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USE OF PMV CONTROL TO IMPROVE ENERGY EFFICIENCY IN COMFORT COOLING APPLICATIONS

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ABSTRACT

Traditionally HVAC systems are controlled based on the sensed air temperature. Human thermal comfort sensation is a result of a number of environmental factors and air temperature alone is not always an adequate parameter for controlling thermal comfort in a conditioned space.

The Predicted Mean Vote (PMV) is a commonly used tool for assessing human comfort rating. It takes into consideration six key factors: metabolic rate, clothing level, air temperature, radiant temperature, air speed and humidity. A PMV based control HVAC system has the potential to provide improved thermal comfort inside a space and to reduce energy consumption by taking into consideration all major thermal comfort variables. Computational Fluid Dynamics (CFD) is used to investigate the combined effect of air temperature, radiant temperature and air velocity on thermal comfort inside a conditioned office space during a hot summer day. An assessment and comparison of two control strategies, PMV based and air temperature based on thermal comfort and energy consumption has been undertaken using a simplified CFD model of office spaces.

NOMENCLATURE

CFD	*Computational Fluid Dynamics*
PMV	*Predicted Mean Vote*
EW	*External Wall*
AW	*Adiabatic Walls*
WHF	*Wall Heat Flux*
RPM	*Rotations per Minute*
HVAC	*Heating, Ventilation and Air Conditioning*
2D	*Two Dimensional*
AFL	*Above Floor Level*
ISO	*International Organisation for Standardization*

INTRODUCTION

Contemporary building design, modern architecture and innovative HVAC systems strategies often result in complex thermal conditions inside a building and effective HVAC control is always a challenge. Thermal discomfort in a typical modern office space is usually caused by high radiation intensity emitted from extensive glazing area, chilled or heated ceilings and floors, radiant HVAC elements and drafts related to ventilation strategies.

It is a standard design practise to set and control air temperature. Such control does not take into account radiative heat exchanged between the human body and surrounding surfaces and cannot respond to frequent changes in air velocity due to different supply air quantities, nor glass surface temperature changes due to exposure to direct solar radiation. Typically one air temperature is set and controlled during the entire cooling or heating season often resulting in poor thermal conditions inside the space and excessive energy use.

An innovative PMV based control system is analysed in this paper. The intention is to investigate the pros and cons of such a control system, to highlight effects of all relevant parameters, to provide the basis for methodology capable of improving thermal comfort inside a conditioned space and to investigate the potential for energy reduction.

MODEL DESCRIPTION

CFD analyses have been undertaken on two simplified office spaces of 14m^2 and 32m^2 with a ceiling height of 2.9m, window area of 4m^2 and external wall area of 7.6m^2. For model description please refer to Figure 1 and Figure 2.

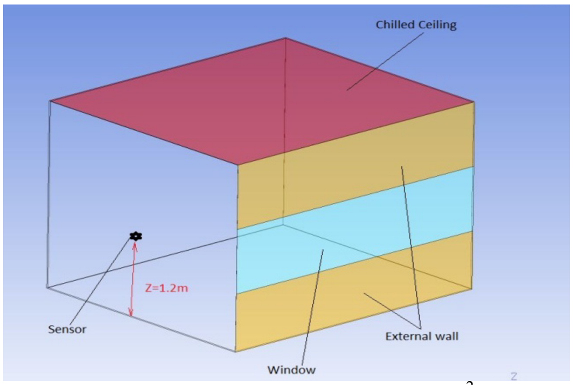

Figure 1: Room Schematic – Small Room 14 m^2

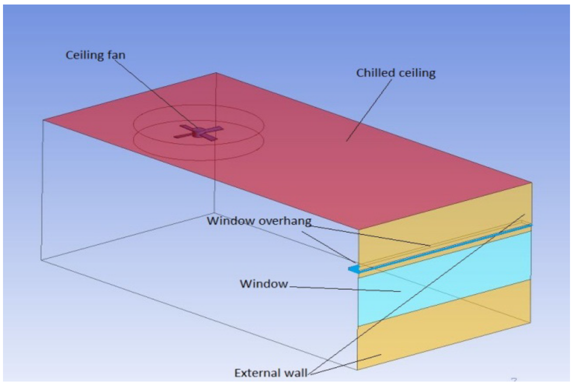

Figure 2: Room Schematic – Large Room 32 m^2

Mesh

From a meshing perspective the CFD analysis can be divided in to two different groups:

- Natural convection analysis, cases without ceiling fan, and
- Forced convection analysis, cases with ceiling fan

For each group an adequate mesh has been generated.

Natural Convection Analysis Mesh

A surface quad mesh of 0.016 m has been created along all room walls. A volume mesh with the first grid size of 0.004 m has been generated along all walls to allow for accurate resolution of surface temperatures, radiative flux and convective wall heat flux. Zhiqiang and Qingyan (2004) report that a first grid size at 0.005 m is reasonably good for the indoor natural convection and that results produced with a first grid cell between 0.002 – 0.005 m agree well with experimental data. A non-conformal hex mesh with a cell size of 0.04 m has been generated throughout the domain resulting in total of 1.5 million cells.

Forced Convection Analysis Mesh

Forced convection analysis does not require the same fine boundary layer resolution as required for natural convection. However a fine resolution mesh has been generated along surfaces with high temperature gradients and potential flow stagnation zones. The same mesh methodology as for the natural convection analysis with the first grid size of 0.004 m has been generated along the external wall, window and chilled ceiling surface. A mesh with the first grid size of 0.04 m has been generated along all internal walls and floor. These surfaces have been modelled as adiabatic surfaces and a small temperature gradient is expected in the boundary layer. Zhiqiang and Qingyan (2004) report that a first grid size at 0.1 m is good for most indoor forced convection flows and that reasonable solutions are provided with a first grid size at 0.05 - 0.1 m for a forced convection flow.

A 0.005 m surface mesh has been generated on the ceiling fan surfaces with a growth rate of 1.2. Hex and tetrahedral meshes with a cell size of 0.04 m have been generated throughout the domain resulting in a total of 2.7 million cells.

The κ- ω based Shear-Stress-Transport turbulence model with Automatic Near-Wall Treatment has been used in both cases.

Mesh Validation

To accurately resolve heat transfer and surface temperatures along the walls and to benefit from automatic near-wall treatment for the κ - ω turbulence model, a high wall resolution mesh with Y+ <= 1 is required (ANSYS CFD technical documentation). Such a generated mesh would not be practical for intensive analysis due to the very high memory requirement. For validation purposes three 2D extruded meshes representing the cross section through the middle of the domain, with 3 cells extrusion width have been generated and compared.

Mesh 1 has been generated to result with Y+ <=1 and at least 10 cells within the boundary layer as recommended (ANSYS CFD technical documentation). An expansion factor of 1.2 has been used for inflation. Mesh 1 has been used as a validation benchmark and reference mesh for assessment of the validity of Mesh 2 and Mesh 3. Mesh 2 has been generated using the same approach and grid size as used for the natural convection analysis mesh. Mesh 3 has been generated for comparison purposes only and it is a simple 0.03 m hex mesh generated throughout the entire domain and without any near wall fine resolution. All cases have been solved for the same boundary conditions and the results relevant for validation are presented below:

Y+ value			
	Mesh 1	Mesh 2	Mesh 3
First cell [mm]	0.30	4.0	30.0
Y+ glass	0.92	4.4	32.1
Y+ ceiling	0.60	2.4	29.8
Y+ EW	0.67	3.4	30.4
Y+ AW	0.12	2.6	35.9

Table 1: Y+ values

Surface Temperature			
	Mesh 1	Mesh 2	Mesh 3
Temperature glass [K]	307.2	306.0	303.2
Temperature ceiling [K]	291.6	292.8	300.8
Temperature EW [K]	303.1	302.6	303.1
Temperature AW [K]	299.3	299.5	301.9

Table 2: Surface temperatures

Wall Heat Flux			
	Mesh 1	Mesh 2	Mesh 3
WHF glass [W/m^2]	15.9	16.4	6.3
WHF ceiling [W/m^2]	-18.1	-18.0	-14.6
WHF EW [W/m^2]	2.5	2.6	1.8

Table 3: Wall Heat Flux

Mesh 1 has a maximum Y+ value of 1 at the top of the window vertical surface where high velocity of buoyancy driven flow is recorded. The average Y+ value along the window is 0.92. With the first grid size of 0.3mm and an expansion rate of 1.2, 10 hex cells are placed within an 8mm thick boundary layer along the wall. In the worst case, along the window vertical surface the Y+ value at 8 mm distance of the wall is Y+= 3.3. Based on this information it can be concluded that more than 10 cells are located within the viscous boundary layer and that Mesh 1 is capable of resolving low-Reynolds number flow. Due to the high memory requirement this mesh has not been used in further analysis but only as a validation benchmark.

Natural convection analyses have been undertaken using Mesh 2. Analysed parameters obtained with Mesh 2 are in close proximity to the parameters obtained using Mesh 1. Results obtained using Mesh 3 indicate high relative difference when compared to results obtained with Mesh 1 and such meshing approach has not been used in this analyses.

Boundary Conditions

Window and external Wall

A wall boundary condition has been used to model the window and external wall. The Heat Transfer Coefficient of 21.9 W/m^{2}K for glass and 0.5 W/m^{2}K for external wall has been specified together with outdoor air temperature. It includes external surface resistance and conduction through the solid material. The analyses have been done using outdoor temperatures of 35ºC and 24ºC to represent peak and mild summer outdoor conditions. Both the external wall and window are assumed to be shaded and not exposed to direct solar radiation.

Chilled Ceiling

A wall boundary condition has been used to model the chilled ceiling. A constant heat flux has been specified for this boundary and a final flux value has been balanced to reach the targeted condition at the sensor location.

Adiabatic Walls

An adiabatic wall boundary condition has been used for all internal walls, floor and ceiling fan walls.

Ceiling Fan Motion

A separate cylindrical fluid domain has been created around the ceiling fan. A rotating motion has been assigned to this domain and the effect of ceiling fan rotation has been simulated using the sliding mesh technique and general grid interface. Analyses have been undertaken for two different speeds of the ceiling fan namely 124 rotations per minute (RPM) and 240 RPM.

Radiation

CFX Discrete Transfer Surface to Surface radiation model with 8 rays and Gray spectral model has been used in all analyses.

SOLVING AND CONVERGENCE

Natural Convection Cases

Steady state simulations have been undertaken for all natural convection cases. Initially energy, fluids, radiation and turbulence were solved to achieve the maximum possible convergence typically being between 1E-03 and 5E-04 for Root Mean Square residuals. After the maximum initial convergence has been reached the cases were solved for energy and radiation only with the turbulence and fluid flow field being 'frozen' inside the domain. Convergence of the final solution has been judged based on three parameters: temperature at the sensor location, average temperature at the height of 1.2m above floor level (AFL) and energy imbalance inside the domain. The cases were assumed to reach convergence once the monitored temperatures stop changing and the energy imbalance within the domain is less than 1%. In order to achieve a higher convergence all cases have been solved with a double precision solver.

Forced Convection Cases

The Discrete Transfer Surface to Surface radiation model could not be solved simultaneously with the ceiling fan domain rotation due to software limitations. Initial transient runs have been solved for energy, fluids and turbulence only and without radiation. Transient cases have been run for a total time of 60 seconds and with a ceiling fan rotation increment of about 3º. After 60 seconds of real time the flow field inside the domain was observed to stabilise. The results obtained in the transient runs with ceiling fan rotation have been used as the initial state for steady state simulations. Steady state simulations have been solved for energy and radiation only with the turbulence and fluid flow being 'frozen' inside the domain. Final convergence has been judged in the same manner as described for the natural convection cases.

MODEL VALIDATION

In order to validate the CFD model the results obtained by ANSYS CFD have been compared with the results produced by advanced dynamic thermal analysis simulation software IES VE. IES VE thermal simulation is based on first-principles mathematical modelling of the heat transfer processes occurring within and around a building. For this purpose a 3D model of the small room for thermal dynamic analysis was generated using IES VE software and relevant parameters used in the analysis have been adjusted to match CFD simulation input data.

The following table presents average internal surface glass and wall temperatures and related heat fluxes into the space obtained for the same outdoor conditions.

	Temperature [K]		Wall Heat Flux [W]	
	Glass	Wall	Glass	Wall
IES VE	304.2	296.0	288.1	43.2
ANSYS CFD	303.3	297.2	277.4	44.7
Relative Difference [%]	0.3	0.4	3.7	3.3

Table 4: Model validation results

The results above indicate a close proximity of the key parameters relevant for CFD analysis.

RESULTS AND ANALYSIS

PMV has been calculated in accordance with ISO 7730 Standard.

$$PMV = TS * \left(MW - \sum_{i=1}^{6} HL_i \right) \tag{2}$$

Where

TS thermal sensation

MW internal human body heat production

$\sum_{i=1}^{6} HL_i$ heat losses

Detailed calculation information for each of the PMV components presented above is provided in the ISO 7730 standard.

All analyses have been carried out for constant values of relative humidity RH=50%, metabolic rate M=70 W/m^2 (M=1.2 MET) and clothing level I_cl=0.11 m^{2}K/W. These parameters have been chosen to represent a typical office conditions.
The PMV thermal sensation scale ranges from -3 to 3 and is defined as follows: +3 hot, +2 warm, +1 slightly warm, 0 neutral, -1 slightly cool, -2 cool and -3 cold.

Small Room

Small room analyses have been carried out for two outdoor temperatures, 24ºC and 35ºC. Four different natural convection cases have been run, each set to achieve different thermal conditions at the sensor location:

Case 1 T=24 ºC
Case 2 PMV=0
Case 3 PMV=0.5
Case 4 PMV=1

The sensor has been located 1.2m above the floor level (AFL) in the middle of the room side wall and 4cm off the wall surface, refer to Figure 1. An internal load, a heat source of 6.9 W/m^3 has been assigned to the room volume.

This equates to an internal load of 20 W/m^2 which is considered to be a typical office internal heat gain.

The obtained PMV contours are presented on the vertical cross section through the middle of the room.

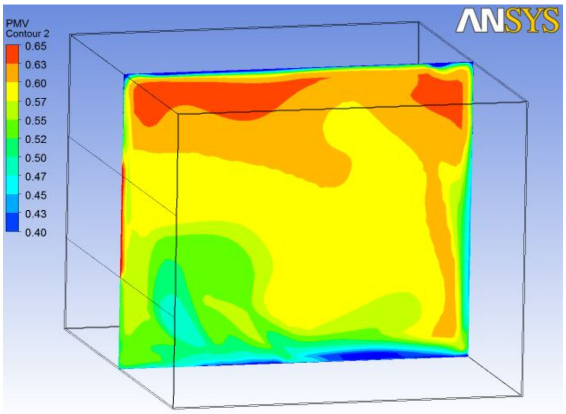

Figure 3: PMV Contours on the vertical cross section – Outdoor T=35ºC, Case 3

Cooling Energy

Results obtained for the four described cases are summarised in the following tables. Controlled sensor parameters are shown in red and Indicative Energy Saving is calculated based on the relative difference to Case 1.

Small Room, Outdoor T=24ºC					
Case	Fan [RPM]	T [ºC] sensor	PMV sensor	Cooling [W]	Indicative Energy Saving
1	0	24.0	-0.4	-321	-
2	0	24.7	0.0	-304	5.3%
3	0	26.6	0.5	-259	19.3%
4	0	28.6	1.0	-202	37.1%

Table 5: Results summary - Small Room, Outdoor T=24ºC

Small Room, Outdoor T=35ºC					
Case	Fan [RPM]	T [ºC] sensor	PMV sensor	Cooling [W]	Indicative Energy Saving
1	0	24.0	0.0	-602	-
2	0	24.0	0.0	-602	0%
3	0	26.1	0.5	-549	8.8%
4	0	28.0	1.0	-497	17.4%

Table 6: Results summary - Small Room, Outdoor T=35ºC

Indicative Energy Saving is used to estimate potential energy saving at given ambient conditions. In order to predict actual energy saving an analysis would need to be undertaken for one calendar year using meteorological weather data.

CFD results indicate that substantial energy savings can be achieved if the PMV based control is used instead of temperature based control. All analysed conditions PMV=0, PMV=0.5 and PMV=1 are considered as acceptable thermal conditions inside an office space. The tabulated energy saving is seen as a result of two key factors: benefit from chilled ceiling radiant temperature accounted through PMV control and increased air temperature adjusted to suit targeted thermal conditions inside the room.

Thermal Conditions at Height of 1.2m

Thermal conditions obtained at 1.2m AFL are a good representation of thermal sensation experienced by people while seated. Average temperatures and PMV at 1.2 m height are in close proximity to the temperature and PMV obtained at the sensor location.

Small Room, Outdoor T=24ºC			Outdoor T=35ºC	
Case	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$
1	24.3	-0.3	24.2	0.1
2	25.0	0.1	24.2	0.1
3	26.8	0.6	26.3	0.6
4	28.6	1.0	28.0	1.0

Table 7: Thermal conditions - Small Room, Outdoor T=24ºC and Outdoor T=35ºC

Small Room with Ceiling Fan

Analyses of the small room with a ceiling fan have been undertaken for the two outdoor temperatures 35ºC and 24ºC and two different rotational speeds of the fan. Two cases have been analysed both targeting 0 PMV at the sensor location:

Case 5 PMV=0, fan with 124 RPM
Case 6 PMV=0, fan with 240 RPM

According to the manufacturer's specification the fan power is 9.7 W for 124 RPM and 50.3W for 240 RPM. The fan power has been taken into account for Indicative Energy Saving analyses.

Obtained PMV contours are presented on the vertical cross section through the middle of the room.

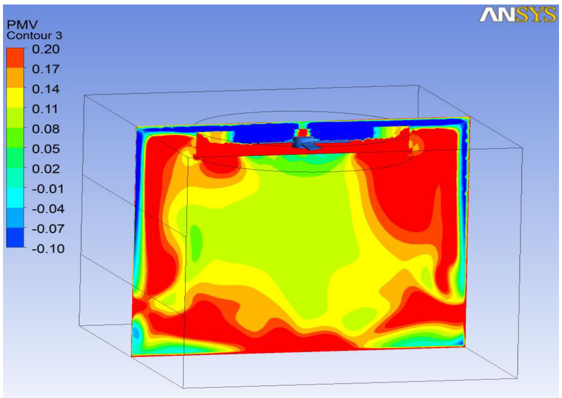

Figure 4: PMV Contours on the vertical cross section – Outdoor T=24ºC, Case 6

Cooling Energy

Results obtained for these two cases are summarised in the following tables and compared with cases 1 and 2. Cases 2,5 and 6 are all controlled to achieve 0 PMV at the sensor location. Controlled sensor parameters are shown in red and Indicative Energy Saving (IES) is calculated based on the relative difference to the energy benchmark Case 1.

Small Room, Outdoor T=24ºC					
Case	Fan [RPM]	T [ºC] sensor	PMV sensor	Cooling [W]	Indicative Energy Saving
1	0	24.0	-0.4	-321	-
2	0	24.7	0.0	-304	5.3%
5	124	25.0	0.0	-155	*48.6%
6	240	25.5	0.0	-139	*41.1%

Table 8: Results summary - Small Room with ceiling fan, Outdoor T=24ºC

Case	Small Room, Outdoor T=35ºC				
	Fan [RPM]	T [ºC] sensor	PMV sensor	Cooling [W]	Indicative Energy Saving
1	0	24.0	0.0	-602	-
2	0	24.0	0.0	-602	0%
5	124	24.9	0.0	-586	*0.1%
6	240	25.7	0.0	-648	*-13.8%

Table 9: Results summary - Small Room with ceiling fan, Outdoor T=35ºC

*Calculated Indicative Energy Saving has been adjusted to account for fan energy consumption

The ceiling fan induced air movement inside the room increases the heat transfer coefficient and related wall heat fluxes. Internal surface resistance decreases due to increased air velocity along the window and external wall (Figure 5). This allows for greater heat conduction through the window and external wall and a more dominant effect of outdoor ambient conditions.

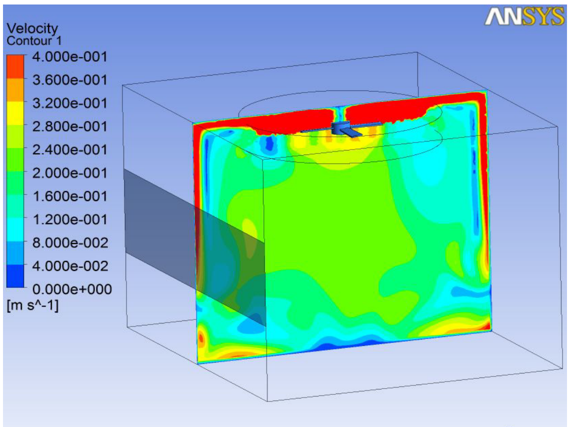

Figure 5: Velocity Contours on the vertical cross section

In cases when the outdoor air temperature is 24 ºC increased air movement along the window and external wall results in increased heat dissipation out of the room and consequently an increased energy saving as shown in cases 5 and 6. In cases when the outdoor air temperature is 35 ºC increased air movement along the window and external wall results in increased heat flux into the room and an increased energy consumption as shown for case 6 where a negative saving has been recorded.

Thermal Conditions at Height of 1.2m

Average PMV value at the 1.2m height has greater value than PMV at the sensor location.

Case	Small Room, Outdoor T=24ºC		Outdoor T=35ºC	
	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$
5	25.1	0.1	25.2	0.3
6	25.6	0.1	26.0	0.2

Table 10: Thermal conditions - Small Room with ceiling fan, Outdoor T=24ºC and Outdoor T=35ºC

The ceiling fan forces air in the radial direction along the chilled ceiling surface and then down along the internal walls. CFD results show that air velocity along the vertical walls and at the sensor location is twice the average air velocity at a height of 1.2m (Figure 7) and that temperature along the vertical walls and at the sensor location is slightly lower than average air temperature at a height of 1.2m. This results in increased levels of PMV (Figure 6) in the central zone of the room.

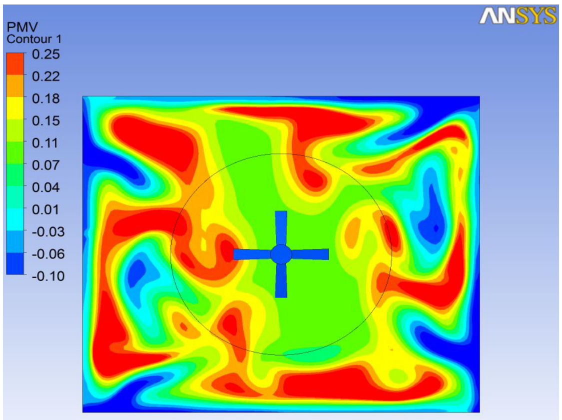

Figure 6: PMV Contours at the height of 1.2m – Outdoor T=24ºC, Case 6

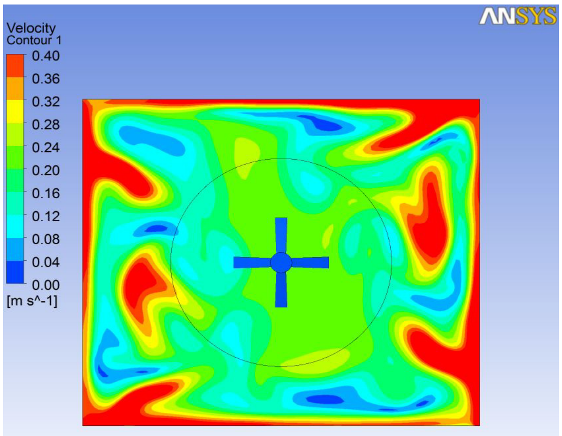

Figure 7: Velocity Contours at the height of 1.2m – Outdoor T=24ºC, Case 6

Large Room

Large room analyses have been carried out for an outdoor air temperature of 35ºC and two different rotational speeds of the ceiling fan; 124 RPM and 240RPM. Four different cases have been run, each set to achieve different thermal conditions at a height of 1.2m:

Case 7 T=24 ºC
Case 8 PMV=0
Case 9 PMV=0.5
Case 10 PMV=1

Large room analyses were carried out to investigate the impact of the ceiling fan on the energy consumption inside the room. The room was extended in order to reduce the impact of air flow on the heat flux into the room from the external envelope. An internal window overhang (Figure 2) is used to reduce potential vertical air flow along the window as observed in the small room analysis (Figure 5).

Cooling Energy

The results obtained for four described cases are summarised in the following tables. Controlled sensor parameters are shown in red and Indicative Energy Saving is calculated based on the relative difference to Case 7.

Large Room, RPM 124				
Case	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$	Cooling [W]	Indicative Energy Saving
7	24.0	-0.2	-755	-
8	25.0	0.0	-733	*2.9%
9	27.5	0.5	-676	*10.3%
10	29.3	1.0	-638	*15.3%

Table 11: Results summary - Large Room, RPM 124

Large Room, RPM 240				
Case	$T_{z=1.2m}$ [ºC]	$PMV_{z=1.2m}$	Cooling [W]	Indicative Energy Saving
7	24.0	-0.3	-838	-
8	25.8	0.0	-791	*5.3%
9	27.5	0.5	-739	*11.1%
10	28.9	1.0	-704	*15.1%

Table 12: Results summary - Large Room, RPM 124

*Calculated Indicative Energy Saving is adjusted to account for fan energy consumption

The increase in cooling requirement simulated for the higher fan rotation speed is due to the higher air velocities along the external wall and window and the higher heat flux into the room.

Thermal Conditions at Height of 1.2m

PMV contours at 1.2m height indicate lower PMV values along the walls in the vicinity of the ceiling fan. This localised effect is due to increased air flow velocities down the vertical walls as explained in the small room analyses section of this paper.

CONCLUSION

Controlling PMV rather than air temperature inside a room provides a base for improved thermal comfort. It prevents over-cooling as simulated and demonstrated in cases 1, 2 and 7. In these cases for the controlled sensor parameter of T=24 ºC negative PMV values are observed.

Ceiling Fan

The ceiling fan analyses indicate that increased air velocity along the external wall and window has a significant impact on the heat conduction through the external room envelope.

The ceiling fan operation and induced air flow along the external wall and window during peak outdoor conditions, T=35ºC increase heat flux into the room. Analyses indicate a potential energy saving can be expected if a radiant HVAC system is used instead of a convective HVAC system during this ambient condition.

For mild outdoor conditions, T=24ºC analyses indicate a greater energy saving in the case of a convective HVAC system and increased air velocities along the external envelope. This has been demonstrated in the cases with the ceiling fan operation.

As observed in the modelling results, selecting the appropriate fan for the room geometry is important as in some cases excessive air velocity will result in greater heat fluxes and hence higher energy consumption.

Energy

Calculated Indicative Energy Saving for the PMV based control system ranges between 0 and 48.6% as tabulated.

The actual energy saving should be calculated based on an annual energy analysis and the final energy saving percentage will depend on many factors including room shape, type of HVAC system, set PMV value climate zone and others.

PMV Control

Compared to typical design practice, with control based on air temperature, the proposed PMV based control:

- Provides control of perceived thermal conditions rather than control of air temperature
- Is capable of providing better thermal conditions
- Has the potential to save a substantial amount of energy

A PMV control strategy could be developed and implemented in practise by measuring air temperature, radiant temperature and relative humidity and based on assumed values of occupant clothing, metabolic rate and air velocity. This could be used to achieve significant saving in energy and higher levels of comfort compared with a temperature controlled air conditioning system.

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