DEFINING THE METHODOLOGY FOR THE NEXT-GENERATION HOT2000 SIMULATOR

Task 1

presented by

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STATEMENT OF WORK

Introduction

The HOT2000 program for residential energy analysis grew out of the National Research Council's HOT-CAN program. HOT-CAN used a monthly heat-balance model whereas the current version of HOT2000 utilizes the bin method. The main motivators for a bin-based model—as opposed to a true simulation model—were CPU speed, CPU cost, and disk-storage requirements (for climate data). However, with modern desktop computers, CPU speeds and disk-storage have—and will continue to—increase at astronomical rates, while costs have plummeted. A bin model can no longer be justified for CPU and disk-storage reasons. Additionally, developing new models for bin programs is generally more difficult and expensive as these are often based on regressions of data generated from multiple parametric runs using more powerful modelling systems.

For these reasons, NRCan is examining the development of a simulation-based engine for HOT2000. This overall project is referred to as the "Next-Generation HOT2000 Simulator". The goal of the project described in this statement of work is to identify the most appropriate methods and existing computer code to use as the starting point for the development of the Next-Generation HOT2000 Simulator.

Context

Any program must be designed around its users' needs. Although this project does not address the user interface, it is critical that the models selected for the simulator not demand data that users cannot provide. As such, some background on HOT2000 users is provided in this section.

HOT2000 is used for modelling low-rise residential buildings: single-family houses, semi-detached houses, and rowhouses. It is primarily used to estimate annual energy consumption and the energy impact of design options. It is used for demonstrating conformance with the R-2000 standard and will soon be used for demonstrating compliance with the performance path of the National Energy Code for Houses.

The primary users of HOT2000 are architects, engineers, builders, auditors, and energy analysts and researchers. They typically have a practical knowledge of building science but limited familiarity with simulation methods. For these users, simplified data entry is a priority. A direct mapping between their knowledge of a building's physics and data inputs is essential. For example, users can be expected to give a detailed description of an envelope construction, including framing details. However, they cannot be expected to describe air-leakage paths, discharge coefficients of air-flow openings, or to select between free-convection surface-heat-transfer correlations.

Users can describe the building in thermodynamic zones as long as these correspond to obvious partitions, such as attic, basement, and main living space. Users cannot be expected to further divide the building into thermodynamic zones although some may for the purposes of modelling, say, baseboard heating with distributed thermostats. Notwithstanding, the program should be capable of considering the impact of such things as passive solar gains in an area not well connected to the rest of the house (eg. sunspace).

Although CPU speeds are a less dominant factor in designing simulation programs than they were in the 1970s and 1980s, run times will still be a factor for users. Some users will perform batch simulations to assess a number of energy conservation upgrades, for example. Therefore CPU requirements must be considered as a factor in this project along with accuracy and ease-of-implementation.

Approach

This project will entail a review, analysis, and synthesis of the literature. The contractor's responsibility is to perform this work—in as unbiased a manner as possible—to assist NRCan to select the most appropriate methods and existing computer code to use as the starting point for the development of the Next-Generation HOT2000 Simulator. The project will not involve any code development nor require the use of existing building simulation programs.

The work will be structured into three topics areas: loads, HVAC, and starting points. These three topic areas, and their specific tasks, are described below. The work performed within each task will be structured in a similar manner and the tasks will be inter-related. The overall structure of work to be performed on each task is as follows:

- The contractor will perform an extensive review of the literature and summarize the findings in a task report.
- NRCan will organize and host a workshop in Ottawa at which the contractor will provide an overview of the task report. The contractor will circulate the draft report, prior to the workshop, to the list of reviewers provided by NRCan. The purpose of the workshop will be to review and discuss the work, draw conclusions, and lay the groundwork for the next task. NRCan staff and others (NRCC, university, private sector) will participate in the workshops.
- The contractor will revise the draft report to incorporate the decisions made at the workshop and to capture and record the main points of the discussions.

INTRODUCTION

The present version of HOT2000 software is an useful tool for the evaluation of overall energy performance of low-rise residential buildings. The software was validated with field measurements and through the comparison with other energy analysis programs. It is presently used by builders, architects, engineers, auditors, researchers and students. However, some important issues for the design of new houses and the analysis of existing ones, such as passive solar effects, controls for HVAC systems, thermal sensation of occupants or indoor air quality, cannot be evaluated with the present version mostly due to the basic limitations of the modelling approach, which is used by the HOT2000 program.

TASK 1a - BUILDING-SIDE PROCESSES

The objective of this task consists in listing all relevant building-side processes required to be simulated in order to meet the user's needs.

Before starting any discussion concerning the most relevant processes, the proposed objectives should be presented:

a. The next generation of simulator will be developed for low-rise residential buildings (single-family detached and semi-detached house, duplex and triplex, and row house). The generic term of <u>house</u> is used in this report to designate a low-rise residential building.

b. All processes relevant to both cold and warm-climate conditions will be simulated to ensure the simulator is appropriate for use in Canada, the USA, Europe, Japan and other countries. The following items must be included:
   - the impact of climatic conditions on thermal behaviour of houses;
   - the construction practices and related codes;
   - the occupants' habits on the energy performance.

   Therefore, the following major energy end-uses must be adequately simulated:
   - heating;
   - cooling;
   - ventilation;
   - domestic hot water;
   - other appliances.

c. The simulator must be an useful tool for:
   - the design of energy-efficient, comfortable and environmentally-friendly houses;
   - the design of energy-efficient systems and sub-systems; for instance, it will allow a

user to design the control for a combination of domestic hot water/space-heating system, and then to evaluate its performance;
- the evaluation of energy performance and quality of the existing houses.

Researchers will probably use the simulator, as they did so far with HOT2000 program, to evaluate the impact of new materials, equipments or controls strategies on the energy performance and the quality of indoor environment (called herein as "building-science research-oriented tool"). However, the simulator will not be developed as a "building-simulation research-oriented tool", where different algorithms should be easily integrated and validated. Other research software should be used for this purpose.

d. This project aims to define the most appropriate mathematical models (engine) to be used by the simulator. The design of interface between the user and the engine (e.g., data input, file management, extraction of data from drawings, graphical presentation of results, optimization, databases) is not considered here.

e. The development of next simulator must make the maximum use of existing capabilities of HOT2000 program, in terms of algorithms, databases and weather files.

f. An optimum trade-off must be found between the extreme detailed description of a house and the simplified description. A high priority should be given to those parameters available through measurements (e.g., blower door test, efficiency of furnaces). It is unlikely that professionals such as architects, engineers, builders, energy auditors or energy analysts will be able to describe in detail parameters such as air-leakage paths or to evaluate the discharge coefficients of cracks in the building envelope.

Presently the HOT2000 simulator evaluates for the whole house, using the bin method:

a. The design heating and cooling loads (kW).
b. The annual and monthly heating and cooling loads (kWh).
c. The monthly energy consumption (kWh/month) and cost ($/month) for satisfying all the inhabitants' needs.
d. The annual energy consumption (kWh/year) and cost ($/year).

The present version of HOT2000 software is an useful tool for the evaluation of overall energy performance of low-rise residential buildings. The software was validated with field measurements and through the comparison with other energy analysis programs. However, the user's expectations and needs have increased over the past years, mainly due to the following:

a. A highly competitive market, which brings: (i) new approaches in the design and construction process, and accordingly new materials and technologies, and (ii) the need for a more complete and accurate evaluation of their impact on the energy performance of houses.

b. Some of the assumptions and the simplifications used 10-15 years ago in simulating the building thermal processes should be revised today, because the fast development in computer hardware over the past few years, and also the expected increase in capabilities in the near future, along with a net reduction of purchasing cost. Moreover, a large knowledge base was developed through laboratory research, on-site measurements, theoretical developments, analysis of the design and construction failures, evaluation of case studies, and demonstration projects.

c. A large number of research simulators are available today, with enhanced capabilities for evaluation of thermal processes. However, some of them are not appropriate for the daily activity of professionals involved in the design process.

Therefore, the time is appropriate for an open discussion about the present needs of those professionals involved with the construction industry, the present capabilities of HOT2000 simulator, the possible developments to be undertaken in the near future, and the expected needs within the next 15 years. Since it is difficult to accurately forecast the potential needs and developments over the next 15 years in order to design the simulator, it is proposed to develop a system with modular structure, which will facilitate the integration of future developments.

An initial list of processes required to be considered in order for the program to satisfy the professionals' needs is presented in Table 1.

Although <u>Task 1a</u> concerns only the building-related processes, the discussion should consider also the need for simulating systems such as radiant floor heating or the decentralized control of HVAC systems. The need for simulating this type of systems could have an important impact on the selection of mathematical models for heating and cooling loads. A preliminary list of HVAC-side processes (systems, equipment, control), which are expected to influence the selection of mathematical models for simulating the building-side processes, is presented in Table 2. The complete list of HVAC-side processes will be presented in Task 2a.

One of the goals of the workshop held in Ottawa on July 18, 1997 was to recommend the possible developments to be undertaken in the near future, and also the expected needs within the next 15 years. Therefore, the last two columns of Table 1 were completed at the end of the workshop, based on the participants' consensus.

Table 1. Relevant building-side processes

	Process	Available in H2K ver.8.x	Must be supported in next-generation simulator	Will likely be necessary within next 15 years
1.	Steady state heat transfer through opaque envelope components, including thermal bridges	x		
2.	Transient heat transfer through opaque envelope components, including thermal bridges	?	x	
3.	Steady state heat transfer through windows and other transparent envelope components (except transmission of solar radiation), including effects of frames and spacers; night insulation (e.g., draperies, venetian blinds)	x	x[3]	
4.	Transient heat transfer through windows and other transparent envelope components (except transmission of solar radiation), including effects of frames and spacers; night insulation (e.g., draperies, venetian blinds)			
5.	Steady state heat transfer through below-grade building components			
6.	Transient heat transfer through below-grade building components	x	x	
7.	Steady state moisture transfer through envelope components			
8.	Transient moisture transfer through envelope components			x
9.	Steady state moisture transfer through below-grade building components (e.g., basement floors and walls)			

10.	Transient moisture transfer through below-grade building components (e.g., basement floors and walls)			x
11.	Air infiltration/exfiltration through unintentional openings; sensible and latent loads	x	x	
12.	Natural ventilation; sensible and latent loads	x	x	
13.	Wind pressure on each component of the exterior envelope		x	
14.	Steady state heat transfer through interior walls, floors and ceilings[1]	x		
15.	Transient heat transfer through interior walls, floors and ceilings		x	
16.	Thermal storage effects of building mass and furniture (including sizing of heating system for intermittent operation)	x	x	
17.	Dynamic material properties (e.g., temperature-dependent thermophysical properties)			x
18.	Time-dependent heat gains from internal sources (occupants, lights, appliances); radiative and convective portions; location within thermal zone		x	
19.	Time-dependent moisture generation by internal sources, and moisture absorption/desorption processes		x	
20.	Time-dependent contaminant generation by internal sources, and contaminant absorption/desorption processes		x	
21.	Condensation on interior surfaces of the envelope components (e.g., wall, roof, window, ceiling, floor)		x	
22.	Condensation within the envelope components (e.g., wall, roof, window, ceiling, floor)			x

23.	Convection heat transfer between the indoor air and the interior surfaces (e.g., walls, windows, floors, lighting fixtures)	?	x	
24.	Convection heat transfer between the external surfaces (e.g., walls, windows, roof, doors) and the outdoor air	?	x	
25.	Longwave radiation exchange between the interior surfaces (e.g., walls, windows, floors, heaters, lighting fixtures)		x	
26.	Longwave radiation exchange between the external surfaces (e.g., walls, windows, roof, doors) and the sky vault	?	?	?
27.	Longwave radiation exchange between the external surfaces and the surroundings	?	?	?
28.	Shortwave radiation on the external surfaces		x	
29.	Transmission of solar radiation through windows and other transparent envelope components	x[2]	x	
30.	Shading of opaque envelope components caused by surroundings (e.g., houses, buildings, trees)		x[4]	
31.	Shading of transparent envelope components caused by exterior shading elements (e.g., overhang, side fins) or by facade obstructions (e.g., recessed windows)	x	x[4]	
32.	Shading of transparent envelope components caused by surroundings (e.g., houses, buildings, trees)		x[4]	
33.	Room air movement and thermal stratification		x[5]	
34.	Zone-to-zone air flow		x	
35.	Convective heat exchange between thermal zones		x	

36.	Coupled interzone air flow and heat and moisture transfer		x	
37.	Coupled interzone air flow and heat, moisture and other contaminants transfer		x	
38.	Variable indoor air temperature (e.g, for evaluating the overheating/underheating in each thermal zone)		x	
39.	Passive solar design - solar windows/direct gain system		x	
40.	Passive solar design - sunspaces		x	
41.	Passive solar design - thermal storage walls/indirect gain systems		x	
42.	Quality of indoor environment - thermal comfort (based on air temperature, relative humidity, mean radiant temperature of interior surfaces, air velocity, occupant's activity and clothing type)		x[6]	
43.	Quality of indoor environment - indoor air quality		x	
44.	Quality of indoor environment - visual comfort (including daylighting)		x	
45.	Design heating and cooling loads for each thermal zone		x	
46.	Heating and cooling loads provided in each thermal zone by the existing HVAC equipment (e.g., baseboard heater, forced air system)		x	
47.	Coupling thermal loads and HVAC system simulation		x	
48.	Location of heat emitters (e.g., baseboard heaters, supply grilles)		x[4]	
49.	Insolation patterns on the internal surfaces (e.g., walls, floors, ceiling)		?[7]	

50.	Heat balance method for space thermal loads		x	
51.	Weighting factor method for space thermal loads			

Notes

1 - HOT2000 can only analyze the heat transfer through one floor (between the main floor and the basement) and through one ceiling (between the main floor and the attic). These are specific cases and not general heat transfer processes between any zones.
2 - Vertical or tilted at any angle.
3 - Multi-layer approach.
4 - Simplified approach.
5 - Only room air movement, with more than one node if it is required. No thermal stratification.
6 - Based on the room air temperature, the mean radiant temperature, and the relative humidity.
7 - After a sensitivity analysis will be performed to evaluate the impact.

Table 2. Preliminary list of HVAC-side processes, expected to influence the selection of mathematical models for simulating the building-side processes

	Process	Available in H2K ver.8.x	Must be supported in next-generation simulator	Will likely be necessary within next 15 years
1.	Radiant floor or ceiling heating/cooling system	x		
2.	Decentralized control of HVAC systems			

TASKS 1b and 1c - TECHNIQUES FOR SIMULATING THE BUILDING-SIDE PROCESSES

The objective of Task 1b is to examine and summarize all available techniques for simulating the relevant building-side processes selected in Task 1a, which are used by existing building simulation programs or are presented in technical publications. The critical comparison of all available techniques is not included in the present project.

For each building-side process, the following items will be presented:

- code of process (e.g., 1-BS), in concordance with Table 1;
- short presentation of available techniques and corresponding references.

The suitability of modelling techniques to simulate the selected relevant building-side processes is assessed in Task 1c. The criteria used in this evaluation are the following:

1- capability to model the relevant building-side processes;
2- accuracy;
3- ease of implementation;
4- ability to integrate HVAC calculation methods;
5- CPU requirements;
6- flexibility for future improvements; and
7- interoperability with other tools.

The absolute qualities of each modelling technique cannot be assessed based only on the literature review. There is no standardized approach for this purpose. Therefore, the suitability of each modelling technique for a particular building-side process is evaluated in this report by comparison, based on published information and previous experience.

The relevant information of both tasks 1b and 1c are combined and presented in this section.

1-BS

Steady-state heat transfer through opaque envelope components, including thermal bridges

Heat transfer through the opaque envelope components is transient in nature. However, to simplify calculations, the heat transfer phenomena are assumed to be independent of time, and then steady-state approaches can be used. The evaluation methods cover the entire possible range, from the over-simplified methods (one-dimensional without corrections for thermal bridging) to the detailed numerical methods, which take into account the three-dimensional effects.

Overall thermal resistance of building envelope sections can be obtained through laboratory or field measurements using one of the following test methods: Guarded Hot Plate, Heat Flow Meter, Guarded Hot Box, and Calibrated Hot Box [ASHRAE 90.1-1989].

<u>Evaluation of overall thermal resistance of an envelope component</u>
When these measurements are not available, the calculation of thermal resistance is performed at steady-state conditions, that is, the effects of heat storage are not considered [ASHRAE 1993]. The temperature of surrounding surfaces equals the ambient air temperature and the film coefficient is constant. The overall thermal resistance of wood frame walls can be calculated by assuming either parallel heat flow paths or isothermal planes. Usually, the parallel heat flow paths method overestimate the thermal resistance, because it does not consider the lateral heat flows, and the isothermal plans method underestimate the thermal resistance, because gives too much emphasis on thermal bridges [Desmarais et al. 1997]. Therefore, an average of results from these two methods gives a good estimate of thermal resistance of an assembly. For structures with widely spaced metal members of substantial cross-sectional area, the zone method can be used [ASHRAE 1993]. This method involves two separate computations: one for a chosen limited portion containing the highly conductive element, and another for the remaining portion of construction. The two results are then combined using the parallel flow method, to estimate the overall transmittance and thermal resistance. Compression of insulation shall be considered in determining the thermal resistance.

ASHRAE Standard 90.1-1989 demands the overall thermal resistance of an opaque envelope assembly to be determined using: (i) a series path procedure with correction for the presence of parallel paths within an element (e.g., the parallel path correction factor for wall cavities with metal studs is given in terms of size and thickness of studs, spacing of frame, and cavity insulation), (ii) isothermal planes procedure, and (iii) zone method.

> <u>*Capability to model the processes listed in Task 1a*</u>
> *This model is limited to the conduction heat transfer process through opaque elements. The convection*

within the wall is neglected. It can be used to calculate the average surface temperature of the wall. It cannot be used to evaluate the local heat flow and the local surface temperature.

Accuracy
Acceptable for the practical applications where the overall U-value is required. It is influenced by the accuracy of pre-calculated corrections factors.

Ease of implementation
Models are simple, available and can quickly be implemented.

Ability to integrate HVAC calculation methods
It can be coupled with most air-type HVAC system models, where the surface temperature is not directly affected by the system operation. It can be used when the overall heat flow through the wall is an acceptable parameter for calculations.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air and moisture transfer through walls, then this model has limited flexibility. If the future improvement represents the development of better correction factors for a large number of thermal bridges, then this model is flexible.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other simulation tools. However, the accuracy of the whole model depends on both components. Hence, if it is coupled with a more detailed model, the accuracy of the whole model is likely to be reduced.

Numerical methods
More detailed calculations of overall thermal resistance can be performed using numerical methods. A grid is defined using one-, two- or three-dimensional, cartesian, cylindrical or spherical coordinate system. The nodal points are defined in such a way to approximate the continuous temperature distribution in the body. An energy balance is written at each node using finite differences methods or finite element methods. A system of linear equations is obtained, where the unknowns are the nodal temperatures. The parameters are deduced from the known geometry, the thermal properties of materials, and the boundary conditions. The boundary conditions can be formulated as (i) known surface temperature, (ii) know heat flow, or (iii) known ambient temperature and surface conductance. Numerical techniques are used to solve the system of linear equations. Examples of such computer programs are KOBRUS86, TRISCO, BISCO, CYLI86, and KOBRA [Standaert, P. 1985; PHYSIBEL 1996], and Heating 7.2 [Kosny et al. 1995a]. The last computer program was used to simulate more than 1,000

metal frame walls, in order to analyze the accuracy of the zone method recommended by ASHRAE for calculations of thermal resistance. Among the main conclusions are the following:

- on average, the ASHRAE zone method gives estimates between -4% and +14% for walls with cavity insulation, and between +4% and +14% for metal stud walls with empty cavities;
- the main reason for inaccuracy of the ASHRAE zone method is the assumption of uniform zone factor for all wall geometries and material configurations; other parameters with an important impact are: thickness of sheathing insulation, stud depth, stud flange size, and stud thickness.

The same program was used to evaluate the steady-state thermal resistance of three masonry wall systems (two-core, cut-web, and multicore units), for different parameters such as clear wall area, corner or wall-ceiling intersection, window header, window sill, window edge, or door header [Kosny et al. 1995b].

Capability to model the processes listed in Task 1a
Depending on the needs, it can model the one-, two- or three-dimensional heat transfer process. It can calculate the local surface temperature, as well as the temperature at different locations within the wall. It can accommodate all the heat and mass transfer processes occurring within the opaque elements (e.g., vertical air movement, condensation), and the temperature-dependent thermophysical properties of building materials.

Accuracy
Relatively high accuracy can be obtained depending of the discretization approach and the methods used for solving the system of simultaneous equations.

Ease of implementation
Numerical techniques for basic heat transfer problems are available in all textbooks. Models are also available, and the source codes could eventually be obtained. The implementation is more a matter of programming time.

Ability to integrate HVAC calculation methods
The coupling of this model with HVAC calculations can easily be done. The problem consists in solving, resulting in a large system of simultaneous equations.

CPU requirements
Although the calculations take more time than the previous model, the model is relatively fast on the today's computers. The CPU requirements increase with the number of nodes used for the discretization of continuous field.

Flexibility for future improvements
It is the most flexible model, since it is based on the first principles approach, and it does not require the correction factors or other functions to be separately calculated. Therefore, any future improvement can be integrated.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can easily be coupled with other tools, without reducing the accuracy of the whole model. It can also be used as a reference model to calculate the

correction factors required by simpler models.

One-dimensional heat transfer model with correction factors for thermal bridges

French standard "Règles Th-G" recommends values for the linear heat loss coefficient, to take into account the thermal bridges in the exterior envelope. A similar concept is proposed by Mahdavi [1993]. The thermal coupling coefficient or "L-value" and the temperature weighting factor "g-factor" are proposed for more precise determination of (i) surface temperatures of complex building elements, (ii) risk of surface condensation, and (iii) heat loss calculations. This concept is capable of properly describing various two- and three-dimensional steady-state heat transfer processes.

$q_l = L_l \cdot \Delta T$ [W/m] for length-related heat flow

$q_a = L_a \cdot \Delta T$ [W/m^2] for area-related heat flow

Therefore, for n building elements in parallel, the overall L-value is equal to ΣL_i.
The temperature-weighting factors "g" provide a means to compute the temperature at any point of the surface as a linear function of any set of boundary temperatures (t_0,,t_n):

$T_{si} = T_0 \cdot g_0 + \ldots.. + T_n \cdot g_n$, where $g_0 + g_1 + \ldots g_n = 1$

For instance, for $T_0 = T_e$ and $T_1 = T_2 = \ldots T_n = T_i$, the surface temperature is given by:

$$T_{si} = g_0 \cdot T_e + (1-g_0) \cdot T_i$$

L-values and g-factors can be determined for different construction configurations, using appropriate numerical simulation methods, and stored in a database. Then, they can be used for any thermal boundary conditions.

A two-dimensional, finite difference computer program was used to predict the steady-state impact of thermal bridges in a typical office building, including a built-up roof system with ceiling fasteners, a roof/wall interface, an insulated masonry cavity wall with metal studs, a floor slab that penetrates wall insulation, and a window/frame interface [Burch et al. 1992]. The heat flow rate per unit length "q" (W/m) for each thermal bridge was calculated by summing the heat flows along the interior surfaces. An excess heat transfer coefficient "H" (W/m·K) for each thermal bridge geometry was calculated using the relation:

$$H = \frac{q}{(T_i - T_o)} - U \cdot W$$

where W (m) is the width of thermal bridge, and U ($W/m^2 \cdot K$) is the thermal transmittance of undisturbed envelope assembly (without thermal bridge). When the excess heat transfer coefficient is multiplied by the temperature difference across the thermal bridge configuration, the result is the additional heat flow attributed to the presence of thermal bridge. Their study indicated the overall building heat transfer coefficient was increase by 24% due to the thermal bridges.

Capability to model the processes listed in Task 1a
It can be used to calculate the average and local surface temperature of the wall, and the corresponding heat flows.

Accuracy
Acceptable for the practical applications. It is influenced by the accuracy of pre-calculated correction factors.

Ease of implementation
Models are simple, and can quickly be implemented. A database of all correction factors presently available should be developed. Other correction factors, relevant to the construction practices, should also be calculated using two- or three-dimensional models, or should be measured, and then integrated within the model.

Ability to integrate HVAC calculation methods
It can be coupled with most air-type HVAC system models, where the surface temperature is not directly affected by the system operation. It can be used when the overall heat flow through the wall is an acceptable parameter for calculations.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air and moisture transfer through walls, then this model has limited flexibility. If the future improvement represents the development of better correction factors for a large number of thermal bridges, then this model is flexible.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other simulation tools. However, the accuracy of the whole model depends on both components. Hence, if it is coupled with a more detailed model, the accuracy of the whole model is likely to be reduced.

2-BS

Transient heat transfer through opaque envelope components, including

thermal bridges

Heat transfer through the opaque envelope components is transient in nature. Several methods have been developed for modelling this complex phenomenon, to satisfy the different needs and resources of users such as designers or researchers. These methods can be classified into the following categories: analytical methods, numerical methods, time-domain response factor methods, frequency-domain response factor methods, cooling load temperature difference methods, and modified bin method.

Analytical methods

A general closed-form analytical solution of transient heat transfer through an exterior building component is not available because of complex boundary conditions and thermal phenomena within the element. From an academic point of view, this approach "is probably the most intellectually satisfying approach, but unfortunately often requires an unreasonable degree of simplification to make the equations tractable" [Day 1982]. Solutions for homogeneous walls or simple boundary conditions for non-homogeneous walls are available in heat transfer textbook [Pratt 1981, Ozisik 1994, Poulikakos 1994].

Capability to model the processes listed in Task 1a
There is no analytical model for solving the complex boundary conditions and thermal phenomena occurring within the building elements. The coupling of heat and mass transfer phenomena is even more complex. The analytical models for simple heat transfer phenomena can be used to validate some other models under simple boundary conditions.

Accuracy
High accuracy for simple heat transfer problems. In the case of complex problems, the accuracy is likely to be lower due to simplifications and assumptions used.

Ease of implementation
Difficulties arise from finding some appropriate analytical models, than to implement them into the source code.

Ability to integrate HVAC calculation methods
It depends on the flexibility to accept boundary conditions imposed by the HVAC calculations (e.g., variation of the solar radiation on the interior surfaces or of the indoor air temperature under different control strategies).

CPU requirements
Generally, it is faster than numerical models.

Flexibility for future improvements

The use of simplifying assumptions limit the application to other problems. Therefore, other models should be developed.

Interoperability with other tools
It depends on the flexibility to accept the boundary conditions imposed by other tools.

Numerical methods
Initial detailed developments have used the numerical solution of one-dimensional heat transfer differential equation, based on finite difference techniques. The continuous domain of building component, where the temperature field and the heat flows are sought, is transformed into a discrete one by properly selecting the grid shape and size. The term lumped-parameters or network of nodes is also used to refer this approach. For each node, a heat balance equation is derived from the governing differential heat transfer equation, by taking into account thermal properties of material around the node and the links with other nodes. If an explicit differencing scheme is used to develop the finite difference "computational molecule" [Ames 1969], the present temperatures at a particular time are expressed exclusively in terms of temperatures at the previous time step. If an implicit differencing scheme is used, the present temperatures are expressed either in terms of other present temperatures or in terms of average of previous and present temperatures. The system of equations which is obtained, is then solved using small time steps to cover the desired period of analysis. This approach has several advantages: (i) it can incorporate most, if not all, heat transfer phenomena of interest for the evaluation of energy performance of buildings (e.g., convection within the wall, phase change of building materials); (ii) it can incorporate linear and non-linear thermal phenomena (e.g., longwave radiation between walls); (iii) it can use constant or time-dependent thermal properties of building materials; (iv) it can use constant or time-dependent surface coefficients; (v) it can easily model two-dimensional and three-dimensional phenomena.

To adequately discretize the continuous domain, especially close to thermal singularities, a large number of nodes is needed, which then increases the computing time. Waters and Wright [1985] recommended the following strategy to distribute nodes within a multilayer wall: (i) place a node on each internal boundary, which will eliminate the truncation error terms in $\partial T^3/\partial x^3$, independently of the node spacing on either side of the boundary; and (ii) place additional nodes within layers in such a way as to approach, as nearly as possible, the condition given by the following relation:

$$n_1 \simeq n_2 \cdot \frac{\beta_{max}^3}{\beta_1}$$

where: n is the number of nodes in each layer, indicated by subscript 1 or 2,

$\beta = \alpha/l^2$ (α - thermal diffusivity, l - thickness of layer), and
$\beta_{max} = \max(\beta_1, \beta_2)$

The development of equations and the solving method should be designed in such a way to avoid the numerical instabilities. For instance, if an explicit method is selected, a small time-step is required, which must be less than an upper limit defined in terms of mesh size and thermal properties of material surrounding the node. Computing time increases linearly with the number of nodes. If an implicit method is selected, a complete set of equations must be solved simultaneously, which normally increases the computing time. However, a larger time-step can be selected without propagating numerical instabilities. Computing time increases as the square or cube of number of nodes. These considerations led to the opinion that this method is time-consuming. Therefore, it was assumed to be less suitable for design or evaluation tools used by practitioners. About 20 years ago, when the first version of most detailed computers programs was developed, this method was not the favoured one, since the computing time for solving the system of equation was too long, due to the limited hardware capabilities of that time.

Some examples of application of numerical methods in the energy analysis programs, for the analysis of heat transfer phenomena in building components, are the following:

1. McFarland et al. [1979] used the Crank-Nicholson implicit method to solve the one-dimensional heat transfer in walls for passive solar design applications. The linear equations are solved using standard linear algorithms. For the non-linear equations, related with natural convection or radiation, an iteration loop is used at each time step to reach the convergence. Number of nodes depends on the building component type. For instance, a central node is used to simulate a water wall, while several nodes are used for a masonry wall. Thermophysical properties do not vary with time or temperature. The simultaneous heat and mass (air, moisture) transfer is not considered.
2. Emery et al [1981] used implicit methods for the one-dimensional heat transfer, with nodes located on boundary planes of each layer composing the exterior wall. Computations are performed only once an hour because weather information is rarely available at more frequent intervals and to limit the duration of calculations. The matrix coefficients were not re-evaluated unless the convective or radiative coefficients changed by more than 50%. Studies performed with time increments of two minutes showed less than 10% difference in heat fluxes and air temperatures for typical building sections. Thermophysical properties do not vary with time or temperature. The simultaneous heat and mass (e.g., air, moisture) transfer is not considered.
3. Clarke [1985a, 1985b, 1986] used the Crank-Nicholson implicit method for the one-dimensional heat transfer, with three nodes per homogeneous layer (one on the middle plane and one on each boundary plane). The simultaneous system of equations is solved through an inversion technique. Presently, the thermophysical properties of building

materials at time t are established on the basis of the latest values at time t-1 [Clarke and Hensen 1991]. The simultaneous heat and mass (e.g., air, moisture) transfer is not considered. Upcoming features will include the time-dependent and moisture-dependent properties [ESRU 1996].

4. French company DIALOGIC developed in 1986 a software called OASIS [Raoust 1992], to design the air-conditioning systems in different climates, and to evaluate the thermal comfort of occupants. The one-dimensional heat transfer through exterior walls is evaluated using an implicit method. Each layer is discretized using four nodes.
5. Zmeureanu et al. [1988a, 1988b] used the Crank-Nicholson implicit method to analyze the one- and two-dimensional heat transfer, with three nodes par layer (on the middle plane and on the boundary planes), and with a time-step of one hour. The Gauss-Seidel iterative method is used to solve the set of simultaneous equations. Thermophysical properties do not vary with time or temperature. The simultaneous heat and mass (e.g., air, moisture) transfer is not considered.
6. Wong [1990] used an explicit finite difference method to analyze the simultaneous one-dimensional heat and moisture transfer in building components. He used three nodes per layer (in the middle plane and on the boundary planes), and constant thermophysical properties.
6. Brau et al. [1992] and Brau [1995], used the nodal method to analyze one thermal zone, by grouping all exterior walls in one node, and all exterior windows in another node. Thermophysical properties do not vary with time or temperature. The simultaneous heat and mass (e.g., air, moisture) transfer is not considered.
7. Burch et al. [1992] used an implicit method for the two-dimensional heat transfer in an exterior wall with thermal bridges. The time-step was one hour. The simultaneous system of equations is solved by the Gauss-Seidel method. Thermophysical properties do not vary with time or temperature. The simultaneous heat and mass (e.g., air, moisture) transfer is not considered.
8. Nakhi [1995] integrated a multi-dimensional conduction modelling into ESP-r program, to simulate thermal bridges; he developed a technique to simultaneously generate an adaptive grid for building elements, and to reduce the computing time and errors.

Some examples of computer programs dedicated to the analysis of dynamic thermal behavior of two- and three-dimensional building components (these are not energy analysis programs) are the following:

1. SECTRA [Standaert, P. 1985; PHYSIBEL 1996], used to analyze the two-dimensional transient heat transfer in rectangular objects, based on the Crank-Nicholson implicit method. The boundary conditions include the time-dependent (i) temperatures, (ii) heat transfer coefficients, and (iii) surface heat flow densities. Time step is defined by the user.
2. VOLTRA [Standaert, P. 1985; PHYSIBEL 1996], used to analyze the three-dimensional transient heat transfer, based on the Crank-Nicholson implicit method. The boundary

conditions include the time-dependent (i) temperatures, (ii) heat transfer coefficients, and (iii) surface heat flow densities.

3. Heating 7.2 [Kosny et al. 1995a], used to analyze the transient heat transfer in one-, two- or three-dimensional cartesian, cylindrical or spherical coordinates. Materials with time- and temperature-dependent thermal conductivity, density, and specific heat can be considered. The boundary conditions which can be defined include (i) given temperatures or (ii) any combination of prescribed heat flux, forced convection, natural convection, and radiation. The boundary condition parameters can be time- or temperature-dependent.

The model reduction techniques can be used to simplify a given mathematical description in such a way that the output calculated by the reduced model does not differ very much from the original model [Rabenstein 1994]. For instance, an initial model of the one-dimensional heat transfer problem can be developed using finite difference techniques and 40 nodes, and then it is reduced to a model composed of only three nodes, which gave almost the same heat flow. The model reduction is based on balanced state space description. This method is applicable to linear building models and to linear sub-models of general nonlinear building descriptions. For a quantitative comparison, a number of cases were identified, where a lower order finite difference model and a reduced model gave approximately the same deviation, of 0.1% to 1%, from the reference model with 20 nodes. However, the required order of the finite difference model was in the range of 9-13, while the reduced order models achieved the same or higher accuracy with a degree of only 4-5. The run time on a workstation SUN SPARCstation ELC for the reduced models was in the range of 19% to 36% of the run time of the low order finite difference models. Some higher level languages such as MATLAB provide powerful commands for model reduction.

Lefebvre [1997] developed the m2^m software package to reduce the size of mathematical model, and to offer rapid and accurate simulations, based on modal representation. This method can be used only when the heat transfer equations are linear (temperature-independent coefficients) and stationary (time-independent coefficients). The objective of modal reduction is to take advantage of the modal dominance and to find a modal model which has the same inputs and outputs as the initial one, but with very few state variables. A general technique consists in ordering the eigenmodes following a particular rule such as (i) the decreasing time constants, (ii) the decreasing energy contribution to the step response, or (iii) the maximum absolute contribution of each mode to all possible input-output couples. The m2^m software package does operations such as:

- builds a finite difference model of the thermal behavior of a building,
- transforms the finite difference model into a modal one,
- reduces the modal model,
- performs the deterministic simulation,
- extract the required data from the modal model to visualize the response spectra, or
- performs the required calculations to visualize the eigenvalues.

For example, an initial finite difference model with 63 nodes was reduced to eight nodes, which led to a reduction of 18% to 62% of the computing time.

Salgon and Neveu [1987] used the modal techniques to analyze the impact of thermal bridges in buildings.

Capability to model the processes listed in Task 1a
Depending on the needs, it can model the one-, two- or three-dimensional heat transfer process. It can calculate the local surface temperature, as well as the temperature at different locations within the wall. It can accommodate all the heat and mass transfer processes occurring within the opaque elements (e.g., vertical air movement, condensation), and the temperature-dependent thermophysical properties of building materials. It can simulate the time-dependent thermal properties and phenomena.

Accuracy
High accuracy can be obtained depending of the discretization approach and the methods used for solving the system of simultaneous equations.

Ease of implementation
Numerical techniques for basic heat transfer problems are available in all textbooks. Models are also available, and could eventually be obtained. The implementation is more a matter of programming time.
Ability to integrate HVAC calculation methods
The coupling of this model with HVAC calculations can be done. The main problem consists in finding an efficient method for solving the large system of simultaneous equations, in order to reduce the CPU requirements. It can use smaller time-steps to accommodate the HVAC controls.

CPU requirements
The CPU requirements increase with the number of nodes used for the discretization of continuous field. The model is slower than other transient heat transfer models. The CPU requirements could be decreased by using the model reduction techniques.

Flexibility for future improvements
It is the most flexible model, since it is based on the first principles approach, and it does not require the correction factors or the response factors to be separately calculated. Therefore, most future improvements can be integrated.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools, without reducing the accuracy of the whole model. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.

Time-domain Response Factor methods

The response factor method is mostly based on research carried out by Mitalas and Stephenson [1967], starting from the following concepts: the response of a linear and invariable system to a unit time-series excitation function is called the unit response function [Stephenson and Mitalas 1967]; linearity implies that the magnitude of the response is linearly related to the magnitude of the excitation; invariability means that equal excitations applied at different times always produce equal responses; linearity and invariability are prerequisites of any superposition method. The use of response factor method is based on two approaches: conduction transfer function (CTF) and room transfer function (RTF) [McQuinston and Spitler 1992, McQuinston and Parker 1994].

The conduction transfer functions are infinite series that relate a current variable to past values of itself and of other parameters considered at discrete time intervals. For instance, the one-dimensional heat flow at the inside surface of an exterior wall at time t ($q_{i,t}$) is expressed in terms of outside surface temperature at times t, t-1, t-2... ($T_{out,t-m}$), inside surface temperature at times t, t-1,t-2... ($T_{i,t-m}$), and previous heat flow at the inside surface at times t-1,t-2... ($q_{i, t-k}$) [Sowell and Walton 1980, Walton 1983, Walton 1984, ASHRAE 1993, BLAST 1991]:

$$q_{i,t}=\Sigma Y_m \cdot T_{out,t-m} - \Sigma Z_m \cdot T_{i,t-m} + \Sigma J_k \cdot q_{i,t-k}$$

where Y_m, Z_m and J_k are the Conduction Transfer Function coefficients.

The above equation, written for all surrounding walls, must be coupled with:

- a heat balance equation at each inside surface, which incorporates (i) the convection heat transfer between the inside surface and the room air, and (ii) the radiation heat transfer between all inside surfaces, the solar radiation received by the surface, and the radiation received by the surface from internal sources (e.g, lights, people, equipment);
- the convection heat balance equation of the indoor air.

A system of simultaneous linear equations is obtained, and solved for each time step over the period of analysis. Since the temperature and heat flow history are not known at the beginning of calculations, an initial history is assumed, and then the calculations are performed for several identical days (usually 3-5 days) in order to eliminate the impact of initial assumptions.

In another largely used approach, the instantaneous one-dimensional heat flow at the inside surface of an exterior wall at time t ($q_{i,t}$) is expressed in terms of sol-air temperature at times t, t-1, t-2...($T_{sa, t-m,}$), heat flow at the inside surface at times t-1, t-2... ($q_{i,t-k}$), and a fixed room air temperature T_R [McQuinston and Spitler 1992, ASHRAE 1993, McQuinston and Parker 1994, DOE-2]:

$$q_{i,t}=\Sigma b_n \cdot T_{sa,t-m} - \Sigma d_n \cdot q_{i,t-k} - T_R \cdot \Sigma c_n$$

where b_n, c_n, d_n are the Conduction Transfer Function coefficients.

The space cooling load is calculated in terms of convection and radiation heat transfer: (i) the convective component of instantaneous heat flow contributes immediately to the cooling load; (ii) the contribution of radiative component of instantaneous heat flow depends of the thermal mass of space and the air circulation within the space. The radiative component of cooling load at time t (Q_t) is calculated in terms of radiative component of instantaneous heat gain at times t, t-1, t-2... ($q_{i,t-m}$), and space cooling load at times t-1, t-2... (Q_{t-k}):

$$Q_t = \Sigma \nu_n \cdot q_{t-m} - \Sigma w_k \cdot Q_{t-k}$$

where ν_n, w_k are the Room Transfer Function coefficients or weighting factors. These coefficients indicate how much of the energy entering a room in a given hour is stored and how fast the stored energy is released during later hours [Kerrisk et al. 1980]. Since the process must be modeled by linear equations in order to apply the superposition principle, the non-linear processes such as natural convection and radiation must be approximated linearly. The second assumption is that system properties are not time- or temperature-dependent, and therefore can be represented by average values over the time of interest. Therefore, this assumption can limit the use of weighting factors in situations where important zone properties vary during the analysis (e.g., variation of film coefficients due to change in heat flow or air velocity, or movement of sunlit area on the interior surfaces).

Finally, the heat extraction rate and the room air temperature are calculated in terms of space cooling load, capacity of cooling system, and throttling range of the control system, by using the Space Air Transfer Function coefficients.

ASHRAE developed a software called TFMTAB to access a database of conduction transfer function coefficients for 41 wall types and 42 roof types [McQuinston and Spitler 1992, Falconer et al. 1993]. The weighting factors are also calculated for several combinations, taking into account the following parameters: zone geometry, zone height, zone location, number of exterior walls, type of exterior wall, percentage of glazing, interior shade, type of partition, floor type and covering, and roof type.

Burch et al. [1992] used a finite-difference model to numerically determine a complete set of Conduction Transfer Function coefficients for five types of thermal bridges usually found in commercial buildings. The heat transfer response to a triangular pulse was obtained by superimposing the responses for three ramp excitation functions, which form a triangular pulse.

Huang et al. [1996] generated a library of two-dimensional response factors for 76 steel-frame wall sections for use with the DOE-2 program. They used a new version of WALFERN model:

first, a finite-element grid is automatically generated, depending on user-specified thicknesses and sizes of wall layers; second, two sets of response factors are computed, one with inside-film-resistance to compute the space loads, and another one without film-resistance to compute the custom weighting factors. Comparisons preformed at the Oak Ridge National Laboratory indicated the model predictions are within 3% of measured data.

Capability to model the processes listed in Task 1a
Most available models can be used to simulate the one-dimensional heat transfer process. The conduction transfer function method allows for the calculation of surface temperature, while the weighting factors method does not. The response factor coefficients are developed separately assuming that all processes are linear, and the system properties do not change with time and temperature. The two-dimensional response factors can also be developed.

Accuracy
For most building materials and operation strategies, if a sufficient large number of coefficients are used, there is no significant difference between the results of this model and those of numerical models.

Ease of implementation
The model is easy to implement. There are already methods available to generate the response factor coefficients. A database should be developed to integrate all available response factors coefficients. Other set of coefficients should be developed, if it is required, for other construction practices.

Ability to integrate HVAC calculation methods
Coupling with HVAC calculations takes place at the boundary of wall. In the case of conduction transfer function method, the thermal conditions of the boundary are well modelled and easy to change. Hence, the model allows for integration using either the sequential or the "simultaneous" simulation of loads and HVAC systems. In the case of "simultaneous" simulation, the main problem consists in finding an efficient method for solving the large system of simultaneous equations, in order to reduce the CPU requirements. In the case of weighting factors method, only the sequential integration is possible.

CPU requirements
This model-type is faster than the numerical models.

Flexibility for future improvements
Most future improvements will require the response factor coefficients to be evaluated again. Moreover, it cannot be used for simulating the time- or temperature-dependent properties.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools, without reducing the accuracy of the whole model. Other tools should use a time-step compatible to that used by this model (generally, one-hour time-step).

Frequency-domain Response Factor methods
Frequency domain techniques can generally be separated into two main categories: (a) the

analytical approach in which a closed form solution is obtained for the variables of interest as a continuous function of frequency, and (b) the discrete frequency approach in which the system response is evaluated at a few frequencies of interest [Athienitis et al. 1987, Athienitis et al. 1990]. The analytical approach facilitates sensitivity studies but it is impractical for models with more than a few nodes. Discrete frequency analysis is computational efficient and easy to program. Weather conditions are approximated by a series of periodic cycles, and can be represented by a steady-state term accompanied by a number of harmonics [Clarke 1982]. The periodic response of building is obtained by superposition of effects of all harmonics. The methodology is primarily based on determination of Laplace transfer functions for the building envelope, assuming the building can be modeled as a thermal network.

Major problems with frequency domain methods were related to the modelling of longwave radiation heat transfer among interior surfaces, the time-dependent properties (e.g., thermal resistance of a window with night insulation), and the modelling of inter-zone heat transfer. Athienitis modeled the longwave radiation by assuming a linear process [Athienitis 1987], and proposed a technique to include the variation of window conductance [Athienitis et al. 1987]. Subbarao and Anderson [1985] introduced "interaction factors" to indirectly account for the longwave radiation heat transfer between interior surfaces.

<u>*Capability to model the processes listed in Task 1a*</u>
Available models can be used to simulate the one-dimensional heat transfer process. It allows the calculation of surface temperature. The response factor coefficients are developed separately assuming that all processes are linear, and the system properties do not change with time and temperature. It can be applied for those applications where the weather conditions can be approximated by series of periodic cycles.

<u>*Accuracy*</u>
Similar accuracy as the time-domain response factor method.

<u>*Ease of implementation*</u>
Only a few models are available. The implementation is more difficult than the time-domain response factor model.

<u>*Ability to integrate HVAC calculation methods*</u>
It can integrate the HVAC calculations.

<u>*CPU requirements*</u>
The model is relatively fast, equivalent to the time-domain response factor model.

<u>*Flexibility for future improvements*</u>
Most future improvements will require the response factor coefficients to be evaluated again. Moreover, it cannot be used for time- or temperature-dependent properties.

<u>*Interoperability with other tools*</u>
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools, without

reducing the accuracy of the whole model. Other tools should use a time-step compatible to that used by this model (generally, one-hour time-step).

Cooling Load Temperature Difference (CLTD) and Cooling Load Factor (CLF) methods

The CLTD/CLF method was initially developed as manual calculation procedure, and used tabulated CLTD and CLF values, which were calculated using the transfer function method. It is based on some approximations that resulted in significant inaccuracies under some conditions [Spitler et al. 1993a]. Research performed under ASHRAE research project RP-472 led to a larger set of weighting factors for different zone types [Sowell 1988a, Sowell 1988b]. For instance, the heat gain through an exterior wall is calculated as

$$q_{wall,t} = U \cdot CLTD$$

The CLTD coefficients are calculated for standard conditions (e.g., July 21, at 24, 36 and 48°N latitude, no exterior shading, maximum outside air temperature 35°C with a daily range of 11.7°C, room air temperature constant at 25.5°C) and tabulated for several wall types (that is, for a group of walls with similar thermal response characteristics) and orientation. The tabulated CLTD can be corrected for different interior and outdoor temperatures. These coefficients were developed based on the Transfer Function method [McQuinston and Spitler 1992, ASHRAE 1993, McQuinston and Parker 1994]. ASHRAE developed a computer program to calculate the CLTD and CLF coefficients, using the electronic tables developed with the Transfer Function method [McQuinston and Spitler 1992]. However, the users are warned to use this approach with caution, when analyzing conditions outside those used for the development of coefficients.

Similar methods use Equivalent Temperature Differences [Carrier 1965], Total Equivalent Temperature Differentials [Trane 1975] or Virtual Temperature Difference [Recknagel 1985].

Capability to model the processes listed in Task 1a

The model can be used to simulate the one-dimensional heat transfer process. It does not allow for the calculation of surface temperature, nor the integration of heat and mass transfer models. Limited capabilities to simulate the time-dependent problems such as thermal storage effects or impact of changes in thermostat settings on the room air temperature or space cooling load. It cannot be applied when the transient room air temperature must be evaluated.

Accuracy

Less accurate than the numerical or response factor methods. The model is not accurate when it is applied for other conditions than those used in the development of CLTD values.

Ease of implementation

The model is easy to implement. Tables for different building elements are available, and a computer *program is available from an ASHRAE Research project, to calculate the CLTD values.*

Ability to integrate HVAC calculation methods
The model was mainly developed to evaluate the overall heat flow through a building element, without planning for future integration. However, if the HVAC model requires only the space cooling loads, then this model can be used. If it requires more detailed information on the wall boundary such as surface temperature, or if it calculates the wall temperature simultaneously in the loads module and the HVAC module (e.g., for radiant systems), then this model is not appropriate.

CPU requirements
Fast calculations.

Flexibility for future improvements
Future improvements could only concern the development of more appropriate CLTD values. However, this improvement will not change the overall approach of this method. It is less flexible than the numerical methods and time-domain response factor method.

Interoperability with other tools
The model was mainly developed to evaluate the overall heat flow through a building element, without planning for future integration. If the overall heat flow is the only information required by other tools, then the model can be coupled with them.

Modified Bin method

This method was initially developed as a simplified energy analysis procedure to fill the void between manual methods (degree-day and equivalent full-load hours) and comprehensive hourly energy analysis methods [Kusuda 1981, Knebel 1983, Knebel 1995]. It is assumed that the transmission load through opaque elements can be expressed as a linear function of outdoor temperature, and is composed of two terms: (i) the transmission due to the temperature difference, using the steady-state U-value, and (ii) the solar contribution, using the 24-hour average solar component of CLTD values, the fraction of possible sunshine, and the U-value. Calculation of transmission load is performed for January and July, and then a linear correlation between peak design temperatures and the corresponding transmission loads is obtained. Some examples of energy analysis programs based on this method are: ASEAM2.1 [Ohadi et al. 1988, Ohadi et al. 1989] and BESA-Retrofit [1985].

Since the procedure is based on time-averaging techniques, it has limited capabilities in accurately dealing with highly time-dependent problems (e.g., thermal mass, deadband thermostat, night setback).

ASHRAE Research project RP-564 developed two simplified methods to be integrated with the modified bin method: (i) a simplified method for calculating solar gains with greatly improved accuracy [Vadon et al. 1991, Balasubramanya 1992], and (ii) a simple approach for treating thermal mass [Claridge et al. 1992b].

Capability to model the processes listed in Task 1a
Limited capabilities to simulate the time-dependent problems such as thermal storage effects or impact of changes in thermostat settings on the room air temperature or space cooling load. It cannot be applied when the transient room air temperature must be evaluated.

Accuracy
Less accurate than the numerical or response factor methods.

Ease of implementation
Relatively easy to implement.

Ability to integrate HVAC calculation methods
The model was mainly developed to evaluate the overall heat flow through a building element, without planning for future integration. However, if the HVAC model requires only the space cooling loads, then this model can be used. If it requires more detailed information on the wall boundary such as surface temperature, or if it calculates the wall temperature simultaneously in the loads module and the HVAC module, then this model is not appropriate.

CPU requirements
Fast calculations.

Flexibility for future improvements
Some developments aimed to make this model able for simulating the thermal mass or solar gains. However, all improvements have tried to accommodate the overall approach of this method, which is not always possible without additional simplifications. Overall, it is less flexible than the numerical methods and time-domain response factor method.

Interoperability with other tools
The model was mainly developed to evaluate the overall heat flow through a building element, without planning for future integration. If the overall heat flow is the only information required by other tools, then the model coupled with them.

Steady-state heat transfer through windows and other transparent envelope components (except transmission of solar radiation), including effects of frames and spacers; night insulation (e.g., draperies, venetian blinds)

The modelling of heat flow must take into account the conduction through panes, the natural convection in the air space between panes and the radiation within the cavity (between panes, frame, and spacers), the forced convection and radiation at the outside surface of glazing, and the natural convection and radiation at the inside surface of glazing. Usually, the thermal storage effects of glazing system are neglected, and the steady-state heat transfer models are used to evaluate its thermal resistance, under constant or variable outdoor and indoor environmental conditions. The transmission of solar radiation will be discussed in section 29-BS.

The overall U-value is either based on measurements in environmental chambers or using guarded plate apparatus, or can be calculated, at standard or reference indoor and outdoor conditions, using area-weighted U-values corresponding to the center of glass (U_{cg}), the edge of glass (U_{eg}) and the frame (U_{fr}), including the effect of space dividers [ASHRAE 1993]. The U-values can be evaluated using: (i) one-dimensional numerical models, (ii) two- and three-dimensional numerical models, and (iii) correlation-type models.

<u>One-dimensional numerical models</u>

Usually, the heat transfer in the centre-of-glass zone is considered to be one-dimensional. An energy balance equation is written for a number of nodes across the centre of glass, and then a system of simultaneous equations is obtained. The solution will give the temperature of each layer. The conduction heat transfer is directly evaluated using the Fourier's law. The natural convection heat transfer in the cavity is evaluated using the correlations for Nusselt number, which depends of Rayleigh number, Prandtl number and the cavity height-to-width ratio. In most cases, the calculation of convection heat flow is based on the one-dimensional correlations, assuming that is independent of the location in the centre-of-glass zone. Arasteh et al. [1989] presented a review of correlations for vertical and tilted windows, heated from below or from above, and a cavity ratio of 40. These correlations were implemented in the WINDOW 4.1 program [WINDOW 4.1 1994]. The results of WINDOW 4.1 program can be easily imported into the DOE-2.1E program, and then used for evaluation of overall energy performance of a house [Reilly et al. 1992]. The thermophysical properties necessary for calculations are also provided for different gases as well as a method to evaluate the properties of mixtures. Wright [1996] analyzed Elsherbiny correlation, and then developed a new correlation based on most recent measurements. This new correlation is useful for analyzing windows with low aspect ratio (short windows, windows with divider bars, or windows with wide pane spacings).

Arasteh et al. [1989] used the Edward's wide-band model to find the transmittance of the infrared absorbing gases (e.g., SF_6 or CO_2) within the cavity, for which the gray gas assumption is not valid.

Capability to model the processes listed in Task 1a
It can evaluate the heat flow through the centre-of-glass zone, the corresponding U-value, and the inside surface temperature. However, it cannot model the heat loss through the edge-of-glass zone and the frame.

Accuracy
High accuracy for the centre-of-glass zone.

Ease of implementation
Models are available in the literature. Source code of computer programs (VISION, WINDOW) could be obtained. Easy to implement.

Ability to integrate HVAC calculation methods
It can integrate HVAC calculations, if the centre-of-glass parameters are accepted by those models, or if the surface temperature is calculated by taking into account the system output (e.g., radiant system).

CPU requirements
Fast calculations.

Flexibility for future improvements
It can incorporate other correlations for natural convection within a cavity, and some corrections factors to take into account the effect of frames and edge-of-glass zone.

Interoperability with other tools
It can be used: (i) as a reference model to generate U-values for the centre-of-glass zone, and then to transfer this value to the next simulator, or (ii) integrated within the next simulator.

Two- and three-dimensional numerical models

However, the air flow within the cavity is more complex, and the one-dimensional model is only a simplification. At a small temperature difference between panes there is a weak unicellular air flow, which is changed into two independent boundary layers close to the panes, if the temperature difference increases. Finally, the two boundary layers could become turbulent, and a second flow within the cavity could be noticed. Wright and Sullivan [1994] developed a two-dimensional model for natural convection of gas within a vertical, rectangular cavity. The differential equations describing the conservation of energy, horizontal and vertical momentum, and mass, respectively, were discretized using a grid aspect ratio $\Delta y/\Delta x = 5$. This choice was view as a compromise to limit computer requirements while avoiding an extreme imbalance in the areas of the vertical and horizontal faces of the finite control volume. After a solution is found for the temperatures and velocity fields, the heat transfer can be quantified, and then the Nusselt number can be evaluated in terms of vertical location. Good agreement was obtained with measured data.

This model was extended to evaluate the temperature and heat flux distribution in a double-pane glazing system [Wright and Sullivan 1995], by adding a model for conduction heat transfer in the solid sections (glazings and edge seals), and another model for the radiation exchange between all surfaces exposed to the interpane cavity. The model was configured to simulate a glazing system under the conditions of a guarded heater plate test. To simulate the thermal radiation within the cavity, each of the four boundaries were divided into finite surfaces. The incident radiative flux H_p at one surface p is calculated in terms of all surfaces from which radiation can reach that surface:

$$H_p = \sum E_{p\text{-}nb} \cdot B_{nb}$$

where $E_{p\text{-}nb}$ is the geometric exchange factor from a neighbour surface nb and the surface p, and B_{nb} is the radiative flux leaving the neighbour surface nb. The hemispheric reflectance was divided into specular ρ^s and diffuse ρ^d components, to be able to simulate more complex situations (e.g., low-e coating), which led to the following relation:

$$H_p = \sum E_{p\text{-}nb} \cdot \rho^d_{nb} \cdot H_{nb} + \sum E_{p\text{-}nb} \cdot \epsilon_{nb} \cdot \sigma \cdot T^4_{nb}$$

The number of variables H_p is reduced by assuming that the centre-of-glass region is very large, and therefore the corresponding surface temperature is constant. Finally, the net flux of radiant energy into the surface p is calculated:

$$S_p = \epsilon_p \cdot (H_p - \sigma \cdot T^4_p)$$

and is included as a fixed energy source in the energy balance of the wall control volumes. Their results indicated that the location of the minimum temperature of indoor pane is at (or very near) the bottom sight line, where the condensation forms usually first.

Laboratory measurements indicated that the edge-of-glass region is limited to 65 mm wide band around the perimeter of the glazing unit [ASHRAE 1993]. Arasteh [1989] developed a correlation to evaluate the edge-of-glass U-value as a function of spacer type and centre-of-glass U-value. Curcija and Goss [1994] analyzed the heat transfer in a complete fenestration system, including glazing and frame, using a two-dimensional numerical model developed based on finite element method. The laminar natural convection heat transfer within the window was modeled using the continuity equation, the momentum equation, and the energy equation. The conduction heat transfer in panes and frame was modeled using the two-dimensional Laplace equation. The radiation heat transfer was modelled using the radiosity matrix method. The convection and radiation heat transfer contribution to the overall surface heat transfer coefficient were evaluated separately. The radiative coefficient was temperature-dependent, and the convective coefficient was either constant or position-dependent. From these results was possible to calculate the centre-of-glass U-value, the edge-of-glass U-value, and the overall U-value. The results of numerical study were in good agreement with measured data. The results indicated also that the two-dimensional effects of edge-of-glass area extend to 102 mm from the sight line, compared with 65 mm currently used. The variable boundary conditions create zones in the vicinity of the edge-of-glass region, which thermally behave different, when compared with the case of constant boundary conditions. Wright et al. [1994] introduced the concept of edge-seal "linear conductance" equal to the rate of heat transfer through a unit length of edge-seal exposed to a unit temperature difference. Values of linear conductance for some typical windows were measured, and then integrated within a two-dimensional numerical model [FRAME 1992].

Smith et al. [1993] presented a review of simulation for one- and two-dimensional window models. They also developed a finite-volume model to simulate the two-dimensional combined heat transfer processes common to window applications. The conservation equations describing two-dimensional heat flow for a rectangular solution domain for laminar flow, steady-state conditions, and no heat generation are: continuity, horizontal and vertical momentum, and energy. The forced convection coefficient on the outside surface is constant, and on the inside surface could be either constant or function of vertical location. For most results, 20 control volumes are assigned to the horizontal direction for the cavity and three to a glass pane. For the vertical direction, 60 control volumes are assigned to the cavity. Doubling the number of control volumes produced U-values that differ by less than 0.2%. For certain limiting cases, the model agrees well with results obtained from one-dimensional calculations and U-values reported in the literature. However, the agreement is highly dependent on the choice of average convection coefficient in the one-dimensional model.

Curcija [1990] developed a simulation model of the three-dimensional heat transfer with spatially varying heat transfer coefficients on the inner and outer panes, and radiant heat exchange between the panes.

Elmahdy [1992] compared the measured U-values of nine high-performance windows with the simulation results using VISION 2, WINDOW 3.1, and FRAME 2.2 programs. He found differences of up to 15%, which are due to several factors such as uncertainty of defining material properties, air leakage through windows, or inaccurate modelling of a window cross section.

McGowan [1995a, 1995b] coupled the mathematical model of convection within the glazing cavity, developed by Wright et al. [1994] with the existing finite-volume method used in FRAME program. The following steps are performed:

- the steady-state velocity field in the glazing cavity, assuming isothermal glazing surfaces and laminar flow, and the temperature distribution across the cavity at the boundary between the edge-of-glass and the centre-of-glass regions are calculated;
- the velocity field is used to determine the convective relationship between nodes in the glazing cavity in the finite-volume network; the previously calculated temperature distribution is used to define boundary conditions at the connection between edge-of-glass and centre-of-glass regions;

- the radiation between the glazing and spacer/frame is neglected.

The results from the FRAME-convection model show the temperature at the lower sightline is 5.6°C colder than that predicted by the FRAME program alone (without considering the convection within the cavity), and the temperature at the upper sightline is 2.2°C warmer.

Capability to model the processes listed in Task 1a
It can evaluate the heat flow through the centre-of-glass, edge-of-glass, and frame zones, the corresponding U-values, and also the overall U-value. It can also evaluate the inside surface temperature of glazing system.

Accuracy
High accuracy can be obtained depending on the discretization approach and the methods used for solving the system of equations.

Ease of implementation
Some models are already available, and the source code could eventually be obtained.

Ability to integrate HVAC calculation methods
The coupling of this model with HVAC calculations can be done. The main problem consists in finding an efficient method for solving the large system of simultaneous equations, in order to reduce the CPU requirements.

CPU requirements
The CPU requirements increase with the number of nodes used for the discretization of continuous field.

Flexibility for future improvements
It is the most flexible model, since it is based on the first principles approach, and does not require the correction factors or the response factors to be separately calculated. Therefore, most future improvements can be integrated.

Interoperability with other tools
It can be used: (i) as a reference model to generate U-values of the centre-of-glass, edge-of-glass, and frame zone, or to generate corrections factors to be used with the one-dimensional model, or (ii) integrated within the next simulator.

Correlation-type models

Sullivan et al. [1992] developed a fenestration performance design tool for residential buildings, called RESFEN, as a method of transferring technology to the construction industry. The program incorporates correlations between energy performance and several window and house parameters, developed through regression analysis of a large database, which was obtained for a prototype house by using DOE-2 simulation program.

Capability to model the processes listed in Task 1a
Limited capability. It evaluates only the overall energy performance of glazing systems.

Accuracy
It is influenced by the accuracy of pre-calculated correlation coefficient or correction factors.

Ease of implementation
A set of correlation coefficients must be developed before the implementation, taking into account factors such as construction practice and Canadian weather conditions.

Ability to integrate HVAC calculation methods
No capability.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement consists in developing better correlation coefficients or covering other types of glazing systems, this model can accommodate it. However, if the future development consists in improving other modelling aspects, then this model does not have the required flexibility.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other simulation tools. If it is coupled with a more detailed model, the accuracy of the whole model is likely to be reduced.

Transient heat transfer through windows and other transparent envelope components (except transmission of solar radiation), including effects of frames and spacers; night insulation (e.g., draperies, venetian blinds)

Generally, the transient heat transfer models for opaque walls can also be used for glazing systems. In addition, the solar radiation absorbed by glazing must be introduced as an internal heat source. The transmission of solar radiation will be discussed in section 29-BS.

Three-dimensional numerical model

ElDiasty [1982] developed a model for simulating the transient, three-dimensional heat transfer in a window system, including glazing and frame. He used an implicit finite-difference technique to discretize the continuous field. The correlation between the outside surface coefficient and the wind speed and direction was developed experimentally. The solar radiation transmitted through a clear glass was evaluated using the Fresnel's equations. The indoor air temperature was assumed constant over the entire period of simulation. The system of simultaneous equations was solved using the Gauss elimination. This results indicated that the impact of window recess is significant on the heat flow through windows.

Capability to model the processes listed in Task 1a
The model is able to simulate processes related to the windows performance (e.g., heat transfer through frame, risk of condensation). However, for the present glazing systems the thermal storage effects are less significant than other thermal phenomena occurring in buildings. Hence, the simulation of transient heat transfer through glazing system should not have the highest priority.

Accuracy
High accuracy can be obtained depending of the discretization approach and the methods used for solving the system of simultaneous equations.

Ease of implementation
Numerical techniques are available, and can be implemented relatively easily (it is more a matter of programming time). The main problem is to find an efficient method for solving the large system of simultaneous equations, in order to reduce the CPU requirements.

Ability to integrate HVAC calculation methods
The coupling with HVAC calculations can be done, using both the sequential or "simultaneous" approach. The main problem is to find an efficient method for solving the large system of simultaneous equations.

CPU requirements
CPU requirements increase with the number of nodes used for the discretization of continuous field.

Flexibility for future improvements
It is the most flexible model. Most future improvements can be integrated.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools. It can also be used as a reference model to calculate the correlation coefficients, the correction factors or the response factor coefficients required by simpler models.

5-BS

Steady-state heat transfer through below-grade building components

Heat capacity of the soil plays a major role in filtering the variation of ground surface temperature. However, in order to simplify the calculations the steady-state models, and the average outdoor and indoor thermal conditions are used. The two-dimensional pattern is usually considered only for heat transfer through the ground, by defining an equivalent U-value. The two- or three-dimensional heat transfer within the walls, including vertical conduction and convection to the top of basement wall, is neglected.

<u>ASHRAE method</u>

The hight of basement walls is considered to be composed of several segments of 300 mm height each. The heat flux through each segment is constant, and varies from segment to segment due to the path length through soil. The heat flux q_s ($W/m^2 \cdot K$) for each depth increment of the basement wall is published by ASHRAE [1993], based on research by Latta and Boileau in 1969, for one uninsulated and three insulated walls (R=0.73, 1.47, and 2.2 $m^2 \cdot K/W$). The heat flux of the basement wall Q (W/K) can be calculated by summing the heat flux through each wall segment q_s multiplied by the corresponding surface area A_s:

$$Q = \sum q_s \cdot A_s.$$

Finally, the total heat loss Q_t (W) through the basement wall is obtained as:

$$Q_t = Q \cdot (T_B - T_{out}),$$

where T_B is the design basement air temperature (K), and
$T_{out} = T_{\text{mean winter air temperature}} - A$

(A is the amplitude of ground temperature about a mean value).

The heat loss through basement floors is considered to be proportional with the length of perimeter P rather than the surface area of floor [ASHRAE 1993]:

$$Q_{floor} = F_2 \cdot P \cdot (T_B - T_{out})$$

<u>Capability to model the processes listed in Task 1a</u>

It can be used to evaluate the overall heat flow through a below-grade component. It cannot be used to calculate the local surface temperature and the local heat flow. The two- or three-dimensional heat transfer *within the walls, including vertical conduction and convection to the top of basement wall, is neglected.*

Accuracy
Low. Many driving factors are integrated within the recommended values of heat flow for a limited number of cases (only for four levels of insulation of the basement walls). The user cannot change values of these driving forces.

Ease of implementation
Model is very simple, available and can quickly be implemented.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC calculations, only if those models require the overall heat flow.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air, and moisture transfer, then this model is not flexible. If better correlations between the heat flow and the driving factors will be developed, then this model can accommodate them.

Interoperability with other tools
Since the model is simple, it can be linked with other simulation tools, provided that its results are useful to other programs. However, it could represent the weak point in a detailed simulation.

6-BS

Transient heat transfer through below-grade building components

By nature, the heat transfer through below-grade building components is transient due to the thermal storage effects of the soil and the time-dependent driving forces (exterior and interior thermal conditions). Some models consider all transient effects, while others consider only the thermal storage effects of the soil. Mathematical methods can be classified as (i) semi-analytical methods, (ii) numerical methods, (iii) frequency-domain response factor, and (iv) correlation-type methods.

<u>Semi-analytical methods</u>
Krarti et al. [1994] presented the results of ASHRAE Research project RP-666, which had the following objectives:

- to develop semi-analytical solutions to predict heat transfer from basement, slabs and crawl spaces;
- to validate these results with empirical data and three-dimensional numerical models;
- to generate response factors for foundation walls and floors for use hourly energy analysis programs such DOE-2.

The time-dependent three-dimensional differential equation governing heat transfer can be reduced to a time-independent three-dimensional Helmholtz equation using the complex temperature amplitude technique. The ground is divided into several rectangular zones, by taking into account the basement geometry, position of insulation, and water table depth. In each zone, the temperature is analytically determined by solving the Helmholtz equation. The temperature profile along the boundary between two zones is calculated using the condition of temperature gradient continuity between zones and Fourier series theory. This model was incorporated within the DOE-2 program to simulate a house with an uninsulated and unvented crawl space, and its predicted basement air temperature was higher by about 1°C in the winter than the temperature predicted using the U-effective and one-dimensional approach of DOE-2 program.

<u>*Capability to model the processes listed in Task 1a*</u>
It can model the process.

<u>*Accuracy*</u>
High accuracy.

<u>*Ease of implementation*</u>
Model is available as a product of an ASHRAE research project.

Ability to integrate HVAC calculation methods
Since the temperature on the boundaries and the corresponding heat flows are calculated, the model can be integrated with HVAC calculations.

CPU requirements
Fast calculations.

Flexibility for future improvements
Theoretically, the model is able to accommodate future improvements. However, they can be limited by the simplifying assumptions used to solve analytically the problem. An evaluation of these assumptions should be performed, along with the list of potential improvements.

Interoperability with other tools
It can be coupled with other tools, or can be used as a reference model to calculate the response factor coefficients or correlation coefficients.

Numerical methods
Sullivan et al. [1985] developed a two-dimensional finite-element model to simulate the heat transfer from basement walls, and then to develop the weekly response factors. The model was integrated within a research version of DOE-2.1B program.

Bahnfleth and Pedersen [1989] used an implicit finite-difference model to study the three-dimensional slab-on-grade heat transfer, where the heat flux at the ground surface was defined in terms of: (i) the solar and infrared radiation fluxes absorbed by the surfaces, (ii) the evaporation at the ground surface, and (iii) the convection due to temperature difference and wind. The domain was discretized using about 10,000 cells. Minimum grid spacing was 0.1 m near the ground surface and slab boundaries, and larger near the outer boundaries. Snow cover was simulated by changing the surface convective and radiative properties. The main conclusions from this study are the following:
- the heat loss is not proportional with the perimeter length (errors of 50% were obtained); it is rather a logarithmic function of the ratio A/P (A is the floor area, P is the perimeter length);
- the thermal conductivity of soil and the ground surface conditions have a strong influence on the floor heat transfer rates, while the thermal diffusivity, far-field boundaries and deep ground conditions do not;
- the ground shadow can increase the floor heat loss by more than 20% under conditions of high evapotranspiration potential.

Forowicz [1994] performed a comparative analysis of various insulation configurations for earth-sheltered building, using a numerical method for two-dimensional transient heat transfer.

Krarti and Choi [1996] developed a simplified procedure to estimate foundation heat losses for all types of insulation configurations, all building shapes, all soil types, and various water table depths. This simplified method is based on the results from the Interzone Temperature Profile Estimation (ITPE) technique, which combines analytical and numerical techniques to obtain two- and three-dimensional solutions [Krarti et al. 1994]. To determine simplified nonlinear correlations for annual heat loss from basement walls and floors, 800 basement configurations were analyzed, using different insulation R-values, geometric characteristics of the basement walls and floors (width, length, and height). The correction coefficients were developed to account for corner effects, based on Mitalas method.

Capability to model the processes listed in Task 1a
It can model the two- and three-dimensional heat transfer processes. It can accommodate all the heat and mass transfer processes occurring through the soil, and is flexible in accepting boundary conditions. It can simulate the temperature-dependent and the time-dependent thermal properties and phenomena.

Accuracy
High accuracy can be obtained depending of the discretization approach and the methods used for solving the system of simultaneous equations.

Ease of implementation
Models are available, and eventually the source codes could be obtained. Moreover, numerical techniques are available in all heat transfer textbooks.

Ability to integrate HVAC calculation methods
The coupling of this model with HVAC calculations can be done using either the "simultaneous" approach or the sequential approach.

CPU requirements
The CPU requirements increase with the number of nodes used for the discretization of continuous field. The model is slower than other transient heat transfer models.

Flexibility for future improvements
It is the most flexible model, since it is based on the first principles approach, and it does not require the correction factors, the correlation coefficients or the response factors to be evaluated again for other conditions.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools, without reducing the accuracy of the whole model. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.

Frequency-domain Response Factor methods

Mitalas [1987] developed a model of three-dimensional basement heat transfer, where the instantaneous heat flux q_n (t) through each segment n (two segments for a basement wall and two segments for the floor) is presented in terms of steady-state term and a sine function:

$$q_n(t) = S_n \cdot (T_B - T_m) + V_n \cdot \sigma_n \cdot T_1 \cdot \sin(\omega[t+\Delta t_n])$$

A two-dimensional finite-element program was used to simulate some representative house foundation systems. The regression analysis of results along with the heat flux model gave the unknown parameters S_n (steady-state heat loss), V_n (periodic heat loss factor), σ_n (amplitude attenuation factor), and Δt_n (time lag of the heat flux relative to surface ground temperature variation). The three-dimensional heat flow due to the corners was accounted for by adding special corner allowance factors C_n, which were computed using data from a test basement.

The BASECALC^{TM} model was developed for the evaluation of basement and slab-on-grad heat losses [Beausoleil-Morrison et al. 1995a, 1995b, and Beausoleil 1996]. The method is an extension of Mitalas method [Mitalas 1987], and used a finite-element model to evaluate the two-dimensional heat-loss factors for central zones, where the impact of corners is negligible. The total below-grade heat loss of the central zone is calculated as the sum of (i) the average heat loss, and (ii) the variable heat loss:

- the average value represents the heat loss from the basement to the water table, plus the annually-averaged heat loss from the basement to the ground surface (steady-state process); the ground-surface temperature is equal to the water-table temperature;
- the variable component represents the heat loss from the basement to the ground surface (transient process, in which the ground-surface temperature varies sinusoidally), minus the annually-averaged heat loss from the basement to the surface; the water-table temperature and the basement-air temperature are equal.

In order to account for the three-dimensional effects in the corner zones, a corner factor F_c was defined as the ratio of the heat loss from the corner zone to the heat loss from the central zone. A correlation-type model was developed relating the corner factor F_c with the foundation's and site's thermal and geometrical characteristics. This approach is called "corner correction method." The evaluation of F_c was done with TRISCO model, under steady-state conditions, with a constant ground-surface temperature.

Sobotka et al. [1995] compared the predictions of heat losses given by several methods: Mitalas's, ASHRAE, European standard, and a two-dimensional finite element program developed by Matsumoto [1992], with the measurements from a test house in Japan. Mitalas's method gave the best agreement. The authors noticed that the implementation in the design process of a three-dimensional model, as one applied in Mitalas's method, is almost impossible due to its complexity. They recommend that future research should concentrate on parametric studies using the three-dimensional model, and then the results should be tabulated or expressed by simplified formula for practical use.

Krarti et al. [1995b] presented an analysis of foundation heat transfer using the transfer functions in frequency domain, and show the filtering effect of the ground at high frequencies. The heat flux through below-grade surfaces is affected only by cycles of more than several days.

> <u>*Capability to model the processes listed in Task 1a*</u>
> *It can model the two- and three-dimensional heat transfer processes, including different combinations of thermal bridges. It cannot evaluate the local surface temperature and heat flow. In the present version, it cannot evaluate the simultaneous heat and mass transfer phenomena.*
>
> <u>*Accuracy*</u>
> *Acceptable accuracy. Correction factors or correlation coefficients are used to account for the three-dimensional effects around the corners of basement.*
>
> <u>*Ease of implementation*</u>
> *Models and source codes are available.*
>
> <u>*Ability to integrate HVAC calculation methods*</u>
> *The coupling of this model with HVAC calculations can be done using either the sequential approach, using the heat flow, or the "simultaneous" approach, if the wall temperature can be calculated. The main problem consists in solving the large system of simultaneous equations.*
>
> <u>*CPU requirements*</u>
> *Relatively high CPU requirements (about 30 minutes CPU-time).*
>
> <u>*Flexibility for future improvements*</u>
> *It can be extended to other below-grade elements, by developing the appropriate correction coefficients.*
>
> <u>*Interoperability with other tools*</u>
> *It can be coupled with other tools, without reducing the accuracy of the whole model. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.*

<u>Correlation-type models</u>

Beausoleil-Morrison and Mitalas [1997] performed 33 000 parametric runs using the BASECALC model, and the results were used to develop a closed-form correlation-type model for calculating both above-grade and below-grade time-dependent heat losses through the foundation walls. The correlation-type model was implemented into the BASESIMP (SIMPlified BASEment model), which can presently model 27 basement and 40 slab-on-grade systems. The same set of four correlation equations are used for all 67 basement systems, each system having its own set of coefficients. All correlation coefficients were stored in ASCII files, to allow an easy extension of the model to other systems. BASESIMP model was incorporated as a subroutine into HOT2000 program, and can also be used as a stand alone program.

Beausoleil-Morrison and Krarti [1997] compared the predictions of foundation heat losses given by two models: (i) the BASESIMP model and (ii) a neural network model. Both models were trained using data from 1080 simulations performed with the detailed BASECALC model. The differences between models were within a few percentages, which indicated that both models can be used for evaluating foundation heat losses, with a slight preference to the neural network model. The authors underlined that it is much faster to develop a neural network model than a traditional correlation-based model.

Capability to model the processes listed in Task 1a
Currently, the correlation-type model can evaluate 67 below-grade elements. It can be used to evaluate the overall heat flow through a below-grade component. It cannot be used to calculate the local surface temperature and the local heat flow. It does not consider the transient indoor temperature.

Accuracy
Acceptable accuracy for modelling the 67 below-grade elements.

Ease of implementation
Model and source code are available. It was already implemented into the H2K simulator.

Ability to integrate HVAC calculation methods
The coupling of this model with HVAC calculations can be done using either the sequential approach, using the heat flow, or the "simultaneous" approach, if the wall temperature can be calculated.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air, and moisture transfer, then this model is not flexible. If better correlations between the heat flow and the driving factors will be developed, this model can accommodate them. However, the development of correlation coefficients for other conditions is time consuming.

Interoperability with other tools
Since the model is simple, it can be linked with other simulation tools, depending on the common information they can share.

7-BS

Steady-state moisture transfer through envelope components

Water vapor diffusion through building materials, under steady-state and one-dimensional flow, can be evaluated using Fick's law [ASHRAE 1993]:

$$w = -\mu \cdot (dp/dx)$$

where: w is the mass of vapor diffusing through a unit area in unit time, μ is the permeability of material, and dp/dx is the vapor pressure gradient along the flow path. Permeability μ may vary along the flow path, and it is function of relative humidity and temperature. To simplify the calculations of the total mass of vapor transmitted WV, an average permeability coefficient called permeance coefficient is used, and it can be found in tables for a given thickness:

$$WV = M \cdot A \cdot \Theta \cdot \Delta p$$

where: M is the permeance coefficient ($ng/s \cdot m^2 \cdot Pa$), A is the area of cross section of flow path (m^2), Θ is the time during which the transmission occurred (s), Δp is the difference of vapor pressure between ends of flow path (Pa).

The steady-state moisture transfer model can be used to simulate the following two phenomena: (i) the total mass of vapor transmitted through a wall assembly, and (ii) the risk of condensation within a wall, based on Glaser method.

Total mass of vapor transmitted through a wall

The resistance to vapor flow of a given material is given by $RV_i = 1/M_i$. The overall vapor resistance of an assembly is equal to $RV_t = \sum RV_i$, and the total mass of vapor transmitted WV is given by:

$$WV = A \cdot \Theta \cdot \Delta p / RV_t$$

Capability to model the processes listed in Task 1a

It is able to model the one-dimensional moisture transfer process. It can be used in the moisture balance equation for the indoor air.

Accuracy

Accuracy depends of the quality of information provided by the user.

Ease of implementation
Model is simple, available, and can quickly be implemented.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models through the moisture balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
The future improvements which could be integrated concern the development of better data on materials permeance. Other improvements are unlikely to be accommodated.

Interoperability with other tools
Since the model is simple, it can be linked with other simulation tools, depending on the common information they can share.

Risk of condensation within a wall (Glaser method)
The risk of condensation at a given location P within a wall assembly (e.g., at the boundary between two layers) can be evaluated based on the following steps:

- calculate the temperature distribution within the wall, and the corresponding saturation vapor pressure;
- calculate the vapor pressure as the difference between the vapor pressure inside the house (assuming the vapor pressure inside the house is greater than outside) and the vapor resistance of all layers between inside surface of wall assembly and point P;
- if the saturation vapor pressure is less than the vapor pressure, there is a risk of condensation at point P;
- the vapor flow WV_1 between the inside surface of wall assembly and the point P, and the vapor flow WV_2 between the point P and the outside surface of wall assembly are calculated; the difference $\Delta WV = WV_1 - WV_2$ ($\mu g/s \cdot m^2$) gives the rate of condensation at point P.

CONDENSE software [1993], which was developed to performed these calculations, is currently used by architects and builders. It uses a large database of thermophysical properties of building materials, and presents graphically the saturation pressure and water vapor pressure curves.

Mahdavi and Lam [1993] developed a computer-aided, graphically oriented tool, called VATRA (VApor TRansmission Analysis) for the analysis and visualization of water vapor diffusion in multi-layered building components. The temperature distribution is calculated hour-by-hour in a steady-state mode, based on Fourier's law, where the thermal conductivity k is calculated in terms of moisture content:

$$k = k_{def} \cdot (1 + Z) \cdot (1 + Z_{def})$$

where: k_{def} is the thermal conductivity of material with the "default" moisture content; Z is the thermal conductivity incremental factor related to the moisture content of material; Z_{def} is the incremental factor at default moisture content. The program presents graphically the saturation pressure and water vapor curves, and indicate the zone of potential condensation. It evaluates also the condensate quantities.

GLASTA program was developed to calculate steady-state and transient heat and vapor transfer through a plane multi-layered construction [PHYSIBEL 1996]. Three different calculation methods may be used:

1. Normal Glaser method: at each interface, the temperature, the saturation vapor pressure, and the vapor pressure are calculated; the vapor fluxes through layers, and the risk of condensation or drying is evaluated; the interface water content changes by the amount of condensation or drying.
2. DIN Glaser method: a small constant amount of water is assumed at the interface where the most condensation occurs; the resulting ratio drying/condensation corresponds to the DIN standard;
3. Extended Glaser method: the calculation takes into account the amount of water at the interfaces, and that contained within the layers themselves.

Capability to model the processes listed in Task 1a
It can model the one-dimensional heat and moisture transfer process through a multi-layer element. It can also evaluate the risk of condensation.

Accuracy
It can be coupled with HVAC models using the moisture balance for indoor air.

Ease of implementation
Model is available, and source code could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models through the moisture balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air, and moisture transfer, then this model is not flexible. If better materials data will be developed, this model can accommodate them.

Interoperability with other tools
Since the model is simple, it can be linked with other simulation tools, depending on the common information

they can share.

8-BS

Transient moisture transfer through envelope components

The moisture transfer occurs within porous building materials under the effect of several driving forces which act on the mixtures of solid matrix, the moisture in voids (water vapor, absorbed and capillary or mechanically bond water, ice) and the dry air. The numerical methods were generally preferred to model these extremely complex phenomena.

Numerical methods

Kohonen [1984] assumed that local thermodynamic equilibrium exists, that is, each phase has locally equal chemical potential. The author developed the constitutive nonlinear equations for one-dimensional heat and mass transfer through porous bodies, using the volume averaging technique: (i) conservation of mass, (ii) conservation of momentum, and (iii) energy balance. Interstitial phase change, capillary condensation and freezing are taken into account by terms which were obtained from applying the average divergence theorem. Because this theorem is not valid on interfacial surfaces of different phases, the surface momentum equation had to be considered in order to take into account the capillary forces. The transport process at the interface between air and a porous body is assumed to occur without accumulation of heat or mass. Four cases were considered at the interface of adjacent layers: (i) there is a perfect hydraulic contact, (ii) no hydraulic contact, (iii) there is so-called surface resistance between layers, and (iv) there is an air gap between layers. The thermal diffusivity and moisture diffusivity coefficients are function of temperature and moisture content. The implicit Crank-Nicholson method is used to discretize the constitutive equations. A computer program called TRATMO (Transient Analysis Code for Thermal and Moisture Physical Behaviour of Constructions) was developed on FORTRAN language for CDC CYBER-173, and validated with an analytical solution and experimental data.

Pedersen [1991] developed a model called MATCH (Moisture And Temperature Calculations for Constructions of Hygroscopic Materials), to simulate the one-dimensional heat and moisture transfer. The vapor pressure was used as the driving potential for the vapor flux, while the suction pressure was used for the liquid flux. The governing Fourier's, Ficke's, and Darcy's equations were discretized using an implicit finite-difference technique.

Karagiozis and Kumaran [1993] used a general research computer model to analyze the two-dimensional heat, air and moisture transport processes through multilayer building envelopes. The transport equations are based on temperature, pressure and water vapor pressure as driving potentials. Darcy's flow equations with Boussinesq approximation for incompressible fluids are used for the convective flows. the balance equations are discretized using a finite difference technique. Although the computer program can model the solar-driven moisture, wind pressure effects, direct rain effect, interface effects, and liquid water moisture movement, they were not included in the study. The results indicated the influence of convective flow within the insulation in the wall cavity to direct moisture to the upper part of the wall.

A detailed model, called MOIST, was developed at the National Institute of Standards and Technology to predict the combined transfer of heat and moisture in a multilayer wall under isothermal conditions [Burch 1993, Burch and TenWlode 1993]. The model includes moisture transfer by diffusion and capillary flow. The following one-dimensional governing equations are used: (i) conservation of mass, and (ii) conservation of energy The effect of energy storage in the dry material and the accumulated moisture is included. Thermal diffusivity, thermal conductivity, and moisture diffusivity are functions of temperature and moisture content. The sorption isotherm function is introduced into the moisture diffusivity term, as well as into the boundary and interface conditions. The non-storage layers such as an air space, glass-fiber insulation or a vapor retarder can also be included. An implicit finite-difference technique was used to discretize the two conservation equations. At each time step, the calculation proceeds by first solving for the temperature distribution, after which the moisture contents is calculated. On a 386 microcomputer, with math coprocessor and 33 MHz clock speed, the total computation time was about 30 minutes, for one year simulation with a time step of one hour.

Capability to model the processes listed in Task 1a
It can modelled the transient one- or two-dimensional moisture transfer phenomena.

Accuracy
Accuracy depends on the material properties defined by the user.

Ease of implementation
Models are available, and source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
Since the models are flexible in defining the boundary conditions, the coupling with HVAC models can be done, using either the "simultaneous" approach or the sequential approach. In the first case, the main problem consists in solving the large system of simultaneous equations.

CPU requirements
Relatively high CPU requirements (one model takes about 30 minutes).

Flexibility for future improvements
It is the most flexible model, since it is based on the first principles approach.

Interoperability with other tools
Since the model is flexible in defining the boundary conditions, it can be coupled with other tools, without reducing the accuracy of the whole model. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.

9-BS

Steady state moisture transfer through below-grade building components (e.g., basement floors and walls)

Mathematical models presented in section 7-BS can be used with appropriate boundary conditions.

10-BS

Transient moisture transfer through below-grade building components (e.g., basement floors and walls)

Mathematical models presented in section 8-BS can be used with appropriate boundary conditions.

11-BS

Air infiltration/exfiltration through unintentional openings; sensible and latent loads

Air infiltration is driven by pressure differences caused by wind, temperature difference between indoor and outdoor air (stack effect), and the operation of appliances [ASHRAE 1993]. The fundamental equation for the air flow rate Q (m^3/s) through an opening of cross-sectional area A (m^2) is given by:

$$Q = C_D \cdot A \cdot \sqrt{(2 \cdot \Delta p/\rho)},$$

where: C_D is the discharge coefficient of the opening, Δp is the pressure difference across the opening, and ρ is the air density. An approximation of air flow rate can be obtained by using the pressure-flow relationship:

$$Q = c \cdot (\Delta p)^n$$

This relation can be transformed into: $\ln Q = \ln c + n \cdot \ln(\Delta p)$; through a least-square regression analysis of a set of measured data (Q_i, Δp_i), the two coefficients c and n are obtained.

For the purpose of this report, the air infiltration models can be classified into two groups: (i) one-zone (single-cell) models, where the air infiltration rate is evaluated for the whole house, and the pressure difference between the thermal zones is neglected; and (ii) multi-zone (nodal-flow) models, where the house is represented by a network of nodes and links; each node represents a thermal zone of uniform pressure, and the link between two nodes represents the airflow between the corresponding two zones.

One-zone (single-cell) models

One simple rule of thumb for calculating the natural infiltration rate (ach) consists in dividing the air change per hour at 50 Pa pressure difference, which is obtained through a pressurization test, by 20. Sherman attributes this rule of thumb to Keonvall and Persily [Sherman 1996].

Tamura [1979] developed an empirical model to evaluate the air infiltration rate of a house by determining separately the infiltration rates caused by stack effect, wind action, and furnace operation, and then combining the three effects. For general application, the following test data from various types of houses are required: (i) neutral pressure level under calm conditions and furnace off; (ii) exterior wall leakage characteristics; (iii) ratio of local to meteorological wind speed; (iv) pressure difference across the exterior wall for different wind exposures; and (v) gas flow rate through chimneys.

Reeves et al. [1979] proposed a generalized computer model to predict hourly air infiltration rates in a residence, using physical variables and theory associated with air infiltration. The following variables were considered: crank length and width, location of neutral zone, pressure difference due to wind and temperature effects, and their interaction. He noticed that air infiltration could not be analyzed by considering wind directional components as originally thought. The correlations needed were developed using data from over 7,000 measurements of tracer gas concentration at six different houses and three apartments.

The air-change method predicts the infiltration rate Q in terms of air change per hour ACH input by the user (based on assumptions or previous measurements), the volume of space VOL, and the wind speed WIND [DOE-2]:

$$Q = a \cdot WIND \cdot VOL \cdot ACH$$

In the residential method, the air change per hour ACH is defined in terms of wind speed, zone temperature T_R, and dry-bulb outdoor temperature T_o [DOE-2]:

$$ACH = a + b \cdot WIND + c \cdot abs[T_R - T_o]$$

where a,b, and c have default values. The infiltration rate is calculated as:

$$Q = d \cdot ACH \cdot VOL.$$

In the crack method, the wind-generated pressure PW, and the stack-generated pressure PST, are added, and used in the pressure-flow relationship [DOE-2]:

$$Q = C \cdot (PW + PST)^n$$

where $PW = abs[\, a \cdot WIND^2 \cdot \cos\alpha]$
$PST = d \cdot P_{atm} \cdot [1/T_o - 1/T_R] \cdot ZHT$

α is the wind incidence angle,
ZHT is the vertical distance measured from the neutral plan,
n=0.8 for delayed walls, 0.66 for windows.

In another model, the user must input the maximum air infiltration rate Q_{max}, and then the hourly air infiltration rate Q is calculated as [BLAST]:

$$Q = Q_{max} \cdot SCH \cdot [a + b \cdot (T_R - T_o) + c \cdot WIND + d \cdot WIND^2],$$

where a,b,c, and d are constants, which can be input by the user.

Walton [1983] used the following empirical model to evaluate the mass flow rate m_{inf}:

$$m_{inf} = (a_0 + a_1 \cdot abs[T_R - T_o] + a_2 \cdot WIND + a_3 \cdot WIND^2)$$

LBL (Lawrence Berkeley Laboratory) model predicts either hour-by-hour or long-term average infiltration [Sherman and Grimsrud 1980]. The complex pressure distributions caused by wind and stack effects are converted into a single pressure across an aperture with the same effective leakage area as in real building. The combined infiltration Q (m^3/s) is calculated by simply superimposing the equivalent pressures induced by wind and stack effects, and assuming an orifice flow model:

$$Q = (Q^2_{stack} + Q^2_{wind})^{0.5}$$

or

$$Q = L \cdot (A \cdot \Delta T + B \cdot v^2)^{0.5}$$

where: L is the effective leakage area at 4 Pa pressure difference, which can be obtained either from a pressurization test of the whole house, or using available tables. The wind parameter B is calculated in terms of three factors: the generalized shielding parameter (it has the greatest uncertainty), the terrain factor (the wind measured at weather station is not usually the same as near the house), and the R-factor (it takes into account the distribution of leakage area between walls, ceiling, and floor). The stack parameter is calculated in terms of: height of neutral level, inside air temperature, and height of house. The comparison between the LBL model predictions and measured infiltration rates over a period of 34 days indicated the half-hour infiltration predictions had an accuracy of 35%, the daily predictions are within 20%, and the weekly predictions are within 10% [Grimsrud et al. 1982].

Lyrberg [1983] evaluated eight models for air infiltration, defined as follows:

- model A: $ach = a \cdot (\Delta T)^\gamma$,
- model B: $ach = a \cdot (v^2)^\gamma$;
- model C: $ach = a \cdot (\Delta T)^\gamma + b \cdot (v^2)^\gamma$;
- model D: $ach = a \cdot (\Delta T)^\gamma + b \cdot (v^2)^\gamma + c$;
- model E: $ach = a \cdot (\Delta T)^\gamma + b \cdot (v^2)^\gamma + c \cdot (\Delta T \cdot v^2)^\gamma$;
- model F: $ach = (a \cdot \Delta T + b \cdot v^2)^\gamma$, with one opening at bottom of winward side and one of the same size at top of leeward side;
- model G: $ach = (a \cdot \Delta T + b \cdot v^2)^\gamma$, with continuous distribution of openings on winward and leeward sides;
- model H: $ach = (a \cdot \Delta T + b \cdot v^2)^\gamma$, with four openings of equal size, located at top and bottom of winward and leeward sides.

He compared the predictions of these models with experimental data from five typical houses (from Canada, UK, US, Belgium, and Germany), and one 4-storey building in the Netherlands. His main conclusions are the following:

1. The most efficient models are of type F, G and H, with an average error of about 20%, when two free parameters are used. If the values of these parameters can be correctly chosen (e.g., through monitoring), these models can predict the air change per hour with an average error of 10 to 15%.
2. To improve the performance of models one must consider rather more complex models that use information about the pressure distribution over the building surface and also its variation with wind direction. If such data are not available, and as long as the experimental error is of the order of 10%, it is doubtful whether any models more efficient that the simple one discussed in his report can be constructed.

The Alberta Air Infiltration Model (AIM-2) uses a superposition technique where the infiltration flow rates due to wind and stack effects are added, and in addition an interaction term is introduced [Walker and Wilson 1990a, Walker and Wilson 1990b]:

$$Q = [Q^{1/n}{}_{stack} + Q^{1/n}{}_{wind} + B_1 \cdot (Q_{stack} \cdot Q_{wind})^{(1/2 \cdot n)}]^n$$

Instead of using the orifice flow assumption (n=0.5), the fan pressurization test is used to determine the coefficients C and n of the pressure-flow relationship. Assuming that the coefficient n has the same value for all leakage sites, the total leakage coefficient C is obtained by adding the corresponding coefficients for floor, ceiling and furnace flue. The distribution of leakages should be estimated by the user; the reader could rise some questions about the accuracy of this approach. However, a study based on eight different estimates of leakage distribution in a house, indicated that the range of predicted infiltration rates was within 14% of the average of the infiltration rates [Walker 1995]. The furnace flue is incorporated as a separate leakage site, at the flue exit height above the roof. The constant B_1 was obtained empirically from measurements of air infiltration. The wind pressure coefficient, the flue cap pressure coefficient, and the wind shelter coefficient were obtained from wind tunnel experiments. However, the estimation of local wind shelter is one major area of uncertainty.

Colliver et al. [1994] presented the development of a database on air leakage through building components, as part of ASHRAE Research Project RP-438. Data from all sources were normalized and presented as Effective Leakage Area at 4 Pa.

The sensible cooling or heating load Q_s (kW) due to air infiltration rate I (L/s) is calculated as [ASHRAE 1993, Walton 1983]:

$$Q_s = 1.23 \cdot I \cdot abs(T_o - T_R)$$

and the latent cooling load Q_l (kW):

$$Q_l = 3.0 \cdot I \cdot abs(W_o - W_R)$$

Liu and Claridge [1995a, 1995b] found, based on measurements in an existing house, that the classic engineering calculations of heat losses neglect the air infiltration heat recovery effects. This heat recovery is caused by the interaction of air infiltration, conduction heat transfer in the building envelope, and the impact of solar radiation of the exterior surfaces.

Capability to model the processes listed in Task 1a
It can model the overall air infiltration rate of a house, and therefore the overall impact on the energy consumption. It cannot be used to simulate the impact of air infiltration on the thermal behavior of each zone.

Accuracy
Acceptable accuracy for practical applications, where the information about the airtightness of the whole house is most likely to be obtained from a blower door test.

Ease of implementation
Models are available. One model (AIM-2) was already implemented into the H2K simulator. Others codes could eventually be obtained.

Ability to integrate HVAC calculation methods
The coupling with HVAC models can be done, using either the "simultaneous" approach or the sequential approach.

CPU requirements
Fast calculations.

Flexibility for future improvements
Models can be improved to better account for the local wind shelter, or the distribution of air leakeages on the exterior envelope.

Interoperability with other tools
It can be coupled with other tools, without reducing the accuracy of the whole model. The interoperability depends on the common data they can share.

Multi-zone (nodal-flow) models
In this type of models, the user has to input the characteristics of each individual opening. The air flow through each opening is calculated on a time-step basis as a function of interior and exterior pressures.

The COMIS program is a multi-zone air infiltration model [Feustel et al. 1989, Haghighat and Megri 1996]. A building is basically modelled by pressure nodes that are interconnected with air flow links. For one time-step, the outside of the building is represented by a fixed boundary condition. The pressures of the internal nodes in the air flow network have to be solved to determine the different air flow rates. The model can simulate crack flow, flow through large openings, single-sided ventilation, cross-ventilation, and HVAC systems.

Capability to model the processes listed in Task 1a
It can simulate the impact of air infiltration on the thermal behavior of each thermal zone, and can take into account the wind pressure distribution on the exterior envelope.

Accuracy
Accuracy depends of the quality of information provided by the user (e.g., type of cracks, dimensions, characteristics).

Ease of implementation
Model is available, and the source code could eventually be obtained.

Ability to integrate HVAC calculation methods
The coupling with HVAC models can be done, using either the "simultaneous" approach or the sequential approach.

CPU requirements
Acceptable CPU requirements.

Flexibility for future improvements
Future work should be done to assess the envelope parameters, as required by this model, based on practical measurements (e.g., blower door test).

Interoperability with other tools
This model can be coupled with thermal analysis models. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.

12-BS

Natural ventilation; sensible and latent loads

The natural ventilation is caused by the pressure difference between exterior and interior, across intentional openings, which is caused by the wind and the temperature difference (stack effect). The resulting air exchange is used for controlling the indoor air quality, pre-cooling the thermal mass of house, and avoiding the space over-heating.

Mathematical models presented in the previous section (11-BS) can be used to evaluate the air exchange rate due to the natural ventilation.

Other models have been specially developed to estimate the impact of natural ventilation on the cooling load of a house. For instance, the LESOCOOL program [Roulet et al. 1996] was developed as a simplified, multi-zone model for passive cooling (e.g., night-time ventilation). Bernoulli equation is used to calculate the pressure difference caused by the wind and stack effects, and the mass air flow through openings. The location of neutral pressure level results from the conservation of mass. In each zone, the equation for conservation of energy is introduced, and coupled with the air flow equations. A system of equations is obtained, and its solution gives the air flow rates, and the indoor air temperature. The program cannot model thin and multi-layer walls. The air flow must follow a single path, and the interactions between zones are limited to the ventilation coupling.

13-BS

Wind pressure on each component of the exterior envelope

The use of a single-node to represent the wind-generated outside pressure is an over-simplification of the real phenomenon, which is generally accepted to avoid modelling the complex pattern of air movement around a house. Moreover, by using this approach it is possible to use the available data from meteorological stations, rather than requiring local measurements of wind pressure.

A detailed model of the air infiltration/exfiltration through the building envelope should take into account the mean pressure created by the wind on various locations on the leeward and windward surfaces. This approach can use empirical models or CFD-based models.

Empirical models

The results of measurements performed on reduced-scale building models in wind tunnels or on real/experimental buildings are usually represented in terms of a dimensionless pressure coefficient C_p at different locations on the exterior walls or roof, as a function of wind direction and the terrain roughness. Charts representing the variation of pressure coefficient on different surfaces of simplified shapes of houses can mostly be obtained from studies of wind loads of buildings. This coefficient along with the approaching wind speed U_H at height H permits to calculate the mean pressure at different locations on the exterior envelope:

$$p_s = C_p \cdot (\rho \cdot U^2_H / 2)$$

Capability to model the processes listed in Task 1a
It can be used with the multi-zone air flow models, to simulate the impact of air infiltration on the thermal behavior of each thermal zone.

Accuracy
Good accuracy, if the pressure coefficients are available for the shape of house under analysis.

Ease of implementation
Model and data for several practical situations are available.

Ability to integrate HVAC calculation methods
It depends on the ability of multi-zone models to integrate HVAC models.

CPU requirements

Fast calculations.

Flexibility for future improvements
Future improvements consist mostly in evaluating the pressure coefficients for other shapes, more appropriate for the construction practices.

Interoperability with other tools
This model can be coupled with multi-zone thermal models.

CFD-based models

Baskaran and Stathopoulos [1993] developed a computer program called TWIST (Turbulent WInd Simulation Technique) to evaluate the mean values of wind-induced pressures on the building envelope. The three-dimensional differential equations of the air flow, the standard k-ϵ turbulence model, and the conservation of mass are discretized using the control volume technique. The Alternative Direction Implicit (ADI) method is used to solve the system of algebraic equations. The predictions of this numerical model compared generally well with experimental data.

Capability to model the processes listed in Task 1a
It can be used with the multi-zone air flow models, to simulate the impact of air infiltration on the thermal behavior of each thermal zone.

Accuracy
Accuracy depends of the number of nodes used to discretize the continuous field around the house.

Ease of implementation
Model is available, and the source code could eventually be obtained.

Ability to integrate HVAC calculation methods
It depends on the ability of multi-zone models to integrate HVAC models.

CPU requirements
High CPU requirements (about 20 minutes CPU time on a VAX mini-computer, for a model using 54,000 nodes).

Flexibility for future improvements
The model is flexible, and the modelling of air movement around the house can be improved, to account for the local wind shelter.

Interoperability with other tools
This model can be coupled with thermal analysis models. It can also be used as a reference model to calculate the correction factors, the correlation coefficients or the response factor coefficients required by simpler models.

14-BS

Steady state heat transfer through interior walls, floors and ceilings

Although the heat transfer through interior elements of a house is transient, the calculations are simplified by assuming the steady-state one-dimensional heat flow model. This assumption can be considered acceptable for the case of continuous operation of the heating/cooling system, with constant thermostat setpoint, and where the air temperature has only small fluctuations around the thermostat setpoint. This approach is mostly used for evaluating the design peak cooling or heating loads. The heat storage in the interior elements of a house, which can have an important effect on the reduction/increase of peak loads, is neglected.

The heat transfer q (W/m^2) through an interior element, from one interior space at design temperature T_{R1} to another space at design temperature T_{R2}, is calculated using the relation:

$$Q = U \cdot \mathrm{abs}(T_{R1} - T_{R2})$$

where U is the heat transfer coefficient ($W/m^2 \cdot K$). If the space 2 is unheated (e.g., attic, unheated basement), the corresponding design indoor air temperature is evaluated based on the heat balance of interior air, taking into account all heat gains or losses. In a similar manner, the design indoor air temperature for uncooled spaces is calculated .

Capability to model the processes listed in Task 1a
This model is limited to the conduction heat transfer process. The convection within the wall is neglected. It can be used to calculate the average surface temperature of the wall. It cannot be used to evaluate the local heat flow and the local surface temperature.

Accuracy
Acceptable for the practical applications where the overall U-value is required. The thermal storage effect is not taken into account.

Ease of implementation
Models are simple, available and can quickly be implemented.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC modelling, when the average heat flow or surface temperature of wall is an acceptable parameter for calculations.

CPU requirements
Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air and moisture transfer through walls, then this model has limited flexibility. If the future improvement represents the development of better correction factors for a large number of thermal bridges, then this model is flexible.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other simulation tools. However, the accuracy of the whole model depends on both components. Hence, if it is coupled with a more detailed model, the accuracy of the whole model is likely to be reduced.

15-BS

Transient heat transfer through interior walls, floors and ceilings

The use of transient heat transfer models allows for considering the thermal storage in the interior elements of a house, and for the variation of indoor air temperature in different zones. It allows, for instance, the evaluation of peak heating load in the morning, following the thermostat night setback, or the evaluation of passive solar heating design approaches.

The most important parameters affecting the performance of thermal mass, and which must be simulated are the following [Balaras 1996]: (i) thermophysical properties of materials, (ii) location within the space with respect to the solar radiation, (iii) air movement (forced ventilation, natural convection), and (iv) insulating materials (e.g., carpets, lightweight furniture). The author presented an overview of computational methods of cooling loads, where the thermal mass plays an important role.

The numerical methods or the time-domain response factor methods, presented in section 2-BS, can be used. In the case of weighting factors method, the air temperature of adjacent space is used as the driving force, instead of the sol-air temperature on the outside surface of the interior element. The room transfer functions are then used to evaluate the extraction rate, and the variation of indoor air temperature. It is important to mention that the weighting factors were developed assuming the system properties are constant, that is, they are not function of time or temperature [DOE-2]. This limitation is important when the film coefficients of the interior elements vary with direction of heat flow, solar radiation, or air movement (direction and velocity).

16-BS

Thermal storage effects of building mass and furniture

Usually, the heating systems are designed on the basis of steady-state continuous operation, and then an increase of capacity of the heating system is allowed to include the intermittent use such as the morning warm-up [Hitchin 1980]. For instance, ASHRAE Standard 90.1 [1989] allows the designer to increase by up to 30% the steady-state capacity for heating, and by up to 10% for cooling. Although a heavy structure cools down more slower than a light one, it also requires a greater input of energy per degree rise of internal temperature.

Therefore, the transient nature of this phenomenon must be modeled to account for its effects on the air temperature and the space thermal load. There are two categories of such models: simplified models (e.g., correlation-type models, semi-analytical) and detailed models (e.g., numerical methods, time-domain response factors methods).

<u>Simplified models</u>

Some examples of simplified models are the following:

- Givoni [1981] used the total thermal time constant (TTTC) of building, which is defined as the heat stored in the building mass per unit of heat transmitted to or from the outside through the elements surrounding the space;

- Barakat and Sander [1982] estimated the seasonal contribution of solar gains through windows G_s to the total seasonal heating load, by introducing the concept of "solar utilization factor η_s," which is expressed as a function of two normalized parameters:
  (i) the gain-load ratio GLR = G_s/(Net heating load in the absence of solar gains), and
  (ii) the thermal mass-gain ratio MGR:
  MGR = C/g_s (C is the thermal capacity of the building interior, g_s is the average hourly solar gain for season).
  The solar utilization factors were derived from a large number of computer simulations for four construction types (light, medium, heavy, and very heavy), with a constant heating setpoint of 21°C, and an allowable rise in room temperature of 2.75°C and 5.5°C;

- Balcomb [1983] developed a method for predicting temperature swings in direct gain buildings. He used the diurnal heat capacity, which is a measure of the effective amount of heat stored during a sunny day and then released at night.

- Hoffman et al. [1984] used the total thermal time constant (TTTC) method to evaluate the impact of thermal mass on the variation of indoor air temperature.

The LESOCOOL program [Roulet et al. 1996] is based on a simple solution of the heat transfer for a semi-infinite medium under the effect of a heat flow step function of amplitude q. The surface temperature of thermal mass varies in terms of thermal effusivity $b = \sqrt{(k \cdot c_p \cdot \rho)}$. The internal air temperature increases according to:

$$\frac{\Delta T_R}{q} = \frac{2}{b} \cdot \sqrt{\frac{t}{\pi}} + \frac{1}{h_c} \cdot [1 - \exp(-\frac{t}{\tau})]$$

where τ is the time constant of thermal mass and interior air.

Claridge et al. [1992] developed a simple model to treat the thermal mass, which was intended to be coupled with the modified bin method. A mass node is assigned to each zone, with an effective "lumped capacitance" C coupled with the envelope resistance R through an appropriate surface resistance R_a. All gains entering the building are treated as radiant gains that have to be absorbed first, and then released into the room air node through the surface resistance R_a. The whole network is subjected to the outdoor temperature T_o, which acts as a driving force for transmission and infiltration loads that interact with the zone through the resistance R. An hourly gain profile is defined. For instance, the infiltration component appears only during the night, when the fan is shut off. The solar load is linearized as a sawtooth gain profile during the occupied period.

Capability to model the processes listed in Task 1a
These models use a global approach to simulate the thermal storage effects, where the surface temperature is not evaluated, or is evaluated using simplifying assumptions (e.g., semi-infinite medium).

Accuracy
Acceptable accuracy for some practical applications. The accuracy depends on the applicability of pre-calculated correlation coefficients to the conditions under analysis.

Ease of implementation
Models are simple, and one source code is available. Other models can be implemented.

Ability to integrate HVAC calculation methods
If the HVAC models use the heat flow from the thermal mass to evaluate the thermal loads, then this type of models can be used. It cannot be used if the HVAC calculations require the surface temperature, in an explicit or implicit way.

CPU requirements

Fast calculations.

Flexibility for future improvements
If the future improvement is understood as coupling with more detailed models for simultaneous heat, air and moisture transfer through walls, then this model has limited flexibility. If the future improvement represents the development of better correction factors for a large number of thermal bridges, then this model is flexible.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other tools, depending on common data they can share. However, the coupling could reduce the potential capabilities of the most detailed model.

Detailed models
Numerical methods presented in section 2-BS can be used to simulate the impact of thermal storage. The Conduction Transfer Function method, presented in sections 2-BS and 15-BS, can also be used to evaluate the thermal storage effects of building mass and furniture [McQuinston and Spitler 1992, ASHRAE 1993, McQuinston and Parker 1994, DOE-2].

17-BS

Dynamic material properties (e.g., temperature-dependent thermophysical properties)

In current applications it is assumed that the material properties such as thermal conductivity or specific heat are constant during the simulation period. This assumption is appropriate for most building materials in use today, and simplifies the calculations. However, there are other materials such as phase change materials or heat-absorbing glass for which the dynamic thermal properties must be considered by the modelling approach.

The numerical methods presented in section 2-BS can be used to model the effect of temperature-dependent thermophysical properties. The mathematical models based on the Response Factor method are not suitable for this application, since the main assumption used in the development of these functions is that the system properties are constant (time- or temperature-independent) [DOE-2].

18-BS

Time-dependent heat gains from internal sources (occupants, lights, appliances); radiative and convective portions; location within thermal zone

Internal sensible heat gains are usually split into radiative and convective components. The convective component contributes immediately to the space thermal load, and the radiative component is first absorbed by the thermal mass, and is released later into the space. The transient nature of heat gains from internal sources can be modeled by taking into account two phenomena:

- the dynamic behavior of the internal sources (occupants, lights); in most detailed energy analysis programs, the internal sources are modelled using a steady-state approach, where the contribution of radiation and convection is constant over the simulation period, and it is imposed by default or defined by the user; there are also some dedicated models, which considers the dynamic behavior of luminaires;

- the dynamic behavior of space (e.g., air temperature, wall temperature, space load); the dynamic behavior of the space is modelled using the heat balance method or the weighting factors method; mathematical models presented in sections 33-BS to 38-BS can also be used to evaluate the impact of location of internal sources within the thermal zone.

Steady-state models of internal sources
When the response factor method is used, a constant distribution of sensible energy is assumed, and then the weighting factors are calculated [DOE-2, ASHRAE 1993]. For instance, the sensible gain from people and equipment is divided by default into 70% radiation and 30% convection. The sensible heat gain from lighting is divided into the following proportions in terms of fixtures and ventilation associated with: (i) suspended-fluorescent unvented - 67% radiation and 33% convection; (ii) incandescent - 71% radiation and 29% convection; (iii) recessed fluorescent vented - 19% radiation and 81% convection.

In another model, based on heat balance method and using the Conduction Transfer method, the user has to define the following components of sensible heat from lights [BLAST 1991]: (i) heat removed immediately from the lighting fixture by the return air vents (default is zero); the remaining part is divided into radiation or convection; (ii) the radiative component is of two types - one of thermal spectrum (20% for fluorescent, and 80% for incandescent), and one of visible spectrum (20% for fluorescent, and 10% for incandescent). In the case of heat gain from people, the default value for radiative component is 70%.

In most models, it is assumed that the radiative component is uniformly distributed over all interior surfaces, except one on which it is installed (e.g., ceiling).

Capability to model the processes listed in Task 1a
These models use a steady-state global approach to evaluate the internal heat gains.

Accuracy
Acceptable accuracy for practical applications. The accuracy depends on the pre-defined contribution of convection and radiation, which is assumed to be independent of the operating conditions.

Ease of implementation
Models are simple, available, and can quickly be implemented.

Ability to integrate HVAC calculation methods
If the HVAC models use the heat flow to evaluate the thermal loads, then this type of models can be used.

CPU requirements
Fast calculations.

Flexibility for future improvements
The model can be improved by defining better coefficients for evaluating the convective and radiative contribution.

Interoperability with other tools
Since it is a simple model, it can easily be linked with other tools, depending on common data they can share. However, the coupling could reduce the potential capabilities of the most detailed model.

Dedicated models of dynamic behavior of internal sources
All energy analysis programs neglect the relationship between the lamp temperature and the input power and light output, which can lead to incorrect calculations of heat gain, cooling load, and electricity consumption. A full-scale test facility has been constructed at the National Institute of Standards and Technology (NIST) to measure lighting levels, electricity demand and consumption, cooling load, temperature of room air and surfaces (lamp, walls, floor) [Traedo 1991]. The test facility is essentially a guarded calorimeter, the design and operation being similar to that of the National Research Council of Canada, which was used to determine the lighting cooling load factors contained in ASTM Standard C-326.

Sowell [1989, 1990] developed a computer model called LIGHTS, as "numerical test cell" to simulate the transient heat transfer between luminaires and a room. Diffuse short- and long-wave radiation is modelled, as is conduction, and surface-to-air convection. The model assumes one-dimensional heat transfer, uniform temperature and radiant flux over surfaces, and uniform air temperature in the room and plenum. The complex relationship between lamp power and lamp wall temperature is modelled using a handbook curve. Heat balances at each surface and air node form a differential-algebraic set of equations that is solved using iterative techniques. The comparison between the numerical model and the experiments at NIST test cell were not encouraging. It was not clear if the significant differences are due to numerical model deficiencies or experimental errors.

Walton [1990] developed a detailed computer model for evaluating the interactions between lighting and HVAC systems.

Sowell and Johnson [1993] developed a lighting heat gain model, with the goal of bridging the gap between the crude lighting models used by energy analysis programs and the detailed models. Since the relationship between the power and light output and the lamp wall temperature is nonlinear, an iterative Newton-Raphson method is used, where the initial guess is the minimum wall temperature. A set of simultaneous linear equations is obtained for the heat balances of lamps, air within the luminaire, lens and luminaire housing. The authors discussed also the possible integration with the weighting factor programs and the heat balance programs.

Available information on thermal load generated by people is based on steady-state data. However, a person can reach the thermal steady state after entering the room only after one hour or more. During this time, the thermal load depends on the prior and new environment, and the new activities. A simulation model, based on a modified version of two-node physiological model of thermal sensation, was developed and validated with experimental data as part of ASHRAE Research Project RP-619 [Jones et al. 1993, Jones et al. 1994]. The new model allows for simulating the thermal sensation of different parts of the body, and to reflect the clothing coverage and the environmental exposure. The new model was combined with a finite-difference model of transient heat and moisture transfer through clothing. The model predicts also the transient nature of heat and moisture loads generated by people on their surroundings.

Capability to model the processes listed in Task 1a
These models take into account the relationship between some relevant parameters affecting the dynamic behavior of internal heat sources.

Accuracy
Theoretically, these models could provide better results, if adequate data are available.

Ease of implementation
Models are available, and the source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
No information available. However, since only a few nodes are used, the computations are expected to be quite fast.

Flexibility for future improvements
The model should integrate better experimental coefficients for convection and radiation heat transfer within the luminaires. It also needs better correlations between the operating characteristics of luminaires.

Interoperability with other tools
It can be used as a reference model to calculate the correction factors, the correlation coefficients, and the response factor coefficients to be used by simpler models. It can also be integrated within the next simulator.

19-BS

Time-dependent moisture generation by internal sources, and moisture absorption/desorption processes

In all energy analysis programs, the moisture balance for room air is calculated at successive time-steps using the moisture generation rate, and the moisture introduced and evacuated from the space through HVAC systems, natural ventilation, and infiltration/exfiltration. The moisture absorption/desorption processes, which normally take place in the building materials or furniture, is neglected. However, several models have been developed for this process, and some of them are presented below.

Kerestecioglu et al. [1990a] used the concept of "effective penetration depth (EPD)" to simulate the simultaneous heat and mass transfer in building components. In the early stages of drying of a wall, the moisture content of the inner regions remains unaffected by the environmental conditions. However, a thin layer close to the surface behaves dynamically and losses moisture to the environment. If the specimen is left in the same environment for a long time, the inner regions of the specimen are also affected. However, for short periods of the order of a few hours the EPD concept can be used. The effective penetration depth for thermal and moisture interactions are empirically determined. For each thermal zone, a set of equations is obtained: (i) one energy balance equation for room air, (ii) one moisture balance equation for room air, (iii) n_1 energy balance equations for furniture, (iv) n_2 energy balance equations for envelope, and (v) $n_1 + n_2$ moisture balance equations for furniture and envelope.
Kerestecioglu et al. [1990b] presented the "evaporation and condensation" theory, a set of spatially distributed equations for modelling combined heat and moisture transport in building materials. The energy and moisture balance equations for the room air are written in terms of (i) the heat and moisture generation, (ii) the infiltration load, and (iii) the heat and moisture taken or released by the solid elements, which are defined in terms of convective heat and mass transfer coefficients. Since the problem is nonlinear, the computer model uses fixed-point and Newton-Raphson iteration schemes. The authors concluded that the theory must be verified with experiments, and experimental transport coefficient data, as a function of temperature and moisture content, are required. The author mentioned that the combined heat and moisture transport is a highly nonlinear phenomenon, it is possible to link this model with an analysis software based on finite-difference or finite-element methods. If the analysis software is based on transfer function, the problem of coupling with this model is more difficult.

Wong and Wang [1990] proposed a model of simultaneous heat and moisture transfer between the solid components and room air, based on the following assumptions: (i) liquid flow is induced by capillary flow and concentration gradients; vapor diffusion is induced by vapor pressure gradients; (ii) during the transport process, the moisture content, the partial vapor pressure, and the temperature are always in equilibrium at any location in the building materials; (iii) for moisture content larger than saturated sorptional content, the vapor pressure is equal to the saturation value; (iv) vapor pressure gradients can be determined from moisture contents by means of sorption isotherms; (v) Fick's law applies; (vi) all mass and heat transfer coefficients are constant; (vii) building materials are homogeneous; (viii) one-dimensional heat transfer through building material is considered; (ix) air infiltration through building material is negligible.

A mathematical model developed by TenWolde [1994] calculates the indoor relative humidity as a function of occupancy, ventilation, and moisture storage characteristics of a house. The model is based on a simple mass balance, including terms for moisture generation Q_g, sorption Q_a, ventilation Q_v, and condensation Q_v. The moisture sorption rate is positive for adsorption (wetting) and negative for desorption (drying). The moisture loss by vapor diffusion through the exterior envelope is neglected because is generally small compared to ventilation losses. The moisture sorption is proportional with $k \cdot (\phi_R - \phi_{R,\tau})$, where k is the sorption constant related to the exposed area of hygroscopic materials, ϕ_R is the room relative humidity, $\phi_{R,\tau}$ is the history of room relative humidity (exponentially weighted back average), and τ is the moisture storage time constant of the room air. The ventilation term is proportional with the difference between the water vapor pressure in exhaust air and the water vapor pressure of outside air entering the house. The condensation takes place, if the surface temperature is below the dew point of room air. The condensation term is proportional to a mass transfer coefficient (set to a default value) and the difference between the room water vapour pressure and the saturation vapor pressure at cold surface temperature. If the surface temperature is above the dew point temperature, but the surface is still wet from previous condensation, evaporation occurs, and the model calculate the rate of drying. By combining all terms, a closed-form model of the room relative humidity is obtained.

Capability to model the processes listed in Task 1a
These models are able to simulate the moisture absorption/desorption processes.

Accuracy
Accuracy depends on the quality of correlation coefficients, developed for different building materials.

Ease of implementation
Models are available, and the source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the moisture balance equation for indoor air. The main problem consists in solving the large system of simultaneous equations.

CPU requirements
High CPU requirements.

Flexibility for future improvements
Since these models are based on the first principles, they are flexible for improvements.

Interoperability with other tools
They can be used as reference models to calculate the correction factors, the correlation coefficients, and the response factor coefficients to be used by simpler models. They can also be integrated within the next simulator.

20-BS

Time-dependent contaminant generation by internal sources, and contaminant absorption/desorption processes

Evaluation of the concentration of contaminant in the room air became an important issue, closely related to the energy performance of a house. It has to be simulated if the operation of HVAC system is controlled by the air quality sensors to prevent high concentration of pollutants in the room air, or if the user evaluates the impact of different construction materials (e.g., paintings, carpets) on the air quality and energy consumption. Under these conditions, the contaminant balance for room air is calculated using: (i) the contaminant generation rate, (ii) the contaminant concentration introduced and evacuated from the space through HVAC systems, natural ventilation, and infiltration/exfiltration, and (iii) the contaminant absorption/desorption processes, which normally take place in the building materials or furniture. A discussion about these models is presented below.

Axley [1991] presented three families of absorbtion dynamics models based on fundamental principles, and assuming constant emission models:

- in the first model, a single control volume was used for the zone air and absorbent (e.g., wall); it was assumed that the absorbate concentration within the absorbent is uniform, and the zone-air remains, at all times, in equilibrium with the absorbent; this assumption is valid when the absorbent is well distributed in the zone, and the zone-air concentration are changing very slow; the time variation of zone-air concentration is calculated, and then the absorbent concentration response;
- in the second model, the zone air and the absorbent were supposed to occupy separate control volumes linked by boundary layer mass transport; it was assumed, again, that the absorbate concentration within the absorbent is uniform, and the air at the surface of the absorbent remains, at all times, in equilibrium with the absorbent; this assumption is valid when the boundary layer diffusion is slow relative to porous diffusion; the time variation of the zone and near-surface air-phase concentrations is calculated, and then the absorbent concentration response;
- in the third model, the absorbent was sliced up into multiple control volumes to enable the modelling of porous diffusion transport; the zone-air is supposed to occupy another individual control volume; the Finite Element Approach was used for writing the mass balance for the absorbent slices: (i) the form of spatial variation of concentration is assumed for each slice, and (ii) the one-dimensional convection-diffusion-adsorption equation was written.

These models can be integrated with existing multi-zone contaminant dispersal analysis methods to model the local sorption dynamics.

Haghighat et al. [1994] review the existing models, which can be classified into two groups: empirical models [Tichenor 1989, Colombo and Bortoli 1992] and physical models [Dunn and Tichenor 1988, Guo and Tichenor 1992]. The coefficients of empirical models do not have always a physical meaning. The available models do not account explicitly for several factors with an important impact. Physical models reflect the mechanisms involved in the process, but sometimes require assumptions which might neglect some important factors. In addition, the physical models require measurements to evaluate some coefficients.

Capability to model the processes listed in Task 1a
These models are able to simulate the contaminant absorption/desorption processes.

Accuracy
Accuracy depends on the quality of correlation coefficients, developed for different building materials.

Ease of implementation
Models are available, and the source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the contaminant balance equation for indoor air.

CPU requirements
High CPU requirements.

Flexibility for future improvements
Since these models are based on the first principles, they are flexible for improvements.

Interoperability with other tools
They can be used as reference models to calculate the correction factors, the correlation coefficients, and the response factor coefficients to be used by simpler models. They can also be integrated within the next simulator.

21-BS

Condensation on interior surfaces of the envelope components (e.g., wall, roof, window, ceiling, floor)

Condensation occurs when the water vapor comes in contact with a surface that has a temperature lower than the dew point temperature corresponding to local moist air [ASHRAE 1993]. Therefore, the model used to evaluate the risk of surface condensation must predict the surface temperature, and the dew point temperature of airflow closed to that surface. If the room air is simulated by a single node, assuming a complete mixing, or the cold surface is simulated by a single node (as in then case of one-dimensional heat transfer), the evaluation of condensation risk is likely to be inaccurate.

The mathematical models able to evaluate the surface temperature, presented in section 1-BS, 2-BS (for opaque walls), 3-BS (for transparent elements), and 5-BS, 6-BS (for below-grade building components) are potential candidates, if the local dew point can be calculated.

ASHRAE committee TC 4.5 initiated the research project SPC 142, to validate some computer models for simulating the heat transfer and temperature distribution on the indoor glass surface of seven types of insulating glazing units. The temperature distribution was used to predict the condensation potential. Zhao et al. [1996] used two different classes of computer programs:

- the FIDAP computer model [FDI 1993] was used to model convective, radiative, and conductive heat transfer within each glazing unit, subject to a prescribed set of convective and radiative boundary conditions; the governing fluid flow equations along with radiative heat transfer equations for non-participating media are solved simultaneously through an iterative approach;
- the THERM [Finlayson et al. 1995] program was used to model the two-dimensional conduction-only heat transfer in the frame and edge-of-glass regions; since the model does not simulate the convective heat transfer in the glazing cavity, the effective conductivity k_{eff} approach was used, where the corresponding value was obtained from the one-dimensional heat transfer analysis with the WINDOW program [WINDOW 4.1 1994]; the locally varying surface heat transfer coefficients were defined on boundaries inside the cavity by using the "variable-h condensation resistance model" developed the same team;

The following boundary conditions were used:

- zero heat flux on the top and on the bottom surfaces, simulating the contact with an insulated surround panel;
- constant surface heat transfer coefficients on the exposed indoor and outdoor surfaces, and constant indoor and outdoor air temperatures.

The results indicated that the THERM/WINDOW models gave good predictions of overall U-value, but less accurate results for the sill and head edge-of-glass sections. This is due to the inability of conduction-only models to account for the convective movement that occurs at the top and bottom of glazing units. The use of "variable-h condensation resistance model" improves the results. The authors concluded "it is necessary to work on the development of computational tools that would incorporate full mathematical heat transfer models that could still reliably be used in everyday engineering practice."

22-BS

Condensation within the envelope components (e.g., wall, roof, window, ceiling, floor)

Mathematical models presented in sections 7-BS and 8-BS can be used for simulating this phenomenon.

23-BS

Convection heat transfer between the indoor air and the interior surfaces (e.g., walls, windows, floors, lighting fixtures)

The convection heat transfer is evaluated by multiplying the convection heat transfer coefficient and the temperature difference between the surface temperature and the room air temperature. Most energy analysis programs use a simplified approach [Gadgil et al. 1980]:

- the convection heat transfer between the indoor air and the interior surfaces is evaluated using a constant convective coefficient, one single value for all locations on the surface;
- the room air has a homogeneous, uniform temperature; it is represented by a single node.

This simplified approach is a part of the overall approach used in most models for energy analysis. The common elements of this overall approach are the following: (i) one-dimensional conduction through building elements, (ii) one single node for modelling the room air temperature, and (iii) solar radiation and radiative component from internal sources (e.g., lights, equipments) are uniformly distributed over the entire floor or over the interior surfaces.

The models used to evaluate the convection heat transfer coefficients can be classified into: (i) correlation-type models, (ii) constant surface-averaged coefficient models, and (iii) CFD-based models.

<u>Correlation-type models</u>

The heat transfer coefficient for natural convection can be calculated using the Nusselt (Nu), Grashof (Gr), and Prandtl (Pr) dimensionless numbers [ASHRAE 1993, Holman 1990, Curcija and Goss 1993]:

$$Nu = a \cdot Ra^{b}$$

where $Nu = h \cdot L/k$; L is the characteristic dimension defined in terms of geometry of the problem, and the location of node where the convective heat flow is evaluated; Ra is the Rayleigh number ($Ra = Gr \cdot PR$);

$$h = k \cdot L^{-1} \cdot a \cdot Ra^{b}$$

<u>Capability to model the processes listed in Task 1a</u>

These models are derived from experimental data, and are able to simulate the convection heat transfer. They can be used to evaluate the local and global convection coefficients.

Accuracy
Good accuracy.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
Better correlation coefficients can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models.

Constant surface-averaged convection coefficient
In the case of air moving along a vertical or horizontal plate, the following simplified relationships are recommended for use:

$h = a \cdot (\Delta T/L)^b$ for laminar convection, and

$h = a \cdot (\Delta T)^b$ for turbulent convection.

where $\Delta T = T_{wall} - T_{air}$, L is the vertical or horizontal dimension of plate.

In other models, the convection and radiation heat transfer coefficients are integrated within the inside surface coefficient.

Spitler et al. [1991] presented results of an experimental investigation of convective heat transfer in enclosure, where the average film coefficient for each surface is defined as follows:

$$h_c = \frac{q}{T_{surface} - T_{return}}$$

The return air temperature was found to be the best choice for a reference temperature, rather than the room air temperature. The correlation for convective coefficient h has the following form:

$$h = a + b \cdot J_n$$

where J is the air jet momentum number. This approach avoids the use of CFD models to determine the room airflows. These correlations were implemented in the BLAST program, and the results indicated the heat transfer coefficients obtained for ceiling diffusers are significantly different of the published values at low flow rates.

Capability to model the processes listed in Task 1a
These models are derived from experimental data, and are able to simulate the global convection heat transfer.

Accuracy
Acceptable accuracy. So far, these models were the most used for energy analysis in buildings.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
Better values can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models.

CFD-based models

Alamdari and et. [1986] compared the surface-averaged convection coefficients obtained from (i) a finite-domain program, (ii) from an intermediate-level model based on the known characteristics of turbulent air-jets, and (iii) the design recommendations given by ASHRAE and CIBSE handbooks. They found a surprisingly good agreement between the values recommended by CIBSE guide with those obtained from the finite-difference model. They wrote: "this was only a fortuitous occurrence, that does not justify the use of these guidelines for enclosures employing other mechanical-ventilation systems."

Curcija and Goss [1993] developed a finite element model, using the FIDAP computer program, version 4.51 [FDI 1993], to model the impact of different fenestration geometries on the indoor surface convection heat transfer coefficient. The new fenestration systems introduced complex two- and three-dimensional effects, for which the one-dimensional analysis modified by a frame factor is less appropriate. The two-dimensional laminar natural convection heat transfer was modelled using the equations of the conservation of mass and energy. The results indicated that there is a large depression in the value of local convective heat transfer coefficient due to either a stagnation flow approaching the frame step or a recirculation flow right after the frame step. Therefore, it is not possible to develop a single local convective coefficient correlation that would accurately describe the heat transfer for any type of fenestration geometry at any location on the indoor surface. However, it is possible to achieve this objective by separating the fenestration into its vertical and horizontal surfaces. The value of convective heat transfer coefficient away from the edge-of-glass region is similar to that obtained for flow over a flat plate. The edge-of-glass region is exposed to a stagnated fluid flow that lowers the heat transfer rate, and therefore reduces the convective coefficient. In addition, the edge-of-glass indoor surface temperature is reduced, which may cause condensation and freezing in cold climate. The results were compared with the constant value of convective coefficient used by WINDOW 4.1 program [1994].

Capability to model the processes listed in Task 1a
These models are able to simulate the local convection heat transfer.

Accuracy
Accuracy depends on the discretization approach and the methods used for solving the system of simultaneous equations.

Ease of implementation
Models are available, and the source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the heat and mass balance equations for indoor air. The main problem consists in solving the large system of simultaneous equations.

CPU requirements
High CPU requirements.

Flexibility for future improvements
Since these models are based on the first principles, they are flexible for improvements.

Interoperability with other tools
They can be used as reference models to calculate the correction factors, the correlation coefficients, and the response factor coefficients to be used by simpler models.

24-BS

Convection heat transfer between the external surfaces (e.g., walls, windows, roof, doors) and the outdoor air

The convection heat transfer is evaluated by multiplying the convection heat transfer coefficient and the temperature difference between the surface temperature and the room air temperature. The models used to evaluate the convention heat transfer coefficients can be classified into: (i) correlation-type models, and (ii) constant surface-averaged coefficient models.

Correlation-type models

Most energy analysis models use a simplified approach, with a combined film coefficient (convection and radiation), which can take either a constant value [ASHRAE 1993, Achterbosch 1985] or variable in terms of wind speed [DOE-2 1982]:

$h_o = a + b \cdot WIND + c \cdot WIND^2$, where a, b, and c are defined in terms of surface roughness.

Walton [1983] used in his model both a simple algorithm, and a detailed algorithm. The simple algorithm uses a combined convective and radiant coefficient in terms of wind speed and surface roughness:

$$h_o = a + b \cdot WIND + c \cdot WIND^2$$

The detailed method combines the natural h_n and forced h_f convection coefficients in one single convection coefficient h_c:

$$h_c = h_n + h_f \cdot W_f \cdot R_f$$

where $h_f = a \cdot \sqrt{(C/A)} \cdot \sqrt{(V_{az})}$

V_{az} is the free stream wind velocity, A is the surface area, and C is the surface perimeter;

W_f = 0.5 for leeward surfaces, 1.0 for windward surfaces; R_f is a roughness multiplier;

$h_n = a \cdot [abs(T_o - T_{air})/b - abs(\cos\phi)]$;

ϕ is the wind incidence angle, and T_o the surface temperature.

The Kimura sixth-floor model:

$$h_c = a \cdot WIND_n$$

is used to calculate the exterior convection coefficient in the WINDOW program [Yazdanian and Klems 1994]. This formula leads to different constant values of h_c for the windward and leeward directions when the wind speed becomes negligible. Therefore, this model suggests that in the absence of wind the convection coefficient on the windward and leeward surfaces are different. The predictions of DOE-2 model and Kimura sixth-floor model are in disagreement with measurements in the MoWiTT field facility. The Kimura fourth-floor model, which has the following form:

$$h_c = a + b \cdot WIND$$

gives the same convection coefficient on the windward and leeward surfaces in the absence of wind. Its predictions and the measurements yield approximately parallel curves, but Kimura's prediction is about 5 $W/m^2 \cdot K$ higher.

Based on measurements, Yazdanian and Klems [1994] developed a new model, called MoWiTT model, which combines the natural and forced convection:

$$h_c = \sqrt{\{[C_t \cdot \Delta T^{1/3}]^2 + [a \cdot WIND^b]^2\}}$$

Taki and Loveday [1996] measured the effect of a rectangular framework on external convection heat transfer from building facades. They found that the convection coefficients are significantly affected by the framework compared with those of either a flat, undisturbed surface, or with those of a surface comprised of vertical-only protrusions. In general, small recess depths increase the convection coefficient h_c compared with the flat surface, but progressive deepening of the recess can generally reduce the value for h_c to less than that for the flat surface. The observed effects are considered to be caused by boundary layer separation at the framework, creating a region of re-circulating flow, followed by re-attachment further downstream. Correlations of the following form were obtained, which are valid for smooth surface elements (glazed, metallic or acrylic) at or near 0.8 m wide by 0.5 m high, surrounded by a rectangular framework, and within the central region of the wider facades of multi-storey cuboidal buildings between the 4th and 8th floor levels:

$$h_c = a + b \cdot WIND + c \cdot D + d \cdot D^2 + e \cdot D_2 \cdot WIND + f \cdot D \cdot WIND$$

where D is the recess depth.

Capability to model the processes listed in Task 1a
These models are derived from experimental data, and are able to simulate the convection heat transfer. They can be used to evaluate the local and global convection coefficients.

Accuracy
Accptable accuracy.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the heat balance equation for the indoor air, and the heat transfer models for the exterior envelope.

CPU requirements
Fast calculations.

Flexibility for future improvements
Better correlation coefficients can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models, to evaluate the heat exchange at the outside surface of envelope.

Constant surface-averaged coefficient models

The combined-film coefficient is given for design winter conditions (h_o = 34.0 W/m^2·K) and for design summer conditions (h_o = 22.7 W/m^2·K) [ASHRAE 1993].

Capability to model the processes listed in Task 1a
These models are derived from experimental data, and are able to simulate the average convection heat transfer.

Accuracy
Acceptable accuracy. So far, these models were the most used for energy analysis in buildings.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, through the heat balance equation for the indoor air, and the heat transfer models for the exterior envelope .

CPU requirements
Fast calculations.

Flexibility for future improvements
Better values can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models, to evaluate the heat exchange at the outside surface of envelope.

25-BS

Longwave radiation exchange between the interior surfaces (e.g., walls, windows, floors, heaters, lighting fixtures)

The net radiant heat flux absorbed by surface i from other surfaces j, separated by a non-absorbing media, is given by [Walton 1983]:

$$q_{R,i} = \Sigma F_{i,j} \cdot (T^4_j - T^4_i)$$

where $\Sigma F_{i,j}$ is the interchange factor between surface i and all other surfaces j. Since this relation is difficult to calculate for rooms with arbitrary geometry or furniture, and the term T^4 leads to non-linear equations, a simpler form is obtained by linearization:

$$q_{R,i} = h_{R,i} \cdot (T_1 - T_2)$$

where $h_{R,i}$ is the radiative coefficient; this coefficient can be constant, independent of temperatures, or variable, and in this case is calculated using the mean radiant temperature models presented below. The radiosity method can also be used to evaluate the radiant interchange among surfaces.

Constant surface-averaged coefficient models

The radiative coefficient can have one value for all interior surfaces, or it can differentiate between surfaces.

Capability to model the processes listed in Task 1a

These models are derived from experimental data, and are able to simulate the global radiation heat transfer between interior surfaces.

Accuracy

Lower accuracy than other models of radiation heat transfer.

Ease of implementation

Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods

They can be coupled with HVAC models using either the "simultaneous" or the sequential approach. In the first case, the main problem consists in solving the large system of simultaneous equation.

CPU requirements
Fast calculations.

Flexibility for future improvements
Better values can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models or thermal comfort models.

Mean radiant temperature model

The "mean radiant temperature" concept was introduced to eliminate the complexity of view factor calculations [Carroll 1980, Carroll and Clinton 1980]: each surface interacts with the mean radiant temperature (MRT), which includes the radiant effect of all other surfaces, instead of interacting to each one separately. The radiant heat flux $q_{R,i}$ between the surface i and all other surfaces in the room can be calculated as:

$$q_{R,i} = h_{R,i} \cdot (MRT_i - T_{s,i})$$

where $h_{R,i}$ is the "radiative conductance" of surface i, $T_{s,i}$ the temperature of surface i,

$$H_{R,i} = 4 \cdot \sigma \cdot T^3_{avg} \cdot [1/F_i + (1-\epsilon_i)/\epsilon_i]$$

T_{avg} can be considered equal to the MRT of the room for the total period of simulation

ϵ_i is the surface emmitance, and F_i is the "MRT view factor" which can be solved iteratively:

$$F_i = 1/[1 - S_i \cdot F_i/(\Sigma S_j \cdot F_j)]$$

The mean radiant temperature MRT_i, as "seen" by the surface i, can be approximated by a surface-weighted average temperature of all other surfaces j:

$$MRT_i = \Sigma h_{R,j} \cdot T_{s,j} \cdot S_{s,j} / \Sigma h_{R,j} \cdot S_{s,j}$$

Walton proposed "that the radiant interchange in a room can be adequately modeled by assuming that each surface "i" radiates to a fictitious surface "fic" which has an area, emissivity and temperature giving about the same heat transfer from the surface as in the real muli-surface case.." [Sullivan et al. 1982]. The net radiant heat flux is given by:

$$q_{R,i} = 4 \cdot \sigma \cdot F_{i\text{-}fic} \cdot T^3_{avg} \cdot (T_{fic} - T_i) = h_{R,i} \cdot (T_{fic} - T_i)$$

where $T_{avg} = (T_{fic} + T_i)/2$

$$T_{fic} = \Sigma(S_j \cdot \epsilon_j \cdot T_j)/\Sigma S_j \cdot \epsilon_j$$

Walton [1983] developed the TARP program using the concept of mean radiant temperature as developed by Carroll; in order to simplify the calculations he used either $T_{avg} = (T_{fic} + T_i)/2$, or $T_{avg} = 23°C$.

Sullivan et al. [1979] used the modified thermal balance technique, which is similar to Walton's mean radiant temperature concept, where:

$$T_{fic} = \Sigma(S_j \cdot T_j)/\Sigma S_j$$

Achterbosch et al. [1985] used a linearized form:

$$q_{R,i} = h_{R,i} \cdot (T_1 - T_2)$$

where $h_{R,i} = 4 \cdot \sigma \cdot F1,2\ T^3_{avg}/(1/\epsilon_1 + 1/\epsilon_2 - 1)$

$$T_{avg} = (T_1 + T_2)/2$$

Capability to model the processes listed in Task 1a
These models can simulate the radiation heat transfer between interior surfaces in rooms, without calculating the angle factors.

Accuracy
Acceptable accuracy, based on the assumption of grey and isothermal surfaces.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models in modelling systems such as radiant floor system, using either the "simultaneous" or the sequential approach. In the first case, the main problem consits in solving the large system of simultaneous equation.

CPU requirements
Fast calculations.

Flexibility for future improvements
.

Interoperability with other tools
They can be coupled with other models such as thermal comfort models.

Radiosity method

The net-radiation method, called also radiosity method, express more accurately the radiant interchange among surfaces within a space [Sullivan et al. 1979]. The radiosity is the sum of energy emitted and the energy reflected (when no energy is transmitted through the surface):

$$J = \epsilon \cdot E_b + \rho \cdot G$$

where J is the total radiation which leaves a surface per unit time and unit area, and G is the total radiation incident on surface per unit time and unit area [Holman 1990, ASHRAE 1993]. A system of N simultaneous equations is obtained for a room with N surfaces:

$$J_i = \epsilon_i \cdot \sigma \cdot T^4_i + (1-\epsilon_i) \cdot \Sigma(F_{i,j} \cdot J_i)$$

These equations are linear if the surface temperatures T_i are known from the previous time step. If not, a non-linear set of equations, including the heat balance at each surface, must be solved. Once the radiosity J_i is known, the net radiant energy lost by each surface is calculated:

$$q_{R,i} = (E_b - J_i) \cdot \frac{\epsilon_i \cdot A_i}{(1-\epsilon_i)}$$

Noboa et al. [1994] developed a model for analyzing the radiant heat transfer in attics containing dusty radiant barriers, using the radiosity method. Saunders and Andrews [1987] developed a model for simulating the transient heat transfer within a room, with radiant and convective heat exchange between a radiant heating panel and room surfaces. They used the radiosity method, where the radiant heat flux leaving a surface is calculated in terms of surface temperature at the previous time step. The convective heat flux on each surface is calculated using the temperature at the previous time step.

Moujaes [1996] developed a numerical model to study the impact of passive radiation barriers in ventilated attics. The transient heat transfer through solid walls is one-dimensional. The radiation heat exchange is evaluated using the non-linear representation. The convective coefficients are obtained using the Nusselt number for natural or forced convection. Therefore, the convective coefficient is not constant. An implicit finite-difference method is used to discretize the field, and a system of 22 equations is solved at each time step.

Winiarski and O'Neal [1996] developed a quasi-steady-state model to predict the heat transfer at all attic surfaces, in order to estimate the reduction in cooling load that would occur with a radiant barrier. The net radiant heat transfer was evaluated using the radiosity method.

Capability to model the processes listed in Task 1a
They can be used to model the radiation heat transfer between interior surfaces in rooms.

Accuracy
More accurate than other models of radiation heat transfer.

Ease of implementation
Models are available, and are easy to implement.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, using either the "simultaneous" or the sequential approach. In the first case, the main problem consits in solving the large system of simultaneous equation.

CPU requirements
Higher CPU requirements than other radiation models.

Flexibility for future improvements
High flexibility.

Interoperability with other tools
They can be coupled with other models such as CFD models. They also can be used as reference models to calculate the correction factors or the correlation coefficients for simpler models.

26-BS

Longwave radiation exchange between the external surfaces (e.g., walls, windows, roof, doors) and the sky vault

This heat transfer phenomenon should be properly evaluated since in the summer it can play a positive role for passively cooling the building by radiation to the night sky. During the winter, the additional heat loss to the night sky must be accounted for in modelling the energy consumption for heating. One type of models uses a constant radiant heat flux, and another type uses the concept of equivalent radiative temperature. They are presented below.

<u>Constant radiant heat flux</u>

One model uses a constant radiant heat flux q_R = 20 Btu/hr·ft^2 between an horizontal surface and the clear sky [DOE-2]. If the sky is covered by clouds, there is no radiation exchange. For partial cloud cover, a linear interpolation is used:

$$q_R = 2\cdot(10 - \text{CLDAMT})$$

where CLDAMT is the clound amount. For a vertical surface, which sees both sky and ground, the net radiant heat flux is assumed to be zero.

<u>*Capability to model the processes listed in Task 1a*</u>
The model can be used for modelling the radiation with the sky vault.

<u>*Accuracy*</u>
Lowest accuracy.

<u>*Ease of implementation*</u>
The model is easy to implement.

<u>*Ability to integrate HVAC calculation methods*</u>
It can be coupled with HVAC models, through the heat balance equation of indoor air, and the heat transfer models for the exterior envelope.

<u>*CPU requirements*</u>
Fast calculations.

<u>*Flexibility for future improvements*</u>
The model could be improved by including correlation-type models, instead of a fixed heat flow.

<u>Interoperability with other tools</u>
It can be coupled with other models such as CFD models, to evaluate the heat exchange at the outside surface of envelope.

<u>Equivalent radiative temperature</u>
The sky may be considered as a global blackbody defined in terms of an equivalent radiative temperature T_c [Berger et al. 1984]. The general expression for T_c is the following:

$$T_c = T_{air} \cdot \epsilon^{o.25}$$

where the ground level air temperature T_{air} is corrected for dew point temperature T_{dp}, and the sky emissivity has the general form:

$$\epsilon = a + b \cdot T_{dp}$$

The atmosphere has a relatively high transmittance to thermal radiation in the 8-14µm "window", but behaves nearly like a blackbody at wavelengths outside this window [Kimball 1985]. The temperature of building surface is calculated from the heat balance using the radiant heat flux from the surface and the sky, within and outside the window. The non-linear equations in T^4 are linearized using an old estimate of temperature:

$$q_R = \epsilon \cdot f \cdot \sigma \cdot T^4 \simeq R_0 + \delta \cdot (T - T_0)$$

where $\delta = 4 \cdot \epsilon \cdot f \cdot T^3_0$. The surface temperature is found through an iterative process, starting with an initial guess for T_0.

For an isolated house, the view factor between a tilted surface and the sky is $F_{sky} = (1+\cos\theta)/2$, and between a tilt surface and ground is $F_{ground} = (1-\cos\theta)/2$ [Walton 1983, Kreider and Rabl 1994, McQuinston and Parker 1994].

Mills (1997) developed a numerical model to study the radiative effects of a group of buildings on a single house. All buildings were identical and were symmetrically situated with respect to others. He used two important parameters to evaluate the impact of other buildings: (i) the solar exposure, which is direct solar radiation intercepted by a building, and (ii) the sky view factor, which was obtained as the weighted average of the sky view factors of each facade. Next developments will include diffuse solar radiation and simplified tree forms.

<u>Capability to model the processes listed in Task 1a</u>
The model can be used for simulating the radiation with sky vault.

Accuracy
More accurate than the previous model.

Ease of implementation
Model is available, and are easy to implement.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation of indoor air, and the heat transfer models for the exterior envelope.

CPU requirements
Fast calculations.

Flexibility for future improvements
High flexibility.

Interoperability with other tools
It can be coupled with other models such as CFD models, to evaluate the heat exchange on the outside surface of houses.

27-BS

Longwave radiation exchange between the external surfaces and the surroundings

Calculation methods presented in sections 25-BS and 26-BS can be used to evaluate the longwave radiation exchange between the external surfaces and the surroundings. Models discussed in section 31-BS can also be used to evaluate the obstruction effect of houses, buildings, and trees.

28-BS

Shortwave radiation on the external surfaces

The total solar irradiation on a surface normal to the sun's rays is composed of (i) normal direct irradiation, (ii) diffuse irradiation, and (iii) reflected irradiation.

If solar data are not available on the weather file, the direct normal irradiation and the diffuse irradiation are obtained using the clear sky conditions, corrected for cloudy conditions [DOE-2]. McQuinston and Parker [1994] presented the ASHRAE Clear Sky model corrected by Galanis and Chatigny [1986].

Reilly et al.[1994] presented a survey of research directed toward the evaluation of impact of the surroundings on the solar loads in buildings. They used the ray-tracing approach to account for specular and diffusive reflection to modify the Radiance program, and then to couple it with the DOE-2 software [Dunne et al. 1995]. The solar radiation absorbed or transmitted by a surface is average over a set of sampling points on that surface, and then transformed into reflection factors, which are then transferred to the energy analysis program.

29-BS

Transmission of solar radiation through windows and other transparent envelope components

The solar heat gain through a window is given by:

$$q = I_t \cdot F$$

where I_t is the total incident solar radiation, and F is the Solar Heat Gain Coefficient (SHGC) [ASHRAE 1993]. The last term is composed of: (i) the solar radiation transmitted through glass and absorbed by interior elements, and (ii) the solar radiation absorbed by the glazing system, and re-emitted to the indoor through convection and radiation heat transfer. The Solar Heat Gain Coefficient is a characteristic of glazing type, and can be calculated as the area-weighted values corresponding to the center of glass ($SHGC_{cg}$), the edge of glass ($SHGC_{eg}$) and the frame ($SHGC_{fr}$), including the effect of space dividers. However, Carpenter and Baker [1992] found that the frame and edge-of-glass heat transfer has a little impact on the SHGC for windows with wood frame. In the case of windows with aluminium or thermally broken aluminium frames the impact is less than 5%. Therefore, the centre-of-glass SHGC can be applied to full view area of window. Since SHGC is function of incidence angle, the solar heat gain can be calculated using the SHGC at normal incidence angle and an off-normal incidence angle factor F_i:

$$q = SHGC_n \cdot F_i \cdot I_t$$

The following methods can be used to evaluate the solar heat gain through windows: (i) shading coefficient method, (ii) CLTD/SCL/CLF method, (iii) modified bin method, (iv) models based on Fresnel's equations, and (v) GLSIM model.

Shading coefficient method

The shading coefficient was introduced to simplify the calculations. It was defined as either (i) the ratio between the Solar Heat Gain Factor of a fenestration and the Solar Heat Gain Factor of a reference glass (double-strength sheet glass with 0.86 transmittance, 0.08 reflectance, and 0.06 absorptance at normal incidence), or (ii) the ratio between the Solar Heat Gain Coefficient of the fenestration and the Solar Heat Gain Coefficient of a reference glass. The advantage of shading coefficient is that for single-pane clear and tinted glass, it is nearly constant with angle of incidence over a wide range of incidence angles [McCluney 1992]. For glazings with optical properties nearly constant over the solar spectrum, the shading coefficient is also almost constant with respect to changes in the spectral distribution of incident solar radiation. For some other glazing systems the shading coefficient may not be constant.

McCluney [1996] recommended that the use of constant shading coefficient should be abandoned for the calculation of solar heat gain for all fenestration systems with strongly spectrally selective glazings, and for those with a variable shading coefficient in terms of angle of incidence. For more exact evaluations, the wavelength-by-wavelength and angle-by-angle calculation method, as used by the WINDOW program, should be used. However, the program does not fully account for certain interferences and other effects in the layers of coated glasses.

Values of shading coefficients for interior shading devices such as venetian blinds and draperies are given by ASHARE [1993]. McCluney and Mills [1993] presented a simple calculation procedure for estimating the effect of vertical, planar, interior shades on the overall solar heat gain factor. The main assumption was that the solar radiation once absorbed by the shade remains in the room, since it is re-emitted at infrared wavelengths to which the glazing is generally opaque. There is also an effect on the radiant heat re-emitted by the glass, since the convection heat flow which was supposed to take place with the room air is partially limited to the space between the glass and the shade.

Capability to model the processes listed in Task 1a
The model can be used for simulating the solar radiation through most common windows.

Accuracy
Acceptable accuracy for most common glazing systems. Lower accuracy for spectrally selective glazings, and for those with a variable shading coefficient in terms of angle of incidence.

Ease of implementation
Model is available, and is easy to implement. Most manufacturers provide the shading coefficient data.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation for indoor air, if these models use the heat flow through windows. It does not evaluate the surface temperature.

CPU requirements
Fast calculations.

Flexibility for future improvements
The shading coefficient can be integrated as a correlation-type model in terms of the angle of incidence and the optical properties at normal incidence.

Interoperability with other tools
.

CLTD/SCL/CLF method

The CLTD/CLF method was initially developed as manual calculation procedure. Solar heat gains through windows are calculated in terms of shading coefficient SC and Solar Heat Gain Factor

SHGF [ASHRAE 1993, Kreider and Rabl 1994]. The instantaneous heat gain due to solar radiation through a window of surface S is given by:

$$q = SC \cdot SHGF$$

The contribution of solar radiation to the space cooling load is obtained by using the Cooling Load Factor CLF, which takes into account the thermal mass of building and how the solar radiation absorbed by materials is released.

$$Q = q \cdot CLF$$

Research performed under ASHRAE research project RP-472 led to a revised methodology, called CLTD/SCL/CLF method, which uses a new term the Solar Cooling Load (SCL), and the solar gain through glass is calculated as [Spitler et al. 1993a]:

$$q = SC \cdot SCL$$

Capability to model the processes listed in Task 1a
The model can be used for simulating the solar radiation through most common windows, using the shading coefficient.

Accuracy
Acceptable accuracy for most common glazing systems. Lower accuracy for spectrally selective glazings, and for those with a variable shading coefficient in terms of angle of incidence.

Ease of implementation
Model is available, and is easy to implement.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, if those models use the heat flow through windows. It does not evaluate the surface temperature.

CPU requirements
Fast calculations.

Flexibility for future improvements
The shading coefficient can be integrated as a correlation-type model in terms of the angle of incidence and the optical properties at normal incidence.

Interoperability with other tools
.

Modified bin method

The modified bin method was initially developed as a simplified energy analysis procedure [Kusuda 1981, Knebel 1983, Knebel 1995]. It is assumed that the solar contribution due to windows can be expressed as a linear function of outdoor temperature. It is calculated in terms of: (i) the maximum solar heat gain factor for the given orientation, (ii) the 24-hour sum of CLF for the orientation, (iii) the fraction of possible sunshine, and (iv) the shading coefficient. Calculation of solar contribution is performed for January and July, and then a linear correlation between peak design temperatures and the corresponding solar contribution is obtained. Some examples of energy analysis programs based on this method are: ASEAM2.1 [Ohadi et al. 1988, Ohadi et al. 1989] and BESA-Retrofit [1985].

Capability to model the processes listed in Task 1a
The model can be used for simulating the solar radiation through most common windows, using the shading coefficient.

Accuracy
The model has the same weaknesses as the shading coefficient model, presented before. In addition, it uses the assumption of linear relationship between solar radiation and outdoor air temperature, which is developed based on two design conditions (January and July).

Ease of implementation
Model is available, and is easy to implement.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Fast calculations.

Flexibility for future improvements
Some developments aimed to make this model able for simulating the thermal mass and solar gains. However, all improvements have tried to accommodate the overall approach of this method, which is not always possible without additional simplifications. Overall, it is less flexible than the numerical methods.

Interoperability with other tools
The model was mainly developed to evaluate the overall heat flow through a building element, without planning for future developments. If the overall heat flow is the only information required by other tools, then the models could be coupled.

Models based on Fresnel's equations

Walton [1983] used in his program two options for calculating the solar heat gain: (i) using the shading coefficient, which was recommended for steady-state calculations only, or when more detailed data are not available, and (ii) using Fresnel's equations along with a set of optical data provided by the user (e.g., normal transmittance and reflectance, index of refraction, extinction coefficient); the angle dependency is calculated by polynomial equations in terms of incidence angle; this second method is used for uncoated panes, reflective coatings, multi-pane windows, and interior shading devices.

Reilly et al. [1992] presented new algorithms for modelling the optical properties of windows, which were incorporated into the DOE-2 program. The total optical properties are calculated from glazing manufacturers optical data. The total solar, visible, and infrared properties for each glazing layer were used to find the total optical properties for the glazing system. The angular dependency of visible and solar transmittance and reflectance of homogeneous (uncoated) glazing layers was developed by Furler [1991], based on Fresnel's equations and Snell's law. The only necessary inputs are the reflectance R_0 and transmittance T_0 at normal incidence, and the glazing layer thickness d, from which the extinction index k is calculated for a given wavelength. The optical properties at normal incidence correspond to a variety of different values for wavelength and index k. It was found that the center wavelengths of the visible and solar spectrum produce effective extinction indices k, which yield reasonable approximations. Therefore, with these effective wavelengths and optical properties at normal incidence, it is possible to approximate the angular dependence of optical properties. The theoretical error of this approximation is less than 1.5%. Although this method uses the total optical properties, for glazings with a strong spectral dependence the spectral calculations should be used, wavelength by wavelength.

Winkelmann et al. [1993] compared two methods used by the DOE-2.1E program to calculate the solar heat gain through windows:

- the use of Window library, which was created by using the WINDOW 4.1 program, takes into account a highly accurate angular dependence; it increases by 50 to 100% the calculation time of LOADS block depending on number of windows;
- the use of shading coefficient is convenient at the conceptual design stage, but leads to an inaccurate angular dependence for multipane glazing; Reilly et al. [1992] found that the constant shading coefficient approach can overpredict the solar heat gain through a window at a given hour by as much as 30% compared with the first approach; the difference is most pronounced for low-transmissivity, highly absorbing glass.

Klems [1994a, 1994b] developed a new method of predicting the Solar Heat Gain Coefficient through complex fenestration systems involving non-specular layers such as shades or blinds. He combined measurements of bidirectional radiative transmittance and reflectance of each layer, performed with a scanning radiometer, with mathematical modelling. The final result is the total directional-hemispherical transmittance of the fenestration system, and the layer-by-layer absorptances, which later can be used to calculate the Solar Heat Gain Coefficient.

Wright [1995] compared the algorithms used by two public domain programs (VISION and WINDOW) for modelling the window solar heat gain, and concluded that most models are identical. The WINDOW program offers optional spectral and directional calculations. The heat transfer models presently differ because of the different correlations used to determine the convective heat transfer coefficients between glazings, and because of some minor differences in fill-gas properties.

Capability to model the processes listed in Task 1a
The model can be used for simulating the solar radiation through windows, using more detailed calculations and optical properties of windows.

Accuracy
More accurate than the previous models.

Ease of implementation
Model is available, and is easy to implement. Several databases are available from manufacturers.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Fast calculations.

Flexibility for future improvements
The model can accommodate new glazing types.

Interoperability with other tools
It can be coupled with other models such as energy analysis programs.

GLSIM model
The optical properties of special glazing (coated low-emissivity, sun reflective, or absorbing tinted glazing) are normally unknown. If the angle-dependent properties of clear glazing are used, the solar gain is described quite accurately up to an incidence angle of about 50°. For higher incidence angles, this approach leads to important errors. Pfrommer et al. [1994] presented a computer model called GLSIM (GLazing SIMulation), which can be used as a pre-processor for any detailed thermal simulation program. The first module calculates the incidence-angle-dependent radiation transfer through coated and tinted glazing. A matrix multiplication approach is used, in which every layer interface and every layer are represented by a separate matrix. The interface matrix describes the electric field amplitude at the interface between two layers. The layer matrix considers the reduction of the field amplitude during the radiation transfer through the layer. The second module calculates the transmittance and absorptance for diffuse sky radiation. The third module calculates the total amount of radiation entering a space by using hourly irradiation data. The authors found that predictions from detailed thermal simulation programs which use simple Fresnel-based recursive calculations to generate glazing transmission properties should be interpreted with caution in the case of special glazings. For instance, the

errors in estimating cooling load reach 19% in the case of tinted grey windows.

Capability to model the processes listed in Task 1a
The model can be used for simulating the solar radiation through special glazings, as well as through common windows.

Accuracy
More accurate than the previous model.

Ease of implementation
Model is available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Higher CPU requirements than for the previous models.

Flexibility for future improvements
High flexibility.

Interoperability with other tools
It can be coupled with other models, if they can share some common data.

30-BS

Shading of opaque envelope components caused by surroundings (e.g., houses, buildings, trees)

Models presented in section 31-BS (shading of transparent envelope components) could also be used in this section.

31-BS

Shading of transparent envelope components caused by exterior shading elements (e.g., overhang, side fins) or by facade obstructions (e.g., recessed windows)

The available models for evaluating the shading caused by exterior elements can be classified into the following three categories: (i) Direct Element Analysis model, (ii) models based on polygon clipping, and (iii) simple models of typical configurations.

Direct Element Analysis model

In this model, the window surface is divided into a matrix of small cells. If the center of each cell lies within the shade, which is defined by the vertices of the shadow cast, then the whole cell is considered to be shaded [DOE-2]. The total shaded area is approximated by adding the area of shaded cells. The accuracy of this procedure depends on the number of elements into which the wall surface is divided.

Capability to model the processes listed in Task 1a
The model can be used for simulating the shading caused by exterior elements.

Accuracy
The model can approximate the shaded area. Accuracy depends on the number of cells used to discretize the window surface.

Ease of implementation
Model is available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Acceptable CPU requirements.

Flexibility for future improvements
.

Interoperability with other tools
It can be coupled with models of insolation patterns on the inside surfaces or the visual comfort.

Models based on polygon clipping

Walton [1983] reduced the three-dimensional problem to a two-dimensional one, by projecting the shadow on the plane of window, considered to be the receiving surface. The area of the overlap between the polygons representing the shadow and the polygon representing the receiving surface is calculated. This area represents the shaded area of window.

Grau and Johnsen [1995] developed a model based on polygon clipping for accurate determination of all sunlit and shaded regions cast by any number of obstructing objects (e.g., overhangs, side-fines, buildings, trees) onto the solar exposed surfaces. The shading objects may be transparent to any degree, from opaque to completely transparent. For each window, the overall reduction factor is defined as the fraction of the direct solar radiation reaching the window, compared with the direct radiation that would reach the window without any shading, as is calculated as the area-weighted average of factors of transparency of each shading objects. A computer program called Xsun was developed based on this model, and it was used as pre-processor for the Danish thermal simulation program called "tsbi3."

Niewienda and Heidt [1996] developed a model to calculate the shaded area of an arbitrarily oriented surface generated by shading elements such as buildings, trees, overhangs and side-fines. The time schedules for transmittance through different shading elements can be defined by the user, which facilitates the evaluation of shading caused by vegetation. The reduction of diffuse radiation due to different types of obstacles is calculated by using the view factors. This model was used as a pre-processor for TRNSYS program [Schnieders et al. 1997].

Capability to model the processes listed in Task 1a
The model can be used for simulating the shading caused by exterior elements.

Accuracy
The model is more accurate that the previous model.

Ease of implementation
Model is available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
The model can accommodate different other types of exterior shading.

Interoperability with other tools
It can be coupled with models of insolation patterns on the inside surfaces or the visual comfort.

Simple models of typical configurations

McCluney [1986] developed an algorithm for calculating the unshaded fraction of a window shaded by a awning or arbitrary length and width and having sidewalls with lower edges making an arbitrary angle with respect to the horizontal. Awnings without sidewalls and simple horizontal overhangs were also included. The model was initially based on five possible shadow configurations. McCluney [1990] improved his previous model, by including the treatment of ground-reflected radiation, the evaluation of shadows crossing the top edge of the window. The new model contains presently nine shadow configurations.

Capability to model the processes listed in Task 1a
The model can be used for simulating the shading caused by some typical configurations of exterior elements.

Accuracy
The accuracy is high, since the model is based on ray tracing approach.

Ease of implementation
Model is available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
Algorithms for other types of exterior shading could be developed and integrated.

Interoperability with other tools
It can be coupled with models of insolation patterns on the inside surfaces or the visual comfort.

32-BS

Shading of transparent envelope components caused by surroundings (e.g., houses, buildings, trees)

Models presented in section 31-BS (shading of transparent envelope components) could also be used in this section.

33-BS

Room air movement and thermal stratification

Presently, the energy analysis programs assume that the indoor air of each zone is very well mixed, and therefore one single node, located in the centre of room, is representative for the indoor conditions. Walton [1982] underlined the fact that one of the major areas neglected by detailed energy analysis programs was the relationship between heat flow and air movement.

The mathematical models used for simulating the air movement in one room can be classified into the following categories [Megri et al. 1996]: (i) one-zone (single-cell) model, (ii) zonal models, and (iii) CFD models.

<u>One-zone (single-cell) model</u>
This model considers the air temperature and contaminant concentration is uniform within the zone, due to well mixing. All convective heat exchanges are modelled with respect to this single node. This approach could lead to over-estimation of the impact of a local, limited convection heat flow, on the room air balance. The vertical and horizontal air temperature gradients are also neglected. Other limitations of this approach concern the evaluation of thermal sensation and the risk of condensation, because the local important parameters (e.g., air velocity, air temperature, mean radiant temperature, relative humidity or dew point temperature) are not available; the calculations are performed with respect to the centre of room conditions.

> *<u>Capability to model the processes listed in Task 1a</u>*
> *The model can be used only for those cases if the assumption of well mixing of the indoor air is acceptable.*
>
> *<u>Accuracy</u>*
> *Acceptable accuracy. Most energy analysis programs use this model. However, the model over-estimates the impact of a local, limited convection heat flow on the room air balance.*
>
> *<u>Ease of implementation</u>*
> *Model is easy to implement.*
>
> *<u>Ability to integrate HVAC calculation methods</u>*
> *It can be coupled with HVAC models, through the heat balance equation for indoor air.*
>
> *<u>CPU requirements</u>*
> *Fast calculations.*

Flexibility for future improvements
It can integrate correlation-type models to link the conditions at different locations within the room with the centre-of-room conditions.

Interoperability with other tools
It can be coupled with other models such as air infiltration models.

Zonal models
These models consider the indoor air volume is divided into a number of elementary cells. They can be used to simulate the thermal stratification, and can be coupled with a thermal model.

Togari et al. [1993] and Arai et al. [1994] developed a simplified model for predicting vertical air temperature distribution in a room. They used a zonal model with two types of elementary zones:

- zones in contact with the exterior walls (the inside surface temperatures are known), for which the wall surface current model was applied; the heat flow between the zone (with temperature T_A) to the wall (with temperature $T_w<T_A$):

  $q = h_c \cdot A_w \cdot [T_A - T_w]$

  generates a descending air current along the cold wall, which has an average temperature T_{avge} equal to the weighted average of wall and zone temperatures:

  $T_{avge} = 0.75 \cdot T_A + 0.25 \cdot T_w$; finally the mass flow air rate is given by:

  $m = 4 \cdot h_c \cdot A_w / c_p$

- interior zones affected by the non-isothermal free jets. The heat flow between two vertically adjacent zones A and B is calculated as:

  $Q_B = C_B \cdot A_B \cdot [T_A - T_B]$

where C_B is a heat transfer factor between these zones, which has a value of 2.3 $W/m^2 \cdot K$ if the temperature of upper zone is higher than that of the lower zone (it assumes a stable temperature stratification). If the wall temperature is not known, a first guess is considered for the indoor air temperatures. A one-dimensional thermal network model is used to evaluate the wall temperature, and the indoor air temperatures are re-calculated. The calculations are repeated until the difference between the air temperatures calculated at iterations i and i-1 is smaller than a pre-defined value. They concluded that additional work is needed mainly in the following directions: (i) to clarify the nature and numerical value of C_B factor, and (ii) to estimate the convective coefficient h_c.

Inard et al. [1996] used a zonal model to predict the air temperature distribution within a room. The indoor air volume was divided into several elementary isothermal zones, which are interconnected by mass air flow. Mass and energy balances were written for each zone to calculate the local temperature. The main problem of this approach is in the evaluation of mass transfer between the elementary zones. Two types of elementary zones were defined: the current zones and the specific flow zones.

- In the case of current zones, it was assumed that the velocities are uniform, and they are mainly governed by variations in the driving pressure. The mass flow between two adjacent zones P and E, in the x or y directions, is evaluated with the following relation

$$m_{P \to E} = \epsilon \cdot \rho_E \cdot C_d \cdot A_E \cdot \left(\frac{2 \cdot |P_P - P_E|}{\rho_E} \right)^{0.5}$$

  where ϵ indicates the direction, and C_d is analogous to discharge coefficient, which must be determined experimentally (they used a value of 0.8). The relation for z direction must include the air density.
- In the case of zones with specific flows (wall non-isothermal horizontal jet, wall thermal plume derived from a local heat source, and thermal boundary layer), the corresponding equation of mass flow rate was written.

The mass flow rate of air crossing a boundary between zones was obtained. By solving the system of non-linear equations, the pressure field was obtained for current zones, and the temperature field for all zones. Vertical air temperature profiles were estimated for three types of heat emitters (baseboard heater, vertical hot water radiator, and radiant heating floor), and compared with measurements. A similar approach was used by Bouia and Dalicieux [1991].

Wurtz [1995] implemented a detailed two-dimensional zonal model in the SPARK environment. By using 9 or 36 elementary zones, he obtained in a few seconds similar results to those from a CFD model called FLUENT, which used 7840 nodes and several hours of simulation.

Capability to model the processes listed in Task 1a
The model can be used for cases where the pattern of air movement within a room can significantly affect the distribution of contaminants or the verticall/horizontal temperature difference.

Accuracy
The accuracy is higher than with the previous model.

Ease of implementation
Some models are available, and could eventually be obtained. Some other models should be developed and

validated.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models. The main problem consists in solving the large system of simultaneous equations.

CPU requirements
Higher CPU requirements than the one-zone model.

Flexibility for future improvements
It can accommodate future models to evaluate the impact of different types of supply grilles or of location of heat sources.

Interoperability with other tools
It can be coupled with models of global or local thermal comfort.

CFD models

A large number of papers were published on the use of Computational Fluid Dynamics (CFD) models for calculating and/or visualizing the air movement in a room. For instance, Zhang et al. [1992] used a CFD code called EXACT3 to simulate the spatial distribution of air velocity, turbulent kinetic energy, energy and temperature in an office room. The averaged three-dimensional Navier-Stokes equations were coupled with the k-ϵ turbulence model, and were solved numerically using the Marker and Cell finite-difference method. Uniform profiles of air velocity, temperature and turbulent kinetic energy were assumed at the diffuser. The diffuser air velocity was specified to provide an equal amount of jet momentum as that measured in the fully developed turbulent region of the air jet. They compared the simulation results with measurements performed in a full scale office room.

Yamamoto et al. [1992] developed a PC-based numerical model to simulate the steady-state, isothermal airflow distribution and contaminant distribution in a room. The model uses the two-dimensional k-ϵ turbulence model. The Navier-Stokes equations and Reynolds stress equations were expressed in the form of vorticity-stream function. The governing equations are: stream function equation, vorticity equation, turbulent kinetic energy equation, and energy dissipation rate equation. These four equations were discretized using the finite-difference approach, and the were solved simultaneously for given boundary conditions. The particle diffusion equation was then solved to evaluate the contaminant distribution. They compared the model with the commercially available detailed FLUENT program.

Alamdari et al. [1986] concluded that the direct coupling of finite-domain models for air movement with a building thermal model is not a realistic approach, because of the increase in computing time. The first model required about 5 hours of CPU time on a DEC VAX 11/785. For comparison reasons, an intermediate-level model based on the characteristics of turbulent wall-jets required only one minute.

Capability to model the processes listed in Task 1a
The models can be used for cases where the pattern of air movement within a room can significantly affect the distribution of contaminants or the vertical/horizontal temperature gradient.

Accuracy
Relatively high accuracy, depending on the discretization approach, and the methods for solving the system of simultaneous equations.

Ease of implementation
Models are available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
Coupling with HVAC models is quite difficult, due to high CPU and memory requirements. Different approaches are used to reduce this problem.

CPU requirements
High CPU requirements, however lower that for CFD models.

Flexibility for future improvements
The models can accommodate future developments.

Interoperability with other tools
Coupling of CFD models with other tools is presently quite difficult, due to high memory and CPU requirements.

34-BS

Zone-to-zone air flow

The thermal performance of a house and the quality of indoor environment are very often influenced by the air movement between the different thermal zones. The energy analysis programs either neglect this phenomenon, or use an over-simplified approach.

The existing mathematical models can be classified into the following groups: (i) correlation-type models, (ii) zonal models, and (iii) CFD models. Some models discussed in sections 33-BS and 36-BS can also be used for simulating the zone-to-zone air flow.

<u>Correlation-type models</u>

Walton developed a computer model able to simulate the thermal loads by simultaneously considering heat transfer and air movement between all rooms in a building [Walton 1982, Walton 1983].

Yamaguchi [1984] developed correlations between the natural convection heat transfer through an opening and the temperature difference ΔT between the two zones:

$$q = a \cdot W \cdot (H \cdot \Delta T)^{1.5}$$

where W and H are the width and height of opening, respectively.

<u>Capability to model the processes listed in Task 1a</u>
These models are derived from experimental data, and are able to simulate the overall convection heat transfer between zones.

<u>Accuracy</u>
Acceptable accuracy. However, it depends on the experimental conditions used for developing the correlation coefficients.

<u>Ease of implementation</u>
Models are available, and are easy to implement.

<u>Ability to integrate HVAC calculation methods</u>

They can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Fast calculations.

Flexibility for future improvements
Better correlation coefficients can be used, based on measurements or detailed simulation of heat transfer and air movement, using reference models.

Interoperability with other tools
They can be coupled with other models such as CFD models.

Zonal models

Wurtz [1995] presented a survey a simple zonal models:

- multi-zone models, where a system of non-linear equations is obtained based on the energy and mass conservation equations; usually a room is modeled by one equation with the assumption of well mixing conditions in the room; a building is represented by a network of nodes and links; COMIS and CONTAM are examples of such models; COMIS is a multi-zone air infiltration model; the wind pressure can be input directly, or the model can calculate them based on wind speed and direction, building orientation, terrain properties; the airflow through cracks is expressed by the power law; CONTAM is a multi-zone model for analysis of air movement and contaminant dispersion;
- multi-room models, which simulate the heat and mass transfer between rooms, without simulation the phenomena occurring in each room (e.g., thermal stratification in each room is not accounted for).

Capability to model the processes listed in Task 1a
The model can be used for cases where the pattern of air movement can significantly affect the distribution of contaminants or the vertical/horizontal temperature gradient.

Accuracy
Acceptable accuracy, provided that the coupling between the elementary cells is correct.

Ease of implementation
Some models are available, and could eventually be obtained. Some other models should be developed and developed.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the heat balance equation for indoor air.

CPU requirements
Higher CPU requirements than for correlation-type models.

Flexibility for future improvements

It can accommodate future models to evaluate the impact of new supply grilles or of location of the heat sources.

Interoperability with other tools
It can be coupled with models of global and local thermal comfort.

CFD models

Schaelin et al. [1993] presented the "method of detailed flow path values" for linking results from a detailed simulation of airflow patterns using a CFD model with a multi-zone thermal model. First, the multi-zone model is used, and the interface parameters (e.g., mass flow, heat flow) are transferred to the CFD model as input boundary conditions. The CFD model is run for a single zone, the local parameters (e.g., temperature, velocity) are evaluated, and the interface values at the flow paths (e.g., mass flow, heat flow) are transferred back to the multi-zone model. This procedure can be repeated two or three times. It is called "ping-pong" technique. The important aspect is that, instead of using only one variable value for each flow path linking one node to others, a separate value from the CFD model is calculated for each path connecting this node with other nodes.

Capability to model the processes listed in Task 1a
The models can be used for cases where the pattern of air movement can significantly affect the distribution of contaminants or the vertical/horizontal temperature gradient.

Accuracy
Relatively high accuracy, depending on the discretization approach, and the methods for solving the system of simultaneous equations.

Ease of implementation
Models are available, and could eventually be obtained.

Ability to integrate HVAC calculation methods
Coupling with HVAC models is quite difficult, due to high CPU and memory requirements. Different approaches are used to reduce this problem.

CPU requirements
High CPU requirements, much higher than for zonal models.

Flexibility for future improvements
Since these models are flexible, they can accommodate future developments.

Interoperability with other tools
Coupling of CFD models with other tools is presently quite difficult, due to high memory and CPU requirements. CFD models can be used as reference models to generate the correlation coefficients or the corrections factors to be used by simpler models.

35-BS

Convective heat exchange between thermal zones

Some models discussed in sections 33-BS and 34-BS can also be used for simulating the zone-to-zone air flow.

36-BS

Coupled interzone air flow and heat and moisture transfer

There is a strong relationship between (i) the airflow pattern, and the position of heat and moisture sources, and (ii) the local values of temperature and moisture content. Therefore, the air flow, heat transfer, and moisture transfer models must be coupled. Some mathematical models presented in sections 33-BS to 35-BS can be used for this purpose.

The major issue of this section is the effectiveness of coupling these models in terms of computing time and memory requirements. The use of a CFD model to simulate all phenomena, at the scale of a house, is not yet a practical solution. Therefore, researchers evaluated other indirect ways to couple the airflow and thermal models. Two main classes of coupling are presented below.

<u>Coupling CFD and multi-zone thermal models</u>

Qingyan (in fact Chen, Q.) and van der Kooi [1988] developed a model called ACCURACY (A Cooling Code Using Room Air Currents), to simulate the room air temperature, contaminant field, and energy requirements. It considers the transient temperature differences between the air near the inside surfaces of building envelope and the middle point of the room ΔT, which can be obtained from hourly results of an airflow model. They found that the direct coupling of an airflow model and an cooling load model will not be acceptable for practical use, because of long computing time. Therefore, they developed an airflow program (CHAMPION SGE) for precalculating airflow patters for typical situation, and this approach was much faster than using a CFD program such as PHOENICS. They concluded:

- if the room air temperature gradient is not important, the one-node model can be used to evaluate the cooling loads;
- if the room air temperature gradient is not function of time, the one-node model with constant temperature gradient can lead to good predictions of cooling loads;
- if the room air temperature gradient is transient, it can be approximated to be proportional to the room cooling load; the best approach is to calculate the room air temperature gradient from airflow pattern.

Chen et al. [1990] presented the coupling of three models:

- an airflow model based on k-ϵ turbulence model, used to calculate the air velocity and temperature distribution in a room, given the room ventilation rates and space loads; the temperature difference ΔT_1 between the controlled point (e.g., in the middle of the occupied zone) and the air at points near the exterior surfaces is also calculated;
- a cooling load model called ACCURACY, used to evaluate the cooling load; the air temperature near the exterior walls is replaced in terms of air temperature at the controlled point and the temperature difference ΔT_1; since the model calculates the wall temperature, it is possible after the simulation to evaluate the temperature difference ΔT_2 between the controlled point and the wall temperature; if the difference ΔT_2-ΔT_1 is large, several iterations between airflow model and the cooling load model are necessary;
- an energy analysis model called ENERK, used to evaluate the annual energy consumption, using the hourly space cooling loads and the extracted air temperature.

Jiang et al. [1992] developed a model to simulate the airflow pattern and the temperature distribution in a two-zone enclosure. In a region near solid surfaces, the wall function method was used. In regions which are far from walls, the k-ϵ turbulence model was used. The Semi-Implicit Method for pressure Linked Equations (SIMPLE) algorithm was used to solve the finite-difference equations.

Yoshino et al. [1993] developed a multi-zone transient model to analyze the effect of improved thermal insulation and airtightness on the house performance. First, the room air temperature and heating load are calculated by the thermal model using initial values of air infiltration rates. Second, the air infiltration rates are calculated by the airflow model, using the latest information about the room air temperature. It was assumed that the air leakages are distributed on the walls, ceiling, and floors, proportional with their respective areas. In addition, the leakages on the walls are distributed according to proportion of windows and doors. Third, the room air temperature and heating load are calculated again, using the latest information about the air infiltration rates. This iterative process is repeated until the convergence for the room air temperature and heating load is reached.

Chen et al. [1995] used an academic program including the following models: (i) the standard k-ϵ turbulence model; (ii) the conjugate heat transfer model, which is used to describe processes that involve variations of temperature within solid walls and air, due to their interactions; (iii) the discrete transfer flux radiation model. Although the results demonstrated the model is a powerful tool to predict the thermal response of rooms, the computing cost is presently too high to be accepted for engineering applications (e.g., the CPU time on a workstation was about 100 hours to perform a six-hour real-time simulation).

<u>*Capability to model the processes listed in Task 1a*</u>
These models can be used to couple the simulation of air flow and heat and mass transfer phenomena.

Accuracy
Relatively high accuracy, depending on the discretization approach, and the methods for solving the system of simultaneous equations.

Ease of implementation
Some models are available, and could eventually be obtained. Some other models should be developed and validated.

Ability to integrate HVAC calculation methods
Coupling with HVAC models is quite difficult, due to high CPU and memory requirements. Different approaches are used to reduce this problem.

CPU requirements
High CPU requirements, much higher than for zonal models.

Flexibility for future improvements
The models can accommodate future developments.

Interoperability with other tools
Coupling of CFD models with other tools is presently quite difficult, due to high memory and CPU requirements. CFD models can be used as reference models to generate the correlation coefficients or the corrections factors to be used by simpler models.

Zonal models

Clarke and Hensen [1991] presented the development of a zonal model to simulate the combined heat and mass flow phenomena in buildings. If the fluid flow simulation is performed independently of the energy analysis, it is assumed that the flows are pressure driven and the buoyancy effects are time invariant. During each simulation time-step, the problem is solved as a steady-state flow of an incompressible fluid through the network of mass flow in the building. If the fluid flow simulation is coupled with the thermal model, an iterative approach is used for each time-step: (i) the thermal model is solved for the latest information about the fluid flows; (ii) the temperature-dependent characteristics (e.g., wall properties) and airflow rates are established based on latest information from the fluid flow and thermal models; (iii) the fluid flow model is solved using the latest information.

Capability to model the processes listed in Task 1a
These models can be used to couple the simulation of air flow and heat and mass transfer phenomena.

Accuracy
The accuracy depends on the discretization approach, and the models used for each elementary cell.

Ease of implementation

Some models are available, and could eventually be obtained. Some other models should be developed and validated.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, using an iterative approach for each time-step.

CPU requirements
Lower CPU requirements than for the CFD models.

Flexibility for future improvements
It can accommodate future developments.

Interoperability with other tools
It can be coupled with other models, using an iterative approach for each time-step.

37-BS

Coupled interzone air flow and heat, moisture and other contaminants transfer

Some mathematical models presented in sections 33-BS to 35-BS, and the approaches used for coupling different models presented in section 36-BS can also be used in this section.

38-BS

Variable indoor air temperature (e.g, for evaluating the overheating/underheating in each thermal zone)

The air temperature of a room changes continuously during the day, due to factors such as operation of HVAC system, internal sources, outdoor conditions, thermal quality of building envelope,people's behavior, or thermal exchanges with other zones. The impact of these factors can be evaluated using models presented in all previous sections. However, for the objective of this project, the discussion will focus on: (i) numerical methods and (ii) time-domain response factor methods.

<u>Numerical methods</u>
The computer models based on heat balance method, which are able to take into account the thermal mass of house, and can modify the time-step, are good candidates to simulate the indoor air temperature (see section 2-BS).

<u>Time-domain response factor methods</u>
The models based on weighting factors can also be used (see section 2-BS). Since these models cannot simulate the time- and temperature-dependent properties, in some cases their use can give erroneous results.

39-BS

Passive solar design - solar windows/direct gain system

Selected models must be able to evaluate the transient phenomena related to the following aspects: (i) the solar heat gain through windows of different types, (ii) the insolation patterns on interior surfaces, (iii) the variation of indoor air temperature, (iv) the surface temperature of all building elements, including interior thermal mass, (v) the time- and temperature dependent properties of building materials, (vi) the time- and temperature-dependent convection heat transfer on the interior surfaces, including air movement and thermal stratification, and (vii) the time- and temperature-dependent radiation heat transfer between interior surfaces.

Evaluation of this design approach is based on models presented in previous sections.

40-BS

Passive solar design - sunspaces

Comments presented in section 39-BS apply also to this section.

41-BS

Passive solar design - thermal storage walls/indirect gain systems

Comments presented in section 39-BS apply also to this section.

42-BS

Quality of indoor environment - thermal comfort (based on air temperature, relative humidity, mean radiant temperature of interior surfaces, air velocity, occupant's activity and clothing type)

The evaluation of thermal comfort must take into account physiological, psychological, and environmental factors. It can be performed using (i) environmental indices, or (ii) detailed mathematical models.

Environmental indices

The most used index for evaluating the thermal sensation of people is the operative temperature, defined as the average of air temperature (T_{air}) and mean radiant temperature (MRT), weighted by their respective heat transfer coefficients (h_c, h_R) [ASHRAE 55 1989]:

$$T_{op}=\frac{h_c \cdot T_{air}+h_R \cdot MRT}{h_c+h_R}$$

ASHRAE Standard 55-1989 recommends the acceptable limits of (i) operative temperature for human comfort in terms of people activity, thermal insulation of clothing, and air velocity, (ii) radiant temperature asymmetry, and (iii) horizontal and vertical air temperature gradient.

Other examples of indices are: effective temperature, skin wettedness, and wet-bulb globe temperature [ASHRAE 1993].

Capability to model the processes listed in Task 1a

This model evaluates the thermal comfort, by calculating the operative temperature and comparing the result with the recommended limits for thermal comfort.

Accuracy

Acceptable accuracy. The range of thermal comfortable conditions were obtained by using the detailed mathematical models.

Ease of implementation
The operative temperature can be calculated if the building thermal model is able to evaluate the mean radiant temperature.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Fast calculations.

Flexibility for future improvements
It can be used to control the HVAC system in terms of the operative temperature, rather than the indoor air temperature.

Interoperability with other tools
It can be coupled with other models, using an iterative approach for each time-step.

Detailed mathematical models
The three fundamental models for simulating the thermal sensation of people are the following:

- Fanger's model [1982], which is a steady-state model, considers the human body represented by one point (or a small element), and takes into account: (i) four indoor environment parameters (air temperature, mean radiant temperature, air velocity, and relative humidity), and two personal parameters (metabolism, and thermal resistance of clothing); Ling and Deffenbaugh [1990] used the Fanger's model to study the impact of several HVAC systems; Kalisperis et al.[1990], Kalisperis and Steinman [1991] used Fanger's model to develop a design model for radiant heating panels; Zmeureanu et al. [1989], Zmeureanu and Fazio [1988] integrated Fanger's model into a finite-difference model to relate the thermal performance of radiant heating panels and of hollow core slabs, with the corresponding thermal sensation of occupants;

- Gagge's model or two-node model [1971], which a transient model, considers the human body represented by two concentric spheres, and takes into account: (i) four indoor environment parameters (air temperature, mean radiant temperature, air velocity, and relative humidity), and two personal parameters (metabolism, mass and height of person, permeability of clothing to water vapor, and thermal resistance of clothing); Zmeureanu [1993] developed a computer model based on Gagge's model, Zmeureanu et al.[1996] integrated the Gagge's model into the DOE-2.1E program, using the Functional Values approach; a thermal comfort prediction tool was developed as part of ASHRAE Research project RP-781, which integrate the Gagge's and Fanger's models [Fountain and Huizenga 1996]; three models for thermal comfort were integrated within the BLAST program: Fanger's, Gagge's and KSU models [BLAST 1991];

- Stolwijk 's model [1971] or 25-node model, which considers the human body represented by 25 nodes: six segments (head, trunk, arm, hand, leg, feet) each one composed of 4 layers (skin, fat, muscle, and core), plus the heart; this model allows for evaluation of

  local thermal sensation; Tellier [1991] integrated the Stolwijk model into the TRNSYS program.

Fanger et al.[1987] developed also a model for evaluating the thermal sensation of people in terms of turbulence intensity, mean air velocity, and air temperature.

Chan and Jiang [1992] integrated the Fanger's models and a one-zone air quality model into an air flow model based on k-ϵ turbulence model to study the indoor air quality and thermal comfort in a classroom.

A simulation model, based on a modified version of two-node physiological model of thermal sensation, was developed and validated with experimental data as part of ASHRAE Research Project RP-619 [Jones et al. 1993, Jones et al. 1994]. The new model allows for simulating the thermal sensation of different parts of the body. The new model was combined with a finite-difference model of transient heat and moisture transfer through clothing. The model predicts also the transient nature of heat and moisture loads generated by people on their surroundings.

Capability to model the processes listed in Task 1a
These models perform detailed modelling of global and local thermal comfort.

Accuracy
Acceptable accuracy. Although the models were developed using both the thermal models and the people perception, some measurements performed in real buildings indicated some differences with respect to the computer simulation.

Ease of implementation
Models and source codes are available. Easy to implement.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, if the mean radiant temperature can be evaluated.

CPU requirements
Fast calculations.

Flexibility for future improvements
It can be used to control the HVAC system in terms of the thermal comfort index, rather than the indoor air temperature.

Interoperability with other tools
It can be coupled with other models, provided they can share some common data.

43-BS

Quality of indoor environment - indoor air quality

The evaluation of indoor air quality must take into account the physiological, psychological, and environmental factors. Several models presented in this report can be coupled with the evaluation of indoor air quality such as 19-BS (time-dependent moisture generation; absorption/desorption processes), 20-BS (time-dependent contaminant generation; absorption/desorption processes), 33-BS (room air movement), 37-BS (coupled interzone air flow and heat, moisture and other contaminant transfer).

There are three main groups of models for evaluating the indoor air quality:

- empirical models, based on statistical analysis of experimental data; these models are not included in this report, since they have limited applications outside the conditions used for their development);

- physical models, based on pollutant mass balance within a control volume; some parameters in the mass balance equation can be replaced with experimentally-derived correlations.

Physical models
The one-compartment model [Wadden and Scheff 1983] is based on the mass balance equation, and uses the following variables when calculates the concentration of pollutant in the indoor air:

a. Room conditions: mixing factor, pollutant generation rate, pollutant decay rate, room volume.
b. Supply air conditions: air infiltration rate, ventilation rate, concentration of pollutant in the outdoor air, efficiency of filter installed on the outside air duct, recirculation rate, efficiency of filter installed on the recirculated air duct.
c. Evacuation conditions: air exfiltration rate, exhaust rate.

The COMIS program is a multi-zone air infiltration model [Feustel et al. 1989, Haghighat and

Megri 1996]. A building is basically modelled by pressure nodes that are interconnected with air flow links. For one time-step, the outside of the building is represented by a fixed boundary condition. The pressures of the internal nodes in the air flow network have to be solved to determine the different air flow rates. The model can simulate crack flow, flow through large openings, single-sided ventilation, cross-ventilation, and HVAC systems. It can predict concentration of pollutants in each zone.

The CONTAM94 [Walton 1995] program simulates the steady-state inter-zonal air movement and contaminant dispersion, assuming the building can be modeled by zones of well-mixed air. Pollutant is added to the zone due to the inter-zone air flow, the generation within the zone, and the reaction of other pollutants. The removal of pollutant from a zone is due to the evacuation of air, the removal from zone, and the chemical reaction with other pollutants. A system of non-linear simultaneous equations is solved. The Institute for Research in Construction, National Research Council of Canada, selected the CONTAM model to be used in the development of an integrated software for indoor air quality analysis [Zhang and Shaw 1996].

44-BS

Quality of indoor environment - visual comfort (including daylighting)

The evaluation of visual comfort must take into account the physiological, psychological, and environmental factors. The models can be classified into the following groups: (i) daylight factors models, (ii) radiosity model, and (iii) ray-tracing models.

Daylight factors models

The DOE-2 daylighting calculations allows the user to evaluate the impact of daylighting-based control of lighting system [DOE-2 1989]. The calculations are performed in the following sequence:

- the contribution of direct light from window is evaluated for two locations indicated by the user; the contribution of diffuse lighting and light reflected by the ground, which enter through the window, and are reflected by walls, ceiling and floor, is also evaluated; a set of "daylight factors" (interior illuminance divided by exterior horizontal illuminance) are evaluated for 20 different solar altitude and azimuth values, for standard clear and overcast sky conditions;
- the hourly daylight illuminance and glare calculation is performed, using the previously derived "daylight factors" and interpolating for sun position and cloud cover;
- the controls systems are simulated, which reduce the electricity demand if the illuminace level corresponds to a pre-defined target.

Clarke et al. [1985] presented the calculation of daylight factors, in terms of (i) the sky component, (ii) the external reflected component, and (iii) the internal reflected component.

The Lumen-Micro program calculates (i) the average illuminace level, using the zonal cavity method (or coefficient of utilization method), (ii) the horizontal and vertical illuminance, using the point-by-point method, and taking into account the interior reflections, (iii) the daylighting effect on the illuminance level, and (iv) the relative visual performance (this version evaluates this index only for pencil task at a 25° viewing angle) [Lumen-Micro 1993]. Light levels due to a combination of daylighting systems and electric lighting systems can be evaluated.

Capability to model the processes listed in Task 1a
These models can evaluate the illuminance level of a house.

Accuracy
Acceptable accuracy. These models do not evaluate the criteria for visual comfort, except the illuminance level and the glare index (for one model).

Ease of implementation
Models are available, and the source codes could be obtained.

Ability to integrate HVAC calculation methods
Some models controls the electric lighting in terms of available daylighting. Therefore, the cooling load is modified. They can be coupled with HVAC models, provided that the CPU requirements are acceptable.

CPU requirements
Fast calculations.

Flexibility for future improvements
They can accommodate future developments.

Interoperability with other tools
The CPU and memory requirements could preclude the coupling with other models.

Radiosity model

The SUPERLITE program uses the radiosity method to account for direct, externally reflected, and internally reflected light, in order to evaluate the interior illuminance level [Hitchcock 1995]. However, there is no evaluation of visual comfort, except the illuminance level. Light levels due to a combination of daylighting systems and electric lighting systems can be evaluated.

Capability to model the processes listed in Task 1a
This model can evaluate the illuminance level of a house.

Accuracy
High accuracy.

Ease of implementation
Model is available, and source code could be obtained.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, provided that the CPU requirements are acceptable.

CPU requirements
Relatively fast calculations.

Flexibility for future improvements
It can accommodate future developments.

Interoperability with other tools
The CPU and memory requirements could preclude the coupling with other models.

Ray-tracing models

The RADIANCE program uses a method based on Monte-Carlo backward ray-tracing to calculate diffuse inter-reflection [Mardaljevic and Lomas 1995]. It has the capability to model complex natural and luminous entities (e.g., sky brightness patterns, luminaire output distribution) and produces photo-realistic images.

Capability to model the processes listed in Task 1a
These models can evaluate the illuminance level of a house.

Accuracy
High accuracy.

Ease of implementation
Models are available, and source codes could be obtained.

Ability to integrate HVAC calculation methods
They can be coupled with HVAC models, provided that the CPU requirements are acceptable.

CPU requirements
Fast calculations.

Flexibility for future improvements
They can accommodate future developments.

Interoperability with other tools
The CPU and memory requirements could preclude the coupling with other models.

45-BS

Design heating and cooling loads for each thermal zone

The design can be performed considering either steady-state or transient processes. using models presented in the previous sections. It can also be performed assuming either isolated thermal zones or coupled thermal zones (in terms of heat and moisture transfer, and airflow).

46-BS

Heating and cooling loads provided in each thermal zone by the existing HVAC equipment (e.g., baseboard heater, forced air system)

The transient nature, the coupling of all thermal zones, and the simultaneous simulation of thermal zones and HVAC system is essential for this topic. Models presented in previous sections can be used to simulate this topic.

47-BS

Coupling thermal loads and HVAC system simulation

Since a house is an integrated system, the evaluation of space thermal loads must take simultaneously into account the heat gains/losses and the energy supplied by the HVAC system. For instance, if the capacity of a furnace is less than the heating load at a particular moment, then the air temperature cannot be kept within the thermostat settings, and the occupants' thermal comfort will be affected.

The following models are used to simulate the coupling between the space thermal loads and the HVAC systems: (i) sequential model, (ii) iterative model, (iii) predictive model, (iv) modular-simultaneous model, and (v) conflation model.

Sequential model

Most energy analysis programs use the sequential approach, where the information is passed only in one direction: from the space load calculation to the HVAC system calculations. There is no feedback to the first block to repeat the calculations when, for instance, the capacity of system is less than the space cooling load.

When the room thermal load is evaluated using the heat balance method, the following relation applies:

$$Q_{int.load} + \Sigma h_i \cdot A_i \cdot (T_i - T_R) + m_{inf} \cdot c_p \cdot (T_{out} - T_R) + Q_{sys} = 0$$

Since the HVAC system is not yet modelled, the term Q_{sys} which represent the system output is approximated by using a control profile in terms of room air temperature [BLAST 1991, Taylor et al. 1991]:

$$Q_{sys} = a \cdot T_R + b$$

The room air temperature is explicitly derived from the above two relations, in terms of coefficients a and b.

Walton [1983] used a more general form for defining the control profile, using always the sequential approach:

- at $T_R=T_{setpoint}$, $Q_{sys}=0$;
- at $T_R=T_p$, $Q_{sys}=Q_{max}$; where $T_p = T_{setpoint} + \Delta T$;

which leads to: $Q_{sys} = Q_{max} + (T-T_p)\cdot Q_{max}/\Delta T$, or

$$Q_{sys} = (Q_{max}/\Delta T)\cdot T + (Q_{max} - T_p\cdot Q_{max}/\Delta T) = c\cdot T + d$$

In the previous relation, the temperature T takes into account both mean radiant temperature and air temperature:

$T = \beta\cdot T_{radiant} + (1-\beta)\cdot T_R$, and β is an user defined radiation fraction.

Capability to model the processes listed in Task 1a
They can model most building-side processes.

Accuracy
Acceptable accuracy for most all-air and air-water HVAC systems. Lower accuracy for the radiant systems or for those air-type systems where other parameters (e.g., outdoor air temperature in the case of an economizer control) have an important impact on the system output.

Ease of implementation
Models are available, and source codes could be obtained.

Ability to integrate HVAC calculation methods
Limited capability to model the coupling with HVAC models. For the all-air and air-water HVAC systems, they use a relationships between the system output and the indoor air temperature. The impact of radiant system on the surface temperature, and consequently on the space load, is not properly evaluated. If the HVAC system has an insufficient capacity, these models cannot evaluate the indoor air conditions (e.g., temperature and relative humidity).

CPU requirements
Acceptable CPU requirements.

Flexibility for future improvements
No flexibility for future improvements in terms of coupling.

Interoperability with other tools
They can be coupled with other models, which provide information for loads calculations (e.g., exterior shading, air infiltration).

Iterative model

Witte et al. [1989] used the Newton-Raphson's method to iteratively calculate the room air temperature so that the sum of all fluxes entering the room should equal zero. Although the successive substitution solution will converge, the number of iterations was very large, about 10 times larger than a standard run. He analyzed also the option to alternate between the building simulation and the system simulation. The system was simulated based on building conditions at the end of previous hour. Then the building was simulated using the system output. This approach caused the system to lag the building by one time step (one hour). In order to reduce the instabilities introduced by this approach, a shorter time step should be used, with the following implications: (i) longer computation time, (ii) new calculation of conduction transfer functions, and (iii) a different way to define hourly events is needed.

Capability to model the processes listed in Task 1a
They can model all building-side processes.

Accuracy
Acceptable accuracy. However, there is a time lag equal to one time-step between the building and system models.

Ease of implementation
Models are available, and the source codes could be obtained. The main implementation concern is the use of an efficient iterative process.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Due to the iterative process, these models have higher CPU requirements than the sequential models.

Flexibility for future improvements
It can accommodate future developments in both the building and system models.

Interoperability with other tools
It can be coupled with other models.

Predictive model

Taylor et al. [1991] modified the room heat balance equation, in order to couple the space loads and system calculations. Instead of calculating the room air temperature from the condition that the difference between loads and system output is zero, they introduced the zone air capacitance $C_z \cdot (dT_R/dt)$ and the system output defined as $Q_{sys}=m_{sys} \cdot c_p \cdot (T_{supply}-T_R)$. The heat balance equation was written as:

$$C_z \cdot (dT_R/dt)=Q_{int.load} + \Sigma h_i \cdot A_i \cdot (T_i-T_R) + m_{inf} \cdot c_p \cdot (T_{out}-T_R) + m_{sys} \cdot c_p \cdot (T_{supply}-T_R)$$

The term dT_R/dt can be replaced with a finite-difference approximation:

$$\frac{dT_R}{dt} = \frac{T^t_R - T^{t-1}_R}{\delta t}$$

and finally an explicit solution is obtained for the room air temperature T^t_R at time t, formulated entirely in terms of lagged quantities at time t-1 (e.g., T^{t-1}_R, T^{t-1}_i, T^{t-1}_{out}, T^{t-1}_{supply}). The authors found that a third order finite difference approximations gives the best results. Time-steps of 0.1 to 0.25 hour are adequate to guarantee stability for most cases where the system response is well behaved. Further, it was assumed that the controller "knows" the net heat flux entering or leaving the room as a function of room air temperature:

$$Q_{load} = Q_{int.load} + \Sigma h_i \cdot A_i \cdot (T_i - T_R) + m_{inf} \cdot c_p \cdot (T_{out} - T_R)$$

If the system can meet the room load (Q_{sys}=Q_{load}), then the desired room air temperature can be maintained. Otherwise, the room air temperature is recalculated. This method is called predictive system energy balance. The authors concluded also that it is possible to calculate the wall surface temperature at time steps intermediate to those which the Conduction Transfer Function series were calculated for by interpolation.

Capability to model the processes listed in Task 1a
They can model all building-side processes.

Accuracy
Acceptable accuracy. However, there is a time lag equal to one time-step between the building and system models. The impact of time lag is reduced by using time-steps of 0.1 to 0.25 h.

Ease of implementation
Models are available, and source codes could eventually be obtained.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
Higher CPU requirements than previous models, since they use smaller time-steps.

Flexibility for future improvements
They can accommodate future developments.

Interoperability with other tools
They can be coupled with other models.

Modular-simultaneous model

Hensen [1992a, 1992b] indicated that the integration of building and plant components for simultaneous simulation is greatly simplified if both use a similar numerical technique. The simulation of building-side processes in the ESP-r model is based on the finite difference method. The "control volume conservation, state-space approach" is appropriate for plant simulation, and can be easily coupled with the finite difference techniques. Therefore, the simultaneous simulation of building-side and plant-side processes can be realized in the ESP-r program by coupling the above approaches. One can obtain several matrices for building energy balance, plant energy balance, plant mass balance, building and plant mass flow network. It is possible to combine all these matrices in one super-matrix. However, the matrix will be difficult to manage, and will have a highly sparse structure. The author preferred, from a practical point of view, the solution of dividing the overall simulation problem into separate sub-systems. Some major advantages are the following: the reduction of memory requirements, and the possibility to use mixed variable time stepping schemes. The matrices required for the problem are solved through an iterative process to accommodate all thermodynamic and hydraulic couplings between equations. This approach is called "the modular-simultaneous technique."

Capability to model the processes listed in Task 1a
It can model all building-side processes.

Accuracy
Acceptable accuracy.

Ease of implementation
One model is available, and source code could eventually be obtained. Other models should be developed and validated.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
No information available. However, one can assume the CPU requirements are similar to other iterative models. The memory requirements are reduced.

Flexibility for future improvements
It can accommodate future developments.

Interoperability with other tools
It can be coupled with other models.

Conflation model

Clarke et al. [1995] coupled the ESP-r energy analysis program with a CFD model, using the conflation approach. When these models are coupled, a large, sparse matrix is obtained, composed of three main types of sub-matrices: (i) one for ESP-r model, (ii) one for CFD model, and another type for the coupling coefficients, that is, coefficients linking the previous two sub-matrices. It was found that is possible to mathematically condense these coupling coefficients towards the matrix equation centre position. It eliminates the sparsity by removing the null matrix elements. Finally, the modified ESP-R and CFD sub-matrices are obtained, and can be solved independently. The overall system balance is achieved through an iterative process. The existing solvers for the ESP-r and CFD models can be used without modifications.

Capability to model the processes listed in Task 1a
It can model all building-side processes.

Accuracy
Acceptable accuracy.

Ease of implementation
One model is available, and source code could eventually be obtained. Other models should be developed and validated.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models.

CPU requirements
No information available. However, one can assume the CPU requirements are similar to other iterative models.

Flexibility for future improvements
It can accommodate future developments.

Interoperability with other tools
It can be coupled with other models.

48-BS

Location of heat emitters (e.g., baseboard heaters, supply grilles)

Comments presented in sections 46-BS and 47-BS, and the mathematical models presented in sections 2-BS, 3-BS, 33-BS to 38-BS, 42-BS apply to this section too.

49-BS

Insolation patterns on the internal surfaces (e.g., walls, floor, ceiling)

The surface temperature of interior surfaces can increase due to the absorbtion of solar radiation, and therefore their contribution to the radiative and convective heat exchange can increase. Usually, the solar radiation through windows is supposed to be uniformly distributed over the interior surfaces, or to be distributed according to fixed percentages defined by the user (e.g., 60% on the floor, 20% on the back wall); this distribution represents an average for the entire simulation period [DOE-2 1982].

A better approach is to evaluate the location of sunlit area on the internal surfaces, for each time-step over the simulation period. Examples of models allowing for this improved simulation, by calculating the solar rays direction, are the following.

BLAST [1991] performs two types of evaluations: (i) all beam sunlight is assumed to be incident on the floor, where is absorbed according to the solar absorbance; the reflected radiation is added to the transmitted diffuse radiation which is uniformly incident on all interior surfaces; and (ii) the amount of beam radiation falling on each surface in the zone is calculated by projecting the solar rays through the window.

Athienitis and Sullivan [1985] developed an algorithm for evaluating the instantaneous solar radiation transmitted through a window and absorbed by each interior surface, calculating the image of each interior surface on the window, based on the direction of solar rays.

Messadi [1990] developed a procedure for identifying the area and geometry of sunlit internal surfaces. The approach is based on a vectorial analysis to locate the direction and position of solar rays in space, and to determine the sunlit area configurations.

Capability to model the processes listed in Task 1a
These models can be useful for evaluating the solar radiation on the interior surfaces.

Accuracy
High accuracy.

Ease of implementation
Models are available, and source codes could eventually be obtained. It can easily be implemented into a building-side model, if the direction of solar radiation for each time-step is available through simulation.

Ability to integrate HVAC calculation methods
It can be coupled with HVAC models, through the building-side model.

CPU requirements
Acceptable fast calculations.

Flexibility for future improvements
They can accommodate future developments such as shading by exterior elements.

Interoperability with other tools
They can be coupled with other models such as shading by exterior elements or thermal comfort.

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