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WINDOW PERFORMANCE FOR HUMAN THERMAL COMFORT

FINAL REPORT TO THE NATIONAL FENESTRATION RATING COUNCIL

FEBRUARY 2006

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TABLE OF CONTENTS

TABLE OF FIGURES

TABLE OF TABLES

EXECUTIVE SUMMARY

Anyone who has ever sat near a cold window on a winter day or in direct sunlight on a hot day recognizes that windows can cause thermal discomfort. In spite of this broad recognition there is no standard method to quantify the extent of such discomfort. The purpose of this study was to:

1. Review the literature to identify relevant work relating to windows and thermal comfort.
2. Develop an improved understanding of the impact of windows on thermal comfort and to propose an analytical method for evaluating this impact. The method could form the basis for a future NFRC window comfort rating method that could be used by both designers and consumers.

Literature review. We identified nearly 200 papers, articles and books that contain relevant information to the topic. This report summarizes the conclusions in some detail and also includes an annotated bibliography of 42 of the most important of these sources. We hope that this review serves as an excellent primer on the ways in which windows affect thermal comfort.

In brief, while there has been considerable work done on the fundamental issues related to windows and thermal comfort (e.g. long-wave radiant heat transfer, solar transmittance, induced convection) there is a lack of research that specifically addresses how to evaluate comfort at a level of detail sufficient to compare one window product to another that could be used as a basis for a rating system.

Analysis. Thermal comfort is influenced by a combination of physical, physiological and psychological factors. ASHRAE Standard 55 defines thermal comfort as “that state of mind which expresses satisfaction with the thermal environment.” Over the last century, considerable research has been undertaken on the factors that influence thermal comfort. The basis for our analysis in this report was simulation rather than physical testing with human subjects. Human subject testing is both expensive and difficult to carry out. Fortunately, models of thermal comfort have been developed that can be used to predict subjective comfort assessment and these models have been validated against human subject studies. The PMV model (Fanger 1970) is the most widely used thermal comfort model. However, this model was developed using tests that were done in uniform thermal environments and windows almost always create asymmetric thermal environments. Because of this and other limitations of the PMV model, UC Berkeley has developed a more sophisticated thermal comfort model that is capable of assessing comfort in non-uniform, non-steady-state conditions (Zhang 2003, Huizenga 2001). This model is capable of predicting *local* discomfort such as what is typically caused by a hot or cold window.

We used the UCB Comfort Model to assess a wide range of conditions that might be created by a wide range of window products. The two most significant aspects of windows and comfort that we focused on are 1) the effect of window surface temperature on long-wave radiation heat exchange between the body and the window, and 2) the effect of solar radiation transmitted by the window and absorbed by the body. With respect to solar radiation, we considered only diffuse radiation based on notion that direct sun falling on the body will cause discomfort in all but the coolest environments and that some action will be taken by the occupant to mitigate direct sun. Diffuse solar still has a significant effect on comfort, though significantly less than direct solar. To simplify the analysis process, we developed a method to represent the effect of diffuse solar radiation as an equivalent rise in temperature of the window that would result in the same overall heat gain to the body from the window. This method, detailed in the report, allowed us to model both effects by changing the window surface temperature.

Another key factor in determining the impact of a window on comfort is geometry. Obviously, the closer a person is to a window, or the larger the window, the greater the impact on comfort.

These two parameters, distance to the window and window size, combine to define the *view factor* between the person and the window. The greater the view factor, the greater the impact on comfort the window can have. We created some standard geometries to define a range of view factors. These included distance from the window of 1m, 2m, and 4m; window sizes from a single double-hung window (1.2m x 1.5m) to a fully-glazed façade; and windows on a single wall or windows on two adjacent walls. To illustrate how the methods we describe in this report could be applied to a comfort rating, we chose geometries with large view factors to calculate possible indices. This was done to clearly differentiate between different window products. In any rating method that might be adopted by NFRC, these assumptions will need to be considered carefully to meet the objectives of the council.

Comfort effects of windows are fundamentally different in summer and winter. The winter effect is largely driven by inside window surface temperature, which in turn is tightly correlated with window U-factor and outside temperature. The summer effect is driven by a combination of the inside surface temperature and transmitted solar radiation, both of which are heavily influenced by the optical properties of the window. As a result, our recommendation is to include both a winter and summer comfort rating, akin to the U-factor (nominally a winter rating) and SHGC (nominally a summer rating) currently used.

Winter Rating. The winter comfort impact is driven by the inside glass surface temperature. As such, we propose that this rating be based on the U-factor of the glass, which correlates well with inside surface temperature for almost all window products. The rating we propose is the minimum outside temperature for which an occupant would still be comfortable sitting near the window. The advantage of this rating is that is easy to understand and also differentiates between typical window types on the market. There are a range of possible configurations and assumptions that could be used (with respect to window size, distance from the window, etc.) but we provide example ratings based on the assumptions described in Table 1.

Distance from the window:	1 meter
View factor:	0.26
Metabolic rate:	1.2 met
Clothing:	0.59 clo (typical indoor clothing)
Inside temperature:	23.5°C

Table 1: Winter rating – Boundary conditions

Based on these assumptions, the winter comfort rating is defined as:

$$\text{Winter comfort rating (°C)} = [-45.3/\text{U-factor (W/m}^2\text{-K)}] + 23.5\text{°C}$$

Example ratings for some typical window types are shown in Figure 1.

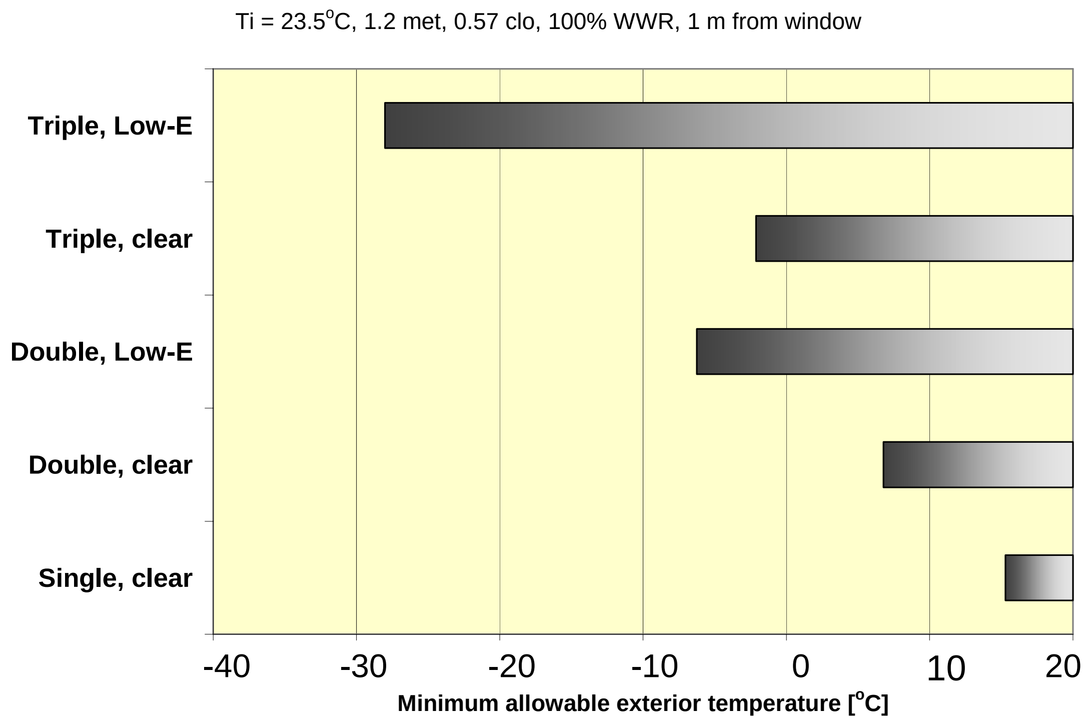


Figure 1: Winter window comfort rating – Minimum allowable exterior temperature

Summer rating: In summer, solar radiation and the optical properties of the window most heavily influence the inside surface temperature of the window. As a result, exterior temperature does not characterize the effect of the window on comfort as it does in winter and using a summer rating akin to the winter rating would have little value. The two important characteristics of a window with respect to warm season comfort are Tsol (direct solar transmittance) and $SHGC_{indirect}$ (defined as SHGC – Tsol, or the portion of SHGC that is absorbed by the window and then retransmitted to the interior). Tsol determines how much solar radiation is transmitted directly through the window that could fall on the body.

In our analysis, we considered two scenarios: (1) a case where the occupant is only exposed to diffuse radiation, making the assumption that direct sun is such a strong effect that an occupant would take action to correct a situation where direct sun was falling on them (by adjusting a blind or shade or even repositioning themselves) and (2) a case where the occupant is exposed to direct solar radiation. $SHGC_{indirect}$ characterizes the increase in the inside window surface temperature due to solar radiation being absorbed by the glazing system. The absorbed energy is then emitted to the inside environment by convection and radiation from the interior glazing surface.

Our analysis shows that for scenario 1 – occupant only exposed to diffuse radiation – $SHGC_{indirect}$ is approximately 4.4 times more important in its impact on comfort as Tsol. However, for scenario 2 – occupant exposed to direct radiation – $SHGC_{indirect}$ is only approximately 2.4 times more important in its impact on comfort as Tsol.

This suggests that a reasonable index that captures the impact of a window on summer comfort could be defined by as a combination of Tsol and SHGC, which we call the *Summer Comfort Index*. This can be thought of as the effective solar gain from the perspective of thermal comfort.

The relative importance of $SHGC_{indirect}$ depends on the assumptions one makes about direct solar radiation.

Scenario 1 – only diffuse radiation: *Summer Comfort Index* = Tsol + 4.4* $SHGC_{indirect}$

Scenario 2 –direct radiation: *Summer Comfort Index* = Tsol + 2.4* $SHGC_{indirect}$

Table 2 shows example comfort ratings for some selected glass types. A lower rating indicates a lower negative impact in comfort. Note that a single absorbing glass (e.g., bronze) and the clear triple glazing have the highest value in the table since they both have a high $SHGC_{indirect}$.

Glazing system	Tsol	SHGC	$SHGC_{indirect}$	*Summer Comfort Index*
bronze, single	0.49	0.62	0.13	1.06
clear, triple	0.49	0.62	0.13	1.06
low-e, double	0.47	0.59	0.12	1.00
clear, double	0.61	0.70	0.09	1.01
clear, single	0.77	0.82	0.05	0.99
low-e, triple	0.34	0.45	0.11	0.82
low-e, selective, double	0.31	0.36	0.05	0.53
low-e, selective, triple	0.25	0.31	0.06	0.51

Table 2: Summer comfort ratings (considering only diffuse radiation) for example glass types

1 INTRODUCTION

Anyone who has ever sat near a cold window on a winter day or in direct sunlight on a hot day recognizes that windows can cause thermal discomfort. In spite of this broad recognition there is no standard method to quantify the extent of such discomfort. HVAC systems are designed to respond to the air temperature sensors, which in fact do not reflect the radiation problems caused by windows. Even when HVAC designers specify dedicated perimeter heating and cooling systems to mitigate window-related comfort problems, they use simplified assumptions that may not solve the comfort problems or that might lead to designs that are energy-inefficient. Window manufacturers promote the positive impact on comfort of high performance glazing, yet they have no real way of quantifying this impact, nor do consumers have a way to compare products with respect to comfort.

The Fenestration chapter of the 2005 ASHRAE Handbook of Fundamentals offers basic guidance about windows and comfort for the designer:

> *"In heating-dominated climates, windows with the lowest U-factor tend to give the best comfort outcomes... In cooling-dominated climates or for orientations where cooling loads are of concern, windows with the lowest rise in surface temperature (for a given SHGC) tend to give the best comfort outcomes."*

This rather limited advice does not provide any explicit way to evaluate whether a given product will produce satisfactory results. No basis for this recommendation is provided, nor does it makes any reference to the wide range of modern products such as low-e glazing, the current standard in high performance glazing systems.

2 OBJECTIVE

The objective of the research project is to develop an improved understanding of the impact of windows on thermal comfort and to propose an analytical method for evaluating this impact. The method could form the basis for a future NFRC window comfort rating method that could be used by both designers and consumers. It would also provide useful information for an improved treatment of the comfort impacts windows in ASHRAE *Standard* 55.

3 LITERATURE REVIEW

This section provides the results of a detailed literature search of work related to windows and comfort. It reviews the studies about the comfort impact from short wave, long wave radiation asymmetry (including the asymmetry from ceilings, walls, windows, and floors), and the comfort limits developed for standards. An annotated bibliography of the most relevant literature is presented in Section 8 and a complete bibliography in Section 9.

3.1 HOW WINDOWS INFLUENCE COMFORT

A window influences thermal comfort in three ways (Figure 2):

- long-wave radiation from the warm or cold interior glass surface
- transmitted solar radiation
- induced air motion (convective drafts) caused by a difference between the glass surface temperature and the adjacent air temperature

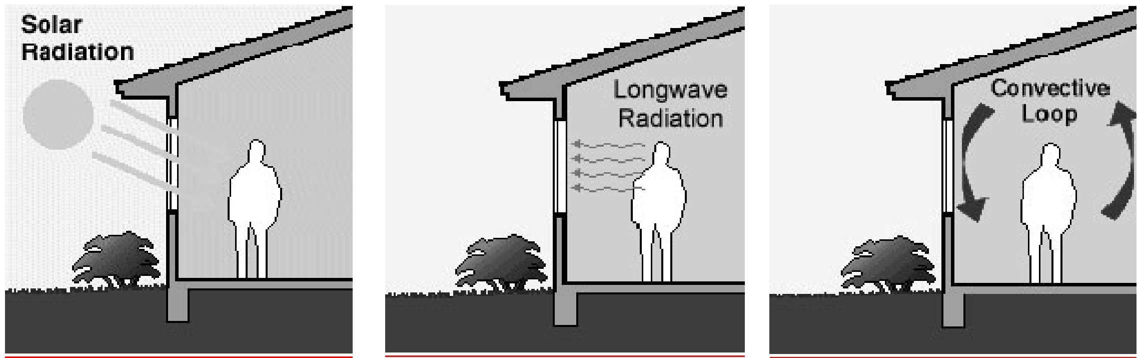


Figure 2: Window impacts on thermal comfort: solar radiation, long-wave radiation, convective drafts

Windows absorb and transmit a significant amount of solar radiation. *Absorbed* radiation influences the temperature of the glass; the inside surface of heat absorbing glass can routinely reach temperatures above 120°F (50°C) in summer conditions, raising MRT by as much as 15°F (8°C). *Transmitted* radiation often causes discomfort if it falls directly on the occupant. A person sitting near a window in direct solar radiation can experience heat gain equivalent to a 20°F (11°C) (Arens et al. 1986) rise in mean radiant temperature. These radiant heating and cooling effects act on the occupant's body *asymmetrically*, causing some parts of the body to be considerably cooler or warmer than a uniform model like MRT can describe. Models need to consider the effect on *local* skin temperature in order to be sensitive to discomfort caused by windows.

The inside surface temperature of a window is heavily influenced by exterior conditions and this temperature can significantly affect the radiant heat exchange between an occupant and the environment. If this heat exchange becomes greater than or less than the acceptable range, discomfort will result. Mean radiant temperature (MRT), defined as the uniform temperature of an imaginary enclosure in which the net radiation heat exchange between the occupant and the enclosure equals the net radiation heat exchange in the actual environment, is commonly used to simplify the characterization of the radiant environment. On a cold day the inside surface temperature can easily drop below 15°F (-9°C) for a clear single pane window and below 40°F (4°C) for a clear, double pane window. If the occupant is sitting sufficiently near the window, MRT could drop to 55°F (13°C) for the single pane case and 62°F (17°C) for the double pane case. Based on ASHRAE *Standard* 55, even the use of the double pane window could result in discomfort.[1] In addition to the MRT effect, a cold inside glass surface can induce a downward draft that increases air movement, contributing to further discomfort.

Understanding the influence of windows on thermal comfort is important not only to help designers create comfortable buildings, it will also help evaluate the benefits of improved windows. Although it is well understood how high-performance windows can reduce building energy consumption, a better understanding of how they affect comfort might lead to even greater savings. For example, a study (Hawthorne and Reilly 2000) suggests that there are significant energy implications to the standard practice of using perimeter duct distribution in houses to mitigate potential discomfort caused by windows. They found that perimeter heating is often not necessary when high performance windows are installed, and that heating energy savings of 10% to 15% could result from installing a simpler, less expensive duct system.

[1] This example assumes an outdoor air temperature of 0°F (-18°C), indoor air temperature of 72°F (22°C), non-window surface temperatures of 72°F (22°C), occupant-window view factor of 0.3, 0.9 clo (standard winter indoor clothing), activity level of 1 met.

3.2 SHORT-WAVE RADIATION

In a cold winter morning, direct sunlight on a person's body may be perceived as a pleasant presence. However, in summer daytime, solar radiation will almost certainly cause discomfort. Short-wave radiation causes thermal discomfort directly when it is absorbed by the body (or clothing) and indirectly by increasing the air and surface temperatures in a space. This latter impact is in theory moderated by the HVAC system. In our analysis, we assume that the HVAC system controls the air and surface temperatures in the space to compensate for solar gain. In practice, HVAC systems rarely achieve perfect control and as a result, solar gain often raises the operative temperature in perimeter zones.

Hausler and Berger (2002) found that when the air temperature is 22ºC or above, direct solar radiation on the body causes discomfort. Schutrum et al. (1968) tested subject thermal sensation with solar radiation transmitted through a window whose glass temperature was separately controlled. When the glass temperature increased from 3ºC to 48ºC (room air temperature increased from 23.7 to 24.3ºC) on a cloudy day, overall sensation elevated 1.1 units (from slightly cool to neutral, 7-point scale,). On a clear day with solar radiation on the body, the window temperature increased to 31.7ºC and the overall sensation became 2.5 units warmer (from slightly cool to above slightly warm).

In summer, thermal comfort is mostly uncorrelated with U-value but is closely related to solar transmittance (Lyons et al. 1999). In fact, solar transmittance is the controlling factor with respect to the effect a particular glazing system will have on comfort. The Tsol of the glass determines the amount of solar gain that is transmitted into the space. Simulations show that replacing single 3mm clear glass (Tsol = 0.83) with double 3mm low-E glass (Tsol = 0.53) can reduce discomfort by more than half (Lyons et al.1999). Sengupta et al. (2005) simulated comfort in a room and the results demonstrated that in summer daytime (780 W/m^2 solar radiation) the change from clear single glazing to clear double glazing did not significantly improve thermal comfort. However, reducing the window size (from 40% to 20% glass-to-floor area ratio) significantly improved the comfortable floor area. Olesen & Parsons (2002) also discuss radiant discomfort and state that "direct solar radiation should be avoided in the occupied zone, by means of building design or solar shading devices".

Because solar transmittance is a major factor in determining comfort, a logical way to reduce solar radiation is to use glazing with a spectrally selective coating. Although glass temperature normally increases because of the higher solar absorptance of the film coating compared to normal glass, the reduction in solar transmittance can be more than 50% (Arasteh et al. 1987, Alvarez et al. 1998, Karlson et al. 1988, Estrada-Gasca et al. 1993a and 1993b, Howthorne and Reilly 2000). Ideally, a spectrally selective coating should have a small effect on visible transmittance, to preserve daylight and views, but will be near-opaque at other wavelengths. This is especially critical in the automobile industry where for safety reasons the visible transmittance must be 70% or above (Bohm et al. 2002, Nair and Nair 1991). Clarke et al. (1998) provided a summary for the performance of the advanced glazing systems that is presented in Table 3.

Insulating glazing	The insulating properties of a window can be improved by including multiple glass layers, by applying low-ε coatings to layers surfaces, or by using a low-conductivity gas such as argon or krypton instead of air (Arasteh *et al.*, 1987).
Spectrally selective coatings	Spectrally selective coatings can be applied to the glass to reduce transmission at specific wavelengths. Of foremost interest are those coatings which give solar control with little adulteration of view, i.e. glazings that have minimum effect on visible light but are opaque at other wavelengths, particularly within the infrared portion of the solar spectrum (e.g. a typical advanced glazing product will have a visible transmittance of 67% for a total solar transmittance of only 37%).
Edge spacers	These provide the seal for multiple glazing and are invariably made of aluminium or steel to provide mechanical strength under thermal stress. Because such materials give rise to a high conductivity thermal bridge, low-conductivity spacers are being developed (Aschehoug and Baker, 1995; Svendsen and Fritzel, 1995).
Insulating frames	Typical double glazing frames have higher *U*-values than the corresponding centre pane *U*-values for low-ε double glazing. With frames comprising some 20–25% of the aperture area, or 10% in the case of a curtain walling system, improved frame systems are required to achieve a low, overall component *U*-value (Beck and Arasteh, 1992).
Variable transmittance	Adjustable glazing systems offers the best prospect for providing an optimum solution as occupant and system needs vary throughout the year and daily. Blinds or louvers operated under automatic or manual control offer one option. An elegant alternative, which is currently under development, is electrochromic glazing whereby the transmittance can be varied by the application of a small voltage, which changes the oxidation state of the electrochromic coating. Both visible and total solar transmittance can be varied from their normal value (say 60%) down to some lower value (say 10%). The time taken to change (from the maximum to minimum values) is typically about 5 min. Because the voltage is only required to effect the change, and need not be sustained thereafter, the power consumption is small. The effective integration of this technology will require the development of appropriate control algorithms (Sullivan *et al.*, 1994).

Table 3: Advanced glazing (Clarke et al. 1998)

In buildings, ways of blocking solar radiation can also be achieved by applying shading elements, such as overhangs or curtains, (e.g. the Phoenix Public Library designed by William Bruder, and the Arup Campus designed by Arup Associates). They both used shading devices and the results are very good in terms of keeping both thermal and visual comfort). There are many studies carried out at Lawrence Berkeley National Laboratory (LBNL) that look at thermal and visual performance of shading systems. For example, a study by Lee et al. (1998) compared the energy and lighting performance of an automated venetian blind with a static venetian system. The automated blind was operated in synchronization with a dimmable electric lighting system to: block direct sun, provide the design workplace illuminance, and maximize view. Through a year-monitoring, they found that the energy consumption (used in cooling and lighting) and peak demand were significantly reduced compared to the static blind.

The impact of direct solar radiation on occupant comfort also depends on the absorptance of clothing skin. Solar absorptance of clothing depends on the color and property of the fabric. Morton and Hearle (1993) give a detailed description of solar properties based on the structure of the fabric. The absorptivity of skin varies with color in the visible and the near-infrared spectra. For visible wavelengths (0.4 – 0.7μm), white skin is about 0.5, while black skin is 0.74 (Houdas and Ring 1982). In the near infrared (0.8 – 2μm), the absoptivity of black skin is also higher than the white skin. For ultraviolet (<0.4μm) and far infrared (>2μm), there is no significant difference. Narita et al. (2001) tested human skin for thermal sensitivity to radiation at different wavelengths and found that human skin is more sensitive to the visible (0.3 – 0.8μm) and middle-infrared (1.7 – 2.3μm) than to near-infrared (0.8 – 1.35μm) wavelengths.

3.3 LONG-WAVE RADIATION

Long-wave radiation from a warm or a cold window affects occupant comfort in two ways. First, it influences the overall radiative heat exchange between the body and the surroundings, affecting the body's heat balance and therefore comfort. Second, even when there is a neutral overall heat balance, local discomfort of one or more body part may result from asymmetric radiation fields near windows.

ASHRAE and ISO standards define a comfort zone for the body based on overall heat balance, however thermal neutrality is not the only condition to ensure thermal comfort. A person may feel thermally neutral for the body as a whole, but he may not be comfortable if one part of the body is warm and another cold. A further requirement to comfort is that no local warm or cold discomfort exists at any part of the body. Local discomfort can be caused by radiation temperature asymmetry such as a warm or cold window, ceiling, wall, or floor; excessive air motion (draft); or vertical air temperature stratification. Comfort standards prescribe limits for these parameters. This section reviews the literature addressing radiation asymmetry caused by ceilings, walls, floor, and windows, and for limits on asymmetry as prescribed in comfort standards.

3.3.1 RADIATION ASSYMETRY FROM WALL AND CEILING

In a uniform environment, the relative contribution of mean radiant temperature on comfort is considered approximately equal to that of the air temperature (Fanger et al. 1980, Fanger 1972), or slightly smaller (McNall and Schlegel 1968, McIntyre and Griffiths 1975). McIntyre and Griffiths (1972) give the relative influence of radiant and convective heat transfer as 0.44 and 0.56, respectively. When the radiation on the body is asymmetric, the geometry and shape of local body parts influence the relative importance of radiation and convection.

Radiation exchange with the surrounding environment is commonly defined in terms of radiant temperature. Radiation from the hemisphere surrounding a plane surface can be defined in terms of 'plane radiant temperature', the uniform temperature of an enclosure in which the irradiance on the plane is the same as in the existing non-uniform environment. The vector radiant temperature (VRT) proposed by McIntyre (1974) is used to measure the asymmetry. It is equal to the difference in plane radiant temperature on opposite sides of a small plane element, when the element has been oriented so as to make the difference a *maximum*. Instead of getting the maximum difference, by fixing the orientation of the plane, Fanger et al. (1980) proposed radiant temperature asymmetry (RTA). It is the difference between the plane radiant temperatures on two opposite sides of a small plane element of a *fixed* orientation. For a heated ceiling, Fanger positions the plane horizontally 0.6m above the floor. Therefore, for measuring the vertical radiant asymmetry between floor and ceiling, the vector radiant temperature (VRT) and radiant temperature asymmetry (RTA) are equivalent. The effects of asymmetric radiation on thermal comfort have been extensively investigated through human subject tests at Kansas State University in the 1960s. Under non-extreme conditions (overall sensation between cool and warm), Schlegel and McNall (1968) found that a wall with 6.7ºC (warm and cool) surface temperature difference from the rest of the surfaces (the chamber was 12 ft and 24 ft in plan and an 8 ft ceiling height, five subjects stayed 3.5 to 4.5 ft from the wall whose temperature was changed, a view factor of 0.2) created no noticeable difference from comfort results under uniform test conditions. The authors then examined the impact of hot and cool walls and ceilings, with a wider range of radiation asymmetries (McNall and Biddison 1970). The cool wall temperature was 9ºC, 11ºC lower than the surface temperature of the rest of the space (the view factor was again 0.2). The hot wall temperature was fixed at 54ºC, 41ºC warmer than the rest of the surfaces, a quite extreme condition. The cool ceiling was 10.5ºC, 16ºC lower than the rest surfaces (view factor 0.12). The hot ceiling was 54ºC, 38ºC higher than the space. The room air temperatures were designed at various levels. A control test was also conducted with a neutral uniform environment (25.5ºC). Each test lasted three hours and the thermal questionnaires were answered in 30-minute intervals. The human subjects were asked both sensation and comfort questions, and the results are presented in terms of sensation and percentage of subjects feeling comfortable (Figure 3). A total of 170 subjects participated in these tests.

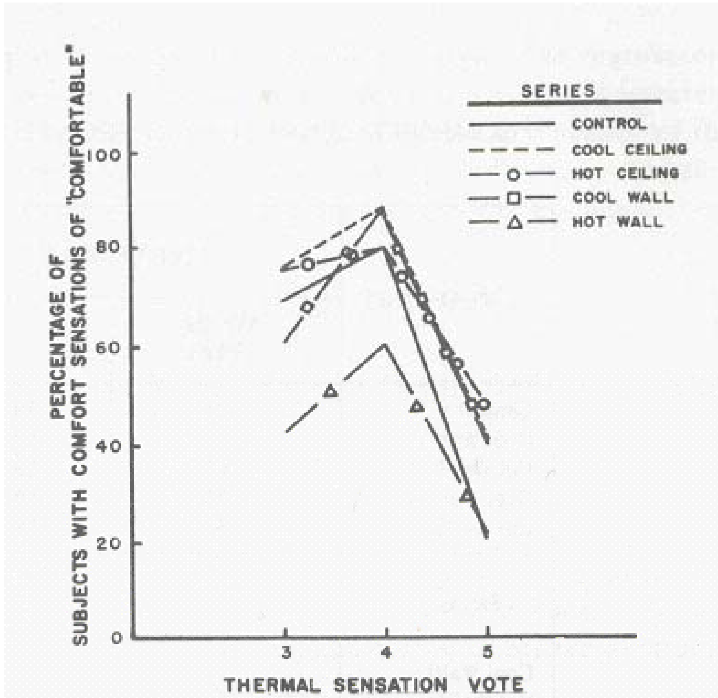


Figure 3: Percentage of comfort in relation to thermal sensation (McNall and Biddison 1970)

In Figure 3, we see that a larger percentage of subjects felt uncomfortable with the hot wall than with the other three radiation asymmetries. The three asymmetries (cool ceiling, cool wall, and hot ceiling) were also more comfortable than the uniform neutral condition (control test). At the neutral overall sensation (value of 4), the percentage of subjects feeling comfortable with the cool wall and cool ceiling is 7% higher than in the uniform control condition.

In order to eliminate discomfort due to whole-body warmth or cool discomfort, the authors used only comfort responses when thermal sensation was neutral for their analysis of impact from asymmetrical radiation. Statistically, only the hot wall has a significantly lower probability of resulting in a "comfortable" vote than those of the control tests. When the overall sensation is neutral, the cool wall as low as 9ºC, cool ceiling as low as 10.5ºC, and the hot ceiling as high as 54ºC, would not cause discomfort due to asymmetric radiation. With the overall whole body thermally neutral, subjects exposed to the 54.4ºC warm wall experienced significant discomfort due to asymmetric radiation. The subjects showed a larger tolerance for the hot ceiling than the hot wall. This might be because in the test chamber, the view factor for the ceiling (0.12) is smaller than for the wall (0.2).

A series of tests were carried out at the Technical University of Denmark (DTU) in the late1970s regarding the impact of radiation asymmetry on comfort. In examining impact from a heated ceiling (Fanger et al. 1980), the test was conducted in a chamber (4.7 x 6 x 2.4 m). A light ceiling (2.2 x 2.2 m) was suspended at a height 2 m above the floor, giving a view factor of 0.11. In examining the impact from a heated/cooled wall (Fanger et al. 1985), 4 panels (2 x 1.6 m), 0.5 m away from the center of the person (view factor 0.25) was installed. The cooled ceiling size was 2 x 1.6 m, plus 2 x 0.9 m vertical sections on both sides (view factor 0.2). The authors reached different conclusions than the KSU tests: the subjects were most tolerant with the warm wall (radiant temperature asymmetry up to 23ºC) and the least with the warm ceiling (radiant temperature asymmetry up to 4ºC).

In the DTU tests, the subjects first stayed in the chamber for an hour. Then the environment was changed five times in the following 2.5 hours to correspond to five radiation asymmetries.

Sixteen females and sixteen males participated in the "cool wall" test (wall temperature from 0.4ºC to 17.8ºC) and eight females and eight males participated in the "warm wall" (32.6 to 70.1ºC), "cool ceiling" (0.8 to 16.0ºC) and "warm ceiling" (34 to 69ºC) tests. The air temperature was adjusted according to the subjects' requests to keep their feelings neutral. The results are presented as the radiant asymmetry vs. percentage of dissatisfied (PPD) (Figure 4). Because people showed more sensitivity to a cool wall than to a warm wall, the authors state that local cooling of the body seems to more frequently cause discomfort than local heating. Comparing the subjects' sensitivity to the warm/cool wall to that of the ceiling, the authors conclude that people are more sensitive to vertical radiation than to lateral radiant asymmetry. To explain the differences between their results and those of McNall and Biddison (1970), the authors propose that in the studies carried out at Kansas State University only the surface temperatures in the chamber were controlled to maintain a constant MRT. The authors explained that the air temperature was not adjusted in a way to keep a neutral temperature for the subjects. Therefore, the discomfort values measured in the Kansas study were due to both the overall discomfort and to local discomfort caused by radiation asymmetry.

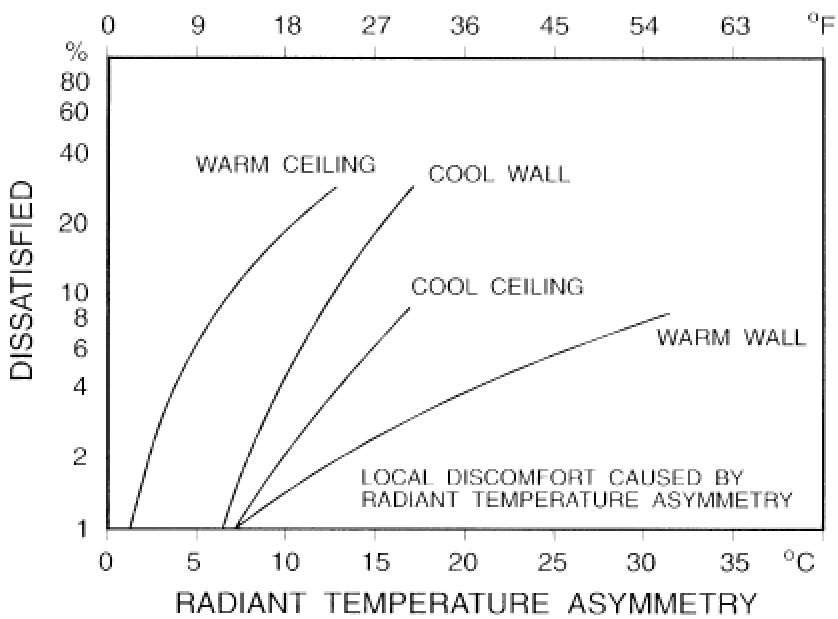


Figure 4: Local thermal discomfort caused by radiant asymmetry (ASHRAE Standard 55 – 2004, same figure as presented by Fanger et al. 1985)

This explanation does not seem sufficient. McNall and Biddison (1970) examined the results by restricting their statistical analysis to data when overall sensation was perceived as neutral, to eliminate the effect of general thermal discomfort. The results still showed that only the warm wall has a significantly lower probability of being "comfortable" than the uniform condition.

A possible explanation lies in the different approaches of the two studies. In McNall's study, when increasing a wall temperature, the remaining five surfaces were simultaneously reduced in temperature so as to keep the same overall MRT. When the warm wall temperature was 54ºC, the rest of the surfaces were at 12.8ºC, while the room air temperature was 25.6ºC. Therefore, subjects felt stronger asymmetry (from both warm and cold walls) than in Fanger's test condition, where the warm wall was 52ºC and the remaining surfaces and the air temperature were kept at 22.3ºC to preserve the neutral sensation. At the cool environment in Fanger's study, the warm wall was perceived as pleasant and therefore was found to have bigger tolerance limits. However, this did not appear in Fanger's warm ceiling test. From Fanger's study, it seems that people are more sensitive to warm ceilings than warm walls. Local discomfort was found either as warm head or cool feet. The room air and the five surface temperatures were low in order to balance the heated ceiling to keep subject neutral. From the results, cool feet discomfort happened more than warm head discomfort when the ceiling temperature was between 34ºC and 52ºC (operative temperatures were the same, around 24.1ºC). It was only with ceiling temperature as high as 63ºC to 69ºC that warm head discomfort occurred more often than cool feet discomfort. So in a way Fanger's resulting limit for heated ceiling was caused partially by cool feet. That is probably why the limit for cool ceiling (14ºC radiant temperature asymmetry, the air and the five surfaces warmer) is higher than the heated ceiling (4ºC radiant temperature asymmetry, the air and the five surfaces cooler) if we assume that people in general are more sensitive to cool feet than warm head. The heated tests cover the radiant temperature asymmetry from 4.5 to 23.6ºC, so in fact the limit of 4ºC radiant temperature asymmetry based on 5% dissatisfied for warm ceiling is not supported by any of the test conditions. There are two other major differences between the two tests. One is that in Fanger's study, subjects stayed in one asymmetric condition for half an hour. During that half-hour, the radiation conditions were changed, and so did the room temperature according to the subjects' requirements. Six subjective surveys were conducted in five-minute intervals. It is not easy to meet subjects' requirements in that short period of time and therefore whole-body discomfort might still exist. Therefore the dissatisfaction might not be totally due to asymmetric radiation. In McNall's study, the test duration was three hours. Another difference was that the comfort rating scales employed in the two studies were different. The questions (comfortable, slightly uncomfortable, uncomfortable, very uncomfortable, intolerable) were asked in McNall's study, while in Fanger's study, if at least twice as much local discomfort was reported during the last three votes, the person was considered uncomfortable.

The ASHRAE (2004) and ISO (1994) standards define the allowable radiant temperature asymmetry categorized as warm/cool ceiling and warm/cool wall. The limits are presented in Table 4. These limits are obtained from Figure 4 with 5% dissatisfied due to radiant asymmetry. These limits are determined based on studies conducted by Fanger et al. (1985, 1980).

Radiant Temperature Asymmetry °C (°F)			
Warm Ceiling	Cool Wall	Cool Ceiling	Warm Wall
< 5 (9.0)	< 10 (18.0)	< 14 (25.2)	< 23 (41.4)

Table 4: Allowable radiant temperature asymmetry (ASHRAE 55 – 2004)

There have been other studies that tried to find the limits for the asymmetric radiation but the results are not consistent. The first investigation of discomfort from a heated ceiling was performed in the 1950s in England by Chrekon (1953). The heated ceiling increased MRT by up to 12ºC, while the air temperature was kept constant. The percentage dissatisfied was higher compared to Fanger's (1980) study. The reason for this is probably that the subjects experienced an overall sensation of warmth because the temperature of the chamber's other surfaces and the air temperature was not changed along with the heated ceiling. Chrekon recommended a limit corresponding to a VRT of about 6ºC. From the early 1970s McIntyre and Griffiths in England did extensive studies on radiation asymmetry and comfort. Griffiths and McIntyre (1974) exposed 24 subjects to a control condition and three different ceiling temperatures. The method is similar to that of McNall and Biddison (1970) where the air temperature was kept constant and the temperature of the rest of the surfaces were adjusted while the ceiling was heated. They found that a vector radiant temperature of 20ºC did not produce any significant worsening of subjective responses when compared with a uniform control condition and therefore a VRT of 20ºC was recommended. This finding is similar to what was found by McNall and Biddison (1970) when the ceiling temperature was 54ºC and the VRT was 24ºC. However this was significantly higher than the results from Fanger (1980), where the radiant temperature asymmetry limit for heated ceiling was 4ºC. McIntyre (1977) did a follow-up study to change both the temperatures of the remaining surfaces and the air while increasing the ceiling temperature (a similar approach to Fanger's), and found that not only did the heated ceiling produce no discomfort, but indeed it was preferred when the ceiling temperature was high and the air and other surfaces were cooler. This confirms our earlier explanation about the difference between Fanger et al. (1985) and McNall and Biddison (1970) regarding the warm wall. We explained that in Fanger's test, when the wall was warm (52ºC), the rest of the surfaces and the air temperature was 22.3ºC, and the warm wall was perceived as pleasant and therefore allows bigger tolerance limits. In McIntyre's similar test configuration (1977), he invited subjects to attribute discomfort specifically to the overhead radiation and the responses show a significant increase of discomfort with ceiling temperature. This is a contradictory result. It appears that people are ready to attribute discomfort to unusual aspects of their environment. For that reason, because people clearly notice the asymmetrical radiation at VRT of 10ºC, although no one was actually more uncomfortable than in the uniform environment, McIntyre and Griffiths (1977, 1975) recommended a VRT of 10ºC as the limit, which is also similar to the recommendations by Schroder and Steek (1973), 9ºC, and Banhidi (1972), who recommended a VRT in the range 8.5 to 13ºC.

Olesen et al. (1972) exposed nude subjects to a lateral asymmetry and found the tolerable limit of radiant temperature asymmetry to be 10ºC. He recommended limits for asymmetry for closed subjects based on heat transfer calculation, which agreed well with the recommendation of a VRT of 20ºC by McIntyre and Griffiths (1972), within the range of recommendations given by Fanger et al. (1985, cold wall: 10ºC, warm wall: 23ºC). Olesen and Nielsen (1981) tested the spot cooling effect from a cold vertical panel (1m x 2m, 0ºC) in a warm environment (30ºC). The cold panel was placed 0.5m behind the back of the subjects (view factor 0.25). The overall sensation was a cooling from 1.68 to 1.12, a 0.5-unit scale reduction. The overall comfort was still around "uncomfortable", from 2.10 to 1.78[2]. The subjects' overall thermal acceptability increased from 38% to 50%, but was smaller than the values for uniform conditions. The author explained that the rather limited increase in acceptability due to cooling was reduced due to the radiant asymmetry. Looking at the local sensation votes, the coldest sensation came from the body parts which were most exposed to the cold radiation: back (-0.18) and neck (0.33), while the non-radiant body parts felt slightly warm: face 1.24, chest 1.26. This suggests it would be preferable if the view factor were also calculated for these most exposed body parts (e.g. back and neck)

Displacement ventilation has the advantage of saving energy and providing better air quality for the breathing zone. However, it is limited in its ability to convectively remove the heat loads encountered in offices. Additional mechanisms such as chilled ceilings may become necessary, which also have the effect of reducing the air stratification in the occupied zone (Feustel and Stetiu, 1995). Kuelpmann (1993) tested the performance of displacement ventilation with a cooled ceiling under several typical configurations and found a radiant temperature asymmetry of 8ºC, smaller than the limits set up by the standard. Hodder et al. (1998) examined thermal comfort in a more sophisticated thermal environment with a chilled ceiling and displacement ventilation. A typical ceiling tile surface temperature is normally in a range of 16ºC to 19ºC. He tested ceiling temperatures of 12.4, 14, 18 and 22ºC and found that ceiling temperatures in this range had no significant influence on overall comfort and local discomfort. His experiments involved eight female subjects because in their early investigations they had found female subjects to be more thermally sensitive to their environment than males. Another study by the

[2] Scale: 1 - comfortable, 2 - slightly uncomfortable, 3 - uncomfortable, and 4 - very uncomfortable

same authors (Loveday et al. 1997, 2002) involving a larger number of subjects (184) also showed that no significant local discomfort happened in the cool-ceiling and displacement-ventilation environments. Kitagawa et al. (1999) examined the effect of humidity and low levels of air motion on thermal comfort under a cool-ceiling environment and found that better comfort votes were obtained in the condition when thermal sensation vote was not neutral but near –0.5.

Summarizing: Many studies have been conducted to define limits for radiant asymmetry caused by cooled/heated ceilings and walls. For heated ceilings, most of the literature suggests a much higher acceptable radiant asymmetry limit (10ºC to 20ºC) than the 5ºC provided by the standard. For warm walls, there is a large difference between the limits provided by McNall and Biddison (1970) and Fanger et al. (1980). Cool ceilings and walls seem to have similar limits. ASHRAE standard 55-2004 defines limits for a warm- or a cool wall (radiant asymmetry temperature <10ºC for a cool wall, <23ºC for a warm wall). Radiant cooling or heating of floors (which will be discussed in a separate section later), ceilings, and walls provide energy-efficient approaches to space conditioning (Roulet et al. 1999). Therefore, these studies about radiant asymmetry and comfort have great value in practical applications. The studies of heated or cooled ceilings also provide useful information for evaluating the impact of skylights.

3.3.2 RADIANT ASYMMETRY FROM WINDOWS

Many studies emphasize the importance of a warm or a cold window on comfort. By simulating the thermal impacts of ten generic glazing systems ranging from single-pane window to high performance window, Lyons et al. (1999) concluded that except in the case where the body is directly in the sun, long-wave radiation to and from the window is the most significant factor affecting thermal comfort. When applying advanced glazing, a secondary phenomenon occurs. The inside glass temperature rises. It causes a positive effect in winter, but increases discomfort in summer. Under NFRC summer test conditions, single bronze glazing is 13ºC hotter than single clear glazing because of the higher solar absorptance, which corresponds to a calculated increase of discomfort from 36% to 45% due to long-wave radiation. Sengupta et al. (2005) and Chapman et al. (2004) simulated window impacts on comfort for eight cases covering different glass areas and window configurations. By displaying PMV contours on a plane 1.25 m above the floor, the authors showed very large variations due to the existence of the windows. In summer with solar radiation, the presence of two windows (40% of the wall area) and one window (20% of the wall area) results in only 7% and 25% of the floor area being comfortable (with PMV within –0.5 to +0.5). Large glazed façades are essential features of modern architecture. Gan's simulations (2001) showed that when outdoor air temperature is –4ºC, room air temperature 21ºC, the radiant asymmetry temperature exceeds 10ºC when the location is 1m from the window (room size 5 x 4 x 3 m with a window 3.5 m wide and 2 m high). That means the area within 1 m distance from the window would not meet the ASHRAE standard. Ge and Fazio (2004) measured the inside glass temperatures of a large glass panel when outdoor temperature was –18ºC (NFRC winter condition), and Montreal's worst winter condition, –32ºC. The inside glass temperatures were 10ºC and 3.8ºC, respectively.

Improving window performance reduces thermal discomfort. Sengupta et al. (2005) showed that in winter nighttime conditions, changing a single-pane window to double-pane greatly improved the comfortable floor area. Gan (2001) examined a series of factors regarding the window properties, sizes, and shapes. For a single-pane window, a 10ºC radiant asymmetry exists at 1m from window at an outdoor temperature of –4ºC. A double-glazed window has the same asymmetry 0.15 m from the window at an outdoor temperature of –10ºC. He also demonstrated that square windows are more likely to cause thermal discomfort than narrow windows and when a large window is replaced by several smaller windows (keeping the same window area), the discomfort is greatly lowered.

Although radiant temperature is as important as air temperature, conventional practice is to use air temperature as the measure for controlling mechanical systems. Gan (2001) recommends that, in circumstances where a large radiant asymmetry exists (e.g. a room with a large window), sensors are more effective if they respond to the combination of air temperature and radiant temperature rather than air temperature alone.

3.3.3 FLOOR SURFACE TEMPERATURE

An early study of foot thermal comfort that concerned the effect of floor material was carried out in 1948 by Munro and Chrenko (1948). The subjects were exposed to air temperatures of 12.8ºC and 18.3ºC for 60 minutes. It was found that the floor material had only a very small influence on the preferred floor temperature for people with shoes. The air temperature was the dominant factor determining foot thermal comfort. After that, Muncey and Hutson (1953) and Muncey (1954) also conducted subject tests and also found no influence of the flooring material on foot thermal comfort.

In early 1950s, a study was undertaken at Kansas State University to determine the effect of floor surface temperature on foot and whole-body thermal comfort. Nevins et al. tested the effect of different floor temperatures on thermal comfort for young men and young women seated (reading) and standing (writing and sorting bibliography cards to simulate light office work) three hours with shoes, and concluded that a floor temperature as low as 15.5ºC and as high as 29.4ºC did not cause significant discomfort when air temperatures was 23.9ºC (Nevins et al. 1958, Nevins et al. 1964, Michaels et al. 1964, Nevins et al. 1967). Except for seated women (Michael et al. 1964), a floor temperature of 32.2ºC did not make the subjects uncomfortable. For seated women, the upper temperature limit was lower, 29.4ºC. Their greater sensitivity could be caused by the women's lightweight cotton smocks exposed bare skin of the lower legs to radiation from the floor, and the lower muscular activity in the legs due to sitting may have reduced blood circulation and the removal of the absorbed radiant gain. Later on the authors did a study to examine floor surface temperature on comfort for elderly. They found that at the same floor temperatures, female elderly feels warmer than the male elderly (Springer et al. 1966). No obvious difference in thermal sensation was seen for male between floor temperature 23.9 to 37.8ºC. For female, no obvious difference in thermal sensation was seen between 23.9 – 35ºC, but a jump in thermal sensation at floor temperature 37.8ºC.

Olesen did studies evaluating floor temperature for bare feet (1977a) and for feet with shoes (1975, 1977b). The tests were carried out for standing and sedentary people. Sixteen subjects (eight females and eight males) occupied the floor for 10 minutes with bare feet and gave an evaluation of their foot comfort. Eighty-five subjects were tested with shoes, keeping their feet on the floor for three hours. For floors occupied by people with bare feet, the author found that the floor material is important. The optimal floor temperature for 10 minutes occupancy ranged from 26 – 29ºC for several typical flooring materials. For floors occupied by people with normal indoor footwear, flooring material had an insignificant effect, the same as found in the early studies mentioned above by Chrenko, Muncey and Hutson, and Muncey. Optimal floor temperatures of 25ºC for sedentary and 23ºC for standing or walking people were recommended. At floor temperatures below 20 – 22ºC the percentage of people experiencing cold feet increased rapidly (Olesen 1975). The results are presented in a figure showing the floor temperature and percentage of dissatisfied (1977b). The optimal floor temperature for seated/standing people ranges from 20 – 28ºC with shoes and 23 – 30ºC with bare feet (Olesen 1997a &b). The minimum temperature can be reduced for higher activity levels. Figure 5 shows a similar figure adopted by the ASHRAE and ISO standards to define acceptable floor temperatures. The allowable range of the floor temperature is 19 – 29ºC for the 10% dissatisfaction criterion. The upper limit is in fact the same as the limit provided by the 1956 edition of the ASHAE Guide.

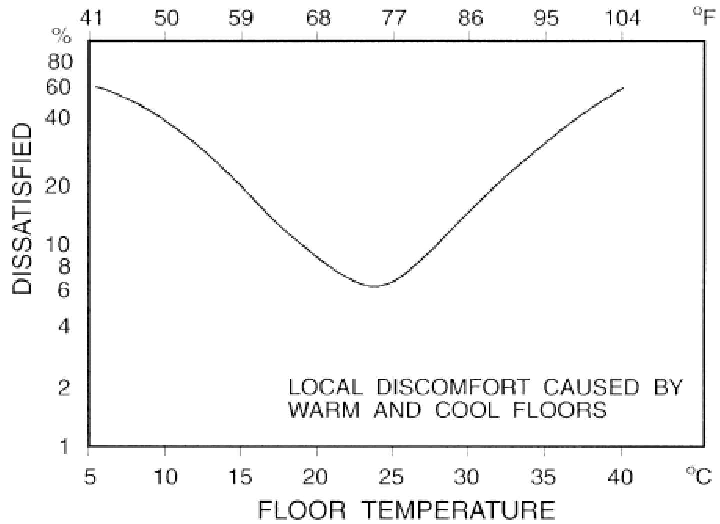


Figure 5: Local discomfort caused by warm and cold floors (ASHRAE 55 – 2004)

A heated floor is regarded as a low-energy heating strategy which provides a high comfort level because of the reduced perception of draft due to the smaller vertical air temperature difference when the floor is heated. Occupants find it pleasant to receive direct thermal radiation from the heated floor (Eijdems 1994, Boerstra et al. 2000, Watanabe 2001). It was popular in Britain during the Roman Empire (Winslom et al. 1949), and is culturally preferred for heating in Korea and Japan (Yoon 1992, Hashiguchi 2004).

Hashiguchi et al. (2004) tested thermal-comfort reactions to floor heating for both elderly and young people and found that the percentage of subjects who felt comfortable was higher for the heated floor (air 21ºC, floor 29ºC) than for the neutral room (25ºC) for both age groups, although the differences were not significant. Sohn (1986) recommends 33.6 – 38.8ºC as the comfortable floor surface temperature, much higher than the value provided by standards. After reviewing a large number of studies about the optimal floor surface temperature in Japan, Zhang et al. (1998) proposed that the optimal floor surface temperature should be within 26 – 30ºC in Japan. The lower limit is also considerably higher than the standards. Katahira et al. (2005) examined thermal comfort in a space with floor heating (no floor surface temperature provided) at air temperatures of 18, 20 and 22ºC, and in air-conditioned space at air temperatures 20, 22, 24ºC. Although the test conditions were generally evaluated as cool and no significant differences were found between the floor-heating and the air-conditioned conditions, the chilliest feeling did go to the 20ºC air-conditioned room. The desire for higher floor temperatures was higher in the air-conditioned spaces than the floor-heating spaces. Tarano et al. (1996, 1997, 2000) conducted a series of studies to examine the comfort impact by heating the sole or legs in a cool environment. One study (1996) showed that in a cool environment (18ºC), a 38ºC hot panel for heating soles (sole sensation was between ‘slightly hot’ to ‘hot’) can remove whole body discomfort. In the study (2000), the authors showed that in a cool environment, a warm thermal sensation of sole at 2 of the ASHRAE 7-point scale seemed to provide the most satisfactory with the sole sensation. In the study (1996), the authors showed that when the lower legs, not only the bottom of the feet, were warmed by both bottom and side heating panels, whole body comfort was achieved in a cool environment (18ºC).

Cooling floors are also considered as an alternative low energy technology for buildings and it has been applied in a large airport (Simmonds 1994, Simmonds et al. 2000), and in residences [Davis Energy Group, personal communication].

3.4 UNWANTED AIR MOVEMENT (DRAFT)

Air motion can be viewed as a pleasant breeze when people are warm, as it is traditionally used for cooling and stimulation in natural ventilated buildings, or it can be considered as an unacceptable cool draft when people are neutral or cool. Draft is defined as an unwanted local cooling of the body caused by air motion. It usually occurs when the body's heat balance is neutral or cold.

3.4.1 DRAFT STUDIES

The sensation of draft from air motion depends on the body's thermal state. The percentage of dissatisfied population is a function of room air temperature (representing the whole-body thermal state), air velocity, and turbulence intensity. Convective heat transfer is roughly proportional to the square root of mean air velocity.

The turbulent intensity, representing the velocity fluctuation which is defined as the standard deviation of the instantaneous velocities divided by the mean velocity, was identified as an important factor on the occurrence of draft sensation (Mayer 1987). Fanger and Pederson (1977) found that the percentage of dissatisfied was much higher for a fluctuating velocity than a constant velocity. Maximum discomfort appears at the air velocity fluctuated at a frequency between 0.3 – 0.5 Hz (Fanger and Pederson 1977, Zhou and Melikov 2002). Fanger et al. (1989) investigated the effect of turbulence intensity on the sensation of draft and developed a draft model (Eq. 1) to predict the percentage of dissatisfied population (DR).

$$DR = (34 - t_a) \cdot (V - 0.05)^{0.62} \cdot (0.37VT_u + 3.14) \qquad \text{(Eq. 1)}$$

The 1989 study was conducted for sedentary people at an air temperature of 23ºC. The draft model was developed by interpolating results from this study and an earlier test (Fanger P.O. and Christensen 1986) done at 20ºC, 23ºC, and 26ºC under moderate to high turbulent intensity.

The air flow in the tests came from behind the subjects, directed toward the back of the neck because this direction was judged to be the most sensitive direction. In the first hour of the tests before the air flow was applied, the subjects modified their clothing to keep themselves neutral. In the following 90 minutes, their clothing was kept constant and the subjects experienced six different levels of velocity from 0.05 m/s to 0.4 m/s, increasing in15 minute time steps. As the velocity increased, the subject's thermal sensation decreased and was lower than neutral.

Because the test conditions were neutral to cool, the draft model should apply only to sedentary, thermally neutral or cool people. It should also apply only to air flows directed toward the back of the neck. Neither of these restrictions are specified in the standards which have adopted the draft model.

The draft model is presented in Figure 6, as it appears in ASHRAE 55 and ISO standards. ASHRAE standard 55 – 2004 also allows elevated air speed to be used to increase the maximum temperature. Figure 7 (from Fountain and Arens 1993) shows the air velocities that can be used to offset the comfort effects of a rise in air temperature. The velocities are much greater than in the draft model. However, this figure can be used only if affected occupants have control of the air speed, usually through an operable window or fan. This leaves Figure 6 covering all other conditions, regardless of the fact that the original tests were conducted under neutral to slightly cool environments and the air motion towards the back of the neck. Field measurements carried out to characterize turbulence occurring in ventilated spaces in a wide variety of buildings by Hanzawa et al. (1987) show that the turbulence intensity is in a range of 10% to 70%. If applying 50% turbulence intensity, the acceptable air velocity is about 0.2 m/s.

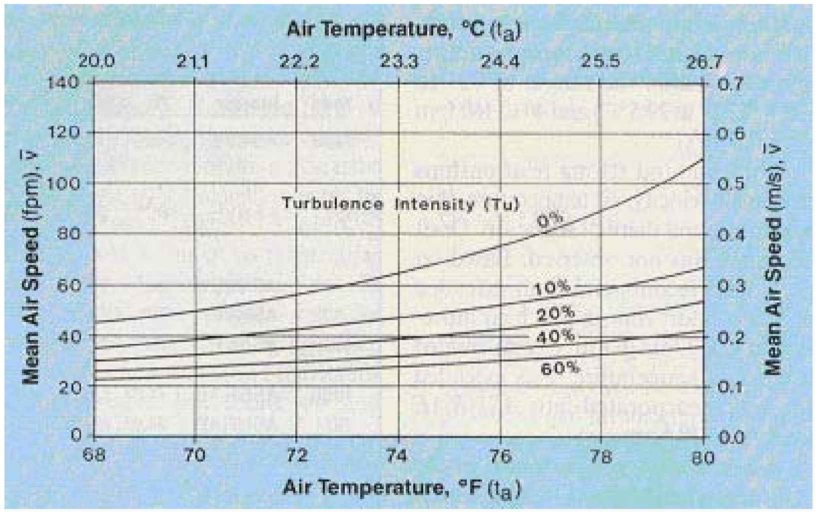


Figure 6: Allowable mean air speed as a function of air temperature and turbulence intensity (Fountain and Arens 1993)

The limit of the air motion and the risk of the draft should be considered under different environments. Under neutral or slight cool environments, the risk of draft sensation is high (Houghten et al. 1938, Fanger at al. 1974, Fanger and Pedersen 1977, Fanger and Christensen 1986, Fanger el al. 1988). Toftum (1994a, 1996, 2003) found that as the overall sensation moves toward cool from neutral, the risk of draft increased two to three times. Toftum (2002) cites the work of Berglund & Fobelets (1987) and states that subjective responses to air velocity and radiant asymmetry are independent. He draws attention to the finding that, in cold weather, "the percent of subjects experiencing a draught approximately doubled in the cool environment as compared to the neutral environment".

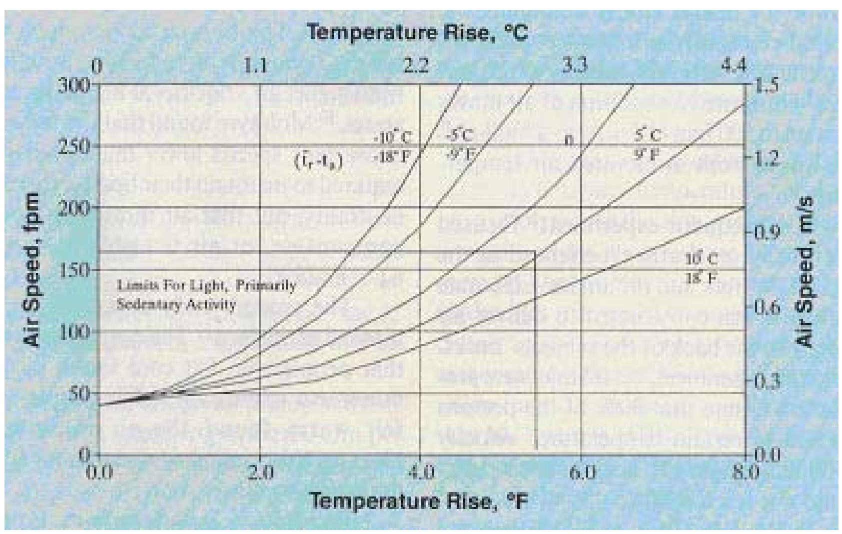


Figure 7: Air speed required to offset increased temperature (Fountain and Arens 1993)

In warmer environments, many studies show that higher velocities (up to 0.8 m/s by Roles et al. 1974, 1983, 1.0 m/s by Scheatzle et al. 1989, Busch 1990, and Tanabe and Kimura 1987, 1.2 m/s by Kontz et al. 1983, 1.6 m/s by Tanabe et al.) were preferable or perceived as pleasant and no unpleasant draft was perceived. A literature review examining air motion, comfort, and standard was provided by Fountain and Arens (1993). The paper provides a figure to consolidate several studies to show the different ranges for preferred velocities and the limits at different environments (Figure 8).

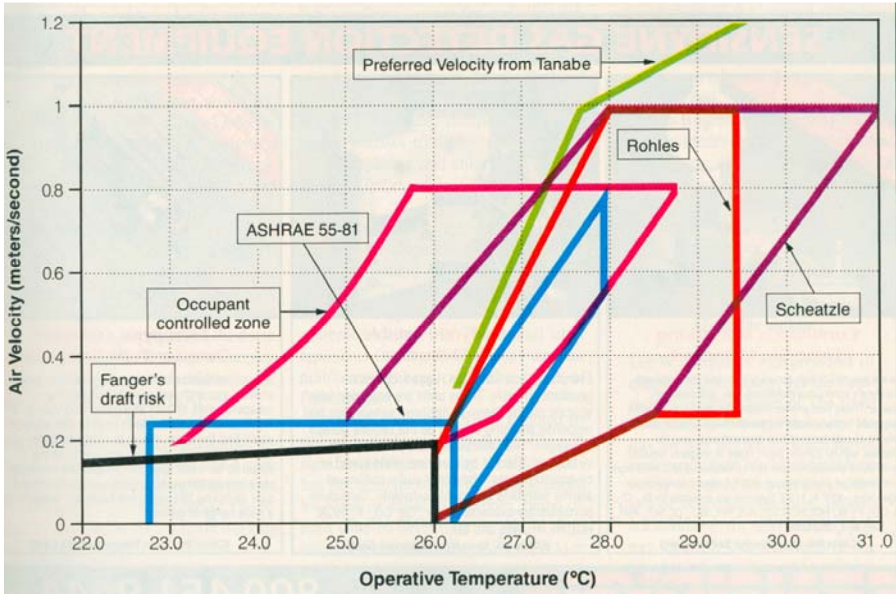


Figure 8: Range of velocity requirements (Fountain and Arens 1993)

The studies of many researchers show that in a warm environment, higher air movement provides comfort. Arens et al. (1998) did a laboratory study where 119 people participated. It found that subjects considered air motion pleasant up to 1 m/s at 29.5ºC, desiring no change to temperature or air movement. It was possible to make people comfortable by air motion up to 1.4 m/s at air temperature up to 31ºC (1 met) or 29ºC (1.2 met). Crossing the entire test conditions (24.5 – 31.5ºC), in most cases, the air velocity was above 0.4 m/s and up to 1.4 m/s, but very few people wanted less air movement. Fountain et al. (1994) found that the air motion can make 91% people comfortable up to an air temperature of 28.5ºC. However, based on the draft model, 63.2% of these comfortable votes would have been predicted to be "unacceptable". At the upper limit of the draft model (0.2 m/s at a turbulence intensity of 40%), 50% of the people wanted more air movement. The authors point out that the draft model is not designed to make the greatest number of people satisfied with the air movement in warm environment, just to protect 20% from being dissatisfied in slightly cool environment. So instead of examining discomfort, the authors defined an index of predicted Percent Satisfied (PS) which can be used to predict the percentage of satisfaction in a warm environment by applying higher air motion. The model is developed based on an experiment that encouraged people to use air motion to make them comfortable. McIntyre did two studies (1978) to examine the acceptable velocity at warm room temperatures. He found that the subjects chose an air motion lower than the value necessary for heat balance, but that the air motion could compensate for air temperature up to 28.5ºC. In another test examining the draft directed at the face (McIntyre 1989), the author found that when the whole body was warm, people considered the air motion pleasant. When the whole body was cool the air motion was considered unpleasant. Toftum et al. (1997) also examined effect of air direction on draft. They found that at an air temperature of 20ºC and 23ºC, the air motion from below was perceived as most uncomfortable. At air temperature 26ºC, air movement from above was perceived as uncomfortable. The authors also recommended taking air direction into account when providing design guidelines for air movement.

Recent studies indicate that the draft model has to be modified to take into account additional parameters such as the activity level, length of exposure and velocity directions (Jorn and Nielsen

1996a & b, Jorn et al. 2000, Griefhana 1999). People with higher activity levels were found to be not so sensitive to draft (Jones et al. 1986, Toftum and Nielsen 1996, Griefahn et al. 2001)). Toftum (1994) proposed an extension of the draft model which specifies much higher velocity corresponding to the high activity level.

Toftum (2004) examined the literature and provides a good summary for the desirable air motion vs. draft sensation under different environmental conditions. "At temperature up to 22 – 23ºC, at sedentary activity and with occupants feeling neutral or cooler, there is a risk of air motion being perceived as unacceptable even at low velocities. In particular, a cool overall thermal sensation negatively influences the subjective perception of air movement. With occupants feeling warmer than neutral, at temperature above 23ºC or at raised activity levels, humans generally do not feel draught at air velocities typical for indoor environments (up to around 0.4 m/s). In the higher temperature range, very high air velocities up to around 1.6 m/s have been found to be acceptable at air temperatures around 30ºC" (Toftum 2004).

3.4.2 DRAFT CAUSED BY COLD WINDOWS

Cool drafts are a common complaint near windows. Although a warm window can also induce air motion, because the air movement is upward and not near the occupied zone, and also because the warm air temperature has little heat removal potential, it has little effect. Most studies focus on the downward air motion induced by a cool window and its impact on comfort (draft). A cold window causes draft discomfort through increased velocity, lowered air temperature, and increased turbulence intensity.

Ge and Fazio (2004) measured velocity and temperature profiles with large tall (3.8m x 6.7m) glass panels. Near the window, the cold window-induced air motion could be as large as 1 m/s (0.6 m above floor). When measured at 1.2 m away from the window, the velocity reduced to 0.15 m/s. The air temperature at 1.2 m away was about 0.8ºC lower than the room air temperature. Near the floor (0.1 m above) and close to the window, the maximum air velocity reached 0.4 m/s. Rueegg et al. (2001) measured velocity profiles due to cold windows. When the outdoor temperature was below –10ºC, a well insulated window (U-value of 1.4 W/m^2 K) created an air movement greater than 0.2 m/s within 0.2m of the window at height 0.13m above the window frame. When the outdoor air temperature was higher but near –10ºC, at 1m away from the window, the maximum velocity was smaller than 0.15 m/s at the height 0.1 m above the floor. People have footwear so normally 0.1 m above floor is considered an appropriate measurement height (Manz and Frank 2004). Manz and Frank also recommend 1m away from the window as the occupied zone. ASHRAE Standard 55 defines 0.6 m away from the window as the occupied zone. Compared with long-wave and short-wave radiations, Lyons et al. (1999) demonstrated that for most residential-size windows, draft effects are generally small. Through CFD analysis of a room 3 x 3 x 5 m with one cold wall, Manz and Frank (2004) found that draft is more critical and caused more discomfort than reduced operative temperature or strong radiation asymmetry. A PPD of 20% due to draft occurs 1 m away from a 15ºC wall and 2m away from a 10ºC wall.

Heiselberg (1994, 1995) developed a set of empirical equations to calculate the maximum velocity and temperature changes of the cold draft along the floor after it flows off the surface and penetrates into the occupied zone. Rueegg et al. (2001) measured the air velocity profile along the floor and compared with the predicted value from Heiselberg's equation and found that the equation provides satisfactory results, even for windows with a sill. Ge and Fazio (2004) compared their velocity and temperature measurements for large tall glass panels with the predictions from the empirical equations developed by Heiselberg. The authors also proved that Heiselherg's equations provide close estimations of the measured data for cases having a frame projection, providing the height of the glazing is used in the calculation. Manz and Frank (2004) used CFD simulations to show that an increase of the internal heat load leads to higher downdraft air speed, and they modified Heiselberg's equation to account for the internal heat load. They explained that the momentum of the buoyancy flow at the heat source boosts the motion of the downward flow near the cold wall. However, Rueegg et al. (2001) measured the velocity profile of a draft near a cold window and found that as the internal heat load increased, the boundary layer thickness increased but the peak velocity decreased. The author's explanation was that the plumes from the heat load spread at the ceiling and circulated down, and mixed together with the draft layer so the temperature of the layer was raised and the draft was reduced.

As the cool air flows downward next to a cold window, the thickness of the air layer to which the vertical motion is confined increases from the top to the bottom. At a certain distance the airflow will become turbulent. By simulation, Manz and Frank (2004) found that in a room (with an entire wall being cold), the turbulent intensity could reach 50%, although a lower turbulent intensity was found in a similar setting (Olesen 2002).

To counteract downdrafts, a common solution is to install a radiator underneath the window, which costs energy (Lyons 1999), but does improve comfort. Gan's simulation (2001) showed that keeping the 10ºC radiant asymmetry, by applying a radiator under the window sill, the discomfort area originally extending to 1m from the window was reduced to 0.75 m from the window. However, care should be taken at ankle level (being hot from the radiator) and cool at the upper body level. With the window performance improved, the effect of the draft is reduced. A few studies have examined the possibility of removing the heating source by increasing the window performance, in some cases finding that it is possible to maintain comfort without additional radiators (Rueegg et al. 2001, Larson and Moshfegh 2002). Measurements by Rueegg et al. also showed that it was critical to insulate the frame of advanced windows. They also tested the effect of increasing the roughness on top of the windowsill, which showed little influence reducing the draft. However, openings on the windowsill (as shown in Figure 9) can significantly reduce the draft because they take up the downdraft and release it again at a lower speed. Heiseberg et al. (1995) found however that with turbulent flow obstacles larger than the boundary-layer thickness can break down the boundary layer to reduce the downward cold draft from large glazed surface. The risk of thermal discomfort due to downdraft was reduced considerably.

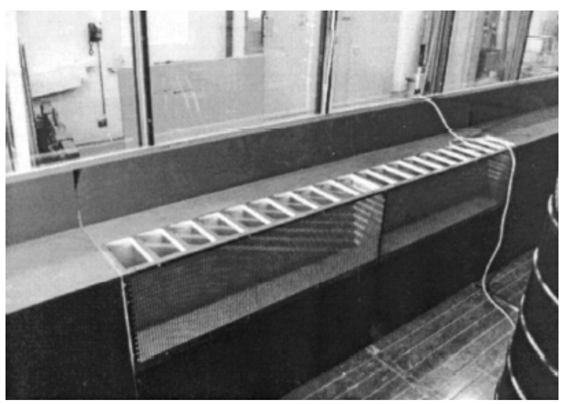


Figure 9: Openings in window sill to reduce draft down flow (Ruegg et al. 2004)

Although the draft model is used in the ASHRAE and ISO standards for any direction or location of air motion, it is based only on tests in which airflow affected the back of subjects' necks (Fanger and Christensen 1986, Fanger et al. 1988). Although the back of the neck may be the most sensitive part of a building occupant, airflows in this direction and location do not happen each time there is measurable air movement. For example, the probability of air coming through an open window and hitting the back of the neck is relatively small given the way people orient themselves. The probability is particularly small in the case of the downward airflow induced by a cold window, where the highest air movement is within inches of the glass or at ankle level along the floor. The sensitivity to draft at ankle is much less than at back of neck (Fanger and Pedersen 1977). Therefore, the draft model should be used carefully in cases with cold windows, although at this time there is no other evaluation method available (except the UCB Comfort Model which we will discuss later). The evaluations by other researchers (Manz and Frank 2004, Rueegg et al. 2001, Ge and Fazio 2004) about the discomfort induced by drafts from cold windows may also need to be reexamined, since they applied the draft model.

3.4.3 OPERABLE WINDOWS

In a general and recurring theme, a number of papers discuss the value of operable windows for increasing summer comfort and reducing air-conditioning energy consumption. Nicol & Humphreys (2002) discuss “adaptive opportunity”, meaning “the ability to open a window, draw a blind, use a fan and so on”, and the beneficial effect this has on an occupant’s perception of comfort. They also remind us that solar control (to reduce PMV) is orientation-specific and therefore so is the specification of the appropriate glazing solar transmittance. McCartney & Nicol (2002), Humphreys & Nicol (2002), de Dear & Brager (2002), and Fanger & Toftum (2002) describe the mechanisms by which operable windows and natural ventilation can offset mechanical cooling.

It may be possible to acknowledge the potential contribution of operable windows in a quantified ‘window comfort rating’. Clearly, correctly-operated windows should improve comfort through much of the occupied space. It may be feasible to include a conservative “operable window/natural ventilation” term in the comfort calculation, if such a quantity is integrated over a year of operation. Any such algorithm would need to be climate-sensitive.

3.5 THERMAL COMFORT MODELS

The most common method for evaluating thermal comfort is the Predicted Mean Vote (PMV) model proposed by Fanger (1973). The PMV method predicts thermal comfort based on overall heat loss from the body.

The thermal comfort index (PMV) is calculated using Eq. 2.

$$PMV = \left[0.303^{-0.036M} + 0.028\right] \cdot L \qquad \text{(Eq. 2)}$$

L represents the thermal load on the body, defined as the difference between the internal heat production and the heat loss to the actual environment for a person hypothetically kept at a comfortable value of skin temperature and sweat rate. The thermal load L is calculated from air temperature, mean radiant temperature (MRT), air velocity, humidity, clothing level and metabolic heat production.

Fountain and Huizenga (1995) developed an ASHRAE Thermal Comfort Tool software to allow consistent calculation of thermal comfort indices including PMV, PPD, and Effective Temperature (ET*). ET* is based on a two-node physiological model of the body, with core and skin as the two nodes (Gagge et al., 1970).

There are several models to predict thermal sensation in uniform but transient environments, such as models developed by Fiala (1998) and Wang (1994). These models predict sensation under transient condition by incorporating a dynamic factor in the model. Ring and de Dear (1991) developed a sophisticated skin model which divides the skin and clothing into 40 layers to calculate thermal sensation response to fast changes in environmental conditions. Tanabe et al (2002) used their 65-node thermoregulation model to investigate a room having a window at one end and/or a cooled ceiling panel. This model incorporates a CFD model developed by

Murakami et al. (1998) and a radiation model (which considers the differing sensitivity of the skin to various wavelengths, Marita et al. (2001)). PMV is used as the thermal comfort evaluation index.

Using these uniform condition models, when windows are warmer or cooler than the air temperature, mean radiant temperature (MRT) has to be determined. MRT is defined as the uniform temperature of an imaginary uniform enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual non-uniform environment. Measured air temperature, globe temperature, and air velocity can be combined to calculate MRT (ASHRAE Handbook 1997). When surface temperatures can be measured, MRT can be calculated from these temperatures along with view factors of the person to each surface. In buildings with windows, the glass and frame temperatures can be calculated by software WINDOW and THERM based on outdoor air temperature and solar radiation. Both WINDOW and THERM are developed by the Windows and Daylighting Group at Lawrence Berkeley National Laboratory (LBNL). There exist other models (e.g. Gan 1994) to deal with the non-uniform radiation field. Gan's model divides non-isothermal walls into isothermal segments and then calculates the MRT based on the surface temperatures of the segments and their view factors. Another way to deal with the long-wave radiation asymmetry is to directly calculate the radiative heat transfer between the person and his/her surroundings based on the view factors, as is done in the UCB Comfort Model (Huizenga et al. 2000). In the UCB Comfort Model, the human body and the environments were divided into thousands of polygons to derive detailed view factors to calculate the radiant heat exchange.

Kansas State University (KSU) produced a program (Building Comfort Analysis Program, BCAP) sponsored by ASHRAE to determine radiant heat exchange of the human body with his/her surrounding (Jones and Chapman 1994, Chapman and De Greef 1997). The current BCAP model links the properties obtained from the WINDOW program and includes the impact from the window frames. The PMV, PPD and operative temperature calculations are conducted for each node point of the room so the results are presented as the contour of PMV, PPD, or operative temperature (Chapman et al. 2004, Sengupta et al. 2005a & b, Chapman and Sengupta 2003). The authors proposed "Penetration depth" and "Fenestration performance map" to evaluate the window performance. Penetration depth is defined as the distance into the room from a window, beyond that thermal comfort condition exists. The fenestration map can be used as a design tool to determine the correct window wall ratio to ensure comfort at a defined distance from the window.

When solar radiation is present, the PMV calculation has to consider this load. One way is to put the solar load as an additional load into the thermal load L calculation, which is used to calculate the PMV as shown in Eq. 2. The alternative way is to calculate an MRT equivalent to surrounding temperatures plus the solar radiation, such as a method provided by Arens et al. (1986). Sullivan provided a linearized expression to calculate the sensitivity of PMV to the solar flux (Sullivan 1986a and 1986b). By applying both Arens and Sullivan's methods, Lyons et al. (1999) demonstrated that every unit of incident solar radiation (W/m^2) (derived from the product of solar irradiance and direct solar transmittance of a window) corresponds to +0.0024 increment in PMV (thermal sensation vote).

The PMV evaluation method treats the entire body as one object. It does not distinguish between different parts of the body. If one side is warm and the other cold, the PMV calculation would calculate a zero thermal load and therefore yield a neutral thermal sensation (PMV=0).

The PMV model is basis for most current standards prescribing methods for evaluating thermal comfort in buildings. The model was developed based on data from uniform thermal environments (Fanger 1970). Because it only calculates the heat transfer for the entire body, it cannot predict local discomfort. Clothing is assumed to cover the entire body uniformly which results in one equal skin temperature across the entire body. Obviously, the local effects of asymmetric conditions, such as an environment with a hot or a cold window, or local air motion around a person's face provided by a fan, are lost in this whole-body model. Calculating MRT in an asymmetric radiation environment averages the influence to the whole body and makes the influence on a specific body part, which has a bigger view factor, less sensitive.

The non-uniformity of the environment with the presence of warm or cold windows not only provides an asymmetric radiant field, but also causes a variation in the air velocity and the temperature over a person's height. Several examples from measurements of a large tall glass panel are shown by Ge and Fazio (2004). Obviously it is difficult to directly apply the PMV method in complex situations with a warm or cold window. In automobile industry, researchers adopted a few ways to deal with the asymmetrical environment. Ingersoll et al. (1992) applied the PMV model individually to head, torso, and feet. Their whole-body PMV is an area weighted average of the PMV from the three body parts. Matsunaga et al. (1993) calculated the weighted equivalent temperature from head, abdomen, and feet and used the equivalent temperature to calculate PMV. Kori et al. (1995) applied the two-node model (which treats the whole body as one uniform unit) to eleven body parts to calculate the SET* individually. The KSU clothing model (Jones and Ogawa 1992) combines the two-node model with a detailed clothing model to deal with radiant asymmetry. Because the PMV model and the 2-node model treat the whole body as one object, all the above methods have difficulties really separating the body into more parts in order to deal with asymmetry. Hagino and Hara (1992) predict whole-body thermal sensation using sensations from a few body parts which experience environmental changes in their test conditions (forehead, upper arm on window side, thigh on window side and instep on window side). Because their experiment didn't examine other body parts, the model can only be used in situations similar to the test condition. Taniguchi et al. (1992) developed a model to predict thermal sensation based on face skin temperatures. Again because the model was based on the test in which asymmetry only appears on the head with air motion, the model can only be used in situations similar to the test condition.

Wyon et al. (1989) proposed to use Equivalent Homogeneous Temperature (EHT) to express heat loss for each body part individually. The EHT for one body part is defined as the temperature in a uniform environment where the heat loss from the body part is the same as his/her heat loss in an actual (non-uniform) environment. The acceptable comfort range for each body part is presented as a "piste"[3] shown in Figure 10(Wyon et al. 1989). Between the upper and lower temperature limits is the ideal profile.

The EHT evaluation method has been used in automobile industry (Bohm et al. 1990, 2004, Nilsson 2005) to address non-uniform environments. EHT defines an acceptable temperature range for each body part, but it does not quantify local comfort levels and the *overall* thermal comfort. In other words, it does not tell how comfortable or uncomfortable a body part feels, and it does not tell the overall comfort when all parts are slightly cool vs. some parts are slightly cool and other parts are slightly warm, all within the acceptable range defined by the piste. In addition, the piste developed from each study only applies to the environmental and the clothing and metabolic conditions tested. So, its use is limited to the specific conditions tested in the studies where it was applied.

[3] so-called because the diagram looks somewhat like a ski run (piste) on a mountainside.

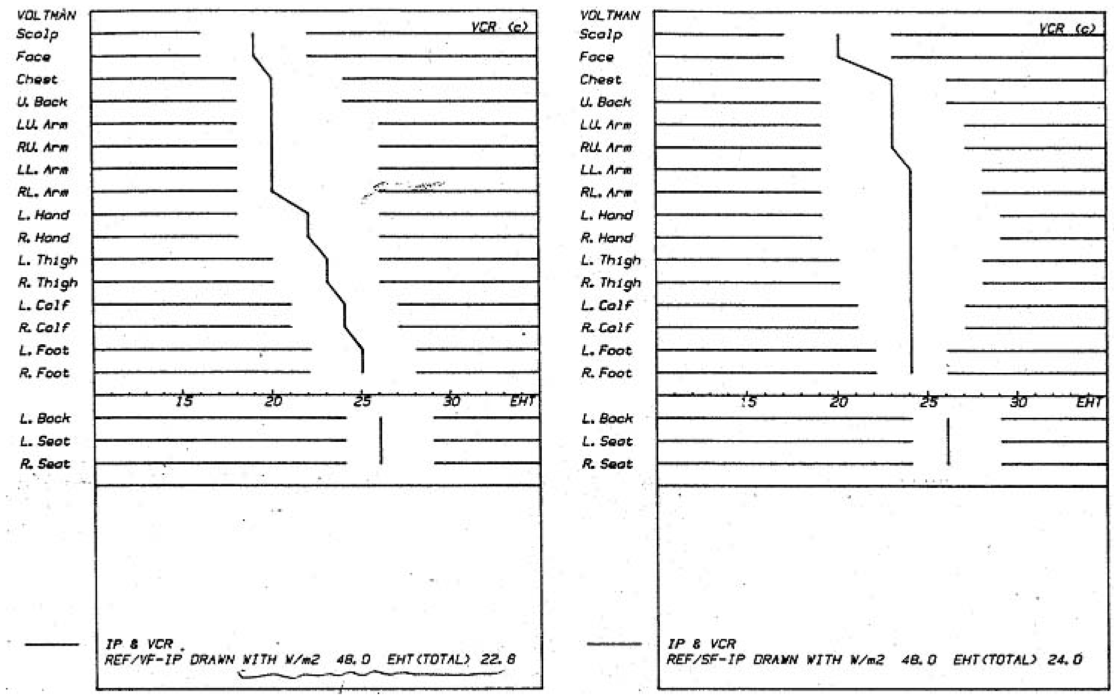


Figure 10: EHT piste and ideal profiles for a driver in winter (left) and summer (right) (Wyon 2004)

3.6 LOCAL DISCOMFORT

Over $80 billion is spent annually in the United States to heat and cool buildings, in order to provide comfortable, productive environments for their occupants. The heating, ventilating, and air conditioning (HVAC) equipment used to create and maintain these indoor environments is energy-intensive, but is not particularly successful at keeping occupants truly comfortable. Studies have shown that about 75% of all occupant complaints in buildings are thermal-comfort-related (Martin et al. 2002). These complaints are largely caused by discomfort in local body parts such as feet, hands, and neck, based on field survey in displacement ventilated office buildings (Melikov et al. 2005). Even in the studies presented early in the literature section for investigating the limits with asymmetric walls and ceilings (Fanger et al. 1980, 1985), and the other study defining the acceptable air temperature stratification (Olesen et al. 1979), the authors all pointed out that the discomfort was due to warm head or cool feet, i.e. local discomfort.

In the highly asymmetric environments such as the ones with large windows or very high or low glass temperatures, the discomfort for local body parts are likely to happen. Identifying how these phenomena of local discomfort occur and removing the driving forces that create them is the key to improve sustained comfort in office buildings. A principal weakness in current building practice is exactly the absence of analytical methods to quantify and evaluate local discomfort in the context of thermally asymmetric environments. Most of the existing analytical models (PMV, Fanger 1970, Fiala 1998, Wang 1994) were all developed for uniform conditions (models developed by Fiala and Wang predict thermal sensation under transient conditions). The UCB Comfort Model is the only model currently existing which predicts thermal sensation and comfort for local body parts.

4 UC BERKELEY COMFORT MODEL

This section introduces the UCB Comfort Model and explains its capabilities in terms of the assessment of thermal comfort in thermally asymmetric environments. A comparison between the comfort indicators provided by the PMV model and the UCB model is presented. The suitability of both models with respect to analyzing the impact of window performance on thermal comfort is discussed.

4.1 MODEL DESCRIPTION

The University of California at Berkeley developed a thermal comfort model (UC Berkeley Comfort Model) that predicts local comfort for different body parts, and integrates local comfort levels to yield whole-body thermal comfort (Zhang et al. 2004). The model is developed based on a large number of human subject tests under different asymmetric and transient thermal environments (Zhang 2003, Huizenga 2004). When simulating an indoor environment with windows, the UCB comfort model predicts the local comfort for all body parts, which is influenced by radiant heat exchange with the window, solar gain, air motion, and non-uniform air temperatures, and provides a whole body comfort level by integrating the local comfort of the various body parts.

The UCB comfort model divides the body into 16 body parts (Figure 11). Each part is divided into core, muscle, fat, and skin layers. An underlying model of the blood circulation system simulates the heat exchanges between the tissue layers, such as muscle and skin. A detailed description of the physiology modeling and the validation results can be found in Huizenga et al. 2001.

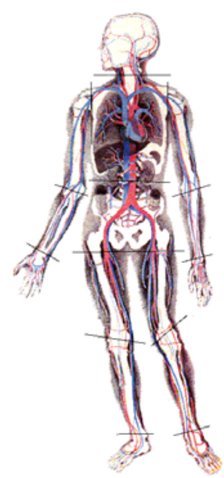


Figure 11: Subdivision into 16 body parts used in UCB comfort model

The model divides the surface of the human body into more than five thousand polygons to calculate the radiation heat transfer between the body and the environment (Figure 12). Therefore, the heat transfer between the body and thermally asymmetric surfaces, such as walls containing windows, can be calculated in great detail. The solar load on the body is also calculated based on the relative geometric position of the sun, the environment, and the polygons mapping the body surface.

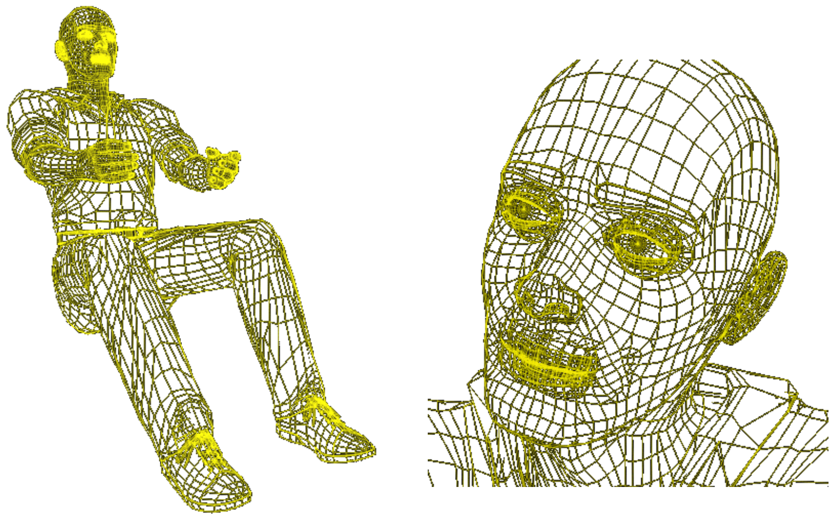


Figure 12: Polygons used to calculate radiation heat exchange in UCB comfort model

Local sensation (e.g. left hand feels cold) is calculated based on the local skin temperature of the body part and the mean skin temperature, which represents the whole-body thermal state. In transient conditions, the derivatives of local skin and core temperatures are included. Local comfort (e.g. face feels comfortable) is calculated based on local and overall sensations. For example, face cooling could be interpreted either as comfortable or uncomfortable depending on whether the whole-body thermal state is warm or cool. Separate local sensation and comfort models are provided for each of the 16 body parts. The local sensation and comfort levels are integrated to get an overall sensation (e.g. whole-body feels hot) and an overall comfort (whole-body is very comfortable). A description of the local and overall sensation and comfort models is presented in Zhang et al. 2004. Figure 13 illustrates the concepts of local sensation and comfort, whole-body sensation and comfort. Because the UCB comfort model simulates the heat transfer between each individual body part and its microclimate in order to predict sensation and comfort for individual body parts, it is well-suited for complex thermal environment where thermal asymmetry and transience exist. It is the currently the only existing model that provides local sensation and local comfort for individual body parts.

Figure 13: Prediction of local as well as overall sensation/comfort in UCB comfort model

Figure 14 shows the interface of the UCB comfort model. The example shown depicts that due to solar radiation (shown in the big image at the center), hands, lower and upper arms, and head feel warmer (right lower bottom image, warmer color presenting warm sensation). As a result, these body parts are uncomfortable (right middle image, yellow to black presenting discomfort from just uncomfortable to very uncomfortable). The overall sensation is shown at the top middle figure (red color representing hot), and the overall comfort is uncomfortable (dark figure on the top left). The overall sensation and comfort levels are also presented numerically next to the figures. The data on the top right shows the values of sensation and comfort for each body part and the whole-body. The data can also show skin, core, muscle, fat temperatures, heat loss, skin wettedness etc based on user's selections. The graph at the middle bottom shows the transient results for the parameters that the user can select.

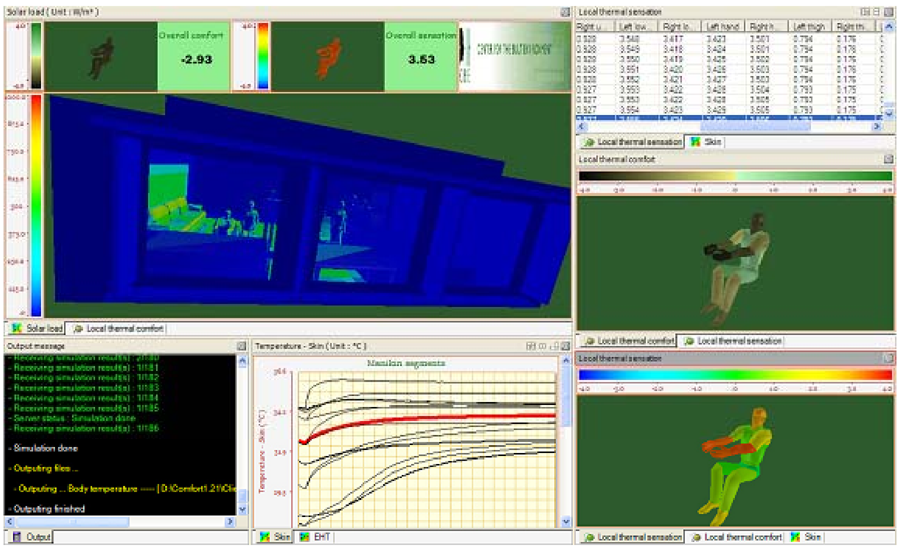


Figure 14: Screenshot of UCB comfort model interface

The model allows users to define their own room and window geometries through a "room editor". Figure 15 shows an example. Because the person can be located at any place, the model can be used to evaluate comfort at any given location of the room, such as a location in a perimeter zone or at a center zone. The surface temperatures of the environment, such as the window or a wall, are individually defined so the window can have the different temperature from the other surfaces. Each surface (such as a wall or a window) can be further divided into small surfaces to assign different component surface temperatures, such as window frame, edge of glass and center of glass temperature. The database with glazing products from WINDOW has been incorporated in the model.

Because of the functionality and the capabilities with respect to the simulation of local discomfort, the UCB comfort model has been used in this research project to predict the effect of window performance on human thermal comfort.

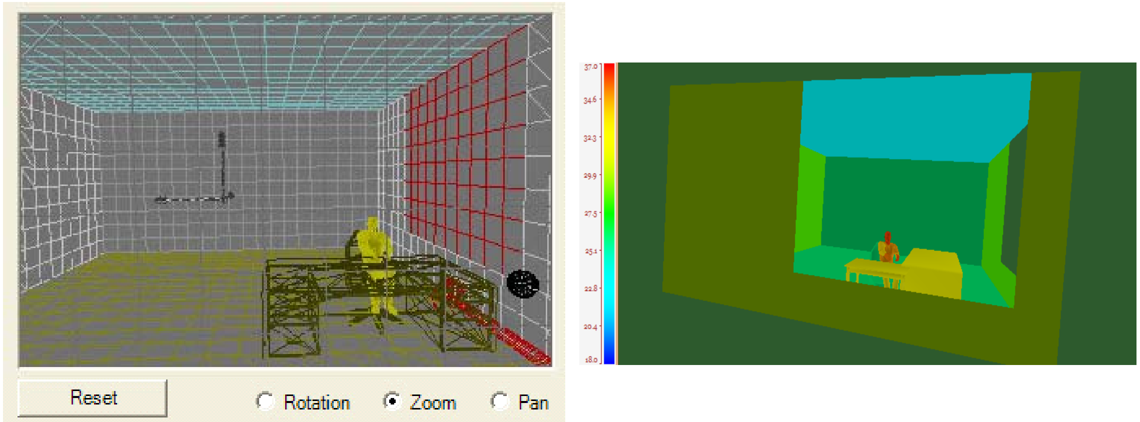


Figure 15: Screenshots of "room editor" and boundary conditions from UCB comfort model

4.2 MODEL COMPARISON

The following paragraphs provide an overview of how the results from both PMV and UCB comfort models compare to each other in particular in assessing the comfort effect of windows.

4.2.1 OVERALL THERMAL SENSATION

The simulations were carried out for a test room measuring 6m wide, 6m deep and 3m high. One elevation of the room is assumed fully glazed (6m x 3m window) and the occupant is positioned 1m away from the window along the center line of the window (Figure 16). The air temperature in the room was set at 25.7ºC and was assumed to be uniform throughout the room. This temperature represents the neutral temperature (i.e. the temperature which yields an overall sensation equal to 0) for a person (1 met, 0.6 clo) seated in the center of a room at uniform temperature (i.e. air temperature equal to internal surface temperature, no windows). We varied the interior surface temperature of the window from –10ºC to 56ºC. The analysis was carried out using both the PMV model and the UCB comfort model and the results are presented in Figure 17 and Figure 18.

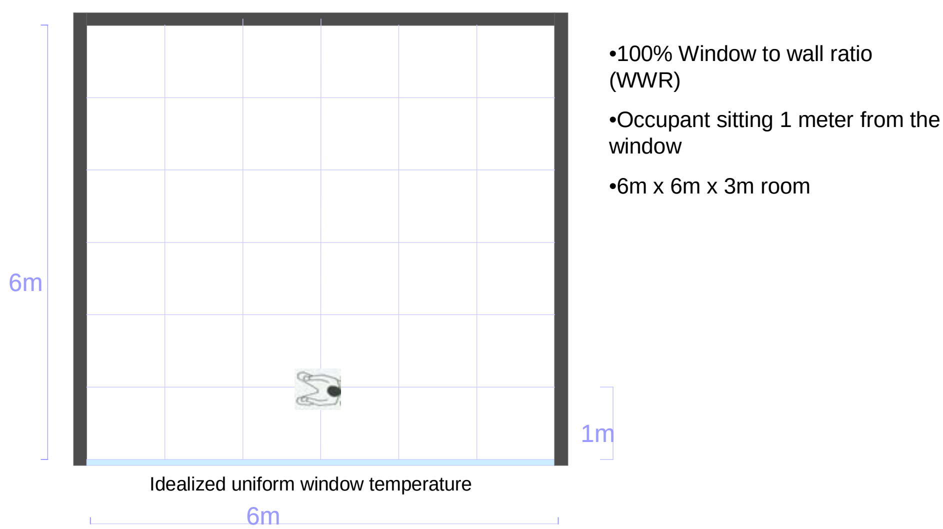


Figure 16: Geometric input data for comfort simulation

When the window surface temperature stays approximately in the center of the temperature range (e.g. between 15ºC and 35º when –0.5 < PMV <0.5), the difference between the PMV and the UCB comfort model is relatively small, less than 0.5 unit of the scale (Figure 17). However, as the window surface temperature becomes more pronounced cold or hot (below 15ºC or above 35ºC), the difference between the two models becomes larger. Because the PMV model only assesses the overall heat transfer between the body and the environment, it is less sensitive to the asymmetry and the local effects caused by the window. The PMV model averages the heat exchange over the whole body and thereby under predicts the effects of thermal asymmetries on comfort. The UCB model calculates the variation in local sensation for the different body parts caused by the window and factors their individual influence into the assessment of overall sensation. The overall heat transfer approach predicts the overall energy balance, but ignores the impact of local discomfort which clearly influences the general assessment of comfort.

The zone in the overall sensation curve which is flattened out (and where the UCB model overlaps with the PMV model) is caused by the adaptation of the thermal receptors in the skin to the environmental temperature. There is a temperature band, centered around the neutral skin temperature (skin temperature under neutral thermal condition, i.e. 25.7ºC), in which the skin temperature can be altered without significantly impacting the thermal sensation. Within this temperature range, the skin is fairly insensitive to thermal stimuli and thermal adaptation occurs. Thermal adaptation occurs more pronounced on the cold side of the neutral temperature than on the warm side (McIntyre 1980, Stevens 1960). This explains why the thermal sensation curve is flattened out more when the window temperature is cooler than the neutral temperature compared to warmer window temperatures.

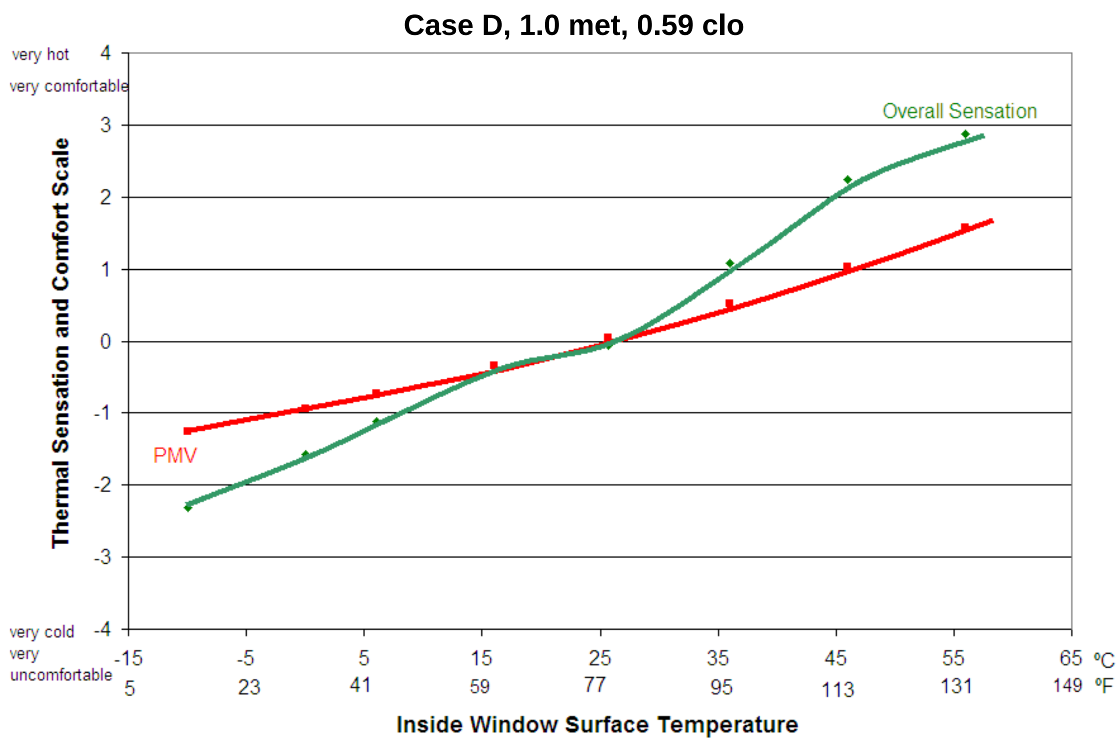


Figure 17: Overall sensation – PMV vs. UCB model

4.2.2 COMFORT LIMITS

The comfort limits for the PMV model are typically defined as –0.5 < PMV < 0.5, which corresponds to a Predicted Percent Dissatisfied (PPD) of 10%. The thermal comfort standards developed by ASHRAE and ISO provide additional limits for the difference between room surface temperatures and the air temperature. For a cold vertical surface, the radiant temperature asymmetry needs to be below 10ºC. For a warm vertical surface, the asymmetry limit is somewhat larger, i.e. 23ºC.

Figure 18 shows the comfort metrics of both the PMV and the UCB model for the boundary conditions described in Section 4.2.1. The PPD curve relates to the PMV curve, in the sense that the PPD attains a minimum value when PMV is equal to 0 (i.e. neutral sensation) and the PPD starts to rise as the PMV moves away from the neutral point. The red shaded area bound by PPD ≤ 10% at the bottom of the graph indicates the limits on the acceptable interior surface temperature of the window. The PMV reaches a value of ±0.5 at 12.5ºC and 37ºC respectively. The comfort limits determined by the UCB comfort model are indicated by the green shaded area at the top of Figure 18.

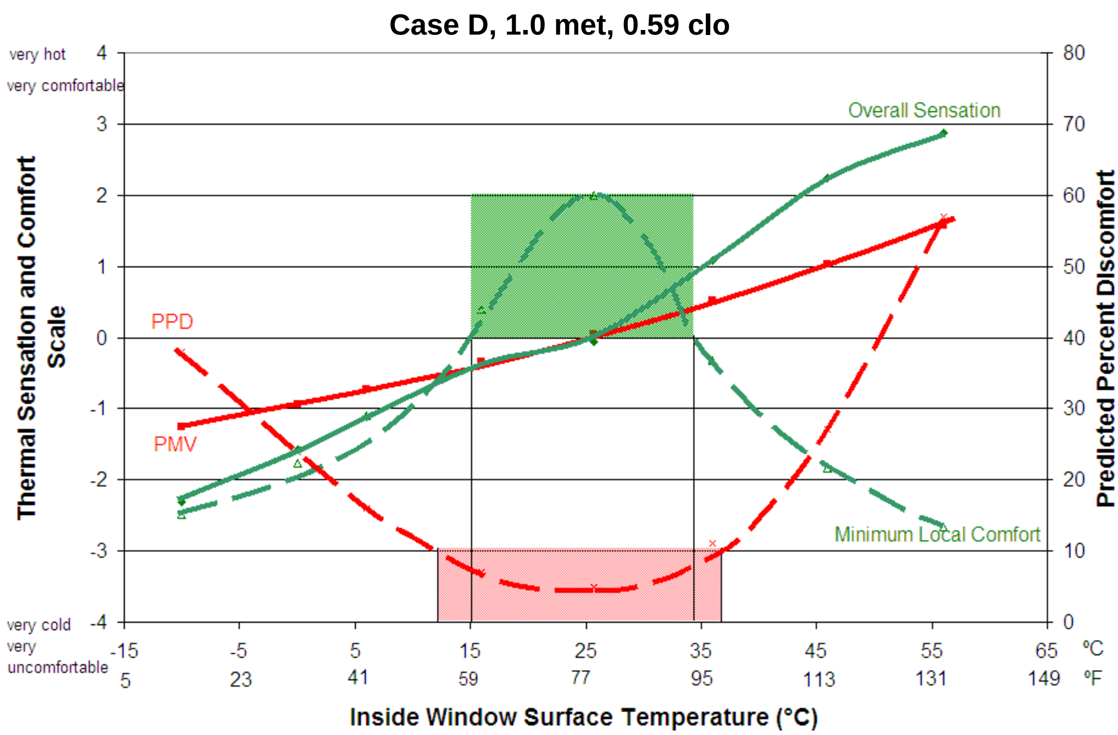


Figure 18: Comfort limits – PMV vs. UCB model

Because studies show that thermal discomfort is mainly caused by local discomfort (Melikov 2005, Wyon 1996), we have defined comfort according to the criteria that 'no *local* discomfort' may occur. The overall comfort value is an integration of the local comfort sensations of all body parts. When a particular body part experiences discomfort, a person might still be overall comfortable. Therefore, an overall comfort does not guarantee no local discomfort happening. However, a situation where no local discomfort occurs does guarantee overall comfort. Therefore, the criterion requiring 'no *local* discomfort' sets stricter limits for acceptable comfort conditions. The UCB comfort limits are defined by the intersections of the curve of the minimum local comfort (of all body parts) and the neutral line (equal to 0). Thermal comfort is guaranteed when the minimum local comfort (of all body parts) stays above zero, i.e. for window surfaces between

15ºC and 34ºC. These comfort limits are about 3ºC higher/lower than the limits provided by the PMV model, i.e. the comfort band decreases by approximately 6ºC.

4.2.3 CONCLUSIONS

Avoiding local discomfort is key to ensure thermal comfort, especially in thermally asymmetric environments such as rooms with cold or warm window surfaces. The PMV model is not only insensitive to the effects of thermal asymmetry, but does also not provide any information regarding possible discomfort of local body parts. The UCB comfort model is currently the only existing model that provides information on overall comfort as well as on local comfort of the individual body parts. Therefore, the model appears appropriate for evaluating thermal comfort in complex thermal environments.

5 ASSESSMENT OF WINDOW COMFORT

This section provides an overview of the primary factors influencing window comfort and the different approaches that can be used to evaluate the impact of different window systems on the thermal comfort in the indoor environment. The UCB comfort model has been used in the various analysis examples, using the criterion 'no *local* discomfort' as the comfort evaluation.

5.1 PRIMARY FACTORS INFLUENCING WINDOW COMFORT

Figure 19 provides a list of the primary factors influencing window comfort, organized according to those factors which impact the view factors between the person and the surroundings, the factors that determine the interior surface temperature of the window and the human factors:

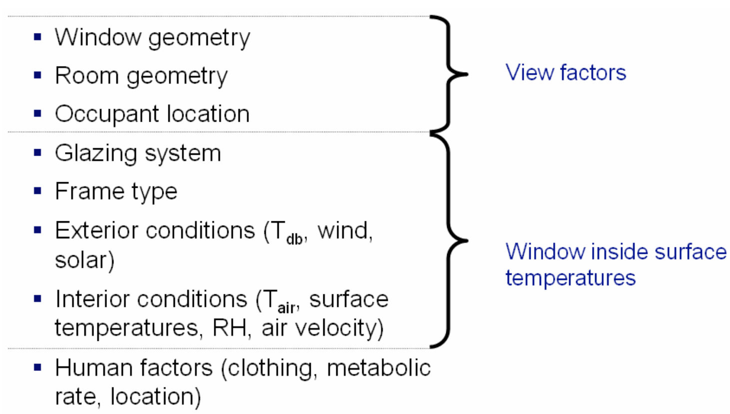
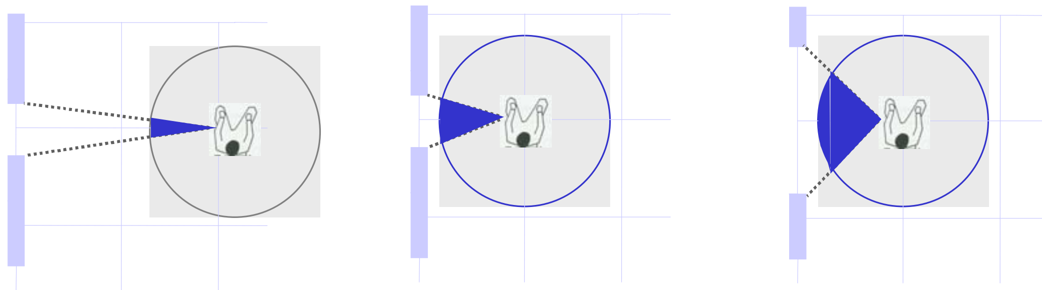


Figure 19: Factors influencing window comfort

To evaluate window thermal comfort, one must evaluate how the window affects the heat balance between the person and the environment. The differential heat exchange across a person's body will result in different skin temperatures for individual body parts and a resulting core temperature, a combination of which invokes different thermal sensations and comfort. As indicated above, there are three areas that affect window thermal comfort:

View factor: The view factor describes how much a person "sees" a surface. It indicates how much the person is influenced by the temperature of a particular surface when calculating the radiation heat exchange between the person and the surrounding surfaces. The bigger the view factor is, the larger the influence on the person. It is determined by the geometry of a surface and its orientation/position relative to the person. Figure 20 shows the qualitative effect of distance and window size on view factor. A more detailed discussion of view factor follows insect Section 5.4.



View factor is used to quantify the amount of radiation energy leaving the body that reaches the window.

View factor is increased by moving closer to the window.

View factor is increased with a larger window.

Figure 20: Schematic diagram illustrating how geometry influences view factor

Window inside surface temperature: The window temperature is affected by the glazing system, frame type, exterior environmental conditions, such as dry-bulb temperature, wind, solar radiation, and interior conditions which include air and surface temperatures, relative humidity, and air velocity. Most window products have an inside surface emissivity of approximately 0.84. Products that have a lower emissivity could be used to reduce heat transfer between the window surface and the occupant. Such products are not common due to difficulties in keeping the surface clean and dry. If the surface becomes dirty, the effect of the low-e coating is reduced. Although the method we describe in this report is fully capable of addressing low-e interior surfaces, we have not included any data on the performance of such products.

Draft caused by cold windows: The sensation of draft due to air motion can be caused by two primary factors: induced air motion from a cold interior window surface temperature and infiltration. Lyon et al. (1999) showed that in general the impact from surface temperature induced draft on comfort is small except in cases with a very tall, low-performance window. A common way to reduce the draft impact on comfort is to install a heater under the window sill, which can easily remove the discomfort caused by the draft.

The impact of infiltration of comfort can be much more significant, however it is extremely difficult to characterize. The effect of infiltration is dependent on whether a space is positively or negatively pressurized with respect to outdoors and also on the details of air flow patterns in the space. These are very difficult to predict and depend heavily on the HVAC system configuration and operation. In most modern window installations, air leakage is generally assumed to be quite low. Based on these reasons, we do not include the effects of air motion in our assessment of window comfort.

Human factors, such as clothing, metabolic heat production: Factors inherent to the human body such as its metabolic heat production and inherent to human behavior such as one's clothing greatly influence a person's state of thermal comfort. The metabolic heat production is a measure of how much heat the body creates internally. The higher one's metabolic heat production, the warmer a person feels. The amount of heat the body produces is closely linked to the level of activity the body engages in. The insulation level provided by the clothing determines how easy it is for the body to exchange heat with the environment (i.e. the rate at which a thermal imbalance can be corrected). The lower the clothing level, the higher the heat transfer rate between the body and the environment. As such, the person feels cooler when the environment is either cool or warm. With respect to the effect of windows on comfort, higher clothing levels will generally lessen the impact. Most of the analysis on this report was done with a clothing ensemble consisting of underwear, slacks, long-sleeve shirt, shoes and socks (0.59 clo). For both clothing and metabolic rate, the most important factor is that they be appropriate for the air temperature being considered. For example, if the clothing or metabolic rate is increased without adjusting the air temperature, the person will be in a warm thermal state. If the effects of a cold window were being considered, the window might actually increase thermal comfort by counteracting the overall warm state of the body. Conversely, if the air temperature was low with respect to clothing and metabolic rate, the body will be in a cool thermal state and the effects of a cold window will be magnified. This balance between air temperature and clothing is probably more important than either the choice of clothing or metabolic rate.

5.2 INTERIOR AIR TEMPERATURE VS. WINDOW SURFACE TEMPERATURE

The impact of a variation in window surface temperature on occupant comfort can be offset by modifying the interior air temperature. As such, the cooling/heating effect of a cold/warm window can be compensated by increasing/lowering the air temperature inside the room.

Figure 21 shows how the minimum local comfort varies as a function of the interior air temperature. The four curves correspond to different window temperatures (i.e. 10, 20, 30 and 40°C), and each curve reveals an interior air temperature at which a maximum for the minimum local comfort is reached. For a window temperature of 10°C, maximum comfort is obtained at an interior air temperature of 25.5ºC, whereas for a window temperature of 40ºC, the optimal interior air temperature drops to approximately 21.5ºC. The shift in optimal air temperatures at which maximum comfort is obtained illustrates how the thermal effect of windows can be offset by a change in air temperature.

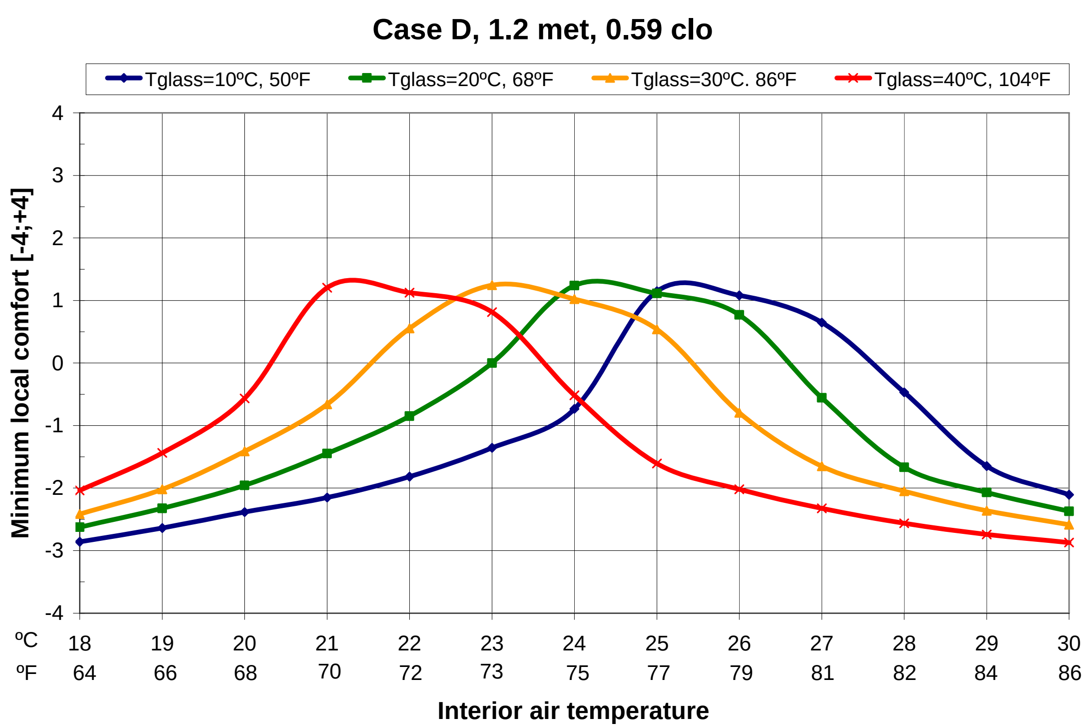


Figure 21: Interior air temperature at which maximum comfort occurs for different window surface temperatures

5.3 IMPACT OF WINDOW TEMPERATURE ON INTERIOR AIR TEMPERATURE

Another way of quantifying the performance of a window with respect to indoor comfort is to assess the change in air temperature in a space that would have the same impact on thermal comfort as the window. The window temperature impact expresses the effect of the temperature difference between the 'actual' window surface temperature and the 'neutral' window surface temperature in terms of an equivalent change in air temperature. The neutral window temperature is defined as the condition where the window temperature is equal to the air temperature that provides a neutral thermal experience for the building occupant.

The extent to which the actual window temperature diverges from the neutral temperature is converted into a change in air temperature which has an equivalent impact on thermal comfort. Hence, the impact of the window on thermal comfort is rectified or accommodated by an equivalent change in indoor air temperature. As such, an ideal window has a value of 0ºC, i.e. it has an actual surface temperature equal to the neutral temperature. The window temperature impact could be calculated for NFRC summer/winter conditions or also for a whole year. The approach could also be used to calculate the energy required to offset the temperature impact (i.e. by changing the heating/cooling setpoint).

Table 5 provides some example data calculated for the typical reference space with geometry case D, a person sitting 1m away from window at 1.2met, 0.9 clo (winter clothing) for winter rating, and 0.6 clo (typical indoor clothing) for summer rating).

Window	Winter rating [°F] [°C]	Summer rating, w/o diffuse solar [°F] [°C]	Summer rating w/ diffuse solar [°F] [°C]
Single, poor frame	-5.1 (2.8)	2.7 (1.5)	5.4 (3.0)
Double, clear, poor frame	-2.7 (1.5)	2.9 (1.6)	5.0 (2.8)
Double, clear, good frame	-2.3 (1.3)	2.2 (1.2)	3.8 (2.1)
Double, selective low-e, good frame	-1.6 (0.9)	1.1 (0.6)	2.2 (2.2)
Triple, clear, good frame	-1.6 (0.9)	2.5 (1.4)	3.8 (2.1)
Triple, selective low-e, good frame	-1.1 (0.6)	1.3 (0.7)	2.2 (1.2)

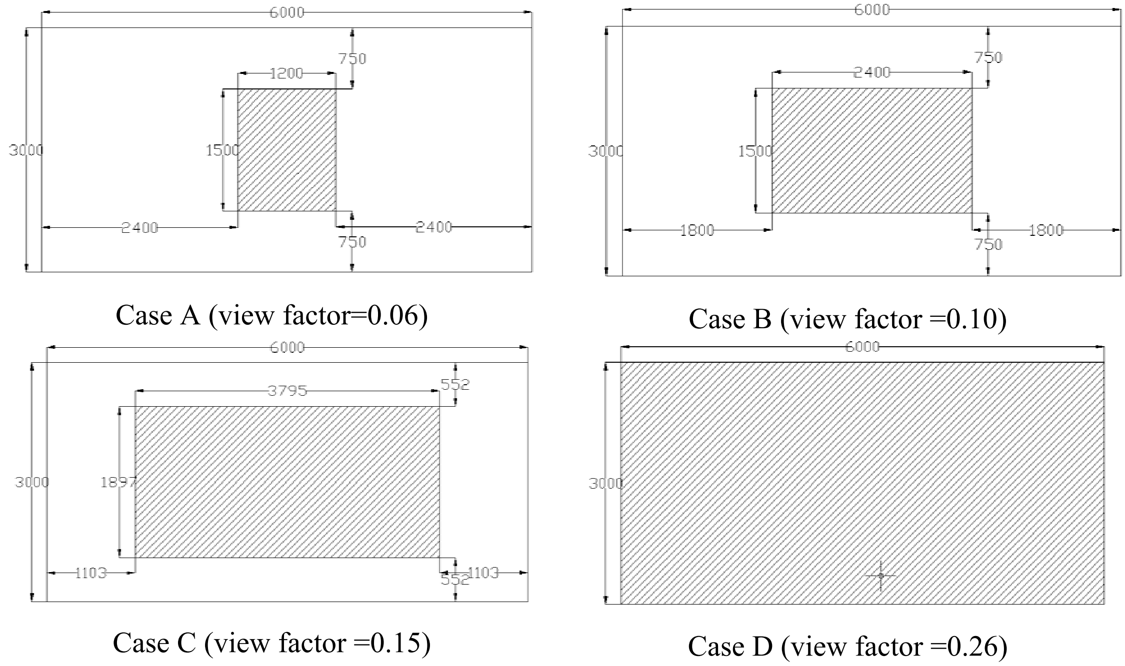
Table 5: Example window temperature impact for different window types

5.4 IMPACT OF GEOMETRY

The following paragraphs explore the impact of geometry on the assessment of thermal comfort, both in terms of the relative position of windows to the building occupant as well as the spatial variation of thermal comfort throughout a room.

5.4.1 WINDOW / ROOM GEOMETRY

The relative position of windows to a person determines the view factor and hence the magnitude of the (long-wave) radiation heat exchange between the person and the windows. We define four window geometries, shown in Figure 22, that are used in the analyses that follow. The most important attribute of each of the geometries is the view factor, shown in Table 6**.**



Case A (view factor=0.06) Case B (view factor =0.10)

Case C (view factor =0.15) Case D (view factor =0.26)

Figure 22: Window geometry for simulated cases

Geometry	Window size	Distance from window			
		1m	1.5m	2m	4m
Case A	1.2m x 1.5m	0.06	0.04	0.03	0.01
Case B	2.4m x 1.5m	0.10	0.07	0.05	0.02
Case C	3.8m x 1.9m	0.15	0.11	0.08	0.03
Case D	6.0m x 3.0m	0.26	0.20	0.16	0.07

Table 6: View factors for reference geometry cases (see **Figure 22**)

For reference, view factors for six addition geometries are provided in **Table 7**. Note that a person sitting in the corner of a façade with a 1.8m high strip window has the same view factor as a fully glazed façade.

The view factors in these tables were calculated according the method proposed by Fanger (1973) and implemented in a computer program. View factors are for a seated person, and the orientation is the average of the four orthogonal orientations with respect to the wall. Although the method can be carried out by hand, it is somewhat tedious. Figure 23 presents a graphical method for calculating view factor for most window configurations.

10 ft

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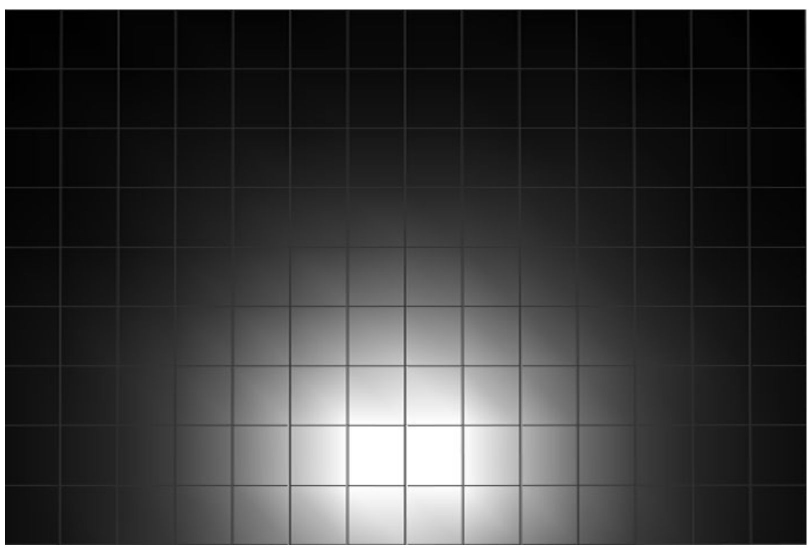
.0006 .0007 .0006 .0005	.0008 .0009 .0008 .0006	.0013 .0012 .0010 .0006	.0019 .0016 .0012 .0007	.0030 .0021 .0013 .0007	.0043 .0025 .0015 .0008	.0057 .0029 .0016 .0008	.0063 .0030 .0016 .0008	.0057 .0029 .0016 .0008	.0043 .0025 .0015 .0008	.0030 .0021 .0013 .0007	.0019 .0016 .0012 .0007	.0013 .0012 .0010 .0006	.0008 .0009 .0008 .0006	.0006 .0007 .0006 .0005

15 ft

(a)



(b)

Figure 23: View factor data for a seated person in one-foot increments for a vertical wall. (a)The arrow at the center column represents the horizontal location for the person. The four numbers in each column represent the view factors for a perpendicular distance from the window of 3', 5', 7' and 10', from top to bottom, respectively. The window(s) can be sketched on this figure and the view factors enclosed by the window added up to determine the view factor for the window. The procedure can be used for multiple wall orientations, summing the view factor for each window. (b) Graphical representation of the tabular data for a distance of 3 feet. Lighter shades represent larger view factors.

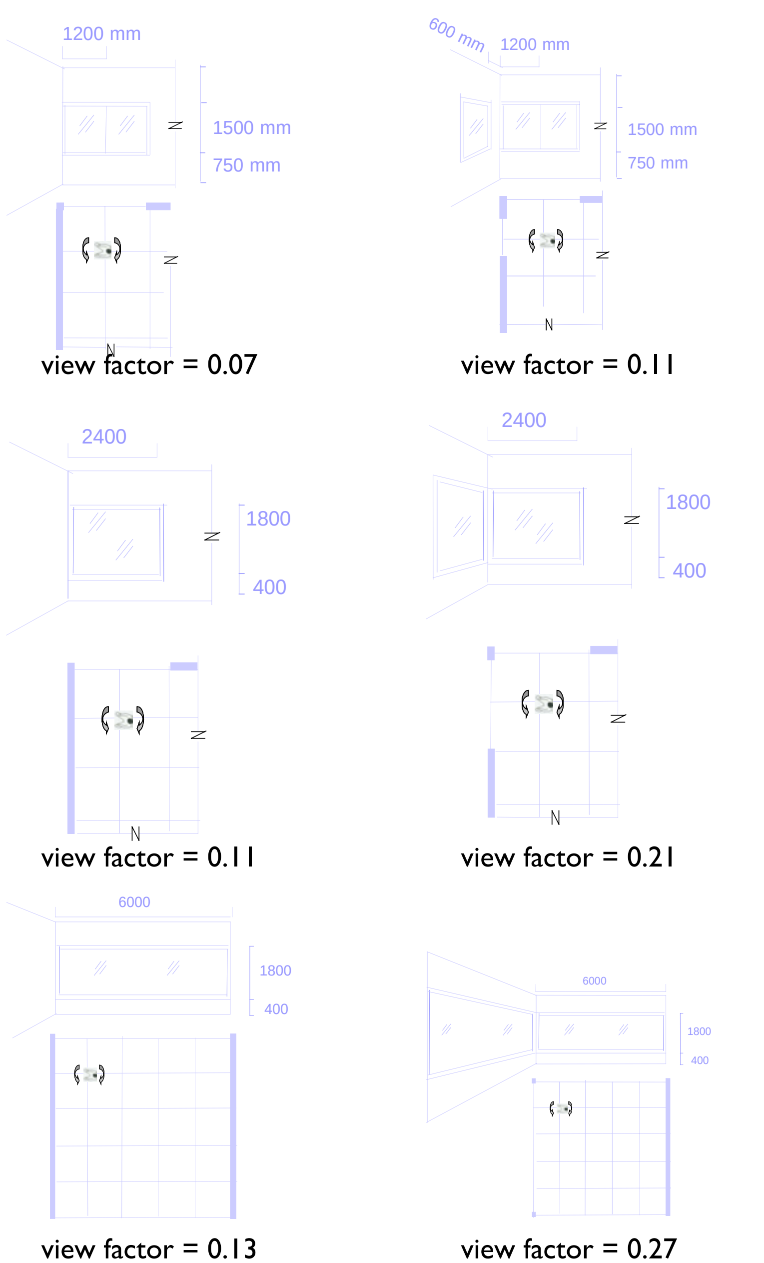


Table 7: Example geometries and view factors

In order to demonstrate the impact of window configurations with the same total window surface area but different window locations, we carried out a comfort assessment for the two geometries shown in Figure 24. Two identical windows, measuring 2m x 1.8m, are positioned either next to each other on one side, or next to each other on the two sides of a corner. The person is sitting (1.0 met) 1m away from the window, wearing typical indoor clothing (0.59 clo). The window has a uniform temperature of -9ºC, which corresponds closely to the performance of single clear glazing under NFRC winter condition (–18ºC outdoor air temperature).

The simulation results show that placing the windows on two sides greatly decreases the overall comfort of the person, because the person feels a lot colder due to the increased view factor to the cold window. The left hand discomfort significantly decreased from –0.6 (left hand sensation –1.5, between slightly cool to cool) to –2.1 (left hand sensation –2.5, between cool to cold). When the windows are located at a corner relative to the person, not only the left hand is more exposed to the cold window, but the entire body feels also cooler, therefore the hand feels cooler, and the cool hand is perceived as much more uncomfortable.

Windows on one side (view factor = 0.11)		Windows on two sides side (view factor = 0.20)	
Left hand sensation	= -1.5	Left hand sensation	= -2.5
Left hand comfort	= -0.6	Left hand comfort	= -2.1
Overall sensation	= -0.7	Overall sensation	= -1.4
Overall comfort	= -0.1	Overall comfort	= -1.3

Figure 24: Effect of window/room geometry

5.4.2 DEPTH OF ZONE OF DISCOMFORT

This approach looks at the distance from the window that is required to maintain comfort for different inside window surface temperatures, and for different window sizes. The closer to the window an occupant is, the bigger the influence from the window on the occupant. As such, we can define the required distance away from the window necessary to maintain comfort as the "depth of zone of discomfort". The geometry of the room is same as shown in Figure 16. Figure 22 illustrates the window geometries for the different window sizes.

Figure 25 indicates that when the window temperature is kept constant, the depth of zone of discomfort increases with increasing window size. When the window size is kept constant, the depth of zone of discomfort increases for more extreme window temperatures.

If we require the zone up to 1 meter away from the window to remain comfortable, stringent limits are required for the interior surface temperature of the glazing for increasing window view factors. The temperature range defined by the dashed vertical lines in Figure 25 decreases with increasing view factor. These temperature ranges are presented further in Table 8. The allowable temperature range proves to be very sensitive to the view factor.

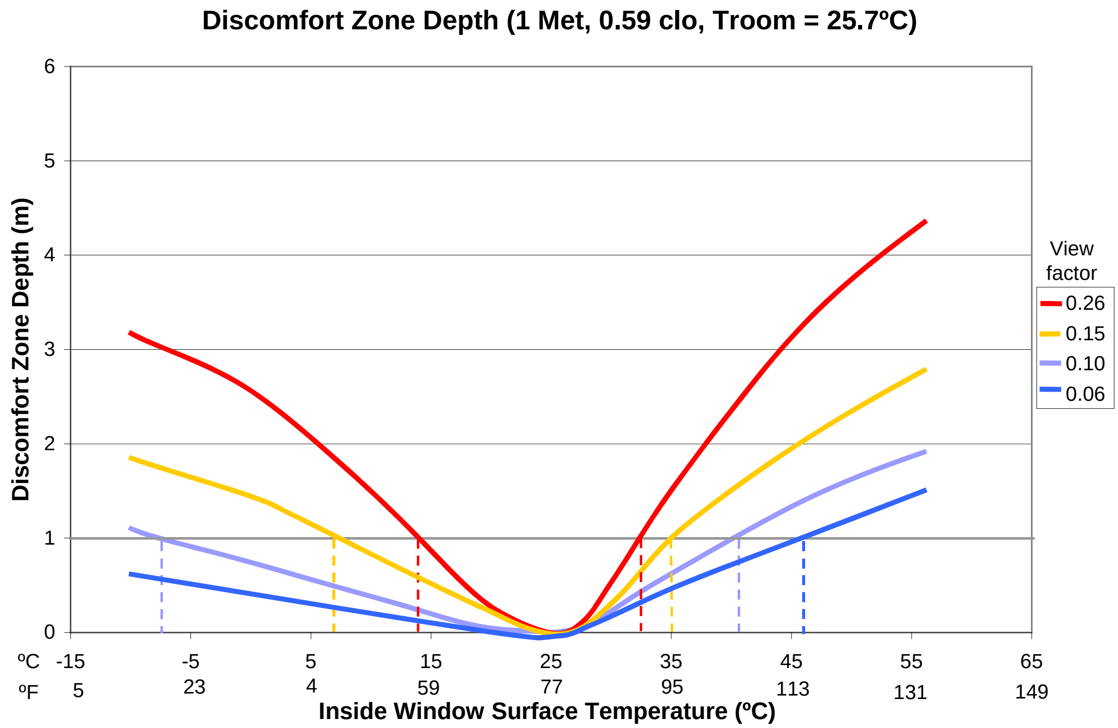


Figure 25: Depth of zone of discomfort for different view factors

For example, to maintain comfort up to 1m away from the window for a view factor of 0.10, the window temperature can be as low as –7ºC and as high as 41ºC. As the view factor increase to 0.15, the range of inside window surface temperatures changes to 7ºC to 35ºC to maintain comfort. When we want to maintain comfort in the space up to 2m away from the window, the limits imposed on the required temperature range are significantly relaxed. For example, for a view factor of 0.26, the required window temperature drops from 14 ºC to 5.5ºC on the cold side and 32.5ºC to 37.5ºC on the warm side.

View factor	Cool window – Minimum allowable Tsi [ºC]	Warm window – Maximum allowable Tsi [ºC]
Case A: 0.06	-20	46
Case B: 0.10	-7	41
Case C: 0.15	7	35
Case D: 0.26	14	32.5

Table 8: Inside window surface temperature required to be comfortable (1m from window)

5.4.3 SPATIAL DISTRIBUTION OF COMFORT

Another way to appreciate the impact of the relative position between person and window is to plot the variation in thermal comfort over the floor plan of a room (Figure 26). We positioned the person along various points on a rectangular grid on the floor plate and evaluated the person's thermal comfort at every location. As such, we obtained an understanding of the special distribution of the thermal comfort throughout the space (in 2 dimensions).

The geometry of the room is same as shown in Figure 16. One of the elevations has a window and three different window sizes are considered (cases B, C and D). The person is wearing typical indoor clothing (0.59 clo), and has an activity level of 1.2 met. The charts plot the minimum local comfort for an interior air temperature of 24°C (summer time) and an averaged window surface temperature of 40°C, representing a tinted glass exposed to solar radiation.

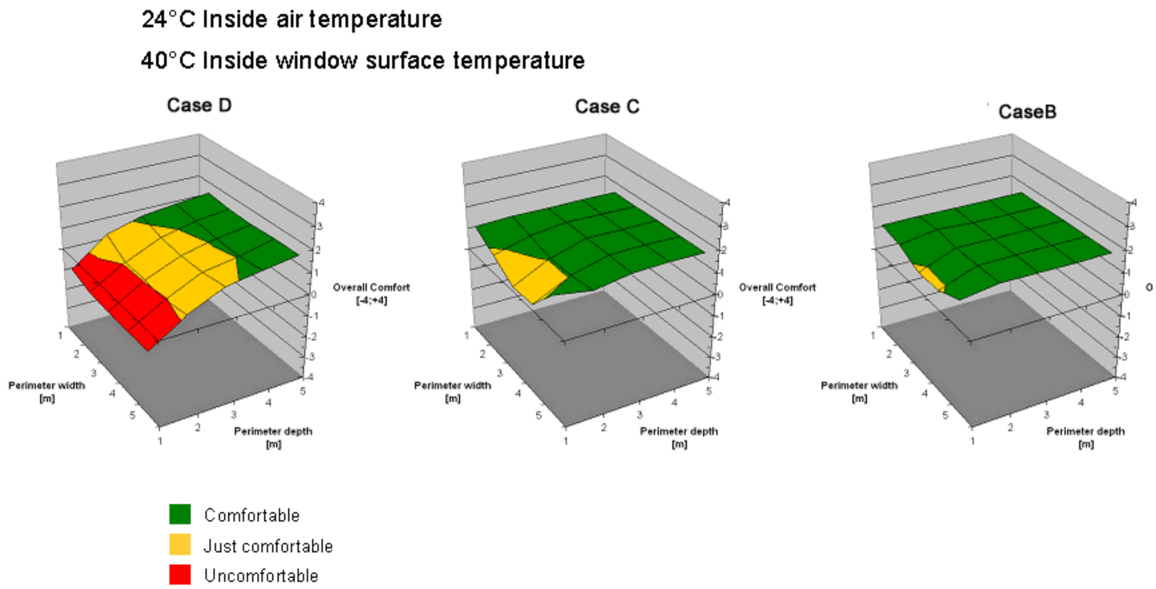


Figure 26: Spatial distribution of comfort

The figures indicate that the larger the window area, the greater the influence of the window on the spatial comfort distribution. A larger window causes the zone of discomfort to penetrate much deeper into the space.

5.5 FRAME AND EDGE EFFECT

We have investigated the impact of the contribution of frame, edge- and center of glass temperatures on the overall assessment of comfort. The objective was to answer the following two questions:

1) Is it important to consider the differences in temperature between the different window components (i.e. center of glass, edge of glass and frame)?
2) Is it an acceptable approximation to use an area-weighted window temperature instead of explicitly modeling each window component with a different temperature?

Figure 27 shows a window and its division into frame, edge- and center of glass. We have used WINDOW to calculate the center of glass temperature and THERM to calculate the edge of glass and frame temperatures.

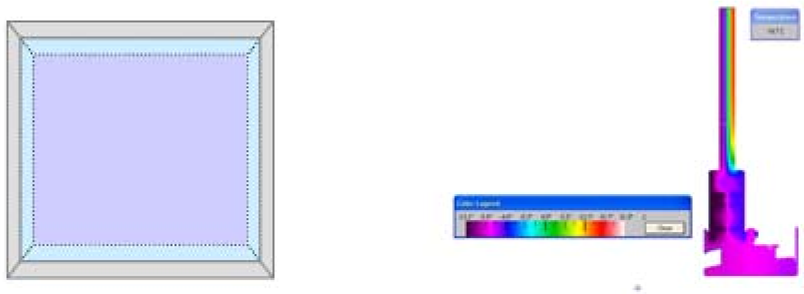


Figure 27: Influence of window components on overall window comfort

The UCB comfort model allows us to model the different window components and define the temperatures for frame, edge- and center glass explicitly. We have compared the predicted comfort values obtained using three different modeling approaches (Figure 28):

- explicit modeling of the three window components with each their own temperature;
- area-weighted approach by calculating an average window temperature;
- modeling the entire window at the center of glass temperature and ignoring the presence of the edge of glass and frame.

Figure 28: Explicit modeling of window components vs. uniform window temperature

The differences between the different modeling approaches have been assessed by carrying out some sample window simulations.

A window unit with the following specifications was used: 1.5m wide by 1.2m high, double IGU, low-e coating, argon filling, 75mm wide aluminum frame. For the first simulations, the premise was to combine a well performing glazing type with a low quality frame to obtain a large temperature difference between the center of the glazing and the frame and hence emphasize the frame effect. The typical room geometry was used as simulation environment and the air temperature was held at 24.5 ^{o}C, which is the neutral temperature for a person wearing typical indoor clothing (0.59 clo), doing light office work (1.2met) while seated along the center line of the window, 1 m away from the window. Three different window configurations were considered (Figure 29):

Table 9 allows the comparison of the results obtained using the three different modeling approaches for the window with a low-performance frame. Standard NFRC winter boundary conditions (outdoor air temperature –18ºC) were used in the calculation of the interior surface temperature of the window components.

The influence of the overall window temperature on the local hand comfort (i.e. the body part most sensitive to environmental changes in this simulation environment) is given in Table 10:

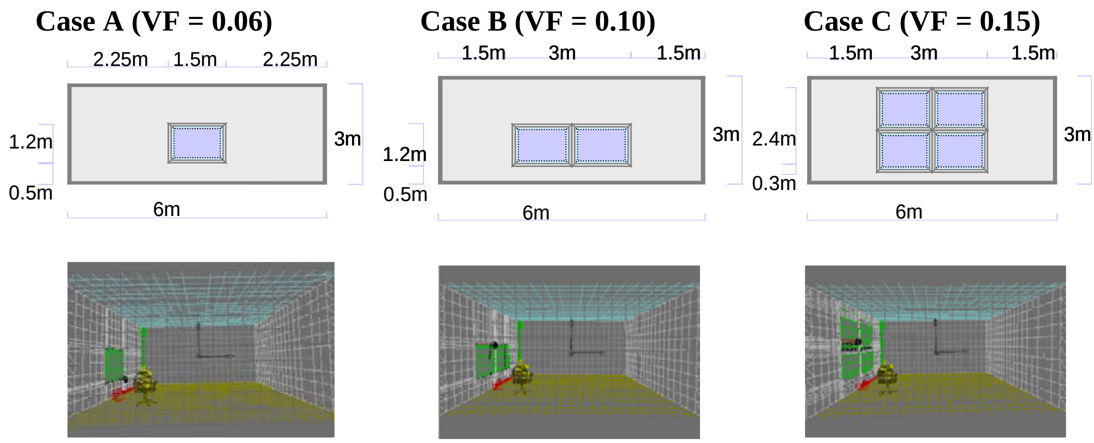


Figure 29: Different window configurations considered for calculation of overall window comfort

Window element	Temperature [^oC]	Dimension
center of glass	14.7	1225mm x 925mm
edge of glass	6	62.5mm depth
frame	-5	75mm width
Overall window (area-weighted average)	9.1	1500mm x 1200mm

Table 9: Calculation of area-weighted window temperature (low performance frame)

	Case A (VF = 0.06)	Case B (VF = 0.10)	Case C (VF = 0.15)
reference case with no window	1.97	1.97	1.97
ignoring effect of frame and edge of glass	1.5	1.3	1.0
explicit modeling of frame and edge of glass	1.4	0.9	0.4
area-weighted average window temperature	1.3	0.8	0.3

Table 10: Local hand comfort for different window configurations (low performance frame)

The simulation results indicate that the different modeling approaches with respect to window temperature yield similar results when the window size is small (case A). However, as the window size increases (cases B and C), the impact of the frame and the edge of glass temperature becomes significant and can no longer be ignored. The lower temperature of the low performance frame starts to impact the overall window temperature compared to the relatively warm center of glass temperature of the good performing glazing system. Local hand comfort is reduced by about 0.5 when considering the effect of the frame and the edge of glass temperature (from 1.3 to 0.9 for case B, and from 1.0 to 0.4 for case C). We note however that the difference between the results obtained with the explicit modeling approach and the area-weighted average approach are small, with 0.1 comfort level reduction for the area-weighted modeling.

The same comparison of analysis results has also been carried out for a window with a good performing frame and the temperature of the window components is shown in Table 11. In this case, the temperature difference between the frame and the center of glass is smaller comparing the window with a poor performing frame.

Window element	Temperature [^oC]	Dimension
center of glass	14.7	1225mm x 925mm
edge of glass	10	62.5mm depth
frame	5	75mm width
Overall window (area-weighted average)	11.9	1500mm x 1200mm

Table 11: Calculation of area-weighted window temperature (high performance frame)

The influence of the overall window temperature on the local hand comfort is given in Table 12. As expected, the impact of the frame and edge of glass on the comfort assessment is smaller. However, we still see that with case C the frame and edge of glass temperatures cannot be ignored (-0.3 reduction in local hand comfort). The comfort levels obtained by explicit modeling of the window components and the area-weighted modeling approach are identical. Again, the simulations were carried out for NFRC winter conditions.

	Case A	Case B	Case C
reference case with no window	1.97	1.97	1.97
ignoring effect of frame and edge of glass	1.5	1.3	1.0
explicit modeling of frame and edge of glass	1. 4	1.2	0.7
area-weighted average window temperature	1.4	1.2	0.7

Table 12: Local hand comfort for different window configurations (high performance frame)

We conclude that (1) the effects of the frame and edge of glass temperature on the overall window temperature and the resulting impact on local comfort should not be ignored, (2) the difference between the results obtained using the area-weighted average window temperature and explicitly modeled window component temperatures is small and (3) the comfort levels calculated using the area-weighted method result in slightly lower comfort compared to the explicit method due to the spreading of the lower frame temperature over a larger area. As such, we recommend adopting an area-weighted calculation of the average window temperature to assess the overall impact of the window on occupant comfort.

5.6 IMPACT OF DIFFUSE RADIATION

5.6.1 GENERAL APPROACH

In typical conditions, when direct solar radiation reaches the body it will almost always cause discomfort. Exceptions to this include cases where the air temperature is very cold, the solar radiation falls on only a small portion of the body, or the solar transmittance of the window is extremely low. [to do: add an example analysis of solar radiation falling on the body]. Because direct solar radiation almost always causes discomfort, we make the assumption in our analysis that the occupant will take steps to avoid this condition by relocating or adjusting blinds or shades. It would be possible to include direct solar radiation in a rating system, although specific assumptions about how much of the body received radiation would need to be made and these are very specific to the details of a particular building. In the rating approach we describe below, we only evaluate the effect of diffuse solar radiation falling on the body. However, we do consider the effect of direct radiation on the glass surface temperature.

A common way to account for the solar radiation impact on the human body is to convert the direct and diffuse solar radiation to an equivalent mean radiation temperature (Arens at al. 1986, Matzarakis et al. 2000). Here we adopted a similar approach, converting diffuse radiation into an equivalent glass surface temperature rise.

The approach entails the calculation of an equivalent temperature increase of the inside surface of the window that would result in the same net heat transfer to the person (by long-wave radiation) as the diffuse radiation. As such, the calculation method accounts for the transmitted diffuse radiation by estimating a temperature increase of the window surface equivalent to the impact of the diffuse radiation.

The approach considers the diffuse radiation arriving from the sky as well as the diffuse radiation reflected from the ground surface (Figure 30). Part of the diffuse radiation incident on the window is transmitted (proportional to the solar transmittance of the glazing) and a portion of the transmitted radiation gets absorbed by the building occupant (proportional to the absorptance of the occupant's clothing).

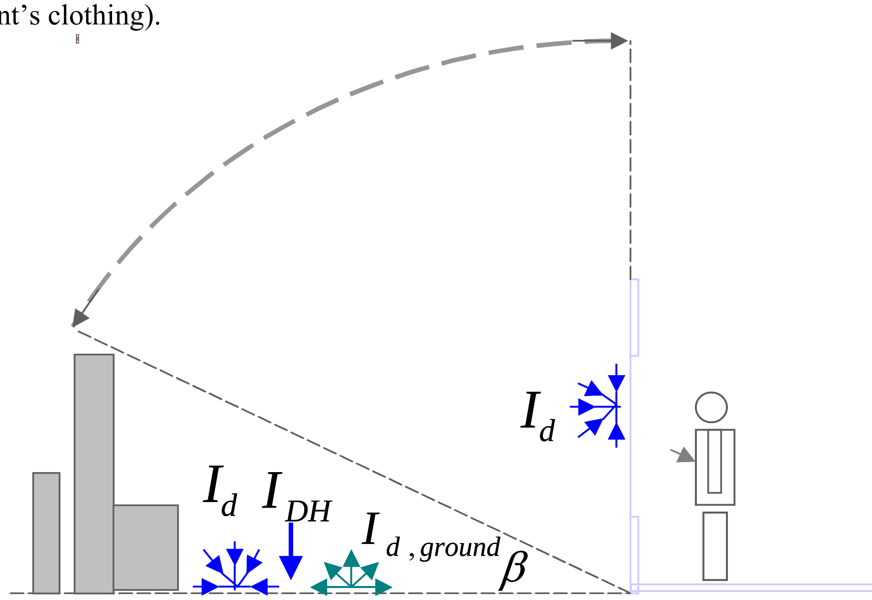


Figure 30: Diffuse solar radiation from the sky and reflected from the ground

The calculation of the glass temperature increase which is equivalent to the impact of diffuse radiation is carried out in three steps:

1. Calculate 'sky' temperature increment of the window assuming the person sees radiation from the sky only;
2. Calculate 'ground' temperature increment of the window assuming the person sees reflected radiation from the ground only;
3. Calculate weighted average of 'sky' and 'ground' temperature increments, factoring in how much a person sees of sky and ground to obtain an 'overall' glass temperature increment.

The calculation process of each of the three steps is discussed further in the following paragraphs.

5.6.2 STEP 1 – DIFFUSE RADIATION FROM SKY

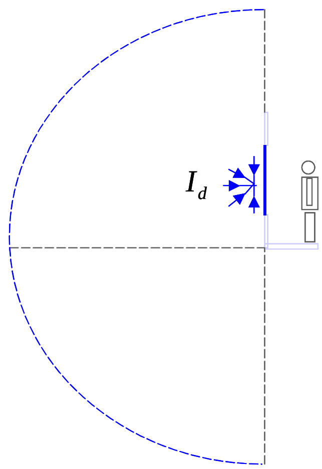


Figure 31: Calculation of diffuse radiation - Assume entire view is sky

The calculation of the portion of diffuse solar radiation coming from the sky which gets absorbed by a person, under the assumption that the entire view (half-sphere) is seen as sky, can be derived as follows:

Diffuse solar radiation	=	I_d	[W/m^2]
Portion transmitted through glass	=	$I_d A_g \tau_{sol}$	(Eq. 3)
Portion absorbed by person	=	$I_d A_g \tau_{sol} F_{g-p} \alpha_{clo}$	(Eq. 4)

With

A_g	=	area of glazing	[m^2]
τ_{sol}	=	solar transmittance of glazing	[-]
F_{g-p}	=	viewfactor glazing - person	[-]
α_{clo}	=	clothing absorptance	[-]

Then,

Radiation between glass and person $= \sigma\varepsilon_g\left[T_g^4 - T_p^4\right]A_g F_{g-p}$ (Eq. 5)

And,

Radiation between glass and person after glass has absorbed diffuse radiation from sky $= \sigma\varepsilon_g\left[\left(T_g + \delta T_{g-sky}\right)^4 - T_p^4\right]A_g F_{g-p}$ (Eq. 6)

Hence,

Radiation increase after glass has absorbed diffuse radiation from sky (Eq. 6 – Eq. 5) $= \sigma\varepsilon_g\left[\left(T_g + \delta T_{g-sky}\right)^4 - T_g^4\right]A_g F_{g-p}$ (Eq.7)

If we say that

Diffuse solar radiation absorbed by person = Increased radiation heat transfer between glass and person

Then,

$$I_d \tau_{sol} \alpha_{clo} A_g F_{g-p} = \sigma \varepsilon_g \left[\left(T_g + \delta T_{g-sky} \right)^4 - T_g^4 \right] A_g F_{g-p} \qquad \text{(Eq.8)}$$

or

$$\frac{I_d \tau_{sol} \alpha_{clo}}{\sigma \varepsilon_g} = \left(T_g - \delta T_{g-sky} \right)^4 - T_g^4 \qquad \text{(Eq.9)}$$

Equation 9 allows the temperature increment due to the diffuse radiation coming from the sky to be written in function of the diffuse radiation and a series of known variables, such as the solar transmittance of the glazing, the absorptance of the clothing and the emissivity of the glazing surface.

5.6.3 STEP 2 – GROUND REFLECTED DIFFUSE RADIATION

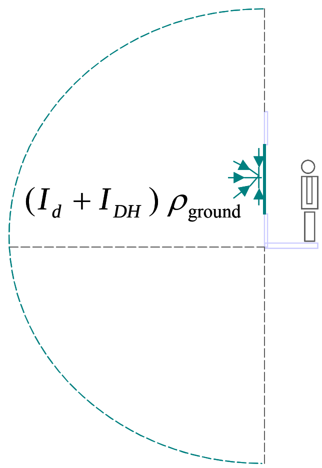


Figure 32: Calculation of diffuse radiation - Assume entire view is ground

The calculation of the ground reflected solar radiation (sky diffuse + horizontal direct) which gets absorbed by a person, under the assumption that the entire view (half-sphere) is seen as the ground, can be derived as follows:

Ground reflected diffuse solar radiation $= \left(I_d + I_{DH} \right) \rho_{ground}$ [W/m²]

Portion transmitted through glass $= \left(I_d + I_{DH} \right) \rho_{ground} A_g \tau_{sol}$ (Eq. 10)

Portion absorbed by person $= \left(I_d + I_{DH} \right) \rho_{ground} A_g \tau_{sol} F_{g-p} \alpha_{clo}$ (Eq. 11)

With

I_{DH} = horizontal direct radiation, which upon reflection of the ground is transformed into diffuse radiation [W/m²]

ρ_{ground} = ground reflectance [-]

Then,

Radiation increase after glass has absorbed ground reflected radiation $= \sigma \varepsilon_g \left[\left(T_g + \delta T_{g-ground} \right)^4 - T_g^4 \right] A_g F_{g-p}$ (Eq.12)

If we say again that

Ground reflected solar radiation absorbed by person = Increased radiation heat transfer between glass and person

Then,

$$\left(I_d + I_{DH}\right)\rho_{ground} A_g \tau_{sol} F_{g-p} \alpha_{clo} = \sigma \varepsilon_g \left[\left(T_g + \delta T_{g-ground}\right)^4 - T_g^4\right] A_g F_{g-p} \qquad \text{(Eq.13)}$$

or

$$\frac{\left(I_d + I_{DH}\right)\rho_{ground} \tau_{sol} \alpha_{clo}}{\sigma \varepsilon_g} = \left(T_g - \delta T_{g-ground}\right)^4 - T_g^4 \qquad \text{(Eq.14)}$$

Equation 14 allows the temperature increment due to the ground-reflected diffuse radiation to be written in function of the diffuse radiation and a series of known variables, such as the solar transmittance of the glazing, the absorptance of the clothing and the emissivity of the glazing surface.

5.6.4 STEP 3 – WEIGHTED GLASS TEMPERATURE INCREASE

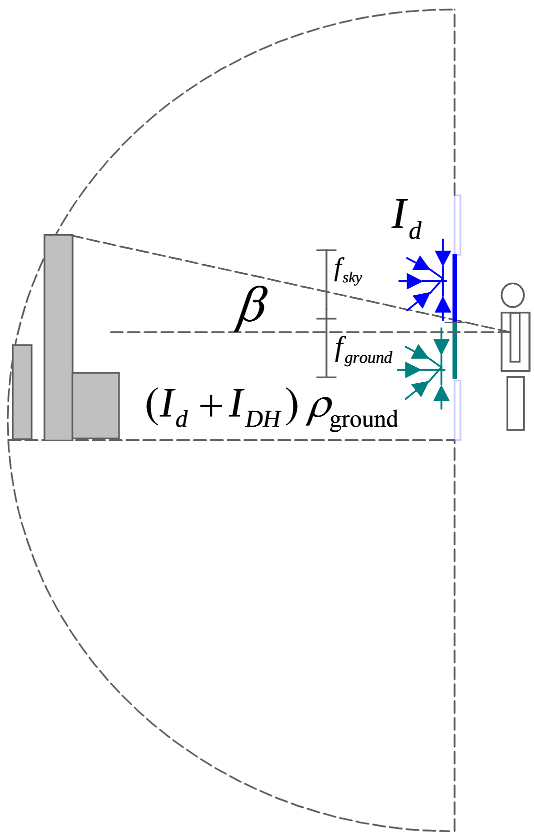


Figure 33: Calculation of diffuse radiation - Weighted glass temperature

The calculation of the weighted glass temperature increase, which combines the effects of both the temperature increment due to the diffuse radiation coming from the sky as well as the ground reflected diffuse radiation, can be summarized as follows:

First, we divide the glazing in two portions: a sky fraction f_{sky} receiving the diffuse radiation from the sky and a ground fraction $f_{ground} = 1 - f_{sky}$ receiving the diffuse radiation reflected by the ground. We obtain a resulting combined temperature increment by calculating the weighted average of both temperature increments calculated in sections 5.6.2 and 5.6.3.

$$\delta T_g = \delta T_{g-sky} f_{sky} + \delta T_{g-ground} f_{ground} \qquad \text{(Eq. 15)}$$

5.6.5 SAMPLE CALCULATION RESULTS

Some sample calculations of the glass temperature increase δT_g [°C] that results in long-wave heat transfer which is equivalent to the impact of short-wave diffuse radiation are provided in Table 13. The following values for the variables are assumed: $f_{sky} = 0.5$, $\rho_{ground} = 0.2$, $\alpha_{clo} = 0.5$.

τ_{sol}	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8
I_d = 165 W/m^2 I_{DH} = 165 W/m^2	1.4	2.7	4.0	5.3	6.6	7.9	9.2	10.4
I_d = 300 W/m^2 I_{DH} = 0 W/m^2	1.6	3.2	4.8	6.3	7.8	9.3	10.7	12.1

Table 13: Equivalent glass temperature increase due to diffuse radiation

The benefits of converting the impact of diffuse radiation in an equivalent glass temperature increase are (1) that the complex impact of diffuse solar radiation is separated from the comfort model and the effect re-inserted in the comfort assessment in a more straightforward way, (2) the approach allows a simpler rating of window comfort while still considering diffuse radiation, and (3) the approach allows the evaluation of windows with curtains. One would only need to provide a curtain temperature (under direct and diffuse solar radiation) to calculate the impact on comfort.

5.7 IMPACT OF DIRECT RADIATION

5.7.1 GENERAL APPROACH

The UCB Comfort Model is capable of including the effect of direct solar radiation in the comfort calculations. The model has a feature to specify the sun position (altitude and azimuth) and the amount of direct and diffuse radiation, and consecutively calculates the sun penetration into the space (including the effect of any shading elements). On the basis of these geometric relations, the amount of solar radiation that is transmitted by the glazing and absorbed by the occupant is calculated. The model allows the specification of the optical properties of the glazing and the absorption of the occupant. The amount of direct radiation is geometrically calculated for the polygons mapping the body surface and the diffuse radiation is evenly spread throughout the space.

5.7.2 SAMPLE CALCULATIONS

Sample calculations were carried out for an occupant positioned in the room shown in Figure 16 and 1m away from a window. The case B geometry (view factor=0.10) was assumed to illustrate the effect of geometry, i.e. the relationship between the person and the sun as seen through a particular window. The window was south facing and the sun was positioned directly south (azimuth = 180º) at 45º altitude. NFRC Summer environmental boundary conditions were assumed (outdoor temperature = 32ºC, indoor temperature = 24ºC and solar radiation = 783 W/m^2). Comfort boundary conditions of 1.2met and 0.6clo were assumed. The glazing was assumed to be clear double glazing with a clear low-e coating on face 3. Other glazing systems were considered in Section 5.8.3. Under NFRC Summer boundary conditions, the interior surface temperature of the glazing is 36.8ºC.

Figure 34 illustrates the direct sun beam penetrating the window and hitting the occupant. The screenshot also shows that the Overall Sensation of the person is +2.76 (very warm) which yields an Overall Comfort vote of -2.40 (very uncomfortable). Although, the applied direct solar radiation is quite high (783 W/m^2), the maximum amount of radiation hitting the occupant (on the left thy) is only about 350 W/m^2 because the clear double low-e glazing only has a solar transmittance of 0.47. This example shows that in most cases, exposure to direct solar radiation tends to cause significant discomfort. The maximum allowable amount of direct solar radiation that can be accommodated by different glazing systems whilst maintaining occupant comfort is explored in section 5.8.3.

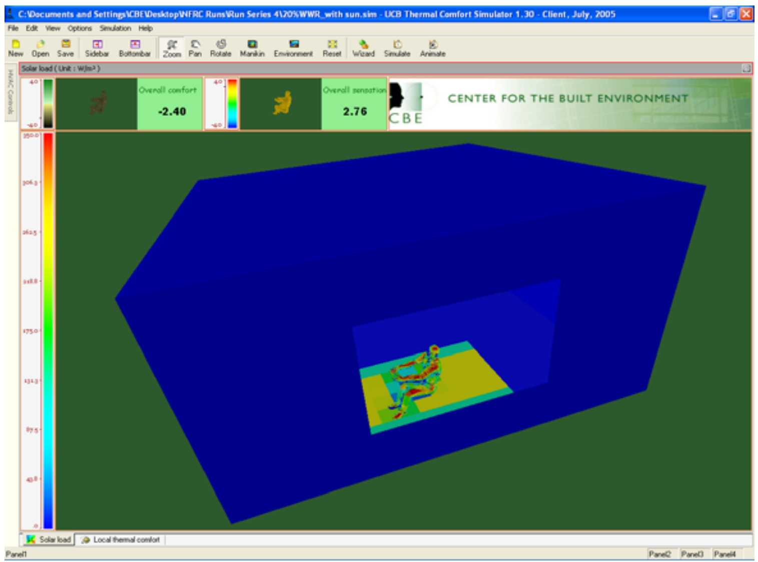


Figure 34: Impact of direct solar radiation, case B geometry

5.8 MAXIMUM ALLOWABLE SOLAR RADIATION

5.8.1 GENERAL CONSIDERATIONS

The dominant factor affecting summer indoor comfort in the vicinity of windows is solar radiation. The impact of solar radiation on window performance is determined by the optical properties of the window. The effect can be divided into two components: transmitted radiation and absorbed radiation. The transmitted radiation impacts how much shortwave radiation reaches the body and the absorbed radiation effects how much the window temperature will increase. The increase of the interior window temperature causes radiant asymmetry as the glazing becomes warm compared to other indoor surfaces such as walls. As such, glazing can become a source of discomfort when it is heated up by solar radiation, because it will act as a radiator emitting heat to the building occupant.

A way to look at the summer-time performance of glazing is to assess the maximum allowable solar radiation that can be tolerated for it to remain comfortable within the room. However, a building occupant will likely be more uncomfortable when he is in the direct path of the sun (exposed to direct solar radiation). In this situation, it is likely that the building occupant will mitigate the discomfort due to the direct exposure to solar radiation by either moving out of the direct solar beam or by pulling a blind down that blocks most of the direct radiation.

As such, our goal was to assess the impact of solar radiation on occupant comfort in two different scenarios. The first scenario assumes that the building occupant is not exposed to direct solar radiation, but is only receiving diffuse radiation transmitted through the glazing and the long-wave radiation due to the glass temperature increase (caused by the glass being exposed to direct and diffuse radiation). A second scenario assumes that the building occupant is exposed to a beam of direct solar radiation penetrating the window and the long-wave radiation of the glazing system which is heated up by the sun.

5.8.2 SCENARIO 1: OCCUPANT NOT EXPOSED TO DIRECT SOLAR RADIATION

This method assumes that the building occupant is not exposed to direct solar radiation and only considers transmitted diffuse radiation and the effect of direct and diffuse radiation absorbed by the glazing system. For this analysis we assumed diffuse radiation to be 20% of the direct radiation, typical of clear sky conditions. Our approach is to calculate a resultant interior surface temperature on the basis of (1) the temperature difference between inside and outside, (2) the increase in glazing temperature due to the exposure to direct solar radiation and (3) the glazing temperature increment due to the impact of diffuse radiation (as per section 5.6). Using this approach, windows can the be characterized in terms of the maximum amount of solar radiation they can tolerate while maintaining comfort for a building occupant seated 1m away from a fully glazed façade (case D).

The maximum allowable solar radiation for a glazing system is calculated in an iterative process:

Step 1: we calculate the interior surface temperature of the glazing (using WINDOW) due to ΔT inside-outside (assuming NFRC summer temperature conditions) and direct solar radiation;

Step 2: we apply 20% of the direct radiation as diffuse radiation and calculate the increase in interior surface temperature δT_g it causes (using the approach explained in Section 5.6);

Step 3: we sum the original interior surface temperature as calculated with WINDOW5 with the temperature increase δT_g due to diffuse radiation to obtain a 'resultant' interior surface temperature of the glazing;

Step 4: we increase the direct solar radiation iteratively (maintaining diffuse radiation equal to 20% of direct radiation) in order to obtain the maximum allowable window interior surface temperature as predicted by the UCB comfort model.

This step-by-step iterative calculation has been carried out for a number of glazing systems exposed to NFRC summer boundary conditions (Ti = 24ºC and Te = 32ºC). Other simulation variables included case D geometry, activity level of 1.2met, typical indoor clothing level of 0.59 clo. These boundary conditions yield a maximum allowable interior surface temperature of the glazing of 36ºC for a person seated 1m away from a very large window.

Table 14 provides a range of glazing systems within a wide band of performances, from single clear glazing to clear and coated double glazing and then to clear and coated triple glazing.

CODE	GLAZING DESCRIPTION
G1	Clear single glazing, 6mm
G2	Bronze single glazing, 6mm
G3	Clear double glazing, air cavity, 6/16/6mm
G4	Low-ε double glazing, air cavity, ε = 0.08 face 3, 6/16/6mm
G5	Spectrally selective low-ε double glazing, air cavity, ε = 0.04 face 2, 6/16/6mm
G6	Clear triple glazing, air cavities, 6/16/6/16/6mm
G7	Low-ε triple glazing, air cavity, ε = 0.08 face 2&4, 6/16/6/16/6mm
G8	Spectrally selective low-ε triple glazing, air cavity, ε = 0.04 face 2 & ε = 0.08 face 4, 6/16/6/16/6mm

Table 14: Range of glazing systems

The maximum allowable solar radiation (in W/m²) is calculated for each of the glazing systems described in Table 14 and the results are presented in Figure 35. The data provides a ranking of the glazing systems in terms of their solar performance expressed as the maximum allowable solar radiation to maintain indoor comfort 1m away from a fully glazed façade. As expected, glazing system G5, which is the double glazing with a spectrally selective coating on face 2, performs best by allowing the highest maximum solar radiation. The tined single glass (G2) cannot tolerate as much solar radiation as the single clear glazing (G1), because it absorbs the solar radiation which causes it to heat up which results in radiant discomfort. A clear low- ε coating does not have a significant impact on the solar performance of double and triple glazing, because it only reduces the solar transmission of the window panes slightly. However, a spectrally selective low- ε coating (G5 and G8) greatly improves the solar performance of the glazing in terms of maximum allowable solar radiation.

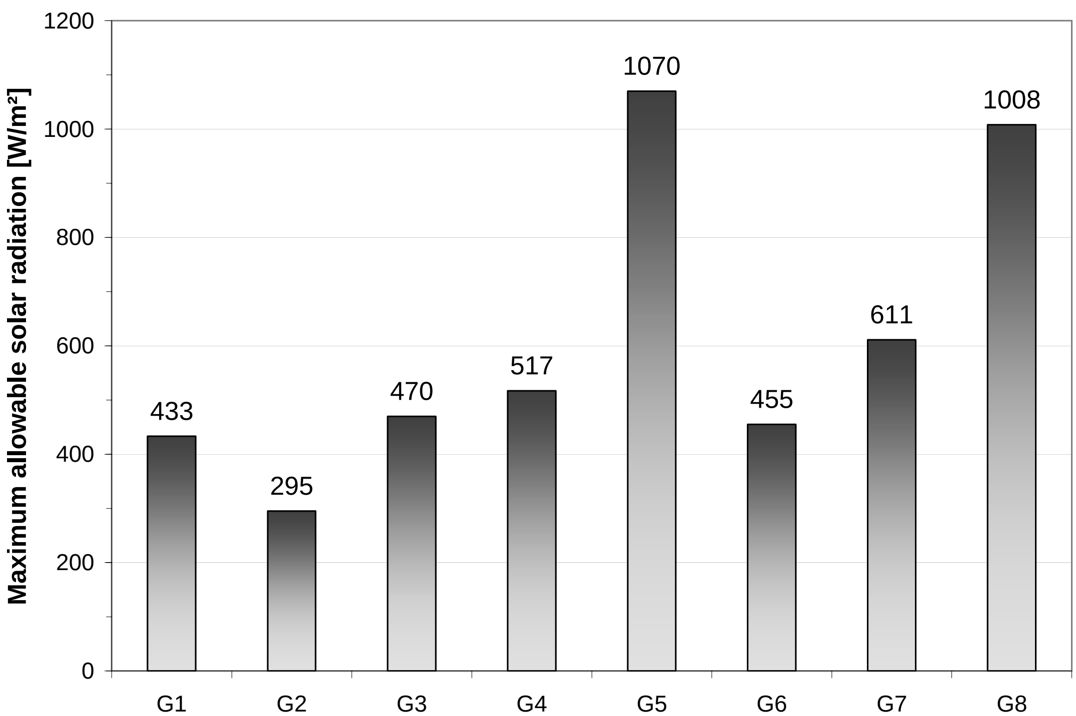


Figure 35: Scenario 1 – Maximum allowable direct solar radiation to maintain indoor comfort

5.8.3 SCENARIO 2: OCCUPANT EXPOSED TO DIRECT SOLAR RADIATION

The performance of the different glazing systems outlined in Table 14 has also been assessed for the scenario where the occupant is exposed to direct solar radiation. The person is positioned 1m away from a fully glazed façade (case D - geometry as per Figure 16). Once again, NFRC Summer boundary conditions (Ti = 24ºC and Te = 32ºC) have been assumed with varying levels of solar radiation (0, 250, 500, 750, 783 W/m^2). The comfort boundary conditions are again an activity level of 1.2met, and a typical indoor clothing level of 0.59 clo. The interior surface temperature of the different glazing systems exposed to different levels of solar radiation was calculated using WINDOW. For every glazing system, we calculated how much direct solar radiation could be allowed whilst maintaining indoor comfort for the occupant. A local comfort above 0 for every body segment was used as the comfort criterion in this evaluation.

Figure 36 illustrates that the maximum allowable solar radiation for the different glazing systems is much lower when considering a scenario where the occupant is exposed to direct solar radiation. However, the relative order of the performance of the different glazing systems is quite similar with only glazing systems G1/G2 (clear/bronze single glazing) and G5/G8 (spectrally selective low-ε double/triple glazing) changing their performance relative to each other.

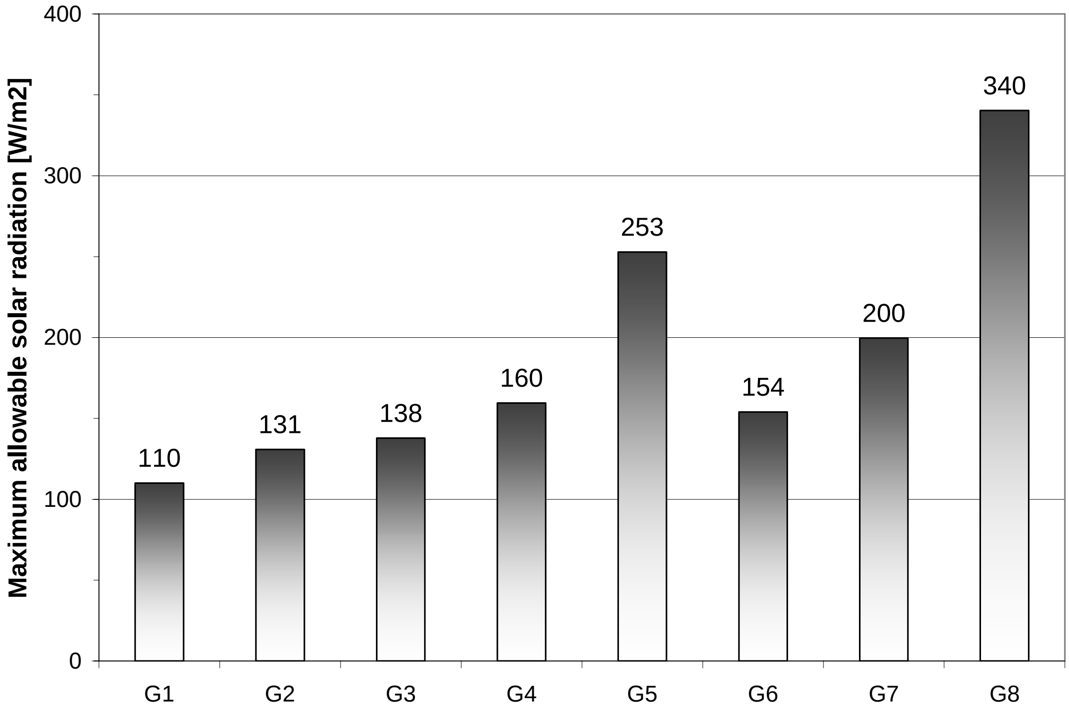


Figure 36: Scenario 2 – Maximum allowable direct solar radiation to maintain indoor comfort

5.8.4 COMPARISON: WITHOUT/WITH DIRECT SOLAR RADIATION

Figure 40 compares the maximum allowable solar radiation in order to maintain indoor comfort for a person seated at 1m away from a fully glazed façade (case D). It is clear that the scenario where the occupant is exposed to direct solar radiation yields discomfort at much lower radiation levels compared to the scenario where the occupant is not exposed to any direct radiation. The case without direct exposure assesses the effect of the temperature increase of the glazing by absorption, combined with the effect of diffuse radiation (assumed equal to 20% of the direct radiation) which is translated into an equivalent temperature rise of the interior glass surface following the method outlined in section 5.6. The case with direct exposure assesses the effect of the temperature increase of the glazing by absorption, combined with the effect of the direct solar radiation that is transmitted through the glazing and absorbed by the occupant. As such, the first scenario puts much more emphasis on those glazing properties that govern the indirect heat transfer of radiation to the indoor space (such as solar absorption and U-value), whereas the second scenario puts more importance on those glazing properties that govern direct transfer of radiation into the space (solar transmission). As mentioned, both approaches lead to slightly different metrics for the maximum allowable solar radiation and hence provide a somewhat different ordering of the solar performance of the different glazing systems with respect to summer indoor comfort.

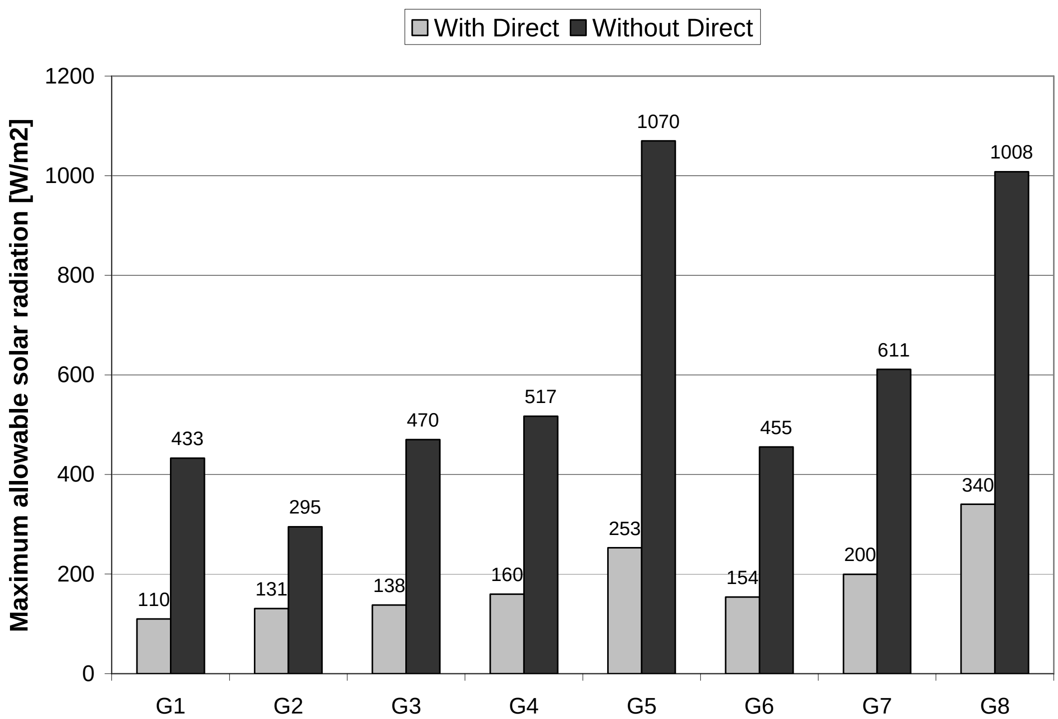


Figure 37: Comparison – Maximum allowable direct solar radiation with/without direct exposure

5.9 WINDOW OPTICAL PROPERTY CHARACTERIZATION

The two effects described above, transmitted direct/diffuse solar radiation and the temperature increase of the window due to absorbed radiation, are characterized by existing rating metrics. The solar heat gain coefficient (SHGC) is defined as:

$$SHGC = Tsol + N\mathit{A}$$

where:
Tsol = overall solar transmittance
N = inward-flowing fraction of absorbed solar radiation
A = solar absorbtance

With respect to thermal comfort, SHGC has the relevant attributes of the window but the relative importance of Tsol and the window surface temperature increase due to the absorbed solar radiation are dependent on a specific condition. Two examples illustrate this point. For a person located very far from a small window, yet with direct solar radiation falling on them, Tsol is the important characteristic of the window with respect to comfort. Since the person is far from a small window, the view factor is small and therefore the longwave radiant exchange is small. Another example is a person sitting very close to a large window, but with no direct sun falling on them. In this case, the surface temperature of the window becomes important. It is a rare case where Tsol becomes unimportant, since even if direct sun does not fall on the body, diffuse radiation would still play a role.

Because Tsol and SHGC are easily obtained values for window products, we can easily determine the quantiy N*A* by the following:

$$N\mathit{A} = SHGC_{indirect} = SHGC - Tsol$$

We use the term $SHGC_{indirect}$ to refer to the fraction of solar energy absorbed by the window and subsequently transmitted to the inside by convection and radiation.

It can be shown that the data in Figure 38, which is the maximum allowable solar radiation calculated for scenario 1 – occupant not exposed to direct solar radiation – from section 5.8.2, can be well predicted by a linear regression of Tsol and $SHGC_{indirect}$ of the form:

$$\text{Maximum allowable solar radiation} = C_1 Tsol + C_2\, SHGC_{indirect} + C_3$$

The result of a linear regression (shown in Figure 38) yields:

$C_1 = -1200$
$C_2 = -5280$
$C_3 = 1600$

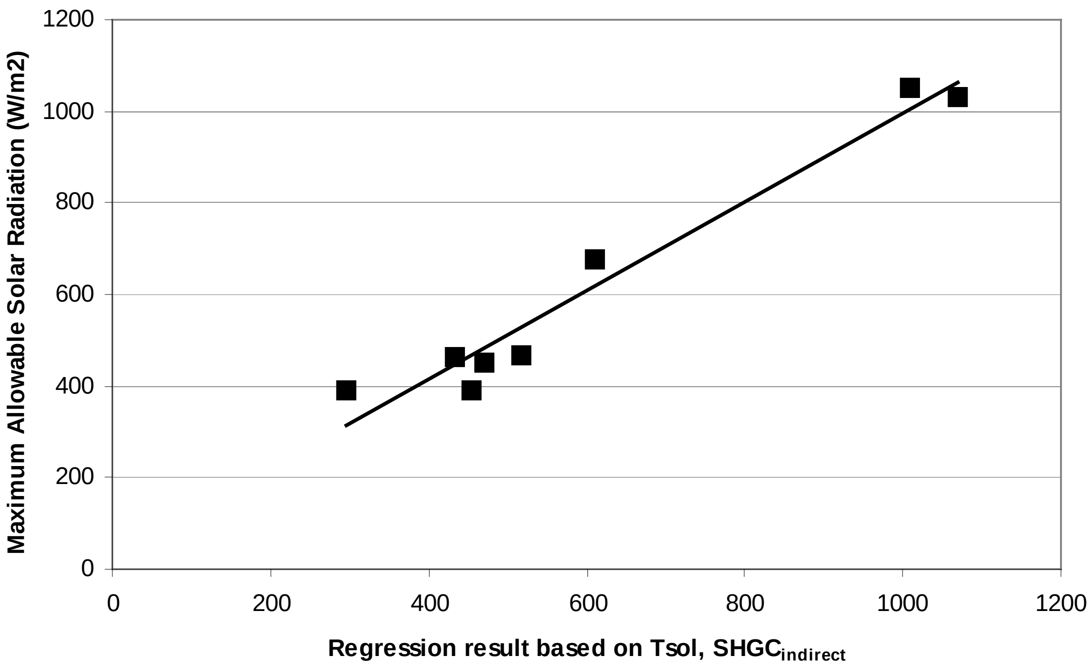


Figure 38: Regression of Maximum allowable solar radiation with Tsol and $SHGC_{indirect}$

This analysis suggests that for the conditions used for the analysis above, $SHGC_{indirect}$ is approximately 4.4 times (5280/1200) more important than Tsol in its impact on thermal comfort. We propose a new index, the **comfort solar heat gain coefficient**, defined as:

$$SHGC_{comfort} = Tsol + k * SHGC_{indirect}$$

The term *k* is a weighting factor on the indirect heat gain, which for the case of diffuse radiation falling on a person 1 meter from a large window (as in section 5.8.2) is of the order 4.4:

$$SHGC_{comfort} = Tsol + 4.4 * SHGC_{indirect}$$

For cases where only diffuse radiation (no direct radiation) falls on the occupant, *k* is very sensitive to the view factor between the occupant and the window. This is because both the transmitted diffuse radiation and the longwave radiation exchange are impacted by the viewfactor. The value of k should be smaller if the person receives direct solar radiation, and smaller still if they receive direct radiation and the view factor window becomes smaller. For scenario 2 described earlier of an occupant sitting 1 meter from a large window in direct sun (section 5.8.3), the value of k is approximately 2.4.

6 RATING METHODS FOR WINDOW COMFORT

In Section 5 we provided a discussion on the several parameters which are involved in the evaluation of window comfort. The following section proposes some methods which could be used to rate windows with respect to thermal comfort, to assess both winter and summer window comfort.

6.1 POSSIBLE ALTERNATIVES

The previous sections have discussed several methods to assess window thermal comfort. However, the list of different performance evaluators was by no means exhaustive. To illustrate this, we have provided a list of possible alternatives for the evaluation of window comfort. Our report has focused exclusively on the performance of windows at a specific point in time, under a static set of environmental conditions.

Some possible point-in-time indices to evaluate window comfort for a specific assessment geometry (space and window size, window and occupant location) are:

- Thermal comfort index (e.g. minimum local comfort) for NFRC summer and winter conditions;
- Required indoor air temperature to achieve comfort for NRFC summer and winter outdoor conditions;
- Window temperature impact expressed as the change in indoor air temperature which has a comfort impact equivalent to the window;
- Minimum distance from the window that a person can still be comfortable in the space;
- Minimum outside temperature that remains comfortable for the building occupant;
- Solar heat gain coefficient for comfort.

These point-in-time indices could also be evaluated over a longer period of time and even converted into annual indices, such as:

- Annual average comfort index;
- Number of hours outside the comfort zone;
- Annual energy required to modify room air temperature to maintain comfort;
- Percent of floor area of a specified room that remains comfortable to a certain level over the year;
- Percentage of time the outside temperature drops below the minimum outside temperature required to maintain comfort.

However, given the difficulties associated with the large number of (debatable) assumptions that would inherently be incorporated into an annual index, we have chosen not to pursue this path further in this report and only focus on point-in-time indices for window comfort.

6.2 WINTER RATING METHOD

The comfort index we propose for the assessment of window comfort in winter conditions is the minimum outdoor temperature that can be sustained while maintaining indoor comfort. The calculation of the minimum outdoor temperature is carried out for a specific set of geometric and environmental boundary conditions. The simulation space is the previously defined standard room measuring 6m wide by 6m deep and 3m high. One of the elevations is fully glazed (case D) and the building occupant is positioned sideways 1m away from the window along the center line of the room. The indoor air temperature is maintained at 23.5ºC and the seated occupant has a clothing level of 0.59 and a metabolic activity level of 1.2met.

To illustrate this approach, we have calculated the minimum allowable outdoor temperature for the 8 glazing systems previously mentioned in section 5.7. Table 15 provides the window properties and performance of the 8 glazing types, as calculated with WINDOW in accordance to NFRC 100-2002. As mentioned, the glazing systems have been chosen to represent a broad range of window performance that can be found in practice.

CODE	GLAZING DESCRIPTION
G1	Clear single glazing, 6mm
G2	Bronze single glazing, 6mm
G3	Clear double glazing, air cavity, 6/16/6mm
G4	Low-ε double glazing, air cavity, ε = 0.08 face 3, 6/16/6mm
G5	Spectrally selective low-ε double glazing, air cavity, ε = 0.04 face 2, 6/16/6mm
G6	Clear triple glazing, air cavities, 6/16/6/16/6mm
G7	Low-ε triple glazing, air cavity, ε = 0.08 face 2&4, 6/16/6/16/6mm
G8	Spectrally selective low-ε triple glazing, air cavity, ε = 0.04 face 2 & ε = 0.08 face 4, 6/16/6/16/6mm

GLAZING	VISIBLE LIGHT			SOLAR ENERGY				U-FACTOR
SYSTEM	Tvis	Rfvis	Rbvis	Tsol	Rfsol	Abs	SHGC	[W/(m^2.K)]
G1	0.88	0.08	0.08	0.77	0.07	0.16	0.82	5.8
G2	0.53	0.06	0.06	0.49	0.05	0.46	0.62	5.8
G3	0.79	0.14	0.14	0.61	0.11	0.28	0.70	2.7
G4	0.75	0.12	0.11	0.47	0.21	0.32	0.59	1.8
G5	0.67	0.12	0.12	0.31	0.32	0.37	0.36	1.7
G6	0.70	0.19	0.19	0.49	0.14	0.37	0.62	1.7
G7	0.65	0.14	0.14	0.34	0.23	0.43	0.45	0.9
G8	0.58	0.14	0.15	0.25	0.33	0.42	0.31	0.9

Table 15: Glazing build-up and performance

The minimum allowable outdoor temperature calculated for the 8 different types of glazing systems are shown in Figure 39. The minimum allowable outdoor temperature is directly related to the U-factor of the window, which is a measure of the heat loss through the window.

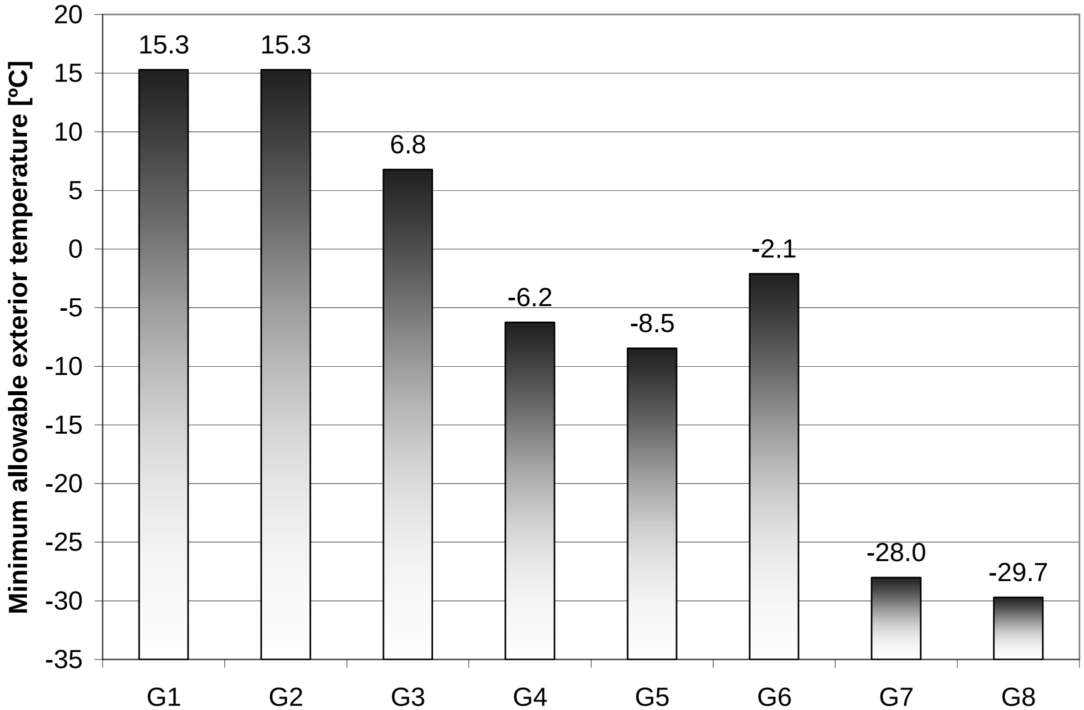


Figure 39: Winter glazing comfort rating - Minimum allowable outdoor temperature

To evaluate whole-window performance with respect to comfort, we have combined the above glazing systems with some typical window frames to obtain an overall window U-factor and the corresponding minimum allowable exterior temperature for the window. The window has been assumed to consist of 20% framing and 80% glazing, but this ratio obviously varies between window geometries. The U-factor of wood framing was assumed to be 2.3 W/(m^2.K), that of vinyl framing 3.4 W/(m^2.K) and of thermally broken aluminum framing 5.7 W/(m^2.K). Ten different window combinations of glazing and framing are shown in Table 16, combining a variation of low and high performance glazing systems with low and high performance window frames.

CODE	WINDOW DESCRIPTION
W1	Clear single glazing, wood frame
W2	Clear single glazing, thermally-broken aluminum frame
W3	Clear double glazing, wood frame
W4	Clear double glazing, thermally-broken aluminum frame
W5	Low-ε double glazing, wood frame
W6	Low-ε double glazing, vinyl frame
W7	Low-ε double glazing, thermally-broken aluminum frame
W8	Low-ε triple glazing, wood frame
W9	Low-ε triple glazing, vinyl frame
W10	Low-ε triple glazing, thermally-broken aluminum frame

WINDOW	GLAZING U-FACTOR	FRAME U-FACTOR	WINDOW U-FACTOR	$T_{ex,min}$
SYSTEM	[W/(m^2.K)]	[W/(m^2.K)]	[W/(m^2.K)]	[ºC]
W1	5.8	2.3	5.1	14.6
W2	5.8	5.7	5.8	15.7
W3	2.7	2.3	2.6	6.2
W4	2.7	5.7	3.3	9.7
W5	1.8	2.3	1.9	-0.4
W6	1.8	3.4	2.1	2.1
W7	1.8	5.7	2.6	5.9
W8	0.9	2.3	1.2	-15.1
W9	0.9	3.4	1.4	-8.8
W10	0.9	5.7	1.9	-0.9

Table 16: Window build-up and performance

Figure 40 displays the results for the calculated minimum allowable outdoor temperature for the different window systems listed in Table 16. In general, the frame performance is lower than the glazing performance (i.e. window framing has a higher U-factor than glazing), except when combining single and uncoated double glazing with a wooden frame. As such, the results show that the minimum allowable outdoor temperature to maintain indoor comfort increases to higher temperatures. For example, the minimum allowable outdoor temperature for glazing system G4 (low-ε double glazing) increases from -6.2ºC for the glazing alone, to -0.4ºC when combined with wooden framing and even to 2.1ºC when combined with thermally broken aluminum framing.

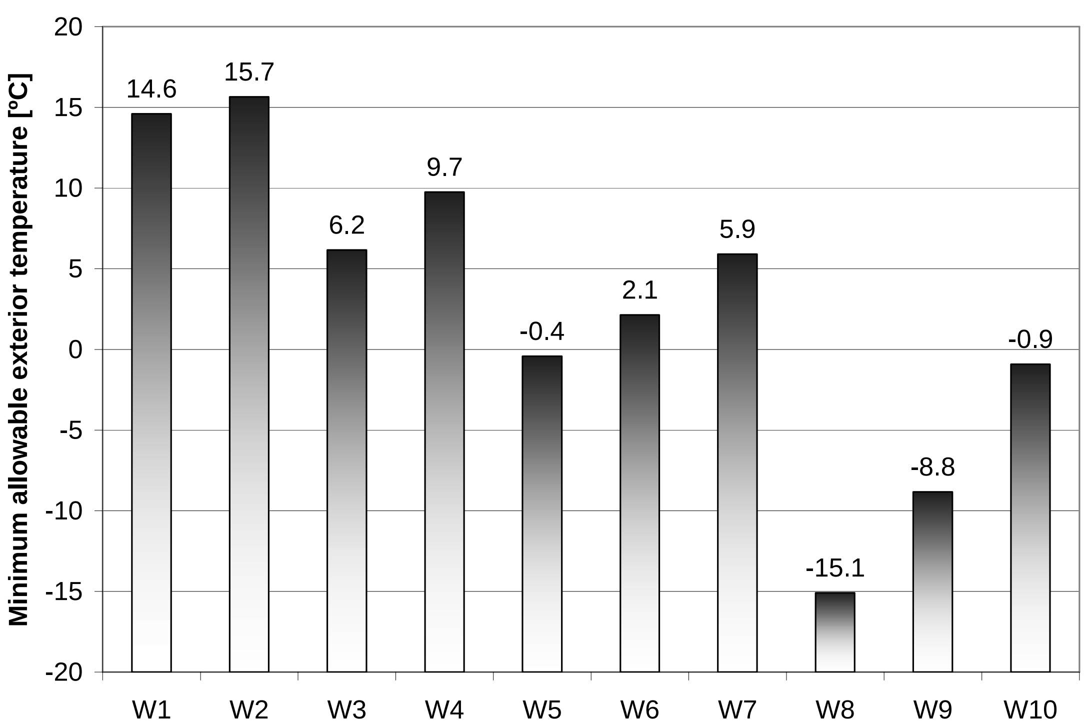


Figure 40: Winter window comfort rating - Minimum allowable outdoor temperature

The direct relationship between the window comfort indices and the U-factor of the windows is illustrated in Figure 41, which provides the regression relationship that allows the prediction of winter window comfort based on the window U-factor.

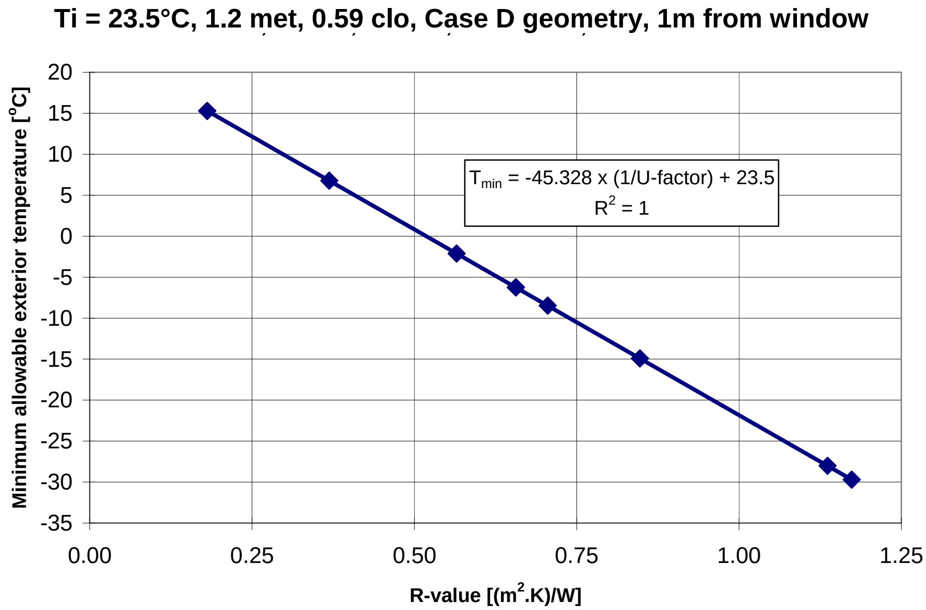
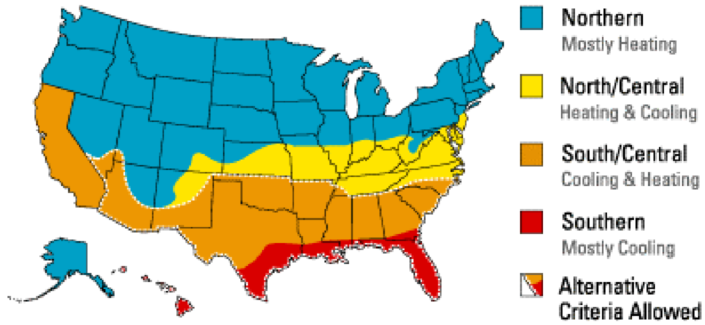


Figure 41: Linear regression between window R-value (1/U-factor) and minimum allowable outdoor temperature

The minimum allowable outdoor temperature index can also be compared to the window performance recommended by the Energy Star program run by EPA/DOE. The Energy Star criteria for residential windows, doors, and skylights are tailored to four Climate Zones, as shown in Figure 42. A product's energy efficiency for a given climate is based on its impact on heat gain and loss in cold weather and heat gain in warm weather.

On the basis of ASHRAE winter time (97.5%) design data, we have selected a number of cities lying within the four Energy Star climate zones to represent the variation in minimum outdoor temperatures experienced within each climate zone. From this minimum winter design temperature we have calculated the window U-factor required to maintain indoor comfort using the approach explained above. We then compared the U-factor we calculated with the window U-factor that is recommended by Energy Star for the different climate zones. The results are presented in Table 17. The maximum window U-factors Energy Star recommends from the perspective of energy conservation are higher than those calculated to maintain indoor comfort for the geometry we used. If a smaller window (and/or a larger distance from the window) was used, windows with higher U-factors would be acceptable.



Windows & Doors

Climate Zone	U-Factor[1]	SHGC[2]	
Northern	≤ 0.35	Any	
North/Central	≤ 0.40	≤ 0.55	
South/Central	≤ 0.40	≤ 0.40	Prescriptive
	≤ 0.41	≤ 0.36	Equivalent Performance *(Excluding CA)*
	≤ 0.42	≤ 0.31	
	≤ 0.43	≤ 0.24	*Products meeting these criteria also qualify in the Southern zone.*
Southern	≤ 0.65	≤ 0.40	Prescriptive
	≤ 0.66	≤ 0.39	Equivalent Performance
	≤ 0.67		
	≤ 0.68	≤ 0.38	
	≤ 0.69	≤ 0.37	
	≤ 0.70		
	≤ 0.71	≤ 0.36	
	≤ 0.72	≤ 0.35	
	≤ 0.73		
	≤ 0.74	≤ 0.34	
	≤ 0.75	≤ 0.33	

Skylights

Climate Zone	U-Factor[1] 2001 NFRC rated at 20°[3]	U-Factor[1] RES97 rated at 90°[4]	SHGC[2]
Northern	≤ 0.60	≤ 0.45	Any
North/Central	≤ 0.60	≤ 0.45	≤ 0.40
South/Central	≤ 0.60	≤ 0.45	≤ 0.40
Southern	≤ 0.75	≤ 0.75	≤ 0.40

[1] Btu/h.ft².°F

[2] Fraction of incident solar radiation.

[3] U-Factor qualification criteria based on 2001 NFRC simulation and certification procedures that rate skylights at a 20-degree angle. Although reported U-Factor is higher than RES97 rated products, energy performance at the ENERGY STAR minimum qualifying level is equivalent.

[4] NFRC certification using the 1997 NFRC procedures for residential windows (RES 97) that rated skylights at a 90-degree angle. Skylights rated under this procedure may be present in the marketplace until March 31, 2008. NFRC labels for products using this procedure state: "RES97 rated at 90 degrees."

Figure 42: Energy Star climate zones and recommended window performance

Table 17 illustrates that the minimum allowable outdoor temperature approach can be used to derive a maximum allowable window U-factor to maintain indoor comfort from a winter design temperature for a specific climatic region. As such, the approach can be used to develop a map of the US displaying the maximum allowable window U-factor to maintain indoor comfort, in parallel to the window performance map provided by Energy Star. The resolution of such map can be improved by considering a larger number of climatic sites. This kind of map would provide more detailed information on the window U-factor that is required to maintain indoor comfort for a set of pre-defined assessment variables.

ZONE	CITY	WINTER DESIGN TEMPERATURE [97.5%] [ºC]	WINDOW U-FACTOR [W/(m².K)]	ENERGY STAR U-FACTOR	ENERGY STAR ALLOWABLE $T_{ex,min}$
N	Bemidji, MN	-32	0.8	2.0	0.7
N	Fairbanks, AK	-44	0.7	2.0	0.7
N	Salt Lake, UT	-13	1.2	2.0	0.7

N	Winslow, AZ	-12	1.3	2.0	0.7
N/C	La Junta, CO	-16	1.1	2.3	3.6
N/C	Norfolk, VA	-6	1.6	2.3	3.6
S/C	Los Angeles, CA	6	2.6	2.3	3.6
S/C	Mt Shasta, CA	-7	1.5	2.3	3.6
S	Tallahassee, FL	-1	1.8	3.7	3.6
S	Key West, FL	14	4.7	3.7	11.2

Table 17: Window comfort assessment vs. Energy Star recommendations

6.3 SUMMER RATING METHOD

In summer, solar radiation and the optical properties of the window most heavily influence the inside surface temperature of the window. As a result, exterior temperature does not characterize the effect of the window on comfort as it does in winter and using a summer rating akin to the winter rating would have little value. The rating method we propose for the summer is the **comfort solar heat gain coefficient**, which is defined as:

$$SHGC_{comfort} = Tsol + k * SHGC_{indirect}$$

The term k is a weighting factor on the indirect heat gain, which value is dependent on the extent to which the occupant is exposed to direct and/or diffuse radiation.

Section 5.8 discussed the summer time performance of glazing systems in terms of the maximum allowable solar radiation which could be accommodated without causing discomfort. The occupant was seated 1m away from a fully glazed façade. Standard NFRC Summer boundary conditions were assumed for external and internal environmental temperature.

Two scenarios were considered: (1) a scenario where the occupant was not exposed to direct radiation but only to diffuse radiation (equal to 20% of the direct radiation), and (2) a scenario where the occupant was exposed to direct radiation. Both scenarios considered the effect of longwave radiation caused by the temperature increase of the glazing due to absorption of solar radiation. Scenario 1 added the effect of diffuse radiation as an equivalent temperature increase of the glazing surface as per section 5.6. Scenario 2 used the functionality of the UCB Comfort Model to directly calculate the direct solar radiation falling on the building occupant for a specific sun position.

The data from section 5.8 was used to derive a correlation between the maximum allowable solar radiation and the optical properties of the glazing (Tsol and SHGC) in section 5.9. On the basis of this correlation, the concept of a *summer comfort index* was introduced which could be calculated from a linear combination of the solar transmission and the regular SHGC.

The weighting factor k varies between the situations whether the occupant is exposed to only diffuse or direct solar radiation. Scenario 1 – considering only diffuse radiation – yields the following relationship:

$$\textit{Summer Comfort Index} = Tsol + 4.4 * SHGC_{indirect}$$

Scenario 2 – considering direct radiation – yields the following relationship:

$$\textit{Summer Comfort Index} = Tsol + 2.4 * SHGC_{indirect}$$

As mentioned, for cases where only diffuse radiation falls on the occupant, *k* is very sensitive to the view factor between the occupant and the window. This is because both the transmitted diffuse radiation and the longwave radiation exchange are impacted by the viewfactor. The value of *k* is smaller if the person receives direct solar radiation, and smaller still if they receive direct radiation and the view factor window becomes smaller.

These relationships allow ranking glazing performance in terms of the *summer comfort index*, once a set of specific boundary conditions is fixed, such as room geometry, window geometry, occupant position and environmental/comfort boundary conditions (met, clo, etc).

Table 18 provides the relevant properties of the different glazing systems and the calculated comfort SHGC for both scenarios, i.e. only diffuse and direct radiation.

GLAZING SYSTEM	OPTICAL PROPERTIES			U-FACTOR	DIFFUSE	DIRECT
	Tsol	SHGC	$SHGC_{indirect}$	[W/(m^2.K)]	*summer comfort index*	*summer comfort index*
G1	0.77	0.82	0.05	5.8	0.99	0.89
G2	0.49	0.62	0.13	5.8	1.06	0.80
G3	0.61	0.70	0.09	2.7	1.01	0.83
G4	0.47	0.59	0.12	1.8	1.00	0.76
G5	0.31	0.36	0.05	1.7	0.53	0.43
G6	0.49	0.62	0.13	1.7	1.06	0.80
G7	0.34	0.45	0.11	0.9	0.82	0.60
G8	0.25	0.31	0.06	0.9	0.51	0.39

Table 18: Comfort SHGC for different glazing systems – diffuse / direct radiation

Figure 43 and Figure 44 plot the *summer comfort index* as linear functions of varying solar transmission and SHGC. Both figures are made up of a series of curves, each curve representing a specific SHGC and varying solar transmission. Physically impossible combinations of SHGC and solar transmission have been excluded by stopping each SHGC curve at its corresponding minimum solar transmission value. For example, a glass with a solar transmission of 0.5 cannot have a SHGC below 0.5.

Figure 43 provides the relationships for scenario 1 – the case where the occupant is only exposed to diffuse radiation, whereas Figure 44 provides the curves for scenario 2 – the case where the occupant is exposed to direct radiation. The actual *summer comfort index* values for the different glazing systems of Table 18 are also shown on Figure 43 and Figure 44. For these types of graphs, it would be possible to color in zones of approximate glazing performance corresponding to the different classes of glazing performance (such as single clear, single absorbing, double low-ε, etc) to represent the variation in performance between products from different glazing manufacturers caused by slight differences in optical properties.

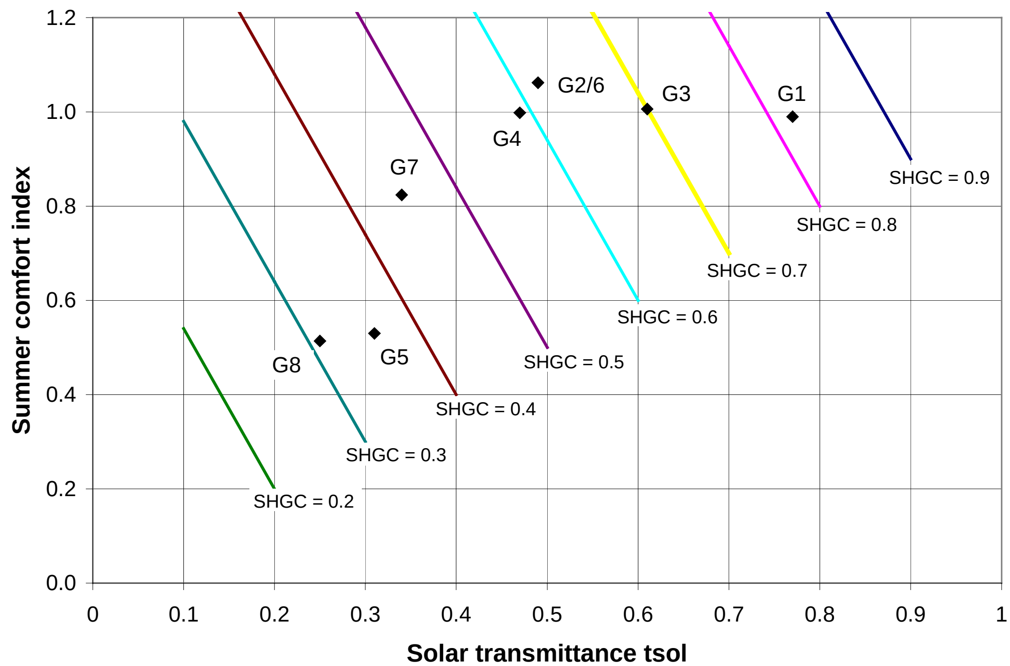


Figure 43: Diffuse radiation – Summer comfort index as a function of solar transmittance and SHGC

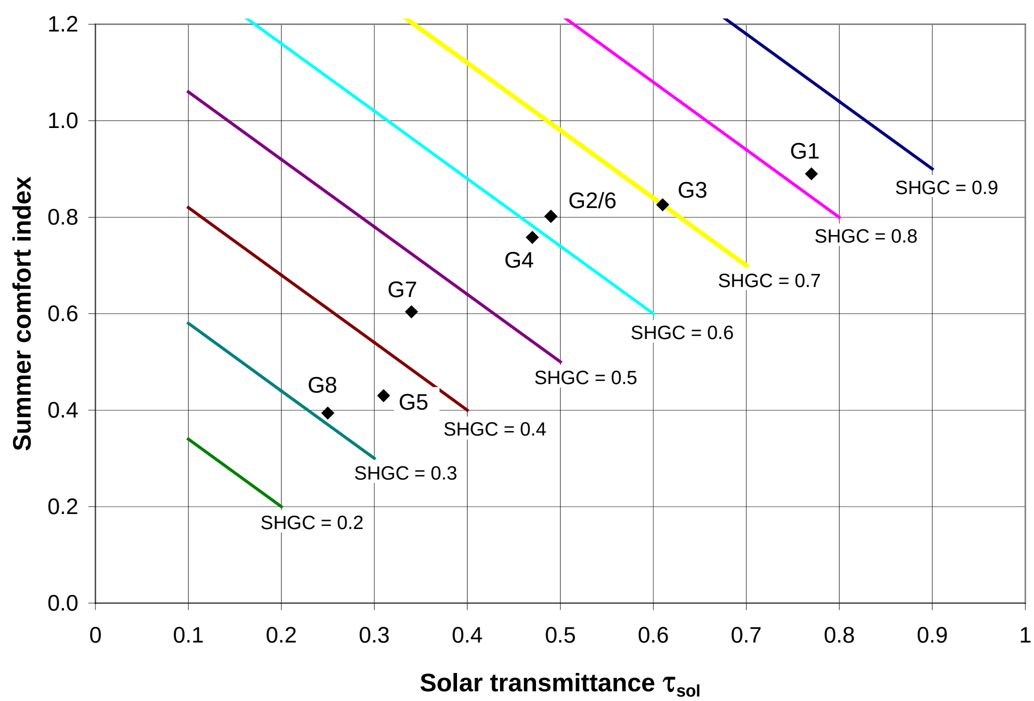


Figure 44: Direct radiation - Summer comfort index as a function of solar transmittance and SHGC

7 SUMMARY

In this report, we provided a thorough literature study on the impacts of windows on thermal comfort and more generally on comfort in asymmetrical environments, such as environments with cool or warm ceilings, walls, or floors.

In evaluating comfort in highly asymmetrical environments, such as the environment with cold or hot windows, local discomfort is very important. The UCB comfort model predicts comfort for 16 body parts, as well as for the whole body. The model is therefore well-suited to evaluate non-uniform thermal environments.

A single index for rating the thermal comfort performance of a window is not practical because the winter and summer performance are dependent on largely unrelated characteristics of the window. We therefore have identified several possible ratings for winter and summer and have recommended one of these for each season.

The winter rating is based on the window u-factor and the summer rating is based on a combination of solar transmittance and solar heat gain coefficient.

For winter, the minimum outdoor air temperature to ensure no discomfort happening for any body part is used as the index. This minimum allowable outdoor air temperature is directly related to the U-factor of the window, and a regression of the window comfort index with the U-value for winter condition is provided. The advantage of this index is that it provides guidance for designers when they choose windows for a specific climate.

In summer, however, the dominant factor is not the outdoor air temperature, but the solar radiation. The solar performance of glazing is determined by the optical properties of the glass, the solar transmission, reflection and absorption. Because the solar absorptance increases the surface temperature of the glass, the index is a combination of the transmittance and the absorptance of solar radiation, which is characterized by SHGC and Tsol (overall solar transmittance).

To illustrate how the methods we describe in this report could be applied to a comfort rating, we chose geometries with large view factors. This assumption will show larger differences between window products than if a geometry with a smaller view factor is used.

8 ANNOTATED BIBLIOGRAPHY

1) **Arens, E.A., Xu, T., Miura, K., Zhang, H., Fountain, M.E. and Bauman, F. 1998, "A Study of Occupant Cooling by Personally Controlled Air Movement". Building and Energy 27: 45 - 59.**

The authors (Center for the Environmental Design Research, UC Berkeley) did a laboratory study of thermal comfort under user-controlled air movement. 119 subjects participated. The results show that subjects considered air motion pleasant up to 1 m/s at 29.5ºC. Across the entire range of test conditions (24.5 – 31.5ºC), the chosen air velocity was in most cases above 0.4 m/s and up to 1.4 m/s, and very few people wanted less air movement. It is possible to make people comfortable by air motion at air temperatures 31ºC (1 met) or 29ºC (1.2 met). Based on the results, the authors proposed a "zone of likely use" for locally controlled air movement, which recommends air movement to provide comfort in warm environments.

2) **Bohm, M., Browen, A. et al. 1990, Evaluation of Vehicle Climate with a Thermal Manikin - The Relationship between Human Temperature Experience and Local Heat Loss, Swedish Institute of Agricultural Engineering.**

This report provides Equivalent Homogeneous Temperature (EHT) pistes for winter and summer conditions. The tests were conducted at the Swedish Institute of Agricultural Engineering, Ultuna, Uppsala. The pistes are widely used in automobile industry to evaluate thermal comfort in highly asymmetrical vehicle environments.

3) **Chapman, K.S., Sengupta, J. 2004, "Window Performance for Human Thermal Comfort". Final Report of ASHRAE Research Project RP-1162, Atlanta: ASHRAE Inc.**

This is a project funded by ASHRAE to evaluation window thermal comfort for human. The authors (Kansas State University) further developed the BCAP (Building Comfort Analysis Program, BCAP) model, which calculates the radiant heat exchange of the human body with his surrounding by calculating radiant intensity field. The current BCAP model links the properties obtained from the Window 5.1 program and includes the impact from window frames. The PMV, PPD and operative temperature are presented as contours.

The report explains the methods to calculate heat transfer of the room in great detail and provides calculation equations. It defines thermal comfort fenestration length as an evaluation parameter to evaluate window thermal comfort. The penetration length is defined as the distance into the room from the fenestration system, beyond which thermally comfortable conditions exist. At the end the authors provides 8 case studies to show the impact of fenestration system on comfort based on the simulation results. The authors proposed "Penetration depth" and "Fenestration performance map" to evaluate the window performance. Penetration depth is defined as the distance into the room from a window, beyond that thermal comfort condition exists. The fenestration map can be used as a design tool to determine the correct window wall ratio to ensure comfort at a defined distance from the window.

4) **Fanger, P.O. 1967, "Calculation of Thermal Comfort: Introduction of a Basic Comfort equation". ASHRAE Transactions 73 (2), 4.0 – 4.20.**

This paper provides a complete set of heat transfer equations to calculate the heat exchange between a human body and his surroundings.

5) **Fanger, P.O. 1973, Thermal Comfort. McGraw-Hill Book Co., New York, 244p.**

This book is based on the five-year research carried out at the Laboratory of Heating and Air-conditioning, Technical University of Denmark, and at the Institute for Environmental Research, Kansas State University, where the author worked for a period in year 1966 – 1967.

The book describes the thermal comfort evaluation indices, Predicted Mean Vote (PMV), and the Predicted Percentage of Dissatisfied (PPD) developed from a large number of human subject tests carried out by the author and other researchers..

The book offers heat transfer calculations between a human body and his surroundings, considers impacts on comfort from many aspects such as body builder, food, circadian rhythm, provides radiant view factors of a person within a room.

6) Fanger, P.O. and Christensen, N.K. 1986, "Perception of Draught in Ventilated Spaces". Ergonomics, 29, 215 - 235.

This study examined the perception of draft at air temperatures 20º, 23ºC and 26ºC, air movement from 0.05 to 0.4 m/s, under moderate and high levels of turbulent intensity. The results are presented as draft chart which predicts percentage of dissatisfied due to draft as a function of air velocity and air temperature. The data from this measurement, together with the data from another study by Fanger et al. (1988) were used to develop the draft model. The model was adopted by ASHRAE and ISO standards to define limits for air velocity and turbulent intensity.

7) Fanger, P.O., Banhidi, L., Olesen, B.W. and Langkilde, G. 1980, "Comfort Limits for Heated Ceilings". ASHRAE Trans., 86 (2), 141 – 156.

The purpose of the study was to determine the temperature limits for heated ceilings, to which people in thermal neutrality can be exposed without feeling discomfort. Researchers from the Laboratory of Heating and Air Conditioning, Technical University of Denmark conducted human subject tests using 8 male + 8 female college students. To eliminate discomfort caused by whole-body thermal imbalance,, neutral condition was maintained by asking subject's temperature preference and making changes of the chamber air temperature accordingly. During the first hour the ceiling was unheated. In the following five half-hour periods the subject was exposed to five different ceiling temperatures ranging from 34ºC to 69ºC (corresponding to a radiant temperature asymmetry from 4.5ºC to 23.6º). The thermal comfort questionnaires were presented in 5 minutes intervals.

If a subject reported twice out of the last three surveys that local discomfort happened, the subject was considered uncomfortable. Local discomfort was found either as a warm head or cool feet. In heated- ceiling tests, because the room air and the rest of the surface temperatures were low in order to balance the heated ceiling to keep subject neutral, the cool-feet discomfort happened more than warm head when ceiling temperature was between 34 – 52ºC (operative temperatures were same, around 24.1ºC). It was when ceiling temperature were between 63 – 69ºC that warm head discomfort happened more than cool feet discomfort. So in a way the limit for heated ceiling was based on cool feet discomfort. That is probably why the limit for cool ceiling (warmer the rest surfaces and air) is higher (14ºC from another paper by Fanger et al. 1985) than the heated ceiling (cooler the rest surfaces and air, 4ºC from this study) if we assume that people in general are more sensitive to cool feet than warm head. The heated ceiling tests cover the radiant temperature asymmetry from 4.5 – 23.6º, which in fact were all outside of the proposed limit, 4ºC for warm ceiling.

8) Fanger, P.O., Ipsen, B.M., Langkilde, G., Olesen, B.W., Christensen N.K. and Tanabe, S. 1985, "Comfort Limits for Asymmetric Thermal Radiation". Energy and Buildings, 8, 225 – 236.

The purpose of the study is to define limits for cool ceiling, hot/cool walls to which man in neutrality can be exposed without discomfort. The researchers from the Laboratory of Heating and Air Conditioning, Technical University of Denmark conducted the study.

A similar method as the study "Comfort Limits for Heated Ceilings" (Fanger et al. 1980) was applied. 16(F) + 16(M) college subjects participated in cool wall test, and 8 (F) + 8 (M) college subjects in warm wall and cool ceiling tests. The subjects first stayed in the chamber for an hour. Then his environment was changed 5 times in the following 2.5 hours to corresponding to 5 radiation asymmetries (cool wall: 17.8ºC to – 0.4ºC, warm wall: 32.6ºC to 70.1ºC, cool ceiling: 16.0ºC to – 0.8ºC). The room air and the rest of the surface temperatures were adjusted based on subject's requirement to keep an overall neutral feeling. The votes were asked every 5 minutes and only the last three (votes on 20, 25, 30 minutes) were used.

The limits are presented as the radiant asymmetry temperature, for cool ceiling, 14ºC, cool wall, 10ºC, warm ceiling 23ºC. Because people seem more sensitive to cool wall than the warm wall, the authors stated that local cooling of the body is more frequently causing discomfort than local heating. Comparing with the larger allowed warm/cool wall then ceiling, the authors concluded that men seem more sensitive to radiation asymmetry from head to feet than from left to right.

The results from this paper and the limit from the heated ceiling are adopted by ASHRAE Standard 55 – 2004 and ISO 3370.

9) Fanger, P.O., Melikov, A.K., Hanzawa, H.and Ring, J. 1988, "Air Turbulence and Sensation of Draught". Energy and Buildings, 12, 21 – 29.

This research was conducted at the Laboratory of Heating and Air Conditioning, Technical University of Denmark. The purpose is to investigate the sensation of draught caused by the air turbulent. Fifty subjects (mostly college students, 25 females and 25 males) were exposed to three different turbulent intensity levels (low turbulence Tu<12%, medium turbulence 20%<Tu<35%, high turbulence >55%) and four air velocities (from 0.05 m/s to 0.4 m/s). The room air temperature was kept at a constant value, 23ºC.

The subjects stayed for an hour during which the subjects were encouraged to change clothing to keep them neutral. Then the air velocity changed every 15 minutes in the following one and half hours to present 6 velocity levels (the participants became cooler as the velocity increased). The subjects answered questions every 5 minutes about whether he felt an air movement, whether it is uncomfortable, and where it was felt.

The study found that at the same air velocity and temperature, airflow with higher turbulence causes more people to feel the draught. Women seem slightly more sensitive to the turbulent intensity than men at lower velocity. A draft model to predict percentage dissatisfied using velocity, air temperature and turbulent intensity was proposed based on this study together with another study conducted at air temperatures 20ºC, 23ºC and 26ºC and under higher turbulence levels (Fanger and Christensen 1986).

Because the draft model is based on data under neutral test conditions (20ºC, 23ºC, and 26ºC for sedentary people), it should only be applied for sedentary people in neutral to cool environment.

The draft model is adopted by ASHRAE and ISO standards to define limits for air velocity and turbulence to eliminate draft.

10) Fanger, P.O. and Pedersen, C.J.K. 1977, "Discomfort due to Air Velocities in Spaces". Proc. of the Meeting of Commissions B1, B2, E1 of the IIR, 4, Belgrade, 289 - 296.

Ten college-age subjects (five females and five males) participated in this experiment to examine draft sensitivity to turbulent intensity. The ten subjects were chosen as the most draft-sensitive from one hundred subjects. The air was designed directly blown to the back of the subject's neck.

The author found that the fluctuation of the air flow is an important factor causing draft. At the same percentage of dissatisfaction, the draft limit for constant flow is much higher than the limit for a fluctuating velocity. The discomfort is maximum at frequencies around 0.3 – 0.5 Hz for air velocity at 0.3 m/s. People are less sensitive to the draft at the ankle than at the neck. When the author blow the air to the bare skin of the ankle, the draft limit is 50 – 100% increased.

11) Fountain, M. and Huizenga, C. 1995, A Thermal Sensation Model for Use by the Engineering Profession. Environmental Anylitics, Piedmont, California, U.S.A.

The authors (Environmental Analytics and Center for Environmental Design Research at UC Berkeley) developed a thermal comfort prediction tool for ASHRAE. In this menu, the authors summarized the thermal comfort evaluation methods, evaluated different comfort models, described the tool developed for this project, and presented the detailed equations to carry out the heat transfer and to predict thermal comfort.

12) Fountain, M.E. and Arens, E.A. 1993, "Air Movement and Thermal Comfort". ASHRAE Journal, August 26 – 30.

This paper reviewshistorical studies of air movement and comfort since the beginning of the 20th century, summarizes the findings. The authors (from Center for the Environmental Design Research, UC Berkeley) argue that the draft model was developed based on laboratory studies at the lower end of the comfort zone (23º). In the higher temperature range (above 23ºC) the draft model was obtained by extrapolations to conditions where data were not collected and where other research is in disagreement.

The authors also provide a figure which shows the preferred air movements at higher temperatures by several studies. The figure shows that the draft model and the ASHRAE standard 55-81 allows the significantly lower velocities than the other studies.

13) Fountain, M.E., Arens, E.A., de Dear, R., Bauman, F. and Miura, K. 1994, "Locally Controlled Air Movement Preference in Warm Isothermal Environments". ASHRAE Transactions 100: 937 - 952.

The objective of the study is to examine the air movement preference in warm environment. Fifty-four human subjects participated in this test (in Center for the Environmental Design Research , UC Berkeley). The room temperature was set at 25.5 – 28.5ºC. The subjects were asked to adjust the air movement from local supply systems to make them comfortable. The thermal questionnaire includes sensation, preference, air movement preference, acceptability, and local sensation.

By providing air motion in this temperature range, the subjects in only 8.8% of the tests (out of 158 tests) responded that their overall sensation was not between –1 and +1, or voted that the present air movement was unacceptable. That means that air motion can make 91% people comfortable up to air temperature 28.5ºC. However, the draft model would predict that 63.2% of these comfortable votes are "unacceptable". At the upper limit of the draft model (0.2 m/s at turbulent intensity 40%), 50% of the people wanted more air movement. The preferred air movement is significantly higher than the draft limit. When the room temperature was 28ºC, 50% people chose air velocity higher than 0.4m/s to make themselves comfortable. The preferred velocity went as high as 0.8 m/s.

The authors pointed out that the draft model is not designed to make a maximum number of people satisfied with the air movement, but rather to protect a small percentage (15%) from feeling draft discomfort. The authors developed a converse model forpredicting percentage of satisfied (PS model).

14) Gan, G. 2001, "Analysis of Mean Radiant Temperature and Thermal Comfort". Building Serv. Eng. Research Technology 22.2: 95 – 101.

The author (Institute of Building Technology, School of the Built Environment, University of Nottingham, UK) examined a series of comfort impacts of windows by simulating more than 20 window design cases. The variables in these 20 cases include the size of the window, shape of the window (height vs. width), type of the glazing, and the effect of a radiator. The author found that for the same size glazing, a tall and a narrow window is preferable than a square window. Replacing a large window with several smaller windows improves comfort. Double-glazing and a radiator installed under a window improve comfort.

15) Ge, H. and Fazio, P. 2004, "Experimental Investigation of Cold Draft Induced by Two Different Types of Glazing Panels in Metal Curtain Walls". Building and Environment 39, 115 – 125.

The authors (Building Envelope Performance Laboratory, Center for Building Studies, Department of Building, Civil and Environmental Engineering, Concordia University, Montreal, Quebec, Canada) measured velocity and temperature profiles with large tall glass panels. Near the window, the cold window-induced air motion could be as large as 1 m/s. When the location was 1.2 m away from the window, the velocity reduced to 0.15 m/s. The air temperature 1.2 m away from the window was about 0.8ºC lower than the room air temperature. The velocity and temperature also vary with the height, so not only radiation, but temperature and velocity are also not uniform. The authors also found that the empirical equations developed by Heiselberg (1994) provide close estimations of the measured data for cases having projecting window frames.

16) **Griffinths, I.S. and McIntyre, D.A. 1974, "Subjective Response to Overhead Thermal Radiation". Human Factors, 16 (3), 415 – 422.**
The authors (Electricity Council Research Centre, Capenhurst, Chester, UK) exposed 24 subjects to a neutral condition and three different ceiling temperatures. The method is similar to McNall and Biddison (1970) where the air temperature was kept constant and the temperature of the rest of the surfaces was adjusted to balance the temperature of the heated ceiling to keep the same MRT. They found that a vector radiant temperature of 20ºC did not produce any significant worsening of subjective responses when compared with a uniform control condition, and therefore a v.r.t of 20ºC was recommended. This finding is similar to what was found by McNall and Biddison (1970) when ceiling temperature was 54ºC corresponding to v,r,t = 24ºC, but significantly higher than the results from Fanger's (1980) where the radiant temperature asymmetry limit for heated ceiling is 4ºC.

17) **Heiselberg, P., Overby, H. and Bjorn, E. 1995, "Energy-efficient Measures to Avoid Downdraft from Large Glazed Façade". ASHRAE Transactions 101 (2), 1127 - 1135.**
To counter-react to the downdraft caused by cold glass, the conventional way is to provide convectors placed down to the façade, but that increases the energy use. This study (Department of Building Technology and Structural Engineering, Aalborg University, DK-9000 Aalborg, Denmark) examined effect from obstacles on the downdraft and found that with turbulent flow and an obstacle larger than the boundary layer thickness, the flow separated from the surface and a new boundary layer was established below the obstacle. The risk of thermal discomfort due to downdraft was reduced significantly.

18) **Heiselberg, P 1994, "Draught Risk from Cold Vertical Surfaces". Building and Environment, 29: 297 – 301.**
In this paper, the author (Department of Building Technology and Structural Engineering, Aalborg University, DK-9000 Aalborg, Denmark) examined the temperature and velocity profiles along the floor which is cause by vertical cold surfaces and penetrates into the occupied zone, and provided the prediction equations. These equations are widely used since they were proposed. The percentage of dissatisfied occupants was also examined. It reduces rapidly after 2 m away from the cold window because of the reduction in maximum air velocity.

19) **Huizenga, C., Zhang, H., Arens, E. and Wang, D. 2004, "Skin and Core Temperature Responses in Uniform and Non-Uniform, Steady-State and Transient Thermal Environments". Journal of Thermal Biology, 29, 549 – 558. Accepted by The 1st Symposium on Physiology and Pharmacology of Thermal Biology and Temperature Regulation, Rhodes Greece, October, 2004.**
This paper describes a large set of human subject tests carried out at the Center for the Built Environment at UC Berkeley. The tests were designed to investigate and develop models to predict human thermal comfort responses under transient and asymmetrical environment. This paper gives an overall description of the experiment set up, together with the skin and core temperature responses of people during the tests.

20) **Huizenga, C., Zhang, H., Arens, E. and Duan, T. 2001, "A Model of Human Physiology and Comfort for Assessing Complex Thermal Environments". Building and Environment 36(6): 691 - 699.**
This paper describes the advanced thermal physiology model that was developed at Center for Environmental Design Research, UC Berkeley. The model is based on the Stolwijk multi-node model, but has included significant changes - considering blood counter-current heat exchange, adding details view factor to calculate the radiative heat transfer of the human body with his surrounding, incorporating the solar radiation load on people. It divides the human body into 16 parts, each part is divided into core, muscle, fat, and skin layers. The model is able to calculate human thermal responses in transient and non-uniform environments.

21) Jones, B.W., Hsieh, K. and Hashinaga, M. 1986, "The Effect of Air Velocity on Thermal Comfort at Moderate Activity Levels". ASHRAE Transaction, 92(2B), 761 – 769.

The study examined the effect by air motion on people with higher metabolic level. The tests were for working people (2.3 metabolic level) with 0.65 clo and 1.09 clo, with velocity 0.2 m/s and 1.2 m/s, temperature over a range. The results show that comfort can be good with high air velocity.

22) Loveday, D.L., Parsons, K.C., Taki, A.H. and Hodder, S.G. 2002, "Displacement Ventilation Environments with Chilled Ceiling: Thermal Comfort Design within the Context of the BS EN ISO 7730 versus Adaptive Debate". Energy and Buildings, 34: 573 – 579.

Displacement ventilation saves energy. However, its ability to balance the heat load is limited. A common method is to apply a chilled ceiling. This introduces a more complex thermal environment. The authors, from Loughborough University and De Montfort University in UK, examined whether the ISO standard is applicable under this condition.

Total 184 subjects participated the tests in a chamber set up with the chilled ceiling/displacement ventilation. The ceiling temperatures were 14, 16, 18, and 21ºC, the supply air temperature from the displacement ventilation was 19ºC. The air supply flow rates were at 2.5, 3.9, 6.0, and 8.0 changes/hour. The subjects stayed in the test chamber for 3 hours to carry out the office work (1.2 met). The PMV predictions and the actual votes were close, so the author concluded that the PMV calculation method can be used in such complex thermal environment.

23) Lyons, P.R.A., Arasteh, D. and Huizenga, C. 1999, "Window Performance for Human Thermal Comfort". ASHRAE Transactions 73 (2), 4.0 – 4.20.

In this study, the authors (Windows and Day Lighting Group of Lawrence Berkeley National Laboratory and Center for Environmental Design Research at UC Berkeley) evaluate the window performance for 10 different window systems by simulation. The simulation results show that except in the situation where the human body is in the direct sun, the long-wave radiation is most important in determining thermal comfort. The impact of draft on comfort for residential size windows is small.

24) Manz, H. and Frank, T. 2004, "Analysis of Thermal Comfort near Cold Vertical Surfaces by Means of Computational Fluid Dynamics". Indoor Built Environment, 13: 233 – 242.

By CFD simulation, the authors (Swiss Federal Laboratories for Materials testing and research (EMPA), Duebendorf) showed that Heiselberg's empirical equations predicting maximum air speed near the floor due to cold windows works well in an empty room. However, with the presencet of internal heat gains, the predicted air velocity is smaller. The reason that the air speed increasedin a room with internal heat load is that the momentum of the buoyancy flows above heat sources boosts the motion of the downward flow near the cold wall. The author provided the modification to the Heiselberg's equations for situations with internal heat load. The simulation results also show that by improving the U-value of the windows, heating devices may not be necessary.

25) McIntyre, D.A. 1980, Indoor Climate. Applied Science Publishers LTD., London, 443p.

The author (Electricity Council Research Centre, Capenhurst, Chester, UK) provides knowledge basically covering all the areas that an indoor thermal environment and air quality study needs to deal with. It includes fundamental topics such as heat transfer, thermoreceptors and thermal sensation, and thermoregulation, to the summary of different indices used to describe thermal comfort, the comparison of laboratory study vs. field measurements.

26) McIntyre, D.A. 1977, "Sensation and Discomfort Associated with Overhead Thermal Radiation". Ergonomics, 20 (3): 287 – 296.

McIntyre (1977) did a following study (after Grinfinths and McIntyre 1974) to study heated ceiling on comfort. In this study, the author increases the ceiling temperature, but reduces both the

air temperature and the temperatures of the rest of the test chamber surfaces, in order to keep the same operative temperature, a similar approach to that of Fanger. He found not only thatthe heated ceiling produced no discomfort, but indeed was preferred when the ceiling temperature was high and the air and other surfaces were cooler. However, in the test, when subjects were invited to attribute discomfort specifically to the heated ceiling, the answers show a significant increase of discomfort with ceiling temperature, a contradictory answer from the pleasantness and discomfort questions in the test. It appears that people are ready to attribute discomfort to unusual aspects of the environments. For that reason, and because people clearly noticed the asymmetry radiation at v.r.t 10º, although no one was actually more uncomfortable than in the uniform environment, McIntyre recommended a v.r.t of 10ºC as the limit, which is also similar to the recommendations by Schroder and Steek (1973), 9ºC, and Banhidi (1972), a v.r.t between 8.5 – 13ºC.

27) McNall, Jr.P.E. and Biddison, R.E, 1970, "Thermal and Comfort Sensations of Sedentary Persons Exposed to Asymmetric Radiant Fields". ASHRAE Transactions, 76 (1), 123 – 136.

The effects of asymmetric radiation on thermal comfort was extensively investigated in Kansan State University in 1960s. In this study, the authors conducted a series of tests to examine the impact from warm and cool walls and ceilings covering large degree of radiation asymmetry. The cold wall temperature reached 9ºC (view factor 0.2) and it was 11ºC lower than the surface temperature of the rest. The warm wall temperature was fixed at 54ºC and it was 41ºC warmer than the rest. The cold ceiling reached 10.5ºC (view factor 0.12), 16ºC lower than the rest. The warm ceiling was kept at 54ºC and the difference from the rest was 38ºC. A control test was also conducted under neutral uniform environment (25.5ºC) in order to carry out the comparison.

Subjects exposed to warm wall had significant small satisfaction rate than the rate in other test conditions. When overall sensation was neutral, radiation asymmetry up to 9ºC wall, 10.5ºC ceiling, and 54ºC wall did not create significant discomfort, but did for the 54ºC warm wall. Under overall neutral condition, people with cool ceiling and cool wall received a 7% high percentage of satisfaction than under neutral condition.

The results are different from the results of Fanger et al. (1985) where the warm ceiling was perceived as the least tolerable while the warm wall had the largest allowable surface temperature. The major cause could be due to the different approaches. In Fanger's tests, both temperatures of the remaining surfaces and the air were lowered to balance the heated ceiling or wall. Therefore, a warm wall in a cool environment was perceived as acceptable and therefore has a larger allowable limit. In McNall's study, only the temperature of the rest surfaces was lowered to balance the heated ceiling or wall to keep the same MRT. The air temperature wasn't adjusted. As the results, the subjects experienced a stronger radiation than in Fanger's test condition. The discomfort could be caused by both warm or cool walls (we cannot tell because this information is not provided in the paper). The reason for a much smaller acceptable limit for heated ceiling in Fanger's study could also be caused by the experiment method. For ceiling temperature up to 52ºC (the rest of the room temperature was 22.3ºC), the discomfort from cool feet occurred more than from warm head. In normal temperature range people are more sensitive to cool feet than warm head. Therefore, the limit found was smaller, but it may not be specifically for the heated ceiling.

The two other differences are the scales to define discomfort and the test durations (please see the report).

28) Michaels, K.B., Nevins, R.G. and Feyerherm, A.M. 1964, "The Effect of Floor Surface Temperature on Comfort. Part II: College Age Females". ASHRAE Trans., 70, 37 – 43.

This test belongs to a series of tests conducted at the Kansas State University to determine the effect of floor surface temperature on foot and whole body thermal comfort. This paper focuses on the studies for college female subjects.

Eighteen female college students participated in the seated tests and eighteen participated in standing tests (with light work). The duration of the tests was 3 hours. The room air temperature was kept at 23.9ºC and floor temperature was changed from 23.9ºC to 37.8ºC. The floor temperature as high as 32.2ºC did not cause much discomfort for standing female. For seated, the upper limit was 29.4ºC.

29) Narita, C., Tanabe, S., Ozeki, Y. and Konishi, M. 2001, "Effects of spectral property of solar radiation on thermal sensation at back of hands". Moving Thermal Comfort Standards into the 21st Century Conference Proceedings, 393 – 400.

This paper showed that human skin reacts differently to different wavelength. The skin is more sensitive to visible (0.30 – 0.80 um) and middle-infrared (1.70 – 2.30 um) than near-infrared (0.80 – 1.35 um). The results are from tests which different wavelength radiation was applied on the back of hands. Twenty subjects participated in the test. It was conducted in the Waseda University and Asahi Glass Co, Ltd. in Japan.

30) Nevins, R.G. and Feyerherm, A.M. 1967, "Effect of Floor Surface Temperature on Comfort. Part IV: Cold Floors". ASHRAE Trans., 73, III.2.1 – III. 2.8

This test also belongs to a series of tests conducted at the Kansas State University to determine the effect of floor surface temperature on foot and whole body thermal comfort. This paper focuses on the studies about the cold floors.

Twenty-four male and twenty-four female college students participated in the 3 hour long seated test. The room air temperature was kept at 23.9ºC and floor temperature changes from 15.5ºC to 23.9ºC. The floor temperature as low as 15.5ºC did not cause serious discomfort for the subjects.

31) Nevins, R.G. and Flinner, A.O. 1958, "Effect of Heated Floor Temperature on Comfort". ASHVE Trans., 64, p. 175.

Starting early 1950s, a series of studies were undertaken at the Kansas State University to determine the effect of floor surface temperature on foot and whole body thermal comfort. This paper focuses on the effect of heated floor on comfort.

College students (108 male, 21 female) were exposed to different heated floors for 60 minutes. The results show that covering air temperature from 18.3ºC to 29.4ºC, the floor temperature range from 26.7ºC to 35ºC did not create significantly effect on sensation. The subject sensation became warmer as the floor temperature increased from 35ºC to 37.8ºC, although the actual value of vote indicated comfort. Therefore, the upper limit is considered as 35ºC.

32) Nevins, R.G., Michaels, K.B. and Feyerherm, A.M. 1964, "The Effect of Floor Surface Temperature on Comfort. Part I: College Age Males". ASHRAE Trans., 70, 29 – 36.

This test belongs to a series of tests conducted at the Kansas State University to determine the effect of floor surface temperature on foot and whole body thermal comfort. The previous study (Nevins and Flinner 1958) had subjects seat in an environment for 1 hour. Now the exposures continued for 3 hours for both seated and standing. This paper focuses on the studies for college male subjects.

Total 45 male college students participated in the tests (25 seated and 21 standing with light work). The room air temperature was kept at 23.9ºC and floor temperature changes from 23.9ºC to 37.8ºC. The floor high as 32.2ºC did not cause much discomfort.

33) Olesen, B.W. 1977, "Thermal Comfort Requirements for Floors". Proc. Of the meeting of commissions B1, B2, E1 of the IIR, Belgrade, 1977/4, 337 - 343.

The study was conducted at the Laboratory of Heating and Air Conditioning, Technical University of Denmark. The objective is to define the limits for floor temperatures. Eighty-five subjects participated in the test, keeping the feet (with shoes) on the floor for 3 hours. The impact from flooring material is insignificant. The optimal temperature of 25ºC for sedentary and 23ºC for standing or walking people are recommended. At floor temperature below 20 – 22ºC the percentage of people experiencing cold feet increases rapidly (Olesen 1975). For 10% dissatisfied, the floor temperature should be within 20 – 28ºC. The ASHRAE and ISO standards specify 19 – 29ºC floor temperature for 10% dissatisfaction based on this study.

34) Ruegg, T., Dorer, V. and Steinemann, U. 2001, "Must Cold Air Down Draughts Be Compensated When Using Highly Insulating Windows?" Energy and Buildings 33: 489 – 493.

The authors (EMPA Swiss Federal Laboratories for Materials Tsting and Research, Duebendorf CH, Switzerland and US Engineering, Wollerau CH, Switzerland) measured air velocity profiles along the floor next to a cold window. They compared the measured data with the predicted values from Heiselberg's equation and found that the equation provides satisfactory results, even for windows with a sill. The measured velocity profile shows that as the internal heat load increased, the boundary layer thickness increased but the peak velocity decreased. The author explained that the plumes from the heat load spread at the ceiling and circulated down, mixed together with the draft layer so the temperature of the layer was raised and the draft was reduced. They also tested effect of increasing the roughness on top of a windowsill which showed little influence reducing the draft. However, openings on a windowsill can significantly reduce the draft because they take up the down draft and release it again at a lower speed. It is possible to remove the heat compliance by increasing the window property as far as the window frame is well insulated.

35) Sengupta, J, Chapman, K.S. and Keshavarz, A. 2005a, "Window Performance for Human Thermal Comfort". ASHRAE Transactions, 111 (1).

The authors (Kansas State University) further developed the BCAP (Building Comfort Analysis Program, BCAP) model, which determines radiant heat exchange of the human body with his surrounding. The current BCAP model links the properties obtained from the Window 5.1 program and includes the impact from window frames. The PMV, PPD and operative temperature are presented as contours.

The authors examined window performance on comfort for 8 cases covering different glass areas and window configurations. In summer with solar radiation, the present of two windows (40% of the wall area) and one window (20% of the wall area) only provides 7% - 25% floor area that is comfortable.

36) Sengupta, J, Chapman, K.S. and Keshavarz, A. 2005b, "Development of A Methodology to Incorporate Fenestration Systems into Occupant Thermal Comfort Calculations". ASHRAE Transactions, 111 (1).

The authors (Kansas State University) described the further development of the BCAP (Building Comfort Analysis Program, BCAP) model. The authors proposed a parameter "penetration depth" to describe the window impact on comfort. The penetration depth is defined as the distance into the room from a window, beyond which thermally comfortable conditions exist.

37) Schutrum, L.F., Stewart, J.L and Nevins, R.G. 1968, "A Subjective Evaluation of Effects of Solar Radiation and Raradiation from Windows on the Thermal Comfort of Women. ASHRAE Transactions, 74 (2), 115 – 128.

The authors (Pittsburgh Plate Glass Co., Harmaville, Pa, and Kansas State University) tested subject thermal sensation with solar radiation transmitted through a window. The glass temperature was separately controlled. The results show that in cloudy day, when the glass temperature increased from 3ºC to 48ºC, overall sensation elevated 1.1 unit. In clear day, the window temperature increased to 31.7ºC and the overall sensation became 2.5 units warmer. In clear day, the glass temperature influence on overall sensation is small because solar radiation is the dominant factor.

38) Toftum, J. 2004, "Air Movement – Good or Bad?" Indoor Air 14: 40 – 45.

The author (from International Centre for Indoor Environment and Energy, Technical University of Denmark) reviewed a large number of literature regarding the impact of air movement on comfort provides a summary. With people feeling neutral or cooler (temperature up to 22ºC to 23ºC, at sedentary activity), there is a risk of draft, even at low velocities. With occupants feeling warmer than neutral (air temperature above 23ºC or at higher activity level), people normally do not feel draft in normal indoor air velocities (up to 0.4 m/s). At high temperature around 30ºC, air

velocity up to 1.6 m/s was fond to be acceptable, although such high velocity may not be desirable for other reasons.

39) **Toftum, J., Zhou, G., Melikov, A. 1997, "Effect of airflow direction on human perception of draught" Clima2000.**
Forty subjects, 20 women and 20 men, were exposed to airflows from five different directions: horizontally towards the front, the back, and the left side and vertically upwards and downwards, at three temperature levels 20, 23 and 26oC. The results showed that airflow direction has an impact on perceived discomfort due to draught. At 20oC and 23oC, airflow from below was perceived as most uncomfortable followed by airflows towards the back and front. At 26oC airflow from above and towards the back caused most dissatisfaction due to draught, but generally only a few of the subjects perceived discomfort at this temperature. The authors also recommended to take air direction into account when providing design guidelines for air movement.

40) **Window 5.1 2001, User Manual, A PC Program for Analyzing Windows Thermal Performance, LBNL – 44789.**
WINDOW 5.2 is a publicly available computer program for calculating total window thermal performance indices (i.e. U-values, solar heat gain coefficients, shading coefficients, and visible transmittances). WINDOW 5.2 provides a versatile heat transfer analysis method consistent with the updated rating procedure developed by the National Fenestration Rating Council (NFRC) that is consistent with the ISO 15099 standard. The program can be used to design and develop new products, to assist educators in teaching heat transfer through windows, and to help public officials in developing building energy codes.

41) **Wyon, D. P., Larsson, S. 1989, "Standard Procedures for Assessing Vehicle Climate with a Thermal Manikin". SAE Technical Paper Series 890049: 1-11.**
David Wyon (Human Criteria Laboratory, National Swedish Institute for Building Research, Gavle, Sweden) proposed a method to evaluate asymmetrical thermal environment – Equivalent Homogeneous Temperature (EHT). The EHT is the temperature of a reference environment where the heat loss for a body part is the same as in the real asymmetrical environment. It is measured by a thermal manikin which shows the dry heat loss and skin temperature for individual body parts. The acceptable EHT range for all body parts are presented as a "piste". The EHT method is widely used in automobile industry to evaluate thermal comfort in asymmetrical environment. The piste only corresponds to the levels of the clothing and metabolic level that tested. It only defines the acceptable temperature range for each individual body part, does not tell local body thermal sensation and does not integrate local body information into the whole body thermal status.

42) **Zhang, H., Huizenga, C., Arens, E., and Wang, D. 2004, "Thermal Sensation and Comfort in Transient Non-Uniform Thermal Environments". European Journal of Applied Physiology, Vol. 92, 728 – 733. Also presented in 5th International Meeting on Thermal Manikin and Modeling, Strasbourg France, September, 2003.**
This paper summarizes the human thermal comfort prediction models developed from the human subject tests carried out at the Center for the Built Environment at UC Berkeley. An overall description of the test was presented by Huizenga et al. (2004). The models predict sensation and comfort under transient and asymmetrical environment. The models have been incorporated into the physiology model developed by the group (Huizenga et al. 2001) so the model now is able to predict the human comfort under complex thermal environments.

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