5 pers/10 m²	10/9	11/10	13/11	15/13	20/18
2 pers/10 m²	13/10	14/11	15/13	18/15	23/19
3 pers/10 m² (meet.room)	18/13	19/14	20/15	23/17	28/21
5 pers/10 m² (high)	28/18	29/19	30/20	33/22	38/26

Example : Normal lighting and appliances (10 W/m²) and 2 persons per 10 m², gives a amplitude heat gain of 15 W/m² and a mean heat gain of 13 W/m²

2.5 Specific solar gain

According to ,\1\, the maximum solar intensity through a vertical pane on a clear day can be approximated to 700 W/m². This figure can be used for facades facing East through South to West. Daily sum for the same facades can be approximated to 4700 Wh/m².

Solar shading in form of venetian blinds, curtains and building extensions can reduce the solar gain considerably. The shading effect for these shading devices is often taken as a constant shading factor. Typical values are 0.1 - 0.25 for external venetian blinds, 0.3 - 0.7 for inside venetian blinds and 0.5 - 0.8 for curtains. These values are for two pane windows with no coating, and has to be adjusted if low emessivity coating, reflective coating, absorbing glass or more panes are used. Shading from building extensions and nearby vegetation or buildings has to be estimated from case to case.

In the room from subsection 2.1, the facade is facing south, and there is inside venetian blinds with a shading factor of 0.5. The normalized amplitude gain can be estimated to :

$$\hat{q}_{sol} = \frac{q''_{sol} A_{win} F_{sh}}{2 \cdot A_{floor}} = \frac{700 \cdot 2 \cdot 0.5}{2 \cdot 12} = 29\ W / m^2$$

The normalized mean solar gain can be estimated to :

$$\bar{q}_{sol} = \frac{Q''_{sol} A_{win} F_{sh}}{24 \cdot A_{floor}} = \frac{4700 \cdot 2 \cdot 0.5}{24 \cdot 12} = 16\ W / m^2$$

Table 5 gives normalized solar gain (amplitude and mean) as a function of window area per m² floor area and total shading factor. Values are valid for facades facing east to west.

Table 5 : Normalized solar gain amplitude/mean heat gain (W/m^2)

Window area pr m^2 / Total shading factor	$0.05\ m^2/m^2$ Low	$0.10\ m^2/m^2$	$0.2\ m^2/m^2$ Normal	$0.3\ m^2/m^2$	$0.5\ m^2/m^2$ High
0.85	15/8	30/16	60/33	89/48	149/80
0.75	13/7	26/14	52/29	79/42	131/70
0.55	10/5	19/10	39/22	58/30	96/50
0.40	7/4	14/8	28/16	42/24	70/40
0.25	4/2	9/5	18/10	26/15	44/25
0.10	2/1	4/2	7/4	11/6	18/10

Example : With normal window area ($0.2\ m^2/m^2$) and a total shading factor of 0.25, gives amplitude solar gain of 18 W/m^2 and a mean heat gain of 10 W/m^2

Total heat gain (daily mean and amplitude) is the sum of the internal gain and solar gain.

2.6 Climatic data

In addition to the maximum solar intensity and daily solar gain treated in the previous subsection, we need to estimate the mean external temperature and its daily variation (amplitude). We also have to estimate the time for maximum heat gain (and external temperature).

The mean external temperature is normally found in meteorological journals. E.g. the highest five day mean temperature for the location in question could be used. This has to be evaluated against the use of the room from case to case.

The external temperature amplitude in Scandinavian climate varies between 5 - 7°C. If accurate information is not available a value of 6 °C can be used. If maximum external temperature isn't corresponding with the maximum heat gain, the temperature amplitude can be reduced a bit (0.5 - 2 °C).

If solar gain is dominating (compared to internal gains), which is one of the main assumptions in the model, the time for maximum heat gain is determined by the facade/window orientation. With daylight saving time (in Oslo) maximum solar gain is occurring : between 12.00 and 13.00 for south facades, between 9.00 and 10.00 for east facades, and between 17.00 and 18.00 for west facades. For other countries adjustment for time zone, longitude and daylight saving time has to be done.

3 Case study; comparison

A room which has been used in validation analysis of a computer program (TeknoSim), will be used as case study here, and compared to results from the widely used simulation program BRIS ,\2\. The room has width , depth and height equal to : 3.6 m x 4.2 m x 2.7 m ($A_{floor} = 15.12\ m^2$, $V = 40.82\ m^3$) . The room has one facade, facing west, with one window ($A_{win} = 1.92\ m^2$, $U_{win} = 2.0\ W/m^2K$). Infiltration is 0.2 ACH, and the room is ventilated continuously with 72 m^3/h (4.8 m^3/hm^2). The room is occupied with one person (9 hours a day), and heat gain from lighting and computer is 270 W (9 hours a day). The supply fan rises the supply temperature 1 °C.

Two different building constructions has been simulated : one heavy room with concrete floor, ceiling and external wall; and one light room with insulated composite construction covered with gypsum boards or particle boards. Partition walls are insulated composite walls with gypsum board in both cases.

Calculation

Normalized total specific loss is calculated to : $H''_{tot} = 2.18$ W/m²K (both cases). Normalized heat capacity for the two cases are calculated to : $C''_{a,h} = 31.2$ Wh/m²K (light room) and $C''_{a,h} = 140.6$ Wh/m²K (heavy room). Timeconstant and time-lag for the light- and heavy room then become : $\tau_{light} = 14.3$ hours, $\tau_{lag,light} = 5$ hours (light) and $\tau_{heavy} = 64.6$ hours, $\tau_{lag,heavy} = 5.8$ hours (heavy). Normalized mean heat gain and heat gain amplitude is respectively : $\overline{q} = 20.4$ W/m² and $\hat{q} = 31.0$ W/m². Mean external temperature and temperature amplitude are respectively : $\overline{T}_e = 22$ °C and $\hat{T}_e = 6.5$ °C . This gives a stationary temperature of : $T_\infty = 31.7$ °C (both cases), and a temperature amplitude for the light and heavy room of respectively : $\hat{T}_e = 1.2$ °C (light) and $\hat{T}_e = 5.3$ °C (heavy). Transient temperature differences are calculated to : $\Delta T_{light} = -10.8$ °C (light) and $\Delta T_{heavy} = -9.9$ °C (heavy). Both cases are simulated for a period of five days.

Simulated operative temperature the fifth simulation day in BRIS is shown in figure 1 (light room) and figure 2 (heavy room), and is compared to calculation with ThermSim (fifth day). The operative temperature in BRIS is used for the comparison, since the calculated temperature in ThermSim is a "merged" room-, surface- and structure temperature.

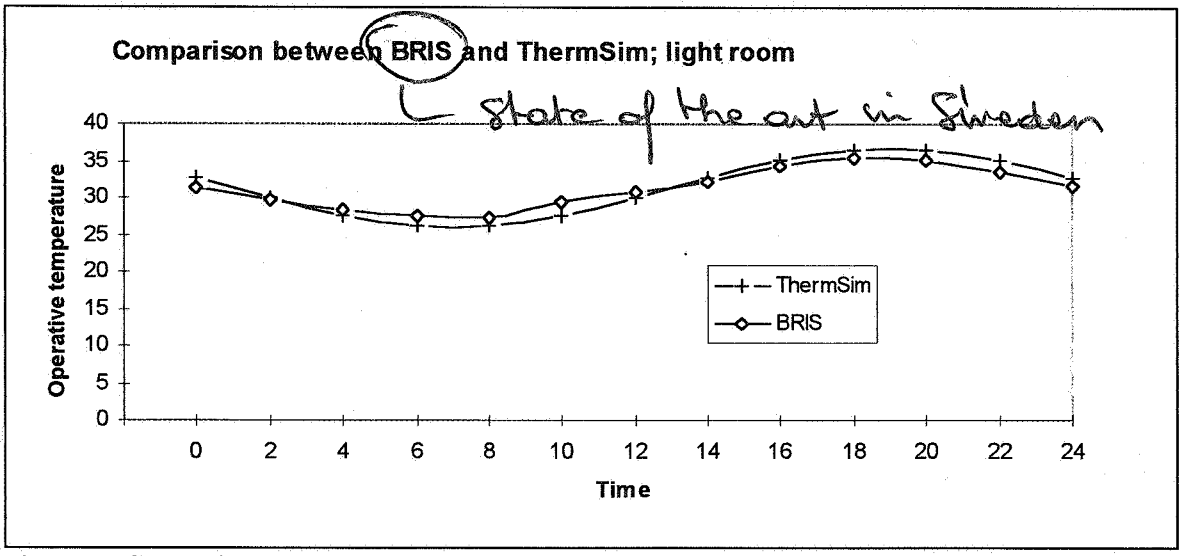

Figure 1 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, light room

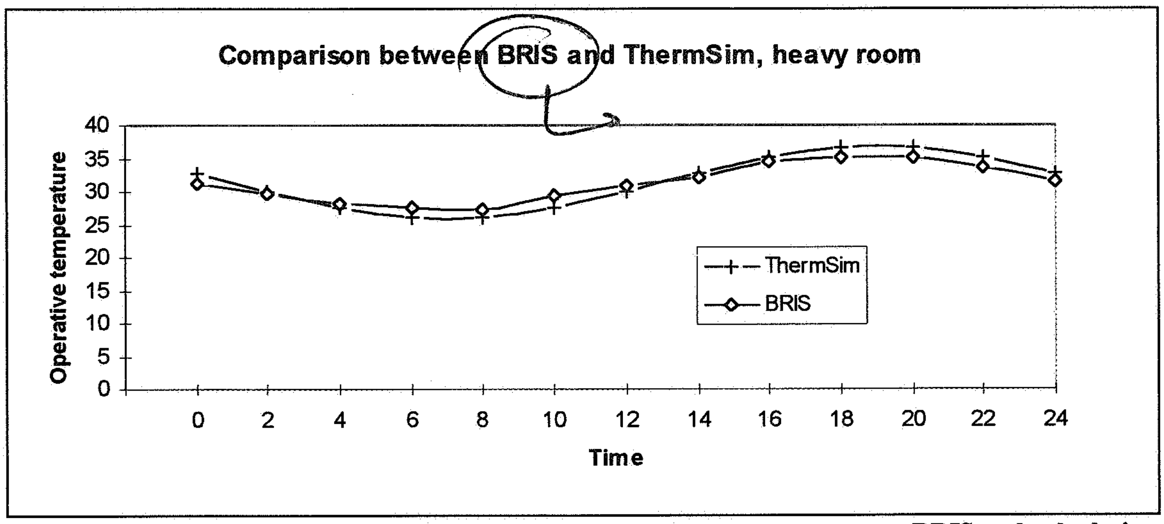

Figure 2 : Comparison between simulation in the advanced computer program BRIS and calculation with ThermSim, heavy room

5 Discussion and conclusions

- Temperature variation simulated with BRIS and calculated with ThermSim is similar, for both the light and heavy room
- Maximum temperature is somewhat higher calculated with ThermSim compared to BRIS (1.2 °C for light room and 1.7°C for heavy room).
- Maximum temperature occur a bit later in ThermSim than in BRIS ((1-2 hours in both rooms). This implicate that the calculated time-lag in ThermSim overestimate the "real" time-lag.
- Diurnal stationary conditions is reached after 5 days in the light room, but far from reached the heavy room (both with BRIS and ThermSim).
- Comparison between ThermSim and BRIS shows good agreement, and ThermSim should therefore be well suited for thermal analysis in a predesign phase of a project
- The simulations and calculations shows that large heat capacity reduce the daily temperature variation to a large extent, and prevent stationary condition being reached during a normal heat wave or a normal working week
- ThermSim is very well fitted for sensitivity analysis, because it only deals with the most important parameters affecting the thermal conditions in the room

References

\1\ Børresen B.A., "Room temperature variation and cooling loads. A simplified calculation method, Tempo", Proceed. : Energy Conservation in the built Environment. CIB, Copenhagen 1979

\2\ Nilsson P.E., Sunström T. " A comparison between the simulation programs TeknoSim and BRIS" (in Swedish), CIT Energiteknisk Analys, Gøteborg 1995