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Numerical Study of Ceiling Fan Utilization Effect on Energy Usage Amount and Residents Thermal Comfort in Cold Season

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ABSTRACT– The considerable portion of produced heat by heating systems in winter, accumulate near the ceiling in the top of the room. If this heat transferred towards the lower parts of room for example by a ceiling fan, temperature in the living zone of the residents will be increased and it can be caused the energy saving. But the utilization of ceiling fan can caused the local thermal discontent due to draught. This article has surveyed numerically the optimum utilization of ceiling fan for energy consuming reduction in the winter. For this purpose, the effects of different factors such as volumetric flow rate, static pressure, rotational speed and blade diameter of the fan on temperature distribution and thermal comfort condition of the occupants have been studied. The results showed significant portion of energy can be saved during the winter by properly designing of a ceiling fan. Using a simple ceiling fan with volumetric flow rate of 0.5 m3/s and static pressure of 8.5 N/m2, will decrease the energy consumption for room heating about the 24 percent.
KEYWORDS: Thermal comfort, Ceiling fan, Local thermal discontent

Introduction

Heating Incorporation is one of the cases that has grown rapidly in recent years, decreases simply the energy consumption and is applicable in old and new buildings. It is one of the best Carbon reduction methods for every building in Carbon Trust classification [1].The general heat layering process is done in all buildings. This process happens because of the bottom and ceiling heat difference and also the wall to wall heat difference. Heat layering occurs due to the warm weather rising from the room bottom towards the ceiling because of its lighter weight than the cold weather. This means that the air conditioning systems must circulate the air continually, keeping the temperature constant within a specified range (Powrmatic, 2014) [2]. Air conditioning systems are charged with warm and cold weather providing and the lost temperature compensation due to the unwanted air circulation within the room area (it is located in 1.5 to 2 m height from the ground). Achieving this purpose accompanied with huge energy consumption and enormous carbon providing. The warm weather will move towards the room ceiling due to this phenomenon and creates a heat difference about 14℃ or more, between the bottom and top of the room in warehouse-like spaces [3]. The more difference between the bottom and top of the room caused the more temperature variance [2]. At the worst situations the heat difference of air layers in a one meter height can rise to 10 ℃ [4]. The mixing phenomenon, increases the in and out temperature difference, caused the warm air exiting accelerate through the ceiling consequently. This heat can be saved and recycled, using an effective air mixing system. This heat can balance the inner temperature and consequently reduces the needed workload of air conditioner systems and their work time duration [3]. Therefore the vertical heat layering destruction is one of the energy consumption reduction that has been used so far in most of warehouses and nonresidential spaces with general high ceiling elevation. In residential areas with a general height about 2.7 to 3 m, sometimes the considerable temperature gradient is created due to vertical heat layering formation in the room. If it was possible that the trapped warm weather at the top of the room being transferred to the bottom in winter, using a fan with specified properties, it is likely to reduce somewhat the energy consumption. The important point in this procedure is that to calculate the fan structural properties so that produced air flow does not cause the local thermal discontent due to its blowing. Sometimes, however the PMV index has an acceptable amount relating to the person comfort but Intended person has local dissatisfaction in some part of his/her body due to blowing phenomenon. Therefore it needs to calculate factors such as the local discontent index caused from blowing (DR) along with PMV index. A lot of studies have been done in fan using for comfort conditions in hot season such as Rohles et al. 1982&1983[5-6], Morton et al., 1985, [7] James et al., 1996 [8] and Ho et al., 2009 [9]. Also there are some studies about the fan utilization for comfort condition origin in cold season such as Aynsley (2005) [10]. He expressed that using the big rotational fans for heated weather mixing in spaces with high ceiling and corridors can save a lot of energy. In that article it has been stated that winter heat saving in warehouse-like buildings equals to 10 percent for each 3 meters distance from the ceiling.

In 2009 the Agviro Company in a research for Anbride[11], calculated the environmental situations and economical frugality in using of an especial Airius stirrer fan in Canada in the winter. This company claimed that has controlled the temperature drop during the night, the time that the most temperature drop is occurred, using the mixing phenomenon caused by utilization of this fan with an especial design. They stated that drop amount is kept constant about 0.5 ℃ no need to rise in temperature of heating system and annual gas consumption will be decreased about the 23.5 percent. In that study the thermal comfort of residents has not been surveyed and that kind of fans are applicable in warehouse-like spaces. Some studies also have been done in using of ceiling fan for thermal comfort creation in cold season for temperature layering in residential spaces such as Ericson (2007). He said in a note, that suitable usage of ceiling fan in summer, only in people presence at room, must be set counterclockwise at average or rapid state but it must be clockwise and slow state in winter. In 2014 a study was done on an especial Airius fan operation at an incubation factory and showed the created heat mixing by this fan, has saved the heating systems consumption about 61 percent (Airius Co., 2014).

Research Method

In this study the effect of heat mixing on energy consumption has been studied in a closed room with 2.75*3*5 m^3 dimension in winter and constant three dimensional condition (fig 1). In this room a human sample is located at the center of room that his heat sense condition has been counted as a scale of thermal comfort situation. There is a radiator under a window at the left side of room and a closed door relates to uncontrolled space with a temperature of 9.5 ℃ at the right side. Three walls that one of them has window are related to the outer winter space of considered climate. The bottom and ceiling of room have been considered without heat exchange due to relation to controlled space. Window and door sutures have been simulated separately and a ceiling fan with 2.3 m elevation from bottom with a LED lamp under it is located at the center of room. In this simulation it has been tried to sample's comfort situations reach to ideal comfort situations in PMV index equal to 0.5. -4 ℃ air temperature and 80 percent relative humidity has been considered for uncontrolled space situations outer winter season. Entered air from the door seam with 5mm, 2 cm and 1 cm width from the top, bottom and door around respectively, has 9.5 ℃ heat and 70 percent relative humidity. Window dimension is equal to 1.5*1.85 m and the wall thickness is equal to 20 cm with 0.36 w/m.k heat resistance. The window's heat resistance has been considered equal to 0.59 w/m.k based on 19th subject data of National Building Regulations. Engine thermal power and lamp thermal power have been counted 10 and 10 w/m.k respectively. Considered analysis is turbulence and used turbulence model is RNG. The fan has been studied in normal and revers ON state with different volumetric flow rate and static pressure equal to 0, 1, 2.5, 5, 7.5, 8.5 and 10 N/m^2. The amount of fan torsion has been counted equal to 0 RPM and also studied in 12, 24, 48, 96 and 120 RPM. The effect of fan diameter changing has been assessed on comfort parameters and the results have been compared. The used net is staggered kind. The sample net was evaluated in various situation and its independent net have 1877722 reticulation element and node number equal to 1926764 and it is a structured type.

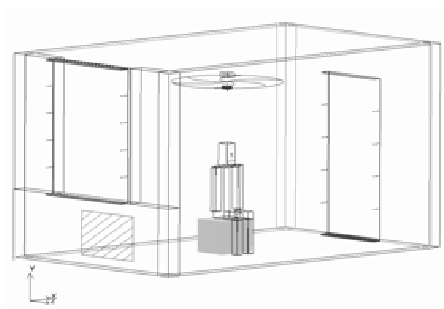

Fig 1. Three dimensional view of study room

Findings

For off mode the minimum limit was gained for the considered standard and it was compared for ON mode in various static pressure and volumetric flow rate. As it can be seen in figure 3 and 4, increasing the volumetric flow rate in constant pressures, the total comfort index amount has been changed. It increases to a distinct amount and decreases after that. This effect will be more considerable along with static pressure increasing.

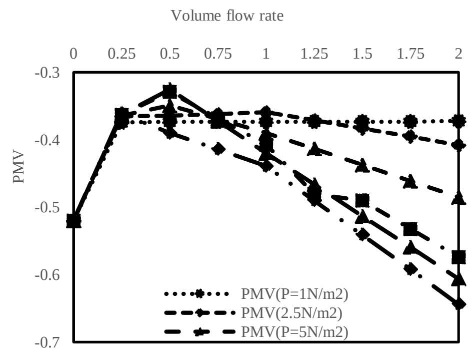

Fig 3. Comparing the effect of volumetric flow rate changes in different pressures on comfort index parameter.

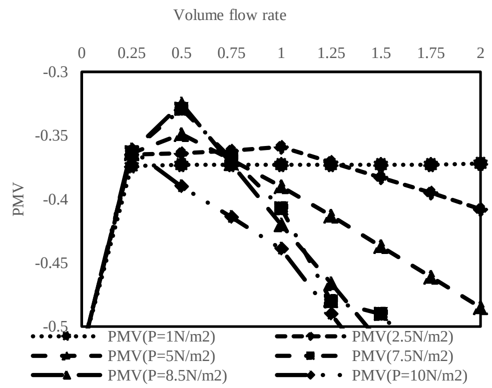

Fig 4. Comparing the effect of volumetric flow rate changes in different pressures on comfort index parameter in standard range.

As it can be seen in figures 5 and 6, at constant flow rate equal to 0.5 m^3/s, the fan in all tested pressure has allowed local discontent less than 20 percent and has had a logical distance from zero and has the better improvement rate than other volumetric flow rates (table 1). Therefore this volumetric flow rate is a good index for comparing the different fan operations in various static pressure. Considering the figure 6, increasing the static pressure at constant flow rate (0.5 m^3/s), to 8.5 N/m^2, the temperature and thermal comfort index are increased and the residents feel less cold than the environment conditions. At higher pressures the local discontent has been raised over the allowed limit and the thermal comfort index gives a downtrend.

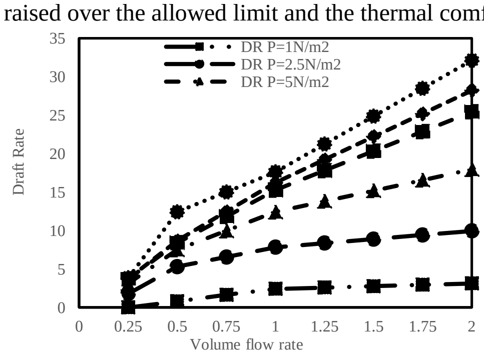

Fig 5. Assessment of volumetric rate effect on local heat discontent parameter caused by blowing of fans with various static pressures.

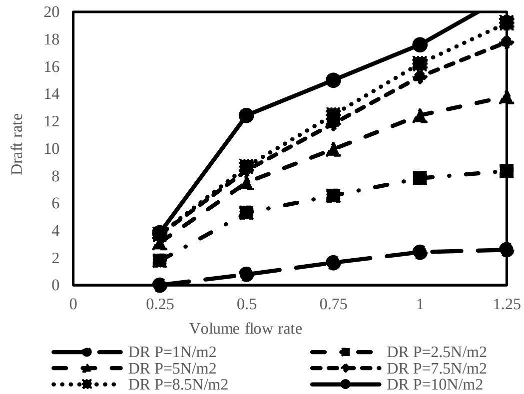

Fig 6. Allowed range of local discontent index because of wind blowing.

This fan has been analyzed and assessed in 9 different static pressure (0 to 12 N/m^2). More pressure have been used in fans with frame and fan coils. The fans with higher pressure has the annoying wind sense for residents in area equal to ceiling fan dimension due to their high turbulence speeds. Therefore the sufficient study has been done until this pressure. As it can be seen in figure 7 the optimum state has been gained and it was indicated that in considered volumetric rate and static pressure equal to 8.5 N/m^2, has the best effect on thermal comfort index in study pressures.

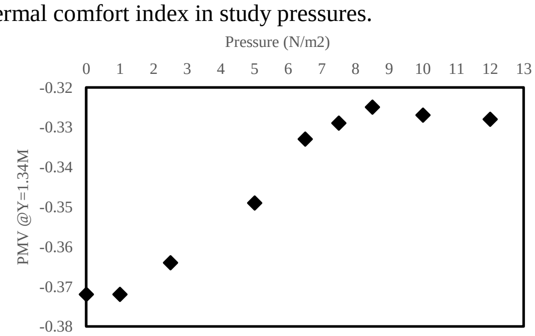

Fig 7. Effect of static pressure changing on comfort index in volumetric rate of 0.5 m^3/s.

Table 1. Compering the comfort situation in off and on fans mode in static pressure of 8.5 N/m^2

Comfort condition improvement percentage (%)	Local discontent due to blowing (%)	General Thermal comfort index	Volumetric rate(m^3/s)
0	0	-0/52	0
15/7	3/72	-0/363	0/25
19/5	8/68	-0/325	0/5
12/75	12/44	-0/3725	0/75
10	16/21	-0/42	1
5/35	19/21	-0/466	1/25
0/7	22/2	-0/513	1/5
-3/95	25/19	-0/56	1/75
-8/6	28/19	-0/606	2

Table 2. Compering the comfort situation in off and on fans mode in static pressure of 0.5 N/m^2

Comfort condition improvement percentage (%)	Local discontent due to blowing (%)	General Thermal comfort index	Static pressure (N/m^2)
15/1	0/1	-0/369	0
14/8	0/79	-0/372	1
15/6	5/31	-0/364	2/5
17/1	7/51	-0/349	5
19/1	8/43	-0/329	7/5
19/5	8/68	-0/325	8/5
13	12/42	-0/39	10

Continuing the study of possible conditions, improving the heat mixing situations, the fan with various torsions was evaluated. As it can be seen in figure 8 and table 3, it was recognized that the fan torsion doesn't have an especial effect on comfort index of room residents.

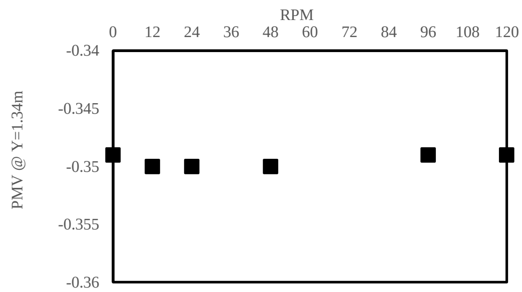

Fig 8. Comparing the effect of torsion on room comfort index in study sample.

Table 3. Comparing the different fan torsion in constant volumetric rate and static pressure

0.5			Volumetric rate (m^3/s)
Comfort condition improvement percentage (%)	**Local discontent due to blowing (%)**	**General Thermal comfort index**	**torsion (RPM)**
17/1	0/1	-0/349	0
17	7/5	-0/35	12
17	7/51	-0/35	24
17	7/5	-0/35	48
17/1	7/5	-0/349	96
17/1	7/5	-0/349	120

The fan with different radius was assessed and as it can be seen in figure 9, the ceiling fan with 0.75 m radii shows the best operation. Then a dimensionless parameter was evaluated for ratio of fan to room area to extend this result for any room with any dimension. As it can be seen in figure 10, it was determined that in ratio of fan sweeping area to room area equal to 0.118, the ceiling fan has the best operation and effect on room residents comfort index.

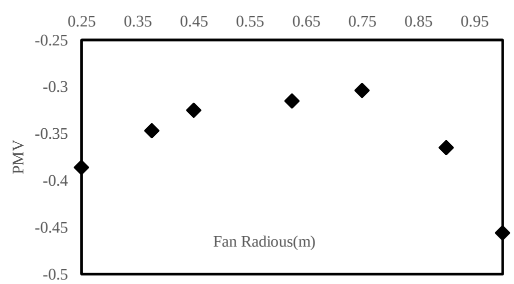

Fig 9. Assessment of radius change on comfort index in study sample.

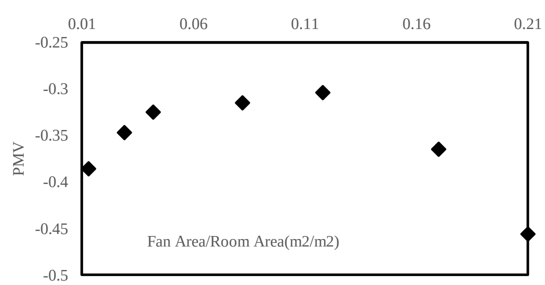

Fig 10. Assessment of radius change on room area on comfort index in study sample.

The minimum comfort with optimum operation was calculated via trial and error method and the energy saving was measured then. As it can be seen in figure 11 it was indicated that ceiling fan usage in distinct optimum pressure and volumetric rate, has improved the room comfort index about the 19.5 percent.

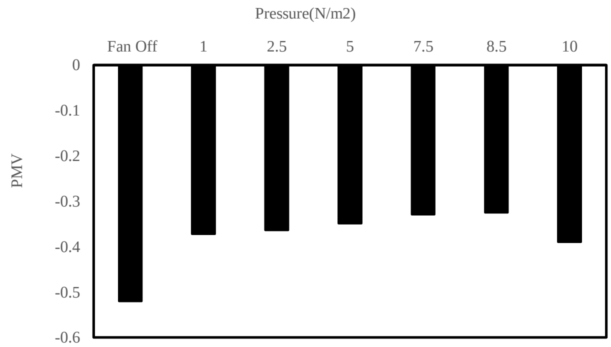

Figure 11. Effect of normal and off fan using on room total comfort index decreasing

The amount of energy saving was calculated by this method. obtained results in figure 12 show that using of ceiling fan in considered optimum volumetric rate and pressure, has made the 23.73 percent energy saving.

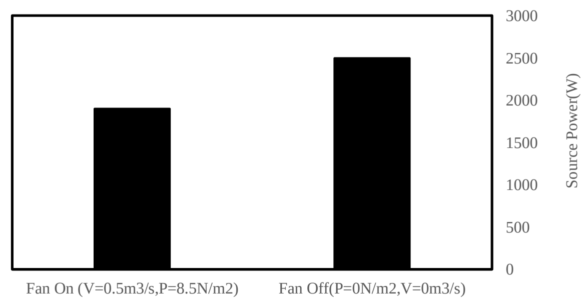

Fig 12. Comparing the utilization of ON and off fans in energy consumption decreasing in room at optimum state of total comfort index.

To better display the changes in room temperature range in elevation direction, the temperature fluctuation profile has been compared for fan ON and off mode in figure 13.

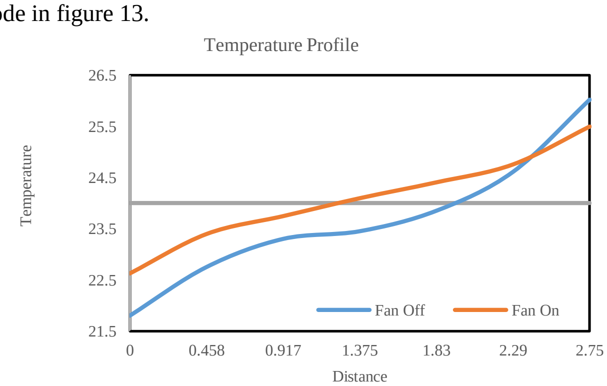

Fig 13. Comparing of temperature profile in elevation direction in fan ON and off mode

Mathematical formulas and relations

Performing this analysis, the air velocity, temperature, room relative humidity and also the local thermal comfort index must be calculated. This act is possible by the simultaneous calculation of mass conservation equations, momentum and the energy of air flow.

The used mass conservation equation in this study is as below:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0 \qquad (1)$$

The momentum transfer equation in i^{th} direction is as below:

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v}\vec{v}) = -\nabla p + \nabla \cdot \left(\vec{\vec{\tau}}\right) + \rho \vec{g} + \vec{F} \qquad (2)$$

Where p is the static pressure, $\vec{\vec{\tau}}$ is the stress tensor and $\rho \vec{g}$ is the gravitational force and $\overrightarrow{F}$ is also the other forces. The stress tensor then calculated as below:

$$\vec{\vec{\tau}} = \mu \left[(\nabla \vec{v} + \nabla \vec{v}^{T}) - \frac{2}{3} \nabla \cdot \vec{v} I \right] \qquad (3)$$

Where μ the molecule viscosity and the second term on the right side is is the volumetric inertial effect. Used turbulence model in this study was the RNG model that is the improved version of two equation standard model. In this model the constants which have been obtained practically are extracted as explicit form. In this model the equations which model the turbulence contain two equation.

The first equation that is the equation of turbulent kinetic energy is as below (Airpak manual, chapter18, 2002):

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\alpha_k \mu_{eff} \frac{\partial(k)}{\partial x_i}\right) + G_k + G_b + \rho\epsilon \qquad (4)$$

Second equation that named the turbulence energy loss rate is as below:

$$\frac{\partial(\rho\epsilon)}{\partial t} + \frac{\partial(\rho\epsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\alpha_\epsilon \mu_{eff} \frac{\partial(\epsilon)}{\partial x_i}\right) + C_{1\epsilon}\frac{\epsilon}{k}(G_k + C_{3\epsilon}G_b) - C_{2\epsilon}\rho\frac{\epsilon^2}{k} - R_\epsilon \qquad (5)$$

In aforementioned equations G_k is the generator term of turbulent kinetic energy based on average velocity gradient, G_b is the generator term of turbulent kinetic energy based on buoyancy force and the α_ϵ and α_k are the inverse of Prandtl number for k and ϵ; $C_{1\epsilon}$ and $C_{2\epsilon}$ are 1.42 and 1.68 respectively. An equation as thermal comfort function is utilized, studying the people thermal comfort.

$$PMV = (0.303 \times e^{-0.036\,M} + 0.028)[(M - W) - 3.05 \times 10^{-3} \times [5733 - 6.99(M - W) - P_a] - 0.42[(M - W) - 58.1$$
$$-1.7 \times 10^{-5} \times M(5867 - P_a) - 0.0014M \times (34 - T_a) \qquad (6)$$

Where M is the metabolic rate of the body in watts, W is the mechanical work in watts, T_a is the environ temperature in Kelvin and P_a is the steam pressure of air in pascal. The heat feeling that Fenger criteria was defined based on it is as below:

-3 Cold
-2 Cool
-1 relatively cool
0 Neutral
+1 relatively warm
+2 Warm
+3 Hot

This criteria has been named as PMV or comfort index, shows the average of several people same feel under the one environmental condition [12].

One other factor as thermal discontent criteria due to blowing, has been considered for comfort condition determination in addition to temperature, relative humidity and thermal comfort index. It is a function of air temperature, air velocity and turbulence intensity. Reaching to a desired comfort criteria, this parameter should be lesser than 20 percent [13]. This parameter is calculated as below [15].

$$D.R = (34 - t_a)(V_a - 0.05)^{0.62}(0.37V_aT_i + 3.14) \qquad (7)$$

Where t_a shows the air temperature in centigrade degree, V_a shows the air velocity in m/s and T_i shows the turbulence intensity in percent.

Result and Discussion

In performed investigations it is shown that:

- In situation that room air is static and trapped air exists at the top of room, the ceiling fan using improves the comfort.
- Using the ceiling fan with volumetric rate equal to 0.5 m^3/s and static pressure equal to 8.5 N/m^2 in constant 0,9 m diameter at the best condition, the room comfort index improves to 19.5 % and room energy is saved about 23.73 % at the best state.
- The fan torsion changes have insignificant effect on comfort parameters related to the room residents.
- Using the ceiling fan with fan area to room area ratio equal to 0.118 m^2/m^2, has an effective performance for study sample in explained volumetric rate and static pressure and can improve the room comfort index to 21.6 %.
- In this research a ceiling fan has been used for winter heating and it was shown that at the best condition for a person stands under the fan, the room energy consumption could be saved to 24%.

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