Thermal comfort in high mean radiant temperature environments

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ABSTRACT

This paper presents a design method to define the settings of HVAC systems in order to provide thermal comfort in high MRT environment in hot climates. The method, firstly, discuss the use of simplified thermal load calculation methods, in face of the specificities of the theme. Then, dynamic heat, air and moisture envelope simulations are performed in order to define the surfaces' internal temperatures. The distribution system, including terminal sizing and positioning, flow rate and temperature, is defined based on CFD simulations. A thermo-physiological model, including short wave radiation, is considered to evaluate the thermal comfort conditions under the sun and in the shadow, during summer and winter days. A commercial single store glazed building in São Paulo, Brazil, is used as a case study.

1. INTRODUCTION

Commercial single store glazed buildings are common in tropical cities, beside its obvious inadequacy to this climate. It happens because the owners demand buildings where the exposed goods can be clearly visualized from the street and side walk. The building will be constructed in São Paulo, Brazil, and it was used as test case for the methodology presented in this paper. The calculated thermal load of this building kind use to be high, since in most of the cases simplified steady-state methods are applied. Those methods assume a uniform air temperature, about 24°C, and neglect complex radiation effect present in those unusually glazed building. The thermal comfort provided for the systems sized that way use to be poor, due to draught on jet areas, as well as the high mean radiant temperature. This scenario is far from the prescribed in ISO 7730 (1994) and ASHRAE 55 (2004).

2. METHODS

Computational fluid dynamics simulations and transient thermal building performance simulations are applied to the studied case, using the thermal comfort level as performance indicator, since it is the aim of air conditioning systems in hot climates. In this context, the concept of thermal comfort has to be revised, in order to provide a useful tool on the assessment of building performance.

The first point is the short term occupancy, since the customers spend usually little time inside the building. In this sense, the thermal comfort tolerance is different of the long term exposure of a working station, describe in the international standards. The second point is the transport mean used by the customers to go to those buildings: mainly by car, exposed to conditions similar to the building interior but highly adapted to his wishes. Based on this, it is assumed that thermal balanced environment is satisfactory, where the customers don't sweat. The third aspect is the variety of shading rates encountered in the city, when compared to the very static indoor environment. The internal environment should provide a similar variety of conditions, in a range assuring the thermal balance environment. The mean clothing insulation is assumed to be higher than the usual in this climate, due to the customer's profile. Concerning the workers, the dress code is adapted to the insulation assumed for the customers, and the work station was placed sheltered from the direct sun light.

Based on this design concept, the first stage in the proposed methodology is to revise the designed glazed building envelope, in order to provide the variability in the building interior spaces, aiming a reduction on the MRT. A set of building envelope simulations with different facades modifications was executed using the commercial package TAS (2004), using the MRT as sensitivity parameter. In the study case, a $500m^2$ car shop, with a suspended volume for administration offices, part of the roof and facade glazing panels were substituted for opaque panels. The proposed modification is incorporated in the architectural design, and assures that 50% of the interior is shaded, like a tree shadow pattern. The result is presented in Figure 1. The use of interior or exterior screens, as well as highly reflexive glasses could provide similar results. The building external area is also optimized by the use of dark tiles, reducing the incidence of reflected sun radiation in the glazed facade. Green low-e glazing is applied in facades less exposed to the streets and side walks, and a reflective glazing is applied in the ceiling. This configuration preserves the transparent glazing in the main facades, and the high light permeability of the building envelope, but it reduces the building solar gains and the short-wave radiation component in the user thermal balance inside the building. Beside the improvements in the building envelope, the peak MRT is about 45°C.

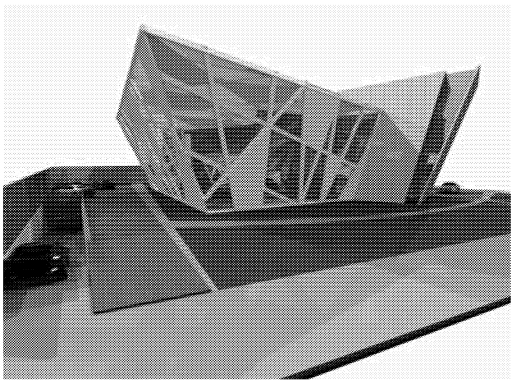

Figure 1: Case-study: glazed building with shading devices

In the second design stage, the air supply system and set-point were studied by steady-state CFD simulations using the commercial package CFX (2003), based on the critical hour concerning MRT. For the final solution, other simulations for typical summer and winter days were also performed. In 15 simulations, two air supply configurations were tests: diffusers at 2,3 meters high (total of 3,5 m^2 air supply), and a displacement system, with 9 air supplies of $1m^2$ each positioned under the cars. Different temperature and velocity set-point were tested, and the results were analyzed regarding the uniformity of the temperature-velocity field in the occupied zone (0 to 1,8 meters high).

The standard k-e turbulence model was used, with scalable wall functions and a second order advection scheme. Buoyancy effects were calculated using Boussinesq approximation. The convergence criteria adopted was 10^{-4} in all equations, and a mesh independence result was obtained after 3 successive refinements (about 1 million cells). The y+ values are between 30 and 100 for most of the solid boundaries. The building surfaces temperatures were obtained in the building envelope simulation, at 13:00 pm. The values found respectively for winter and summer conditions are: roof 38/50°C; external glazing facades 26/37°C; floor 21/33°C, cars 30/40°C. The other surfaces were considered adiabatic. Figure 2 shows an external view of the simulated volume, and also the surface of the unstructured mesh used. The suspend office volume was omitted in the CFD simulation, because it has an independent air supply system. The thermal comfort analyses were carried out with the air velocity and temperature provided by the CFD simulation, and the MRT calculated for several point in the building interior, based on the surface temperatures provided by the building envelope simulation.

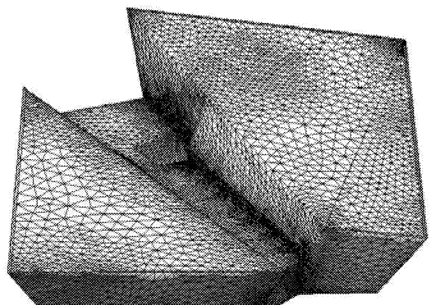

Figure 2: External view of the simulated volume, with the unstructured surface mesh.

3. CFD RESULTS

Figure 3 shows a section of one simulation with the 2,3 m height air supplies, using an inlet velocity of 1 m/s and a 14°C temperature. In all simulated set-point, the presence of high speed (0,6m/s) was evident in the jet region. The comfort analyses will show this solution is unviable.

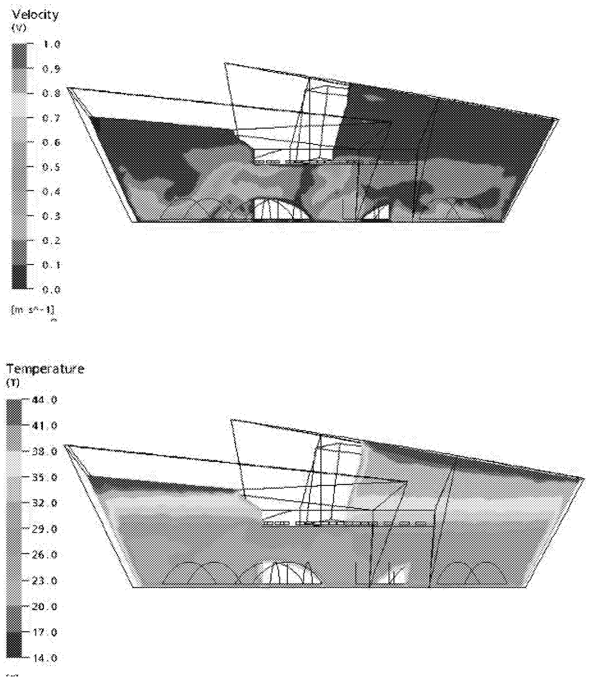

Figure 3: Velocity and temperature field in a representative section (“ceiling” air supply)

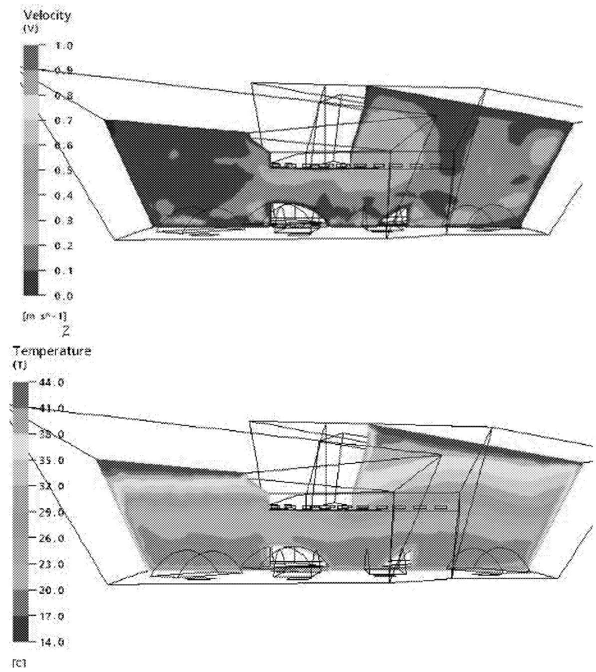

Figure 4: Velocity and temperature field in a representative section (displacement ventilation)

Figure 4 shows the results for the best displacement ventilation setting, with 1m/s inlet velocity, at 17°C. The temperature-velocity field obtained is very uniform, as expected in a displacement ventilation system. The velocities vary in a range between 0 and 0,3m/s, while the temperatures vary between 18°C and 20C. The cars shelter the users from the high air speed used in the inlet, allowing the use of the very high ventilation rates prescribed for this methodology. The ventilation is made using a plenum in the basement, and the grills are metal grades that can support the cars weight and impose minor pressure losses in the system. As expected, the volume near the 7m height ceiling is quite hot, but it doesn’t affect the thermal comfort in the occupied area, and saves energy just conditioning the occupied zone. The temperatures near the facades are also high, but it clearly doesn’t represent a thermal comfort issue, because those areas are seldom used. Figure 5 shows a temperature contour plot in a representative section for winter conditions. The same uniform pattern was achieved using a higher temperature set-point (19°C) and a lower inlet velocity (0,5m/s). The comfort analysis will show good results for this settings.

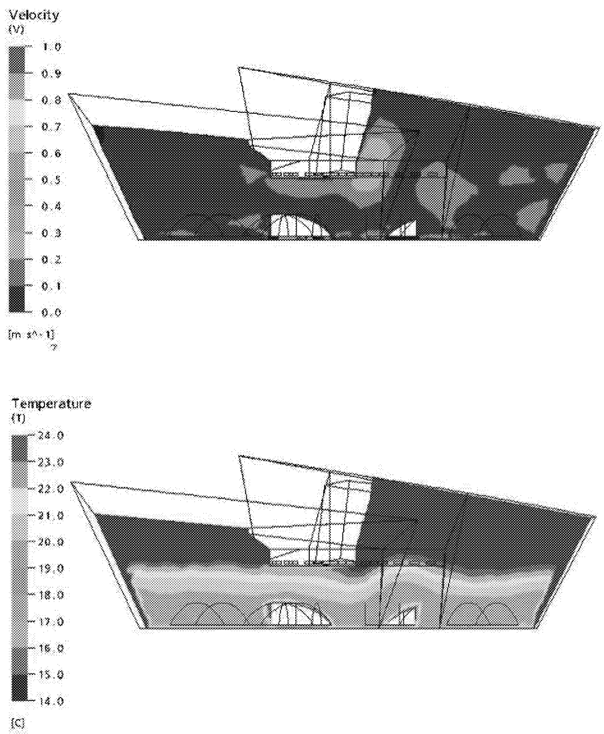

Figure 5: Velocity and temperature field, using displacement ventilation, during winter.

Based on the temperature difference between the imposed inlet and the calculated outlet, and the imposed flow rate, the peak sensible thermal load was calculated in 120kW. The result is half of the obtained using steady-state methods, and is in agreement with the dynamic building envelope simulation.

4. THERMAL COMFORT

The adopted model is based on the thermal physiological balance between the human body and the environment, considering: individual characteristics (activity, position, clothing), environmental characteristics (albedo and emissivity of surfaces) and local climate characteristics (direct, diffuse and reflected solar radiation, air temperature, humidity, vapor pressure, atmospheric pressure and air speed). The equation that expresses the balance is:

$$S = M + C + L + R + E + Res$$

where: S: heat storage in the body, M: metabolic rate; C: heat exchange on the skin by convection; L: heat exchange by long wave radiation; R: heat gain by short wave radiation; E: heat loss by evaporation at skin surface; Res: respiration heat loss.

According to Blazejczyk (2002) and Berdhal (2001), long wave radiation and emissivity of sky can be calculated by the equations.

$$L = Irc*(0.5*Lg+0.5*La - (\varepsilon p*\sigma*(Tp\hat{}4)))$$

$$Lg = \varepsilon piso*\sigma*(Tgr\hat{}4)$$

$$La= \varepsilon sky*\sigma[(Tar\hat{}4)*(0.82-0.25*10\hat{}\hat{}(-0.094*vp))]$$

$$Irc= hc'/(hc'+4*\varepsilon p*\sigma*Tar\hat{}3)$$

$$hc'= (0.013*Pat-0.04*tar-0.503)*0.53/[Icl**(1-0.27*Vrel\hat{}0.4)]$$

$$Vrel = Var + 0.0052*(M-58)$$

$$\varepsilon sky= \varepsilon csky *(1+0,0224*n-0,0035*n\hat{}2 + +0,00028*n\hat{}3)$$

$$\varepsilon csky = 0,711 + 0,56*(Tdp/100) +0,73* *((Tdp/100)\hat{}2)$$

$$Tdp = C3*(Ln(RH)+C1)/\{C2-(Ln(RH) +C1)\}$$

where: L: long wave radition exchange between the body the ground (Lg) and the atmosphere (La); Irc: clothing insulation reduction coefficient; tar: air temperature (°C), Tar: absolute air temperature; Tp: skin absolute temperature (K); Tgr: ground absolute temperature (K); vp: vapor pressure (hPa); Pat: atmospheric pressure (hPa); Vrel: relative air speed (m/s); Var: wind speed (m/s);σ: Boltzman constant (5.67*10^(-8)); εpiso: ground emissivity (0,97); εp: skin emissivity (0.97); εsky: sky emissivity; Icl: clothing insulation, (ISO 8996, 1990); n: nebulosity (n=0 to clean sky and n=1 to completely cloudy sky); εcsky: emissivity of clean sky; Tdp: absolute dew point (K); C1= C2*Tar/(C3 +Tar); C2= 17,08085; C3= 234,175; RH= relative humidity (0-1).

For the short wave radiation absorbed by the body, Blazejczyk (2004), based on empirical data, suggest different methods, according to the solar radiation data available. Having the global solar radiation data, the sun height and the nebulosity, the thermal gain by short wave radiation can be estimated by:

$$\text{If } h\leq 10 \quad R=1.4*Rad\,(0.546-0.224*\ln(h))* *(1-0.01*alb)*Irc$$

$$\text{If } h>10 \quad R=1.4*Rad*(0.04+5.166/h)* *(1-0.01*alb)*Irc$$

where: alb: albedo of clothing (dimensionless); h: sun height (degrees); Rad: global solar radiation (W/m2).

Kuwabara (2004) proposes the following equations for the consideration of long wave (L) and short wave (R) thermal radiation exchanges between the body and the surroundings.

$$Q=R+L= L-(Rdir +Rrefl +Rdif)$$

$$L= La+Lg+Lt$$

$$La=\varepsilon p*\sigma*((Tp^4)-(Tsky^4))*fref*fcl*FFuc$$

$$Lg= \varepsilon p*\sigma*((Tp^4)-(Tgr^4))*fref*fcl*FFup$$

$$Lt= \varepsilon p*\sigma*((Tp^4)-(Tt^4))*fref*fcl*FFue$$

$fcl = 1+0.31*Icl$ (ISO 9920,1995)

$$TER=(FFuc*Tsky^4+FFupTgr^4+FFut*Tt^4+ +((Rdir+Rdif+Rrefl)/(\varepsilon p\sigma*fref*fcl)))^{0.25}$$

$$Rdir = \alpha*fp*fref*fcl*R$$

$$Rdif = \alpha*FFuc*fref*fcl*R$$

$$Rrefl = \alpha*FFut*fref*fcl*\rho*R$$

$$Tsky=Tar*[(0,8 + ((Tdp-273)/250)))\hat{}0,25$$

where: fcl: clothing factor; fref: radiant effective area of the body (0,71); fp: projected area factor; FFuc: form factor between the body and the sky; FFut: form factor between the body and the built surroundings; TER: effective radiant temperature; Tt: absolute temperature of the surrounding surfaces.

The projected area factor (fp), according to Fanger (1972), expresses the portion of the body that receives the direct solar radiation. It is function of body position (sitting /standing) and sun position (height and azimuth in relation to the body). Considering the critical situation, in which the azimuth between the sun and the body is zero, it is possible to estimate, for standing position, the value of "fp" using the following equation. In the case of sitting position, "fp" can be estimated by: if $h > 45°$, $fp = 0,22$; and if $h \leq 45°$, $fp = 0,3$.

The form factor FF expresses the geometrical relation of the thermal radiant exchange between the body and the surroundings. According to ISO 7726 (1998) FF can be determined bt the following equation.

$$fp = -0,0037h + 0,4153$$

$$FF = Fmax*(1-e\hat{}(-((a/c)/T)))*(1-exp(-(b/c)/G))$$

where: $T=A+B*(a/c)$ and $G=C+D*(b/c)-E*(a/c)$. The indexes a, b and c can be determined using Fanger (1972).

Table 1 presents the other necessary values for calculating the form factor, considering different situations regarding the person (sitting or standing) and the given surface (vertical or horizontal).

Table 1 – Values for calculating FF, according to ISO 7726(1998)

Situation	Fmax	A	B	C	D	E
Sitting/ Vertical surface	0,118	1,216	0,169	0,717	0,087	0,052
Sitting/ Horizontal surface	0,116	1,396	0,130	0,951	0,080	0,055
Standing/ Vertical surface	0,120	1,242	0,167	0,616	0,082	0,051
Standing/ Horizontal surface	0,116	1,595	0,128	1,226	0,046	0,044

In order to consider the superficial temperatures of the envelope, the data from the transient thermal building performance simulations were used, which were the same data used for the dynamic simulations, respectively for winter and summer conditions: roof 38/50°C; external glazing facades 26/37°C; floor 21/33°C, cars 30/40°C. The consequent mean radiant temperature found is approximately 37ºC in the summer and 28ºC in the winter. A relative humidity of 50% was adopted, value commonly used in Brazilian air conditioned environments. The metabolic rate considered was one of $M=74W/m^2$, referring to a standing person with low activity, for instance, talking. Applying ASHRAE (2004), an 80% of thermal environment satisfaction was adopted. As a consequence of this assumptions, Table 2 and Table 3 show the air temperature results found for summer and winter, under the sun and in the shadow, for two different clothing ensembles (0,5 clo, typical clothes in Brazil; and 0,75 clo, executive suit used in Brazil), in function of different air velocities (0,1 and 0,2 m/s).

Table 2 – Air temperature results found for different air velocities (0,1-0,2m/s).

Summer	Icl (clo)	Ta max (ºC)	va (m/s)
Shadow	0,5	19,0	0,1
		21,0	0,2
	0,75	18,5	0,1
		20,0	0,2
Sun	0,5	14,0	0,1
		15,5	0,2
	0,75	13,0	0,1
		14,0	0,2

Table 3 – Air temperature results found for different air velocities (0,1-0,2m/s).

Winter	Icl (clo)	Ta max (ºC)	va (m/s)
Shadow	0,5	26,0	0,1
		27,5	0,2
	0,75	24,0	0,1
		25,5	0,2
Sun	0,5	18,0	0,1
		20,5	0,2
	0,75	16,5	0,1
		18,5	0,2

5. CONCLUSION

Considering the research done, one may conclude that the use of displacement ventilation, associated with high ventilation rates and low temperature set-points, in order to provide thermal comfort in high MRT environments in hot climates, even in glazed building with direct exposure to the sun, is viable.

On the other hand, one must admit that the architectural design of such buildings is not concerned with energy saving and sustainability issues. Nevertheless, the user requires a minimally healthy environment. As a consequence, although the best solution would be ideally a complete review of the architectural design concepts, in order to reach an architecture that is really developed to hot climates; this research provides a way of repairing, at least, the thermal discomfort of the users of such buildings that are climatically inappropriate.

REFERENCES

ISO. ISO 7730. (1994) Moderate thermal environments: determination of the PMV and PPD indices and specification of the conditions for thermal comfort. Genève: ISO.

ASHRAE. (2004) ANSI/ASHRAE Standard 55, Thermal Environmental Conditions for Human Occupancy, Atlanta: ASHRAE.

EDSL. (2004) TAS Theory. EDSL.

CFX. (2003) CFX 5 Solver Models. Diddcot Oxfordshire: CFX COST.

Blazejczyk, K. (2002) Menex. Warshaw: Inst. of Geography and Spatial Organization.

Santamouris, M. (2001) Energy and climate in the urban built environment. London: J&J.

Kuwabara et al. (2002) Evaluation of thermal sensation in urban environment.

Fanger, P.O. (1972) Thermal Comfort. New York: Mc Graw-Hill.

ISO. ISO 10551. (1995) Ergonomics of the thermal environment: assessment of the influence of the thermal environment using subjective judgment scales. Geneve: ISO.