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Thermally Acceptable Temperature Drifts can Reduce the Energy Consumption for Cooling in Office Buildings

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The energy consumption and cost in large office buildings can be reduced by allowing the indoor environment to deviate from the optimum comfort conditions to situations which are still acceptable by most occupants. One such strategy is to allow the indoor temperature to increase in the afternoon in summer. A computer program was developed to predict the thermal sensation and discomfort under transient conditions, and the detailed computer program BLAST was used to evaluate the indoor air temperature, space cooling load, chilled water consumption and seasonal energy use. This paper presents a comparison performed for a reference office building in Montréal between a conventional design, with constant set-point temperature during the operating hours, and a new design, where the indoor temperature is allowed to increase after 3:00 p.m. The results show a reduction of seasonal consumption for chilled water between 34 and 40%, and a reduction of energy budget for HVAC systems of 11%.

NOMENCLATURE

A_b surface area of body (m^2)
CLO thermal insulation of clothing (clo)
$C_{p,B}$ specific heat of body (kJ kg^{-1} °C^{-1})
$C_{p,b}$ specific heat of blood (kJ kg^{-1} °C^{-1})
C_{RES} convective heat loss by respiration (W m^{-2})
CS convective heat loss (W m^{-2})
$DISC$ index of thermal discomfort
dT_{CR} change in core temperature (°C h^{-1})
dT_{SK} change in skin temperature (°C h^{-1})
E_{DIFF} evaporative heat loss by diffusion (W m^{-2})
E_{RSW} regulatory sweating heat loss (W m^{-2})
E_{RES} evaporative heat loss by respiration (W m^{-2})
$E_{RSW,REQ}$ regulatory sweating heat loss at thermal comfort (W m^{-2})
E_{SK} evaporative heat loss (W m^{-2})
$Facl$ ratio of clothed to naked surface area of a person
h_c convective heat transfer coefficient (W m^{-2} °C^{-1})
h_e evaporative heat transfer coefficient (W m^{-2} °C^{-1})
h_r radiative heat transfer coefficient (W m^{-2} °C^{-1})
i_m moisture permeability
k thermal conductance between core and skin (W m^{-2} °C^{-1})
M metabolism (W m^{-2})
m body mass (kg)
P_A partial pressure of water vapour at ambient conditions (kPa)
P_{SK} water vapour pressure at saturation (kPa)
Q_{CRSK} heat transfer from core to skin (W m^{-2})
RS radiative heat loss (W m^{-2})
S_{CR} heat storage in the core compartment (W m^{-2})
S_{SK} heat storage in the skin compartment (W m^{-2})
$SKBF$ blood flow rate (kg s^{-1} m^{-2})
t time (s)
T_A ambient temperature (°C)
T_B body temperature (°C)
$T_{B,C}$ cold set-point of evaporative regulation zone (°C)
$T_{B,H}$ hot set-point of evaporative regulation zone (°C)
T_{CR} core temperature (°C)
T_{CL} temperature of clothed body surface (°C)
T_R mean radiant temperature (°C)
T_{SK} skin temperature (°C)
$TSENS$ index of thermal sensation
V air velocity (m s^{-1})
W work (W m^{-2})
w skin wettedness
α ratio of skin to total mass of body

INTRODUCTION

AMONG the physical factors influencing human thermal comfort in indoor environments, the most used for the design process and operation management of energy-efficient and comfortable buildings are: air temperature, mean radiant temperature, air movement and humidity. There are several control strategies of indoor air temperature that can significantly reduce the energy use in large office buildings, while providing to the occupants thermal conditions within acceptable limits. One strategy is to modify the cooling set-point temperature of HVAC systems in such a way to let the indoor air temperature increase in the afternoon in summer. Since the indoor temperature does not increase suddenly, because of thermal inertia of building materials and furniture, through an appropriate control strategy one can obtain a rate of change and a final indoor air temperature, which are acceptable to most occupants. This approach can be used in existing buildings, where the most cost-effective energy conservation measures have been already applied, and the building owners do not invest in new technologies unless the payback period is less than 2–3 years.

McNall *et al.* [1] noted that very significant energy savings in both heating and cooling are possible by using control strategies which allow ambient temperatures to drift upon the perimeters of thermal comfort zone. Fleming [2] indicated that energy savings up to 10% can be obtained in office buildings by controlling the indoor air temperatures to reach the limits of the comfort zone.

Since 1975 ASHRAE Standards [3, 4] encourage engineers to design HVAC systems that would let the indoor temperature swing within the comfort limits (rather than be kept at a constant set point) in order to reduce the energy consumption. ASHRAE Standard 55-1981 [5] specifies the limit values of factors such as air temperature, surface temperature, humidity and air velocity, which are needed for thermal comfort. For example, the thermal comfort zone of office workers (sedentary or near sedentary occupants) for summer conditions is defined by the following boundaries:

—at dew point temperature = 16.7°C, the operative temperature should be 22.6–26°C,
—at 1.7°C dew point temperature, the operative temperature should be 23.3–27.2°C.

ASHRAE Standard 55-1981 allows the operative temperature to temporarily deviate beyond the limits of the comfort envelope, as it states that [5]:

> Slow rates of operative temperature change (approximately 0.6°C/h) during the occupied period are acceptable provided the temperature during a drift or ramp does not extend beyond the comfort zone by more than 0.6°C and for longer than one hour.

Nelson *et al.* [6] suggested that maintaining an office 1°C cooler in winter can lead to savings of about 5% of the annual heating cost.

Schiller and Arens [7], Schiller *et al.* [8] and Schiller [9] conducted a study on environmental conditions and occupant comfort in ten office buildings from San Francisco Bay area to assess the thermal comfort, and also to compare the people's perception of the thermal conditions with the recommendations of ASHRAE Standard 55-1981. About 78.2% of measured parameters in winter and 52.8% in summer fell within the comfort zones recommended by this standard. Analysis of the results also indicated:

(1) An acceptability of 80 to 85% to the thermal comfort conditions for both heating and cooling seasons.
(2) Neutral temperature (defined as the temperature at which the greatest percentage of people are experiencing neutral thermal sensation) was 22.0°C in winter and 22.6°C in summer, that is workers preferred indoor conditions slightly cooler in winter and summer than those established by the ASHRAE Standard 55-1981, and by the Fanger's model. They also found the measured thermal sensation was warmer than that predicted by thermal comfort indices such as Predicted Mean Vote (PMV) or TSENS (Thermal Sensation).

They finally concluded that current standards and practices for controlling the thermal environment in office buildings need to be re-analyzed.

Griffiths and McIntyre [10] evaluated the human response to the change of indoor temperature at a rate between 0 and 1.5°C/h for six hours. They found the maximum rate of temperature change should not exceed 0.75°C/h, provided that the maximum deviation from the mean comfort temperature is lower than 2.25°C. Later, McIntyre and Griffiths [11], indicated there is no appreciable difference in terms of thermal comfort between an indoor environment with a constant temperature and one with a temperature change of 0.5, 1.0 and 1.5°C/h, provided that the maximum deviations from neutral temperatures is not larger than 2.25°C.

Berglund and Gonzalez [12] performed a study with 12 college-age subjects to evaluate the effect of temperature drifts as an energy conservation strategy in existing buildings. They tested subjects dressed in three types of clothing, with thermal resistance of 0.5, 0.7 and 0.9 clo, each of whom experienced seven rates of temperature change (0, ± 0.5, ± 1.0 and ± 1.5°C/h). Their findings indicated that at faster rates of temperature change (i.e. 1.0 and 1.5°C/h), the permissible deviation from neutral temperature was larger than that for 0.5°C/h rate of change. They noticed that the upper limit of indoor air temperature as recommended by the ASHRAE Standard 55-1974 [13] was too conservative, as the subjects accepted temperatures higher than 27°C. In another experiment, Berglund and Gonzalez [14] found that an increase of indoor temperature at a rate of 0.6°C/h for eight hours to a maximum temperature of 27–27.2°C was acceptable to more than 80% of the subjects wearing typical summer clothing.

Rohles *et al.* [15] studied the response of sedentary people to three types of thermal environment: cool, comfortable and warm. They concluded that the way an individual responds to his thermal environment depends upon the length of time he has been exposed to that environment and the temperature he experienced just prior to exposure to that environment, which implies thermal comfort models should consider the transient rather than steady-state conditions.

Hensen [16] emphasized that the thermal conditions in a building are seldom steady-state due to the thermal interaction between the building structure, climate, occupancy and HVAC system. For situations with a larger variation of room air temperature, the Fanger model which is based on steady-state and neutral conditions does not seem to be appropriate for use. The literature review led him to the conclusion that the recommended change rate of operative temperature of 0.6°C/h [5] should be applied during daytime and in an upward direction only. He found no evidence why the rate of change of 2.2°C/h, accepted for cyclical changes, cannot be applied for temperatures drifts and ramps, or eventually to be exceeded.

The above research works indicated that reasonable air temperature drifts can be accepted by the office workers, and can lead to the reduction of energy consumption for the HVAC systems. An HVAC engineer facing the alternative of selecting this operating procedure should know more about the impact of temperature drifts on both thermal acceptance and energy consumption. Since this information is not available from experiments in existing large office buildings, an alternative and yet single source is the computer simulation. Although it does not provide absolute results, because of highly complex thermal, physiological and psychological phenomena, the computer simulation can predict the average behaviour of both people and HVAC systems.

This paper presents a comparison based on computer simulation between a conventional design of an office building in Montreal, with constant set point ambient temperature during the operating hours, and a new design where the indoor temperature is allowed to increase after 3:00 p.m. To perform this comparison, a computer program called TCM was developed to predict the thermal sensation and discomfort, based on J. B. Pierce's two-node model, and then was validated by comparison with responses of subjects under different transient conditions and with results from another computer program. The detailed computer program BLAST was used to evaluate the indoor temperature and the energy consumption, and the TCM program to predict the corresponding thermal sensation.

MATHEMATICAL MODEL OF THERMAL COMFORT

To analyze the thermal comfort under transient conditions, J. B. Pierce's two-node model [17] was selected, which consider a person as composed of two concentric thermal compartments representing the skin and core of the body. The temperature within each compartment is assumed to be uniform, so there are only two variables: skin temperature (T_{SK}) and core temperature (T_{CR}). The metabolic heat generation is assumed to occur within the core, and then is lost by conduction through a massless conductor to skin, by respiration, and by convection through blood, which is circulated between the two compartments. The skin compartment loses heat by convection, radiation, evaporation of sweat and diffusion of water vapour to the indoor environment. The model is used to evaluate the temperature of skin, of core, and of the whole body, and then to predict the thermal response of occupants to given conditions of the indoor environment. Since there is not available on the market a software incorporating the two-node model, we developed a FORTRAN program for IBM-compatible micro-computers, by compiling the mathematical models published by ASHRAE [18], Berglund [19, 20], Doherty and Arens [21], Gagge *et al.* [17, 22] and Lotens [23].

The rate of change of temperature in each compartment ($\mathrm{d}T/\mathrm{d}t$) can be expressed in terms of the rate of heat storage and the thermal capacity of the body:

$$\frac{\mathrm{d}T_{CR}}{\mathrm{d}t} = \frac{S_{CR}\cdot A_D}{(1-\alpha)\cdot m\cdot C_{p,B}} \tag{1}$$

$$\frac{\mathrm{d}T_{SK}}{\mathrm{d}t} = \frac{S_{SK}\cdot A_D}{\alpha\cdot m\cdot C_{p,B}}. \tag{2}$$

The specific heat of body is assumed to be 3490 kJ/kg °C.

The rate of heat storage in the core and skin compartments are given by:

$$S_{CR} = M - W - (C_{RES} - E_{RES}) - Q_{CRSK} \tag{3}$$

$$S_{SK} = Q_{CRSK} - (CS + RS + E_{SK}). \tag{4}$$

The main mathematical models of heat transfer processes between the two compartments and between the person and indoor environment are presented below.

(a) Sensible convective heat loss by respiration:

$$C_{RES} = 0.0014\cdot M\cdot(34 - T_A). \tag{5}$$

(b) Latent evaporative heat loss by respiration:

$$E_{RES} = 0.0173\cdot M\cdot(5.87 - P_A). \tag{6}$$

(c) Heat transfer from the core to the skin by convection through blood circulation, and by conduction through the body tissue:

$$Q_{CRSK} = (k + C_{p,b}\cdot SKBF)(T_{CR} - T_{SK}). \tag{7}$$

The thermal conductance between the core and skin compartments is evaluated to be 5.28 W/m² °C, and the specific heat of blood to be 4187 J/kg °C.

(d) Convective heat loss between clothed body and ambient air:

$$CS = h_c\cdot Facl\cdot(T_{CL} - T_A). \tag{8}$$

(e) Fractional increase in body surface attributed to clothing:

$$Facl = 1 + 0.3CLO. \tag{9}$$

(f) Heat transfer by convection is the maximum value from the following formulas:

$$h_c = 8.6\cdot V^{0.53} \tag{10}$$

$$h_c = 3.1 \tag{11}$$

$$h_c = 5.66\cdot\left(\frac{M}{58.2} - 0.85\right)^{0.39}. \tag{12}$$

(g) Radiative heat loss between clothed body and indoor surfaces:

$$RS = h_r\cdot Facl\cdot(T_{CL} - T_R). \tag{13}$$

The linearized radiative heat transfer coefficient h_r is almost constant for typical indoor temperatures and equal to 4.7 W/m² °C.

(h) Evaporative heat loss from the skin:

$$E_{SK} = wh_e(P_{SK} - P_A). \tag{14}$$

(i) Fraction of skin surface covered by water w, which is called skin wettedness:

$$w = 0.06 + \frac{0.94E_{RSW}}{E_{MAX}}. \tag{15}$$

(j) Evaporation heat transfer coefficient:

$$h_e = 16.5\cdot i_m\cdot h_c. \tag{16}$$

(k) Saturated water vapour pressure on the skin surface:

$$\ln P_{SK} = [C_8/T_{abs} + C_9 + C_{10}\cdot T_{abs} + C_{11}\cdot T_{abs}^2 + C_{12}\cdot T_{abs}^3 + C_{13}\cdot\ln(T_{abs})/1000 \tag{17}$$

where $T_{abs} = T_{SK} + 273.15$; $C_8 = -5.8002206\ 10^3$; $C_9 = 1.3914993$; $C_{10} = -4.8640239\ 10^{-2}$; $C_{11} = 4.1764768\ 10^{-5}$; $C_{12} = -1.4452093\ 10^{-8}$; $C_{13} = 6.5459673$.

(l) Clothed body surface temperature (T_{CL}):

$$T_{CL} = \frac{\dfrac{T_{SK}}{0.155CLO} + Facl \cdot (h_c \cdot T_A + h_r \cdot T_R)}{\dfrac{1}{0.155CLO} + Facl \cdot (h_c + h_r)}. \quad (18)$$

(m) Thermoregulatory control processes (rate of blood flow, sweating and shivering) are governed by temperature signals from the skin and core, which are assumed to be proportional with the difference between the actual temperature and the corresponding set-point value for neutral condition:

(1) Warm signal from the core

$$WSIG = \begin{matrix} 0 & T_{CR} \leqslant T_{CR,N} \\ T_{CR} - T_{CR,N} & T_{CR} > T_{CR,N} \end{matrix}. \quad (19)$$

(2) Cold signal from the core

$$CSIG_{CR} = \begin{matrix} T_{CR,N} - T_{CR} < T_{CR,N} \\ 0 \quad T_{CR} \geqslant T_{CR,N} \end{matrix}. \quad (20)$$

(3) Warm signal from the skin

$$WSIG_{SK} = \begin{matrix} 0 & T_{SK} \leqslant T_{SK,N} \\ T_{SK} - T_{SK,N} & T_{SK} > T_{SK,N} \end{matrix}. \quad (21)$$

(4) Cold signal from the skin

$$CSIG_{SK} = \begin{matrix} T_{SK,N} - T_{SK} & T_{SK} < T_{SK,N} \\ 0 & T_{SK} \geqslant T_{SK,N} \end{matrix}. \quad (22)$$

(5) Warm signal from the body

$$WSIG_B = \begin{matrix} 0 & T_B \leqslant T_{B,N} \\ T_B - T_{B,N} & T_B > T_{B,N} \end{matrix} \quad (23)$$

where:

$$T_{CR,N} = 36.80^\circ\text{C}; \quad T_{SK,N} = 33.70^\circ\text{C};$$

$$T_{B,N} = 36.49^\circ\text{C}.$$

(n) Mean body temperature (T_B) can be determined by the weighted average of the skin and core temperatures:

$$T_B = \alpha T_{SK} + (1-\alpha) T_{SK} \quad (24)$$

where alpha (α) is the fraction of total body mass attributed to the skin compartment:

$$\alpha = 0.1 + \frac{0.00028}{SKBF + 0.0011}. \quad (25)$$

(o) Rate of blood flow depends on the skin and core temperature deviations from their respective setpoints:

$$SKBF = \frac{6.3 + 200\,WSIG_{CR}}{3600(1 + 0.5CSIG_{SK})}. \quad (26)$$

(p) Rate of regulatory sweating heat loss:

$$E_{RSW} = 114.2 \cdot WSIG_B \cdot e^{WSIG_{SK}/10.7}. \quad (27)$$

The equations 1–27 are used to calculate $\mathrm{d}T_{CR}/\mathrm{d}T$ and $\mathrm{d}T_{SK}/\mathrm{d}T$, and then the temperature of skin (T_{SK}), of core (T_{CR}), and of body (T_B), by integration from the initial conditions ($T_{CR,0}$, $T_{SK,0}$) through the interval t:

$$T_{CR} = T_{CR,0} + \int_0^t \left(\frac{\mathrm{d}T_{CR}}{\mathrm{d}t}\right)\mathrm{d}t \quad (28)$$

$$T_{SK} = T_{SK,0} + \int_0^t \left(\frac{\mathrm{d}T_{SK}}{\mathrm{d}t}\right)\mathrm{d}t. \quad (29)$$

The index of thermal sensation (*TSENS*) is defined in terms of deviations of the mean body temperature from the cold and hot set-points of the evaporative regulation zone:

$$TSENS = \begin{matrix} 0.4685(T_B - T_{B,C}) & T_B < T_{B,C} \\ 3.995(T_B - T_{B,C})/(T_{B,H} - T_{B,C}) & T_{B,C} \leqslant T_B \leqslant T_{B,H} \\ 3.995 + 0.4685(T_B - T_{B,H}) & T_{B,H} < T_B \end{matrix} \quad (30)$$

where: T_B = mean body temperature; $T_{B,C}$ = cold set-point for evaporative regulation zone

$$T_{B,C} = (0.194/58.15)(M - W) + 36.301 \quad (31)$$

$T_{B,H}$ = hot set-point for evaporative regulation zone

$$T_{B,H} = (0.347/58.15)(M - W) + 36.669. \quad (32)$$

The thermal sensation is then evaluated using the following 11-point scale:

+5 intolerably hot
+4 very hot
+3 hot
+2 warm
+1 slightly warm
0 neutral
−1 slightly cool
−2 cool
−3 cold
−4 very cold
−5 intolerably cold.

The index of thermal discomfort (*DISC*) is equal to *TSENS* when T_B is below its cold set-point $T_{B,C}$, and is related to skin wettedness when body temperature is regulated by sweating:

$$DISC = \begin{matrix} 0.4685(T_B - T_{B,C}) \quad T_B < T_{B,C} \\ 4.7(E_{RSW} - E_{RSW,REQ})/(E_{MAX} - E_{RSW,REQ} - E_{DIFF}). \\ T_{B,C} \leqslant T_B \end{matrix} \quad (33)$$

The rate of regulatory sweating during the conditions of thermal comfort is:

$$E_{RSW,REQ} = 0.42(M - W - 58.15) \quad (34)$$

The thermal discomfort is then evaluated using the following 6-point scale:

±5 intolerable
±4 limited tolerance
±3 very uncomfortable
±2 uncomfortable and unpleasant
±1 slightly uncomfortable but acceptable
0 comfortable.

VALIDATION OF THE TCM COMPUTER PROGRAM

Generally, the validation of a model by comparison with measurements is a difficult process, because of the assumptions, simplifications and default values used in the development of the computer program. In this particular case of simulating the thermal sensation of people in office buildings, the task seems to be even more difficult. To better assess the accuracy of model as well as its limitations, one should compare the predictions and measurements first for some physiological factors (e.g. T_{SK}, T_{CR}), and second for people perception of thermal environment (e.g. *TSENS, DISC*). Wissler [24] recommended that a model can be accepted if the predicted and measured values agree within 2°C for mean skin temperature, or within 0.5°C for central temperature. The comparison of thermal sensation can sometimes lead to large differences, which are generated not only by the weaknesses in the mathematical modeling, but also in the experimental techniques used to assess the perception of indoor environment.

In this paper, the results of the TCM program are compared with:

(1) Measurements of skin temperature on subjects, following a step-function change in ambient temperature, as presented by Young *et al.* [25].
(2) Measurements of skin temperature on subjects, under cyclic variation of ambient temperature, as presented by Grivel *et al.* [26].
(3) Thermal sensation of subjects under different transient conditions, as published by Berglund and Gonzalez [12].
(4) Results of computer program developed by Gagge *et al.* [22].

In all experiments and computer simulations presented below, the mean radiant temperature was equal to the ambient temperature.

Young *et al.* [25] measured the skin temperature of seven male subjects after a step-function change of ambient temperature in the environmental chamber from 24°C to 5°C. The following parameters of experiment were used in the computer simulation: body mass 79.4 kg, body surface area 1.98 m^2, thermal insulation 0.1 clo, relative humidity 0.30, metabolism 60.0 W/m^2, $i_m = 0.1$, air velocity 0.10 m/s. The measured skin temperature presented in Fig. 1 is an average of measurements at three locations (forearm, chest and calf). The program estimates are acceptable, as differences between measurements and predictions are smaller than 2.0°C. The largest difference occurs after 30 minutes, when the measurements show a sudden modification of the decrease rate of skin temperature, while the computer program predicts a uniform decrease of temperature.

Grivel *et al.* [26] measured the skin temperature of six male and six female subjects in a climatic chamber under cyclic variation of ambient temperature. In the first set of measurements, the subject had only minimum clothing (pants and bras), with a thermal insulation of 0.1 clo. The ambient temperature was kept constant at 28.8°C for one hour, and then was first decreased, followed by an increase and finally decreased again at a rate of 2°C every 8 minutes. The minimum air temperature was 20.8°C, and the maximum 34.8°C. The following experimental parameters were used in the simulation: body mass 63.1 kg, body surface area 1.72 m^2, thermal insulation 0.1 clo, relative humidity 0.50, metabolism 60 W/m^2, $i_m = 1.0$, and air velocity 0.20 m/s. In the second set of measurements, the subjects had additional clothing (cotton trousers and shirts, wool socks and sandals), with a thermal insulation of 0.6 clo and moisture permeability i_m of 0.45. The initial ambient temperature was 25.3°C and the cyclic variation had the same sequence and rate of change as in the first experiment. The minimum air temperature was 17.3°C, and the maximum 33.3°C. The skin temperature, as an average of measurements at four locations (chest, upper arms, thigh and calf) is compared in Fig. 2 (naked subject) and Fig. 3 (clothed subject) with the predictions of the TCM program. Predictions and measurements are in good agreement, with differences smaller than 1.5°C.

The predicted thermal sensation and discomfort levels given by the TCM program were compared with the

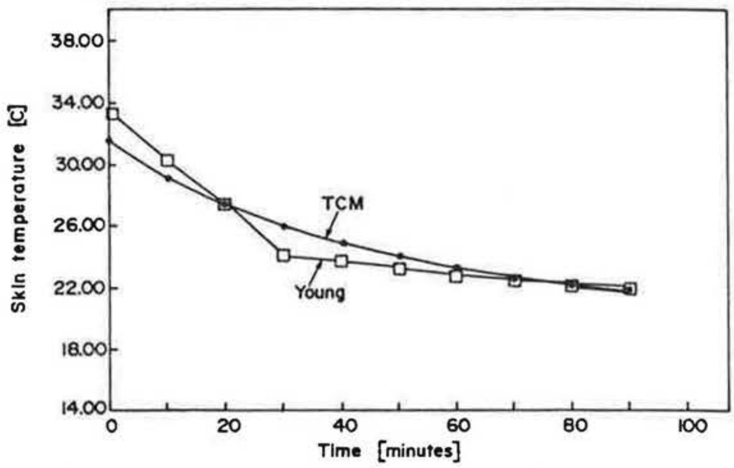

Fig. 1. Variation of the skin temperature due to a step change in ambient temperature. Comparison between the measurements on subjects [25] and the predictions of the TCM program.

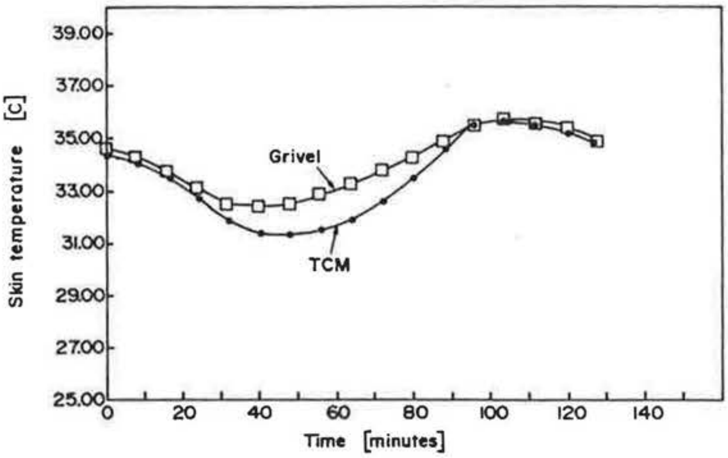

Fig. 2. Variation of the skin temperature of a naked person under a cyclic variation of ambient temperature. Comparison between the measurements on subjects [26] and the predictions of the TCM program.

results obtained by Berglund and Gonzalez on 12 college age students in a test chamber [12]. The students remained in a test chamber at 25°C for a half hour, after which the temperature was changed using an either positive or a negative ramp: 0.5, 1.0 and 1.5°C/h. The subjects had three different types of clothing, with thermal insulation of 0.5, 0.7 and 0.9 clo. The main parameters of interest are the following: body mass 68.3 kg, body surface area 1.81 m^2, relative humidity 0.45, metabolism 70 W/m^2, and air velocity 0.1 m/s. Berglund and Gonzalez developed a multiple linear equation from all experimental data, relating the mean thermal sensation to the clothing insulation value and indoor temperature for all the temperature ramps:

$$TSENS = 0.305T_A + 0.996CLO - 8.08. \quad (35)$$

A similar relationship was developed using the predictions given by the TCM program:

$$TSENS = 0.166T_A + 0.935CLO - 4.403. \quad (36)$$

The comparison between the two linear relationships for a thermal insulation of 0.5 clo is presented in Fig. 4. One can notice that for ambient temperatures between 22 and 28°C, both measurements and predictions show similar thermal sensations. The measured neutral temperature, at which $TSENS = 0$, is 24.86°C, while the predicted value is 23.71°C.

Values of thermal discomfort presented by Gagge *et al.* from computer simulations [22] were compared to thermal discomfort levels predicted by the TCM program (Fig. 5) for a similar environment: body mass 70 kg, body surface area 1.8 m^2, relative humidity 0.50, metabolism 73.0 W/m^2, thermal insulation 0.57 clo, and air velocity 0.20 m/s. The moisture permeability was assumed to be 0.45 [18]. Thermal discomfort estimates from both models show good agreement.

Finally, one can conclude the TCM program provides acceptable estimates of the thermal response of people, and therefore can be used in the next section to evaluate the effect of temperature drifts on the thermal comfort of occupants in an office building.

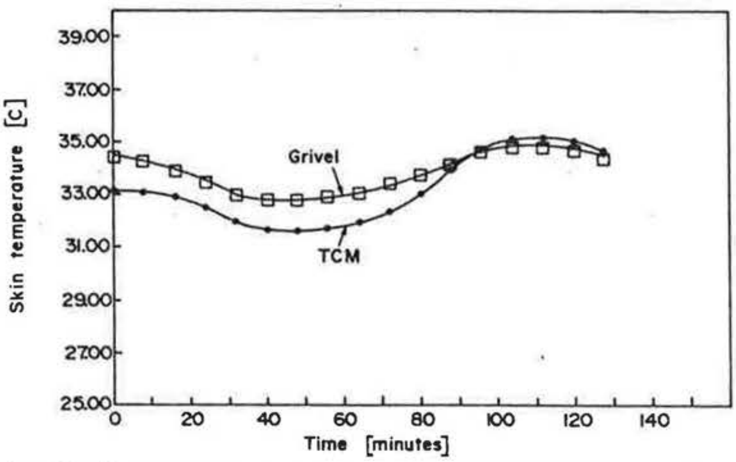

Fig. 3. Variation of the skin temperature of a clothed person under a cyclic variation of ambient temperature. Comparison between the measurements on subjects [26] and the predictions of the TCM program.

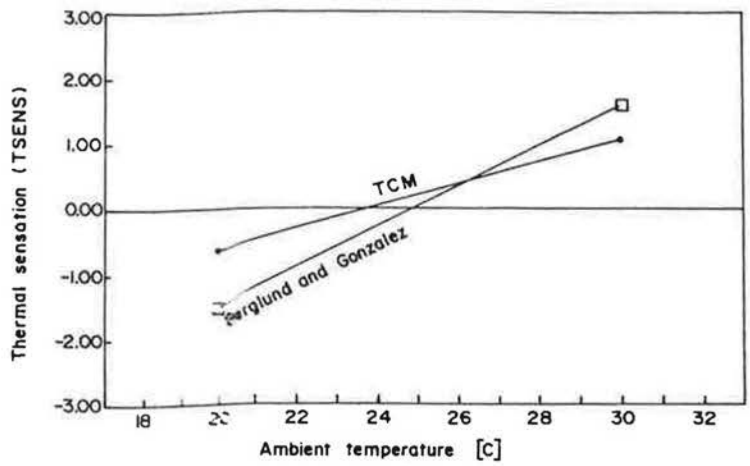

Fig. 4. Comparison between the predictions of the TCM program of the thermal sensation index (*TSENS*) and the measurements on subjects [12].

EVALUATION OF THE IMPACT OF AIR TEMPERATURE DRIFTS

A model of an office building is developed in accordance with the ASHRAE Standard 90.1-1989 [4], Energy Cost Budget Method, and to the regulation concerning energy conservation in new buildings in the province of Québec [27] (Appendix). Each floor is divided for calculation purposes into five thermal zones, as recommended by the ASHRAE Standard 90.1, as follows: four perimeter zones, having a depth of 4.5 m from the exterior wall, and one interior zone. Hence, the interior zone of each flow has 440.0 m^2 surface area, and represents 49% of the total floor area. As the interior zone plays a major role on the energy consumption for cooling, due to the important internal heat gains (lighting, office equipment, people), this paper emphasizes the impact of the proposed control strategy on the thermal comfort and space cooling load of that zone. The hourly and daily values presented for one interior zone can be considered as representatives for all ten interior zones of the reference building. It is worth mentioning that each thermal zone is not controlled by an independent HVAC unit, but there are two central VAV systems with perimeter reheat serving the entire building. First air-handling unit is located on the fifth floor and supplies air to floors 1 through 5, and the second unit is located on the roof and supplies air to floors 6 through 10. Each system uses one central cooling coil. Therefore, in addition to the analysis of thermal factors of a typical interior zone, this paper presents the impact of the proposed control strategy on the energy used by the HVAC systems for the whole building.

First, the base model of the office building is simulated using the BLAST (Building Load Analysis and Systems Thermodynamics) program developed by the U.S. Army Construction Engineering Laboratory [28], to evaluate the indoor temperature and energy consumption. Then, different control strategies for the indoor environment are proposed, where the air temperature is allowed to increase in the afternoon, as a measure of energy conservation which can be implemented in office buildings

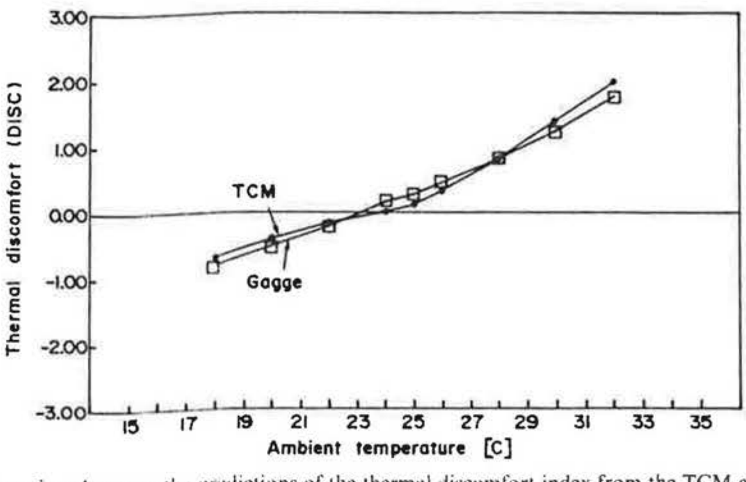

Fig. 5. Comparison between the predictions of the thermal discomfort index from the TCM and Gagge's program [22].

in Montréal in summer (June to September). The thermal comfort of occupants for each proposed operating condition is evaluated using the TCM program, and only those strategies which lead to a comfortable thermal environment are selected for the next step, where the energy consumption is evaluated using the BLAST program. Although several other design alternatives were expected to provide thermally acceptable indoor environments, they were eliminated from the analysis since the peak electrical demand and the energy consumption increased in the morning.

The following results are presented below:

(1) Hourly values of thermal parameters for a typical interior zone (ambient temperature, thermal sensation, thermal discomfort, space cooling loads, chilled water load) on a hot summer day.
(2) Monthly values of chilled water consumption and demand for the two HVAC systems operating in the building.
(3) Seasonal energy consumption of the HVAC systems.

Hourly analysis for a hot summer day

A hot summer day in 1980 in Montréal with a high average daily temperature (high = 27.4°C, low = 20.8°C) was selected for the analysis.

The hourly space temperatures in the interior zone of the reference building are illustrated in Fig. 6, where the two curves represent the summer space temperatures for the base case, and for the proposed design. The following convention was used in those drawings indicating the hour of day: 15:00 corresponds to the time interval starting at 15:00 and ending at 15:59.

During the highest occupancy period (9:00 a.m.–6:00 p.m.), the space temperature is relatively constant at 24.6°C for the base case, and for the proposed design is approximately 25.2°C from 9:00 p.m. to 3:00 p.m. However, after 3:00 p.m. the space temperature is allowed to drift to about 27°C for the rest of the occupied period.

Since the overnight temperature for the proposed design is about 2°C higher than the base case, the HVAC system is turned on two hours earlier (5:00 a.m.) to reduce the early peak cooling demand (Fig. 7). The peak space cooling load for the base case is 9.1 kW, and occurs at 6:00 p.m., while for the proposed design it occurs three hours earlier, and the maximum value is 9.3 kW, that is 3.2% larger than for the base case. The daily total cooling load of the base case design is 111.8 kWh, while for the proposed design is 109.3 kWh, that is 2.2% less.

In order to evaluate the energy and cost savings of the proposed design, the designer must also analyze the load on the cooling coil of the HVAC system rather than the space cooling load only. The chilled water consumption presented in Fig. 8 represents the amount of energy evacuated from the system by both central cooling coils. The electrical energy used by the chiller is obtained by dividing the cooling coil load with the coefficient of performance (COP), operation which is carried out by the program. The two curves are almost identical until 3:00 p.m. The peak load of the proposed design (260.8 kW or 26.08 kW/floor) is 5.5% smaller than that of the base case (276.0 kW or 27.6 kW/floor), and both occur at 11:00 a.m. The energy savings occur only after 3:00 p.m. when the space air temperature for the proposed design is allowed to drift. The daily total chilled water consumption of the proposed design is 1866 kWh or 186.6. kWh/floor, or 35% less than for the base case (2869 kWh or 286.9 kWh/floor). The energy savings of the cooling coil are 1003 kWh/day or 100.3 kWh/(day floor) compared with 2.5 kWh/day reduction of the space cooling

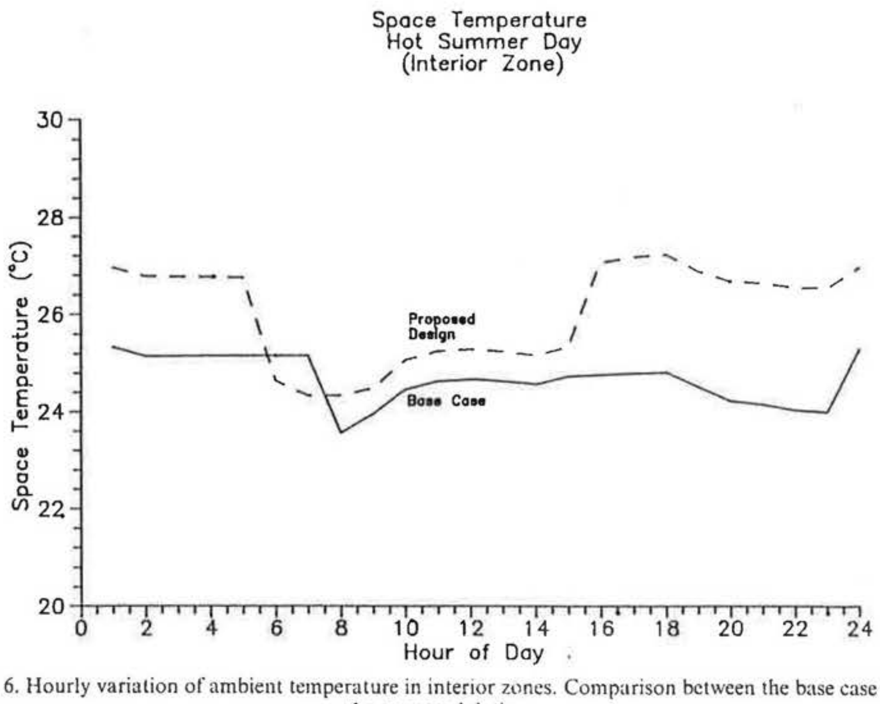

Fig. 6. Hourly variation of ambient temperature in interior zones. Comparison between the base case and the proposed design.

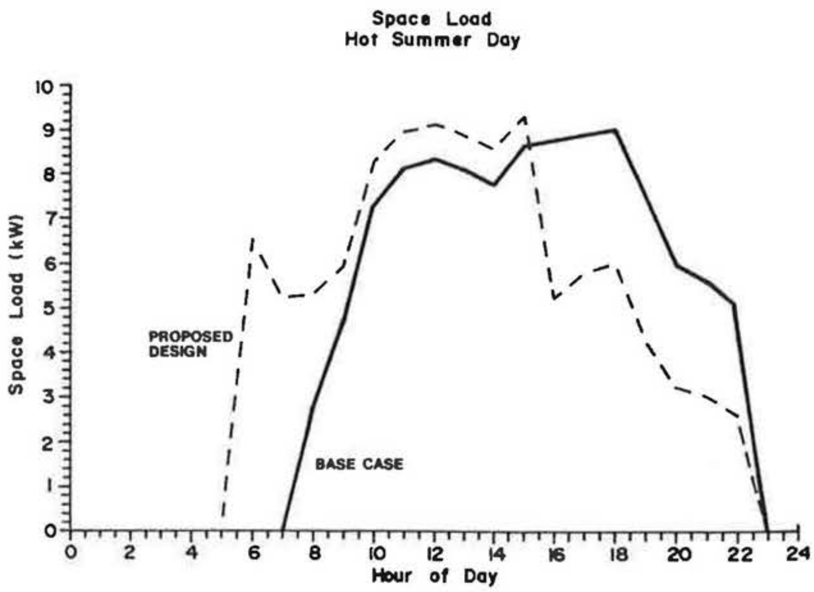

Fig. 7. Hourly variation of space cooling load of interior zones. Comparison between the base case and the proposed design.

load of a typical interior zone. When the proposed control strategy is used, the daily space cooling load of an interior zone is 109.3 kWh, while the daily cooling coil load is 186.6 kWh/floor. The difference is due to the impact of perimeter zones and the operation of HVAC units. For instance, since the automatic reset of the supply air temperature is not used, the mixed air entering the cooling coil is always cooled down to 13°C, before it is supplied to all zones, even some of them require a higher supply temperature. Therefore, the analysis of energy savings in buildings with large HVAC systems must be mainly based on the analysis of these systems (e.g. type, operation, control). The variation of the space cooling load is not a reliable indicator of the improvement of energy performance, since it does not take into account the control and operation of different components of HVAC systems.

The TCM program is utilized to predict the occupant's response, which are assumed to wear typical summer clothing ensembles ($CLO = 0.5$), with the ambient environment at 50% relative humidity and air velocity of about 0.1 m/s. The thermal sensation of the building

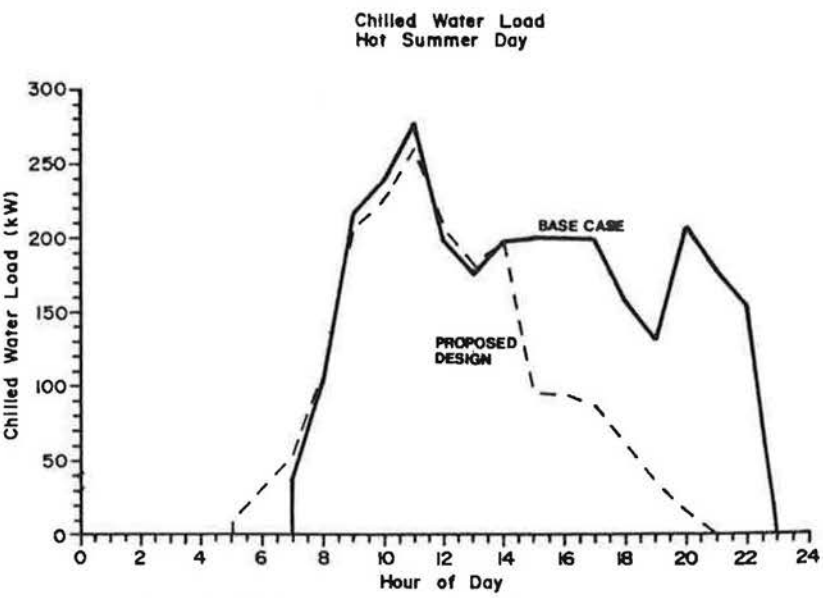

Fig. 8. Hourly variation of chilled water consumption. Base case: peak demand = 276.0 kW, daily consumption = 2869.0 kWh. Proposed design: peak demand = 260.8 kW, daily consumption = 1866.0 kWh.

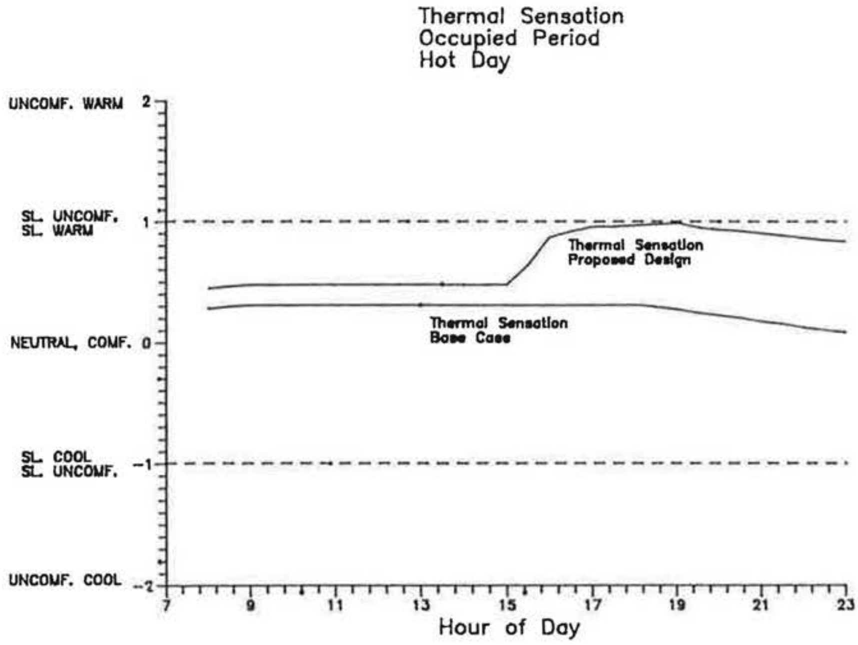

Fig. 9. Predictions of thermal sensation of building occupants in interior zones. Comparison between the base case and the proposed design.

occupants for the base case is neutral ($TSENS = 0.3$) (Fig. 9). The proposed design is assumed to keep the temperature relatively constant at 25.1°C from 9:00 a.m. to 3:00 p.m. During this period the thermal sensation index is 0.50. When the temperature in the space is allowed to drift to 27°C, the TCM program predicts that the indoor environment becomes slightly uncomfortable ($TSENS = 1$) around 7:00 p.m. Since a thermal sensation value of 1.00 represents the upper limit of warm thermal comfort (i.e. 80% thermal acceptability), and this limit is not exceeded, the proposed design can be accepted. Moreover, as indicated by Benzinger [29], the set-point of human thermoregulatory system is not constant, but increases during the day, which leads to a higher acceptability of warmer indoor environment in the afternoon. Figure 10 illustrates the variation of thermal dis-

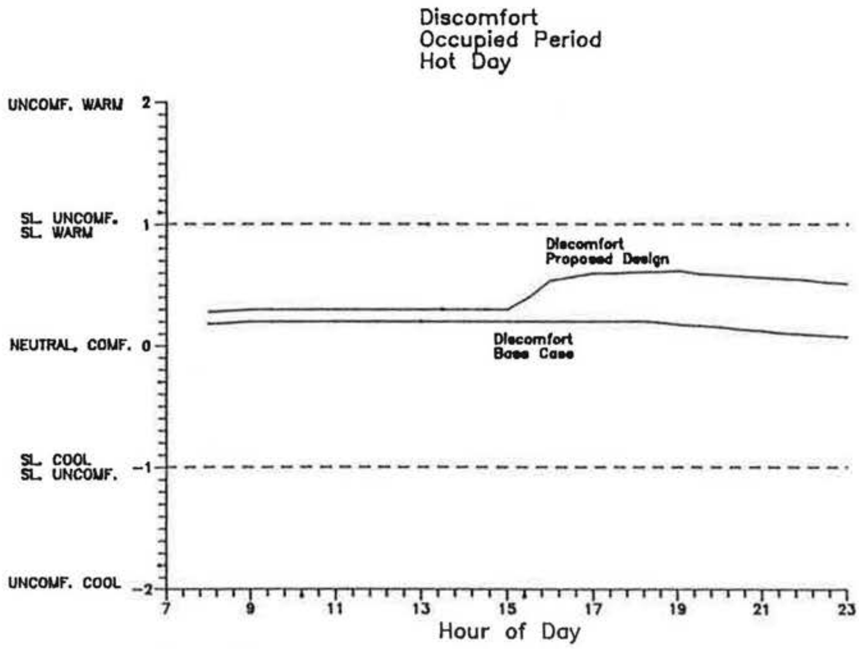

Fig. 10. Predictions of thermal discomfort of building occupants in interior zones. Comparison between the base case and the proposed design.

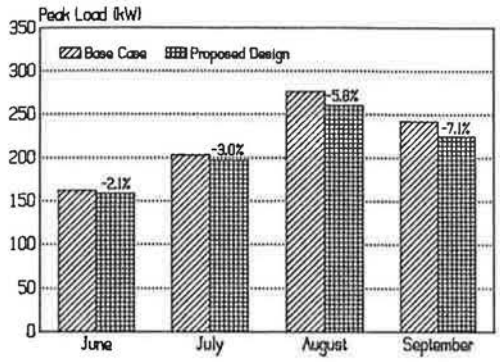

Fig. 11. Monthly peak demand for chilled water. Comparison between the base case and the proposed design.

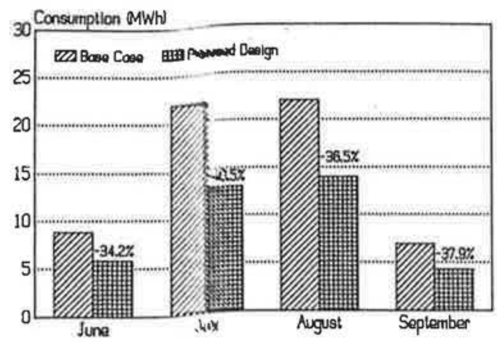

Fig. 12. Monthly energy consumption for chilled water. Comparison between the base case and the proposed design.

comfort index (DISC) under the same conditions. For the proposed design, the thermal discomfort index before the temperature drift is 0.3, and after is about 0.6.

Therefore, the TCM program indicated that the air temperature drift up to 27°C during a hot day did not create uncomfortable thermal conditions for occupants.

Seasonal consumption of chilled water

The proposed control strategy of ambient environment can lead to daily savings of about 35% for chilled water. Since it is important to evaluate the peak demand and chilled water consumption for the entire cooling season (June to September), Fig. 11 presents the monthly peak demand, which is reduced by 2.0 to 7%, and Fig. 12 the chilled water consumption, which is reduced by 34 to 40%.

Seasonal energy consumption

Table 1 specifies the seasonal energy consumption for heating, cooling, and electricity for the base case and the proposed design. The seasonal energy consumption for cooling is substantially reduced by 69250 kWh (27%) by the proposed design. The electrical consumption increased by 7000 kWh (2.7%) as the HVAC system is scheduled to start two hours earlier than in the base case, the fans would consequently operate for a longer period of time. The heating energy consumption increased by 4642 kWh. Finally, the energy budget of the HVAC systems is reduced by 11%.

The computer simulations also demonstrated that the reduction of the space cooling load during an accepted temperature drift does not necessarily indicate energy savings in a large office building. In some circumstances the chilled water demand and consumption may increase since the HVAC system is not appropriately designed for this energy conservation strategy. For example, if the discriminator control (which resets cold deck temperature based on the zone with the largest cooling demand) is replaced by a fixed set-point control, then the chilled water demand and consumption increase for the proposed design (Fig. 13), although the profiles of indoor

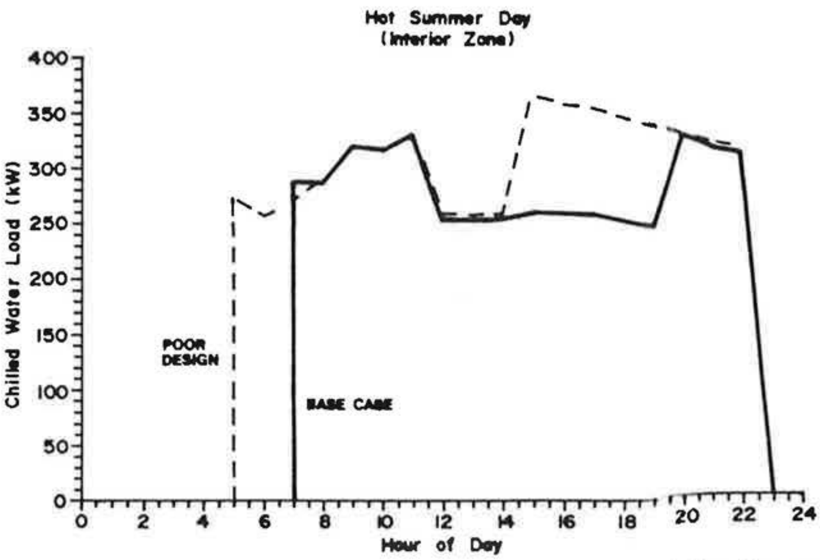

Fig. 13. Hourly variation of chilled water consumption for an inappropriate design. Base case: peak demand = 329.8 kW, daily consumption = 4511.0 kWh. Proposed design: peak demand = 363.9 kW, daily consumption = 5537.0 kWh.

Table 1. Seasonal energy consumption per component

System load	Base case	Proposed design	Difference
Heating (kWh)			
Perimeter	4 379	8 727	+4 348
Interior	44	338	+ 294
Cooling (kWh)			
Perimeter	198 300	151 600	−46 700
Interior	59 700	37 150	−22 550
Electricity (kWh)			
Perimeter	124 700	126 500	+1 800
Interior	136 700	141 900	+5 200
Energy budget for all systems (kWh/m^2)	58.28	51.89	−11.0%

temperature and space cooling load remain the same as in Figs 6 and 7. Consequently, the chilled water consumption increases by 22%, while the peak demand increases by 10% on a hot summer day.

CONCLUSIONS

Important energy savings can be obtained in office buildings in summer by allowing the indoor temperature to increase in the afternoon, to reach or even exceed the limits of the thermal comfort zone. This paper presents a computer simulation, performed for both thermal comfort and energy consumption, as a practical way to evaluate the effect of this energy conservation measure. The thermal inertia of a building plays an important role in the variation of the indoor air temperature, and the results can only be applied to buildings with similar characteristics. Moreover, the type and operation of HVAC systems can seriously affect the energy consumption, as a reduction in the space cooling load does not automatically lead to a reduction of energy use. Further analysis is needed to evaluate the best strategy for different types of HVAC systems in large office buildings.

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APPENDIX

DESCRIPTION OF REFERENCE OFFICE BUILDING

Location: Montréal.
Size: 30×30 m, 10 stories high, total floor area = 9000 m^2.

One interior and four perimeter zones are defined on each floor.

Occupancy: 7 persons/100 m^2;
Lighting: 22 W/m^2;
Equipment: 8 W/m^2.

Building envelope:

(i) Insulated spandrel glass wall construction, $U = 0.313$ W m^{-2} °C^{-1};
(ii) Built-up roofing, $U = 0.204$ W m^{-2} °C^{-1};
(iii) Double-pane windows, $U = 2.22$ W m^{-2} °C^{-1} and shading coefficient SC = 0.60. Glazing-to-wall area ratio is 0.31;
(iv) Air infiltration rate occurs on the perimeter zones only while the HVAC system is on and is evaluated to be 0.19 L s^{-1} m^{-2} of the gross exterior wall.

HVAC system:

(i) Two central VAV systems with perimeter reheat, one serving floors 1 through 5, and second for floors 6 through 10;
(ii) Operating hours: 7:00 a.m. to 11:00 p.m. on weekdays only;
(iii) Ventilation rate: 10 L s^{-1} $person^{-1}$;
(iv) Heating set-point temperature = 21.7°C, cooling set-point temperature = 24.5°C, night setback temperature = 13°C;
(v) Dry-bulb economizer system;
(vi) Centrifugal chiller, COP = 3.0;
(vii) Hot water boiler using natural gas, efficiency = 60%.