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Corrigendum

Corrigendum "Temperature control of a cabin in an automobile using thermal modeling and fuzzy controller" [Applied Energy 97 (2) (2012) 860–868]

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The authors regret that Navid Kashaninejad's name was not included on the above mentioned paper and would like to correct this omission. The authors would like to apologize for any inconvenience this may have caused to the readers of the journal.

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Temperature control of a cabin in an automobile using thermal model and fuzzy controller

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ABSTRACT

This paper covers thermal modeling of a cabin in an automobile to find and control the air temperature by the means of fuzzy controller. Then, in the first step, the thermal and ventilation loads were estimated then the equations of dry air mass and energy conservation as well as internal components of a cabin were derived and solved simultaneously. The performance of the proposed thermal modeling of a cabin was compared with the experimental hot room test. In the next step, to maintain the thermal comfort of a cabin and controlling the two effective parameters (blower outgoing air velocity and the circulated air percentage), a fuzzy controller was applied. Results showed that when using a fuzzy controller, the temperature control of [illegible] took shorter time period and as a result, the time spent for ventilating and cooling the cabin as well as [illegible] consumption are reduced.

1. Introduction

Controlling thermal comfort of passengers in a cabin is important. The temperature and thermal comfort control of a cabin for an automobile would be possible by using proper controller. Matsui et al. [1] used modern control theory for thermal analyses of an automobile cabin and determined the governing equations of the model by means of statistical methods. Lim et al. [2] employed a microcontroller which continually recorded the temperature of a cabin and sent the controlling commands to the compressor in order to maintain the cabin's thermal comfort for passengers. Davis et al. [3] used a [illegible] controlling system due to nonlinear and complex nature of governing equations in thermal modeling of a cabin. Wei and [illegible] [4] designed the control system of a cabin for an automobile [illegible] intelligent methods. Durovic and KovaZeviC [5] used artificial neural network under several weather conditions to control the temperature of a cabin. Aeatrice et al. [6] analyzed the dynamic model of a refrigeration cycle to analyze the thermal comfort of a cabin and to control the air conditioning system in state space. In his work, the moisture produced by passengers was measured by a sensor, while the governing equations were just applicable to 100% circulated return air passing over the evaporator. Qi and Deng [7] introduced a fuzzy controller for modeling the thermal comfort inside the cabin of a car in which only the inlet air volume rate was considered for air temperature control. Farzaneh and Tootoonchi [8] used a fuzzy controller with temperature feedback to supervise the thermal comfort of a cabin.

In the first part of this paper the thermal and ventilation loads were estimated, and then the mass and energy balance equations for dry air as well as the balance equations for internal components of a cabin were derived and solved simultaneously. The proposed cabin thermal model was validated by hot room test experimental data. Then a fuzzy controller was applied to the model to control the cabin's temperature. The velocity of air passing through the blower and the position of the circulating air vent were controlled by this type of cabins air condition controller. The modeling would help designers to adapt the air conditioning system of an automobile with the design of an internal temperature controlling system.

The following are the contribution of this paper into the subject:

1. Effect of pollution level in an outside weather.
2. Control the air circulation ratio with a fuzzy controller.

2. Thermal modeling and governing equations for a cabin

The cabin thermal modeling was performed by using the Lumped Capacitance Method in which the temperature distribution was assumed to be spatially uniform. The variation of temperature and relative humidity of the air inside the cabin was estimated using dynamic equations of operating conditions which were influenced by: solar radiation load received by outer surfaces and windows of an automobile, convection and conduction heat transfer loads of a cabin, effect of sensible and latent heat as well

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Nomenclature

A	surface area (m^2)
A_D	skin surface area (m^2)
bp	blower power (W)
c_{mass}	specific heat of internal component (J/kg K)
E_{diff}	diffusive thermal load (W)
E_{sw}	sweating thermal load (W)
G_{cb}	beam solar radiation (W/m^2)
G_{cd}	diffuse solar radiation (W/m^2)
G_{cr}	reflection solar radiation (W/m^2)
h	heat transfer coefficient (W/m^2 K)
I_{cl}	thermal resistance of clothes
i	enthalpy (kj/kg)
i_{fg}	latent heat of saturated water at the skin temperature (kj/kg)
k	thermal conductivity (W/m K)
LH	latent heat
lhs	left hand side
m	mass (kg)
$\dot{m}$	mass flow rate (kg/s)
M	metabolic rate (J/m^2 s)
np	number of passengers
Nu	Nusselt number
p	pressure (Pa)
p_{wl}	saturated pressure of vapor at skin temperature (Pa)
p_{wa}(Pa)	saturated pressure vapor (Pa)
Pr	Prandtl number
q''	heat transfer flux (W/m^2)
Q	heat transfer rate (W)
R_b	ratio of beam radiation on a tilted surface to that [illegible] izontal surface
Ra	Raleigh number
Re	Reynolds number
rhs	right hand side
SH	sensible heat
T(C)	temperature
V	velocity (m/s)
x	thickness (m)

Greek letters

ζ	circulation ratio
ω	absolute humidity or humidity ratio (kg H_2O/kg dry air)
ρ	density (kg/m^3)
μ	weighting parameter
β	angle
τ_s	absorption coefficient
α_s	transmission coefficient

Subscripts

a	air
atm	atmosphere
w	water
evap	evapora[illegible]
conv	conv[illegible]on
s	su[illegible]
comb	[illegible]bi[illegible]n
cond	conductio[illegible]
i	inlet
i[illegible]	interior
[illegible]ss	internal components
[illegible]x	mix[illegible]
[illegible]	win[illegible]
re[illegible]	res[illegible]tion
sw	[illegible]ating
diff	diffusive
[illegible]	skin
v	vapor
cir	circulation
inf	infiltration
AC	air conditioning

as the inlet cold air from the air c[illegible]ing system [illegible]1). Therefore, four nonlinear differenti[illegible] equa[illegible] for the ai[illegible] inside the cabin were constructed in[illegible]ing mass b[illegible]e of dry air (Section 2.1), mass balance of w[illegible] vapor (Section [illegible] energy balance of inside air (Section 2.[illegible] [illegible]d ene[illegible] balance for the interior components of a cabin (Se[illegible] 2.4[illegible]

2.1. Dry air [illegible]ance e[illegible]tion [illegible] cabin

Con[illegible]ring th[illegible]abin of an automobile as a control volume, the followin[illegible]ss[illegible]tion for the dry air was applied:

$$\frac{dm_a}{dt} = (1-\zeta)\text{[illegible]} + \dot{m}_{a,inf} \quad (1)$$

where the ratio of mass flow rate of circulating air to the air flow rate passing through the evaporator is defined as:

$$\zeta = \frac{\dot{m}_{a,cir}}{\dot{m}_{a,AC}} \quad (2)$$

The term on the left hand side (lhs) of Eq. (1) shows the transient change of air mass in the cabin. The terms on the right hand side are as follows; ($\dot{m}_{a,AC}$) is the cabin inlet air mass flow rate passing through the evaporator, ($\dot{m}_{a,cir} = \dot{m}_{a,AC} \cdot \xi$) is the recirculation air mass flow rate exiting from the control volume, and ($\dot{m}_{a,inf}$) is the mass flow rate of the infiltration air.

The cabin infiltration air passes through the gaps of doors and windows due to a slight vacuum pressure inside the compartment (for which there is not enough information in the open literature to estimate this mass flow rate directly). It was assumed that the absolute humidity of the infiltration air was the same as the humidity of the atmospheric air.

2.2. Water vapor mass balance equation in a cabin

The evaporator exit air (sum of inlet fresh air and the circulated air going into or coming back to the evaporator inlet), has a considerable amount of humidity which its level increases due to aspiration of passengers. Considering the cabin as a control volume, the mass balance equation for water vapor can be shown in the following form:

$$\frac{d(m_a\omega)}{dt} = (\dot{m}_a\omega)_{inf} + (\dot{m}_a\omega)_{AC} + \dot{m}_{v,human} - (\dot{m}_a\omega)_{cir} \quad (3)$$

The (lhs) term of Eq. (3) is the transient change of inside cabin water vapor mass, the first term on (rhs) of Eq. (3) is the infiltration water vapor mass flow rate $(\dot{m}_a\omega)_{inf}$, the second term in (rhs) is the water vapor mass flow rate of air passing through the evaporator, the third (rhs) term is the water vapor produced by human aspiration and the fourth (rhs) term is the outlet water vapor mass flow rate from the cabin (by circulating air).

As explained in the last paragraph of Section 2.1, $\omega_{inf} = \omega_{atm}$.

By substituting Eq. (1) into the lhs term of Eq. (3), the mass conservation equation for water vapor content of air in a cabin was obtained as:

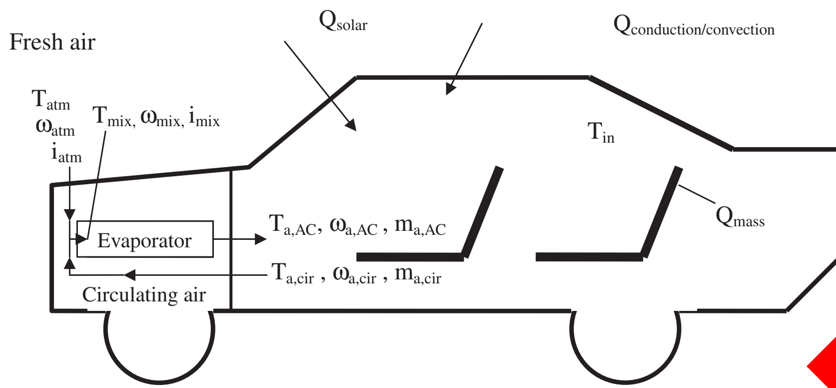

Fig. 1. The schematic diagram of cabin for an automobile.

$$\frac{d\omega}{dt} = \frac{\dot{m}_{a,AC}[\omega_{AC} - \omega] + \dot{m}_{a,inf}[\omega_{atm} - \omega] + \dot{m}_{v,human}}{m_a} \quad (4)$$

2.3. Energy conservation equation for air in a cabin

Considering the cabin as a control volume, energy balance for the air in a cabin would be:

$$\frac{d(m_a i)}{dt} = \sum \dot{Q} + (\dot{m}_{a,AC} \cdot i_{AC}) + (\dot{m}_{inf} \cdot i_{inf}) - (\dot{m}_{a,cir} \cdot i_{cir}) \quad (5)$$

The cabin air thermal loads have three main sources: solar the[illegible] load passing through windows and bodywork of vehicle ($\mu \cdot \dot{Q}_s$[illegible]), thermal load of convection ($\dot{Q}_{conv}$) and conduction ($\dot{Q}_{cond}$) h[illegible] transfer from bodywork of the vehicle and the thermal load of pa[illegible]sengers ($\dot{Q}_{human}$), thus:

$$\sum \dot{Q} = \mu \cdot \dot{Q}_{solar} + \dot{Q}_{human} + \dot{Q}_{cond} + \dot{Q}_{conv} + \dot{Q}\text{[illegible]} \quad (6)$$

In Eq. (6), $\dot{Q}_{mass}$ is the heat transfer ra[illegible] between [illegible] air and the internal components in a cabin w[illegible] can be est[illegible]ed from $\dot{Q}_{mass} = h \cdot A_{mass} \cdot (T_{mass} - T_{in})$. Th[illegible]ara[illegible]r μ in Eq. (6) is a weighting parameter to indicate the fract[illegible]f solar load which was absorbed by the air in [illegible]bin. Therefore [illegible]$\mu)\,\dot{Q}_{solar}$ was absorbed and dissipated via [illegible]in internal compon[illegible]ts. By substituting Eq. (1) in the (lhs) [illegible]q. (5) [illegible] energy conservation equation for the air in a cabin wa[illegible] obtained from:

$$\frac{di}{dt} = \frac{\sum \dot{Q} + \dot{m}\text{[illegible]}[i_{AC}\text{[illegible]}] + \dot{m}_{a,}\text{[illegible]}i]}{m_a} \quad (7)$$

2.4. Energy ba[illegible] for the internal components of a cabin

To account for t[illegible]eat capacity of internal components of a cabin acting as a heat source or heat sink which transfer heat to the air, a lumped mass was used to represent all the cabin internal components. Considering internal components of a cabin as a control volume then:

$$(1 - \mu)\dot{Q}_{solar} - \dot{Q}_{mass} = (mc)_{mass}\frac{dT_{mass}}{dt} \quad (8)$$

Replacing $\dot{Q}_{mass}$ in Eq. (8) by its definition ($\dot{Q}_{mass} = h \cdot A_{mass} \cdot (T_{mass} - T_{in})$), the equation for temperature variation of cabin internal components was obtained as:

$$\frac{dT_{mass}}{dt} = \frac{(1 - \mu)\dot{Q}_{solar} - h \cdot A_{mass} \cdot (T_{mass} - T_{in})}{m_{mass}c_{mass}} \quad (9)$$

Eqs. 1, 4, 7, and 9 are [illegible] ma[illegible]nd energy balance differential equations which w[illegible] solv[illegible]nd m_a ([illegible]in air mass), ω (cabin air humidity), [illegible] cabin ai[illegible]thal[illegible], T_{mass} (interior mass temperature).

2.5. Thermal analysis o[illegible]assing through the evaporator

[illegible] and ω_{AC} which are needed in Eqs. (7) and (4) respectively w[illegible] estimated [illegible] knowing the cooling capacity $\dot{Q}_{evap}$ (which wa[illegible]btained fr[illegible] of refrigeration cycle model). By considering the [illegible]orator [illegible] control volume, and by applying water vapor mass a[illegible]y balance equations for humid air passing through [illegible]e evaporator (Fig. 1), i_{AC} and ω_{AC} were obtained from the follow[illegible]rvation equations:

$$\dot{m}_{a,AC}\omega_{mix} = \dot{m}_{a,AC}\omega_{AC} + \dot{m}_w \quad (10)$$

$$i_{AC} = i_{mix} - \frac{\dot{Q}_{evap}}{\dot{m}_{a,AC}} - \frac{\dot{m}_w i_w}{\dot{m}_{a,AC}} \quad (11)$$

$\dot{m}_w$ is the condensed water vapor in the evaporator (drain) when humid air passes through its cooling surface. i_{mix} and ω_{mix} in Eqs. (10) and (11) are the enthalpy and absolute humidity of the evaporator inlet air (after mixing the fresh air and the circulating air, Fig. 1). These parameters were obtained by applying the mass conservation equation for water vapor and the energy conservation equation for mixing air process at the evaporator inlet:

$$\omega_{mix} = \frac{\zeta \cdot \dot{m}_{a,AC} \cdot \omega_{cir} + (1 - \zeta)\dot{m}_{a,AC} \cdot \omega_{out}}{\dot{m}_{a,AC}} \quad (12)$$

$$i_{mix} = \frac{\zeta \cdot \dot{m}_{a,AC} \cdot i_{cir} + (1 - \zeta)\dot{m}_{a,AC} \cdot i_{out}}{\dot{m}_{a,AC}} \quad (13)$$

3. Modeling the thermal loads of air in a cabin

The cabin of automobiles is exposed to the transient environmental conditions which change the thermal loads of solar radiation, conduction and convection heat transfer rates as well as the thermal load for passengers. Fig. 1 shows the schematic diagram of a cabin and its thermal loads.

3.1. The solar thermal load

The solar thermal load for a cabin was estimated from HDKR [9] homogenous model. The total absorbed solar energy by the cabin surfaces is explained as follows [10]:

$$G_{total} = G_{cb}\cdot R_b + G_{cd}\left[\frac{1+\cos\beta}{2}\right] + \rho_g[G_{cb}+G_{cd}]\left[\frac{1+\cos\beta}{2}\right] + G_{cr} \quad (14)$$

where the total solar thermal load for cabin was estimated from:

$$\dot{Q}_{solar} = G_{total}\tau_S A_{win} \quad (15)$$

$\dot{Q}_{solar}$ includes solar thermal loads absorbed from the total window surface area including rear window, windshield and side windows, and it is found as follows:

$$\dot{Q}_{solar} = \dot{Q}_{front,win} + \dot{Q}_{rear,win} + \dot{Q}_{side,wins} \quad (16)$$

3.2. The convection and conduction heat transfer thermal loads

The magnitude of convection thermal load depends on internal and external surface positions, ambient temperature as well as the vehicle speed (or air speed). Assuming a warmer air in the surrounding environment, the following heat transfer mechanisms occur [11]:

(a) The forced convection heat transfer between ambient warm air and external cabin surface.
(b) The conduction heat transfer between interior and exterior layers with the air gap in between.
(c) The natural convection heat transfer between interior surfaces of the cabin and the conditioned (cool) air in a cabin.

The amount of the above mentioned loads depends o[illegible] the boundary conditions such as automobile interior and exte[illegible]face temperatures, air velocity and solar load. Therefore fo[illegible]bin convection and conduction load analysis the cabin was di[illegible]ed into six separate sections included windshield, re[illegible]window, [illegible] windows, doors, roof and floor. For each of th[illegible]ention[illegible] sections, the effective parameters such as air[illegible]ocity, t[illegible]peratur[illegible] material specifications, slope and type of m[illegible]ial w[illegible]uted.

In estimating the thermal loads res[illegible]ting [illegible]nvectio[illegible]nd conduction heat transfer, the typica[illegible]uivalent [illegible]mal resistance between outside and inside temp[illegible]es as is sho[illegible] Fig. 2 was applied.

Considering the radiati[illegible] absorbed [illegible]exterior surfaces, the thermal balance at a po[illegible] (node) with $T_{s,o}$ [illegible]mperature (at the exterior surface of the[illegible]icle) [illegible]n be express[illegible]d as:

$$q''_{conv,out} + \alpha_S G_S = q''_{cond} \quad (17)$$

Alternatively i[illegible] equiv[illegible]t form[illegible]

$$h_o(T_{out} - T_{s,o}) + \alpha_S G_S = \frac{T_{s,o} - T_{s,i}}{\sum \frac{x_i}{k_i}} \quad (18)$$

where h_o i[illegible] outside air heat transfer coefficient.

From the [illegible]y balance at a node with $T_{s,i}$ temperature,

$$q''_{cond} = q''_{conv} \quad (19)$$

or in an equivalent form:

$$\frac{T_{s,o} - T_{s,i}}{\sum \frac{x_i}{k_i}} = \frac{T_{s,i} - T_{in}}{\frac{1}{h_{in}}} \quad (20)$$

where q''_{conv} is the heat flux exchanged between the vehicle exterior surface metal (with $T_{s,o}$ temperature) and the ambient air temperature (T_{out}).

The outside air forced convection heat transfer coefficient h_o for horizontal exterior surfaces (cabin's roof and floor) was estimated from the following empirical equation [11]:

$$\mathrm{Nu} = \left[0.037(\mathrm{Re})^{0.8}\right](\mathrm{pr})^{0.33} \quad (21)$$

For the natural convection on horizo[illegible] inter[illegible]urfaces such as a roof, air in a cabin is under the s[illegible]e, hence, t[illegible]emperature of the roof is higher than the temper[illegible] of the ai[illegible] a cabin, then for surfaces heated from th[illegible]ove [illegible]

$$\mathrm{Nu} = 0.27\mathrm{Ra}^{1/4} \quad (22)$$

and for the natural co[illegible]io[illegible] horizontal internal surfaces such as cabin's floor, f[illegible]hic[illegible] the ca[illegible] is above the floor and its temperature i[illegible]ver than th[illegible]or [illegible] number was estimated from [11]:

$$\mathrm{Nu} = \begin{cases} 0.54\mathrm{Ra}^{1/4} & 10^4 \leqslant \mathrm{Ra} \leqslant 10^7 \\ 0.15\mathrm{Ra}^{1/3} & 10^7 \leqslant \mathrm{Ra} \leqslant 10^{11} \end{cases} \quad (23)$$

[illegible] the natural convection on inclined surfaces such as wind screen [illegible]eld and rea[illegible]indow:

[illegible]r Ra > 10[illegible]

$$\mathrm{Nu} = \left\{0.825 + \frac{0.387(\mathrm{Ra})^{1/6}}{\left[1 + \left(\frac{0.429}{\mathrm{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 \quad (24)$$

And for Ra < 10^9:

$$\mathrm{Nu} = 0.68 + \frac{0.67\mathrm{Ra}_L^{1/4}}{\left[1 + (0.492/\mathrm{Pr})^{9/16}\right]^{4/9}} \quad (25)$$

3.3. The sensible and latent loads of passengers

The human thermal load was estimated from [12]:

$$\dot{Q}_{human} = \mathrm{np}(\mathrm{SH} + \mathrm{LH}) \quad (26)$$

where, np is the number of passengers, SH is the sensible heat and LH is the latent heat for the light activity (sitting situation for passengers).

Furthermore there is an increase in humidity produced by passengers due to the three factors m_{res}, m_{sw} and m_{diff}:

m_{res} is the respiration factor which is a function of metabolic rate (M) and was obtained from the following experimental relation [13]:

$$\dot{m}_{res} = 0.43 \times 10^{-6}\left(\frac{\mathrm{kg\ m^2}}{\mathrm{J}}\right)\cdot M \quad (27)$$

$\dot{m}_{sw}$ is directly proportional to the latent thermal losses from the skin, related to the body temperature regulating mechanism, this parameter was estimated from:

$$\dot{m}_{sw} = \frac{E_{sw}}{i_{fg}} \quad (28)$$

where the latent thermal losses from the skin (sweating thermal load, E_{sw}) was computed from [14]:

$$E_{sw} = 0.07(M - 58.12)\cdot A_D \quad (29)$$

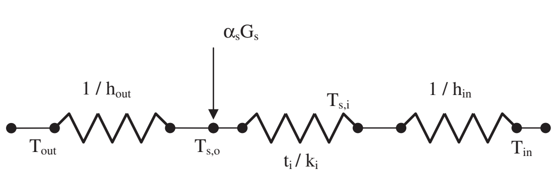

Fig. 2. Equivalent thermal resistance of bodywork for an automobile.

$\dot{m}_{\text{diff}}$ is the spread water vapor mass flow rate from the skin, determined by diffusive thermal load E_{diff}, divided by the latent heat of the saturated water at the skin temperature (i_{fg}):

$$\dot{m}_{\text{diff}} = \frac{E_{\text{diff}}}{i_{\text{fg}}} \tag{30}$$

The procedure of computing E_{diff}, requires some input parameters obtained from experimental data such as thermal resistance of clothes (I_{cl}), vaporizing heat transfer coefficient, and perspiring rate. A proposed equation for computing E_{diff} is [14]:

$$E_{\text{diff}} = 0.508 \times 10^{-3}(p_{\text{wl}} - p_{\text{wa}}) \cdot A_{\text{D}} \tag{31}$$

where p_{wa} is the corresponding saturated pressure of water vapor at the cabin air temperature, and p_{wl} is the corresponding saturated pressure of water vapor at the skin temperature derived from:

$$p_{\text{wl}} = 256T_{\text{sk}} - 3373 \tag{32}$$

where T_{sk} is the skin temperature of one person which has the following relation with the metabolic rate of that passenger:

$$T_{\text{sk}} = 35.7 - 0.02275M \tag{33}$$

In case of existing perspiring rate and ambient temperature in vicinity of human thermal comfort zone, it can be assumed that the perspiration will be totally vaporized. Hence, the resultant equation for estimating the water vapor produced by passengers is:

$$\dot{m}_{\text{v,human}} = \text{np} \times (\dot{m}_{\text{res}} + \dot{m}_{\text{sw}} + \dot{m}_{\text{diff}}) \tag{34}$$

The humidity load from passengers increases the relative humidity of the air in a cabin and may cause a non-comfort air [illegible]dition in the cabin which should be taken care of by the cabin [illegible] conditioning system.

3.4. Solving the system of governing conservation eq[illegible]

Four coupled nonlinear ordinary differenti[illegible] equatio[illegible] including Eq. (1) (mass conservation of dry air in a c[illegible]) E[illegible] vapor mass conservation for air in cabi[illegible] Eq. (7) [illegible]rgy balance for air in cabin) and Eq. (9) (energy ba[illegible]e of intern[illegible]mponents in the cabin), constructed the lum[illegible] model of [illegible]ysis for the cabin air. Four differential [illegible]uation[illegible] ere solved simultaneously using a classical f[illegible]h-order Ru[illegible]kutta method to obtain the variation of dr[illegible] mass absolute [illegible]idity, enthalpy, and temperature of air [illegible]he cab[illegible]t each time step.

The relations of abs[illegible] hu[illegible]ity with temperature for air in a cabin are as follows:

$$T_{\text{in}} = \frac{i - 25\text{[illegible]}\,\omega}{1.00\text{[illegible]}1.805\text{[illegible]}} \tag{35}$$

where, i is [illegible] a cabin.

4. Fuzzy control[illegible]

Fuzzy controller can model the nonlinear relation between inputs and outputs [15]. Because of the nonlinear nature of differential Eqs. 1, 4, 7, 9 describing the desired output (temperature) their use of coefficients varying with time and the complex behavior of the system, conventional controllers like PID are not practical [3].

A controller is being applied to adjust the air temperature (set by passengers) in a cabin. Therefore, the difference between the real temperature and the adjusted one (temperature error) is an input to the controller. Another input signal for the controller may be the air pollution level in a cabin by controlling the amount of circulated air. The block diagram of such a controller is illustrated in Fig 3.

The error of input temperature is defined as:

$$\text{error} = T_{\text{cabin}} - T_{\text{desired}} \tag{36}$$

where T_{desired} is the desired temperature adjusted by passengers and T_{cabin} is the cabin's actual air temperature (obtained by solving the set of Eqs. 1, 4, 7, and 9) which changes due to the existing air conditioning system.

The pollution factor (a number between 0 and 1) determines the level of outside air pollution. Pollution factor equals to 1 represents the maximum pollution and equals to 0 means clean air. Relative clean air is defined for a controller as the air with a pollution factor of less than 0.4, while a relative pol[illegible]air has a factor of 0.4 and more. In this section, four stag[illegible]ap[illegible]g a fuzzy controller system are discussed.

4.1. First step: fuzzification

The first step is to d[illegible]mine [illegible] physical i[illegible]puts and attribute proper degree to in[illegible]ts b[illegible]bership [illegible]ctions. Considering a system with six [illegible]s each [illegible] depe[illegible]ncy on resultant inputs conversions to [illegible]re between [illegible]. These six rules are:

(1) If the tempera[illegible]error is small and the pollutant level is [illegible]en the blow[illegible]ower and the circulated air proportion [illegible] be low.
[illegible] If the temperature error is medium and the pollutant level is low, then [illegible] blower power and the circulated air proportion [illegible]ill be me[illegible]um.
(3) [illegible]erature error is large and the pollutant level is low, [illegible] the blower power and the circulated air proportion will be high.
[illegible]he temperature error is small and the pollutant level is high, then the blower power will be low and the circulated air proportion will be very high.
(5) If the temperature error is medium and the pollutant level is high, then the blower power will be medium and the circulated air proportion will be very high.
(6) If the temperature error and the pollutant level are high, then the blower power will be high and the circulated air proportion will be very high.

It is worth mentioning that if the pollution level is not defined in the first moments of operation, the controller will maximize the amount of circulated air because of large temperature error and as a result, the rate of fresh air would be minimized. As time passes gradually, due to the error reduction, the amount of fresh air increases and when the temperature of the cabin reaches the desired level, the rate of fresh air entering the cabin will reach its highest level. Hence, if the pollution level of outside air (like smelly gases and smoke) is high, it can have dangerous effects on the cabin passengers.

Each of the above concepts such as small error and high power of blower have to be clarified, by using triangular membership functions presented in Figs. 4–7:

As illustrated in Figs. 4–7, the range of variation in variables can be shown as in the following numbers:

Range of temperature error variation: −6 to 25.
Range of pollution level variation: 0–1.

4.2. Second step: fuzzy inference

When the inputs are fuzzificated, the degree for each section will be found which satisfies the rules. Here Min Max method used for valuing the above if-then rules.

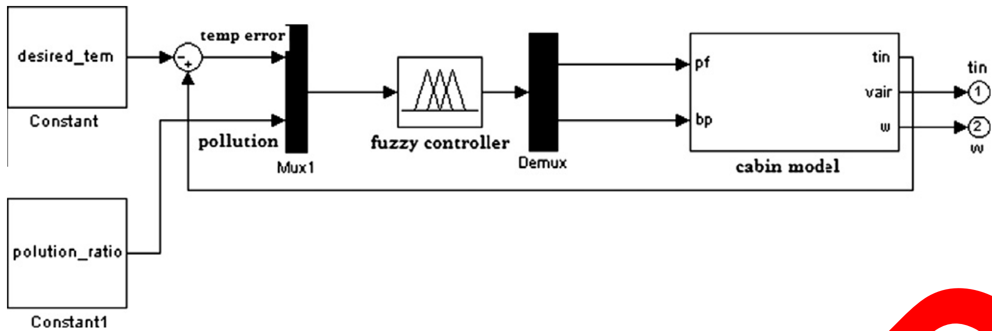

Fig. 3. Using fuzzy controller as well as temperature feedback in air conditioning system of automobile.

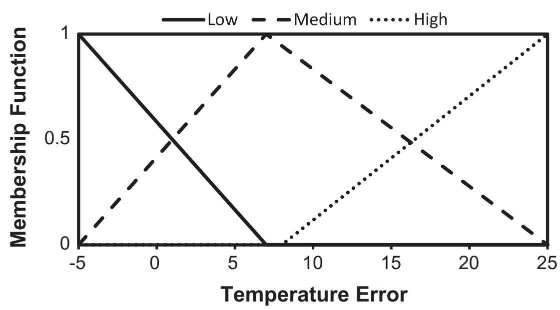

Fig. 4. Membership functions for fuzzy controller input (temperature error).

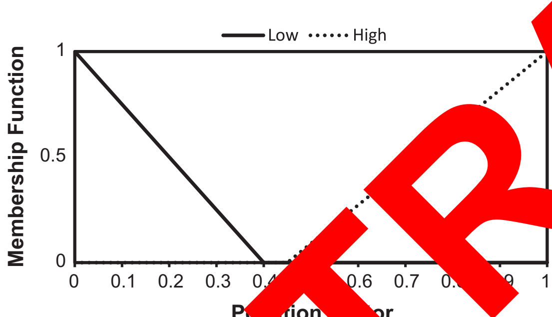

Fig. 5. Membership functions for fuzzy controller input (pollution level).

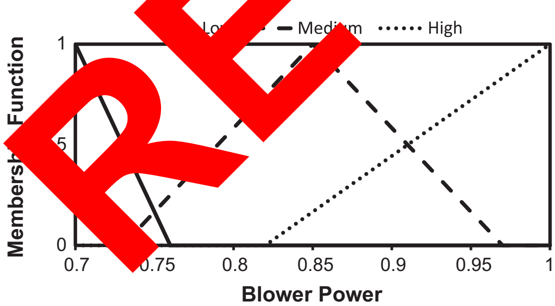

Fig. 6. Membership functions for first output of fuzzy controller (blower power).

4.3. Third step: blend and make

Each fuzzy inference about air conditioning system of automobile will lead to a certain command. Here the first rule reduces the blower power and the circulated air proportion. The second rule maintains the magnitudes at a medium level and the third rule increases them. Finally the maximum results among the defined rules will be implemented.

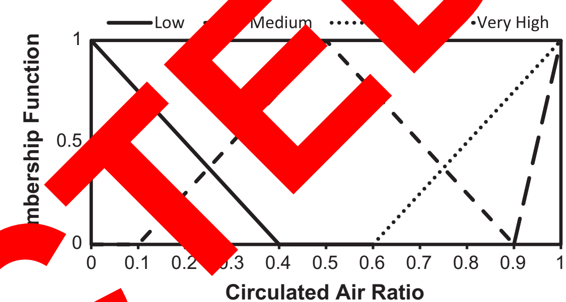

Fig. 7. Membership functions for second output of fuzzy controller (proportion of circulated air).

4.4. Forth step: defuzzification

The fuzzy results drawn from previous steps cannot be run by the output system and they should be defuzzificated. In Fig. 8, every three rules are put together to show how each rule attributed and blended to a fuzzy collection which its membership function allocates a weight to each output.

5. Case study

In order to analyze and evaluate the accuracy of the results for the proposed cabin thermal model, an automobile with the characteristics of its cabin and air conditioning system components shown in Table 1 was selected. Fuzzy controller has been used for making thermal comfort under Mashhad weather conditions with pollution factor of 0.3, without any passengers at a constant velocity of 40 km/h.

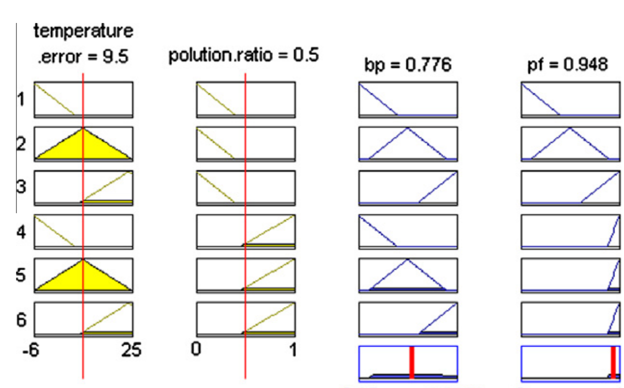

Fig. 8. Control of operation and inference of rules in the case the input error is 20°.

Table 1
The geometrical specifications of components for an automobile in hot room test.

Title	Type	Geometry
Cabin sizes		2300 mm × 1300 mm × 1190 mm
Compressor displacement volume	Rotary vane	120 cc
Condenser	Parallel flow	545 mm × 319 mm × 16 mm
Expansion valve	Thermostatic	
Evaporator capacity	Laminated	2600 W
Engine displacement volume		1300 cc

6. Results and discussion

6.1. Model verification

In order to verify the numerical results of the simulation model, a real automobile with specifications introduced in Section 5 and Table 1 was tested at the following conditions:

(1) Initial air temperature of the cabin was maintained at 60 °C.
(2) The data were recorded 10 min after the car cooler was powered on.
(3) The power mode of the blower was adjusted at high rate.
(4) The velocity of the car was maintained at 40 km per hour.
(5) Measurements of temperature and absolute humidity were taken at the suction vent of the air conditioning system.
(6) Measurement of temperature was taken place at the front dashboard air inlets.
(7) Number of passengers inside the cabin was zero.
(8) Data was recorded at every 2 s for 10 min.

Fig. 9 shows the cabin air temperature variatio[illegible] time. I[illegible] the first 10 min of the starting period, there w[illegible] abo[illegible]3.39 °C average difference value between modeling a[illegible] experi[illegible]tal cabin air temperatures. Regarding the fact that [illegible] mode[illegible] sumed that the temperature is not de[illegible]dent [illegible]cation and average temperature is used (based o[illegible]mpact cap[illegible]) and with respect to thermal loads predictio[illegible]ncertainty o[illegible]asuring devices and variation of outside te[illegible]pera[illegible] the error seen in this figure is acceptable.

6.2. Temperature

Fig. 10 show[illegible] var[illegible] of [illegible] air temperature of the evaporator d[illegible]g a[illegible]onditi[illegible]art up. In the first 20 s, the slope of t[illegible]eratur[illegible] eduction [illegible] steep. This is due to the high temperat[illegible]f ai[illegible] the evaporator tubes at the first stages of coo[illegible]ocess which leads to chilled air with a temperature of 12 °C. [illegible]r 20 s, the temperature changes very slowly (because of therm[illegible]ads increase) so that the air temperature coming out of the evaporator ranges from 7 °C at the end of 20 s to 5 °C finally. Thus, the mean temperature of the cabin does not change after this stage. The temperature distribution over the cabin versus time is shown in Fig. 11.

6.3. Humidity

The variation of absolute humidity with time is similar to temperature distribution which is shown in Fig. 12; On the other hand, the absolute humidity of the evaporator exit air is a function of temperature of the outlet air. As a res[illegible] observed that the absolute humidity declines until it [illegible]ches to [illegible]ecified value. The variation of absolute humidity [illegible] time shows [illegible]imilar trend as in the evaporator exit air temperat[illegible]

In Fig. 13 the relative hu[illegible]y varia[illegible] versu[illegible]me is illustrated. By reducing the air [illegible]perature o[illegible], the relative humidity increases. Te[illegible]ature [illegible]uction [illegible]tinues until the 100th second. The reduc[illegible] of [illegible]ative humidity continues until t = 100th and chan[illegible] very [illegible] after th[illegible]

One of the m[illegible] factors in [illegible] [illegible]mfort control for automobile passenge[illegible] velocity of t[illegible]nning air through the cabin. This parame[illegible] is co[illegible]lled by a blower. Fig. 14 shows the variation of the volumetric flow [illegible] of air passing through the blower. In the first [illegible] of simulation, [illegible]ause of the large difference between the [illegible]sired temperature and the real temperature of the cabin (here is [illegible] °C), the vol[illegible]etric flow rate of air passing through the blower is a[illegible] maximum [illegible]vel (the range of temperature variation has been defi[illegible] from −[illegible] 25 for controller and in this situation, temperature d[illegible] about 10 °C at the beginning of the simulation. [illegible] the maximum power of the blower is 0.103 m^3/s not the real max[illegible] 0.24 m^3/s). Reducing this temperature difference gradually, [illegible]ntroller will decrease the volumetric rate of air. One of the advantages of this controller is the prevention of hitting high velocity air to the passengers' face (storm phenomenon).

The other controlled parameter is the amount of circulated air (or the fresh air). Two cases were considered in this paper.

The first case: As shown in Fig. 15 the pollution level of the surrounding ambient is low and the amount of circulated air reduces with time (the percentage of fresh air increases).

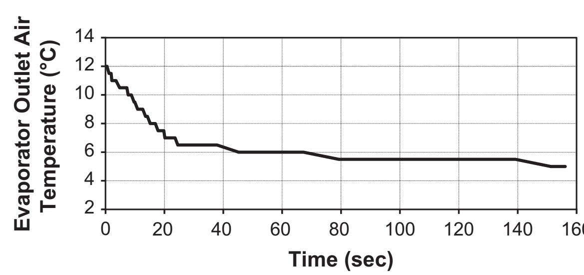

Fig. 10. Variation of exit air from evaporator.

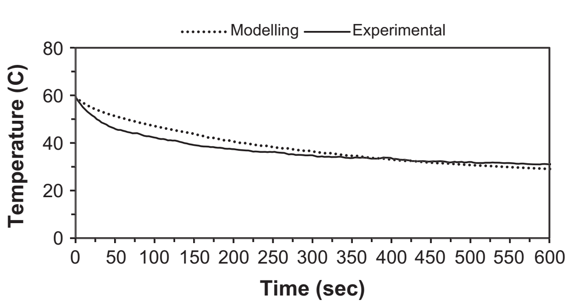

Fig. 9. Comparison between results drawn from modeling and experimental tests.

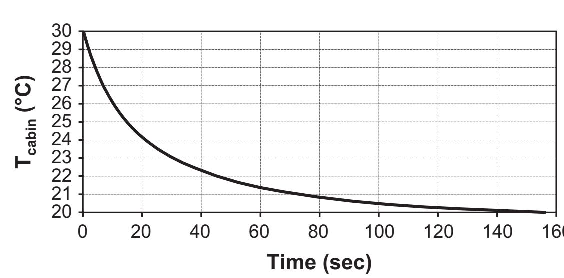

Fig. 11. Distribution of air temperature inside cabin of automobile.

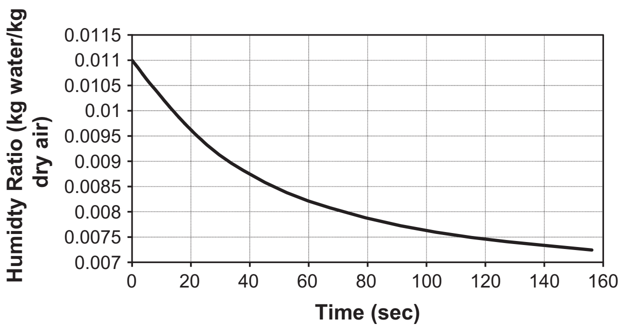

Fig. 12. Variation of absolute humidity of air inside cabin.

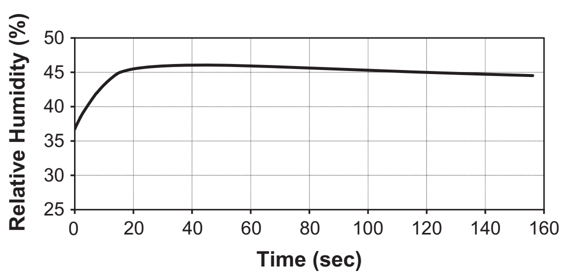

Fig. 13. Variation of relative humidity of air inside automobile's cabin.

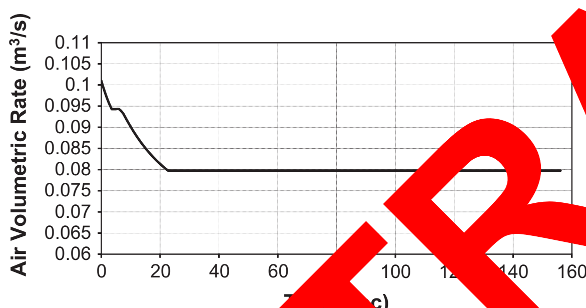

Fig. 14. Variation of air vo[illegible]metric rate at [illegible]lower outlet (m^3/s).

The higher perce[illegible] points [illegible]rculation, due to the evaporator lower thermal load, [illegible]s temperature decreases faster. At the start up (for exampl[illegible] about [illegible]°C temperature difference), the fuzzy c[illegible] adjust[illegible] the [illegible]ximum circulated air percent points t[illegible]% (w[illegible]e tempe[illegible]re difference was about 25 °C – and o[illegible]e air t[illegible]erature was 45 unit, the desired temperature was 20-u[illegible] th[illegible]pe[illegible]vel at the beginning was about 100% circulated a[illegible] With reaching to a constant temperature difference, the amount o[illegible]lated air did not change (43%).

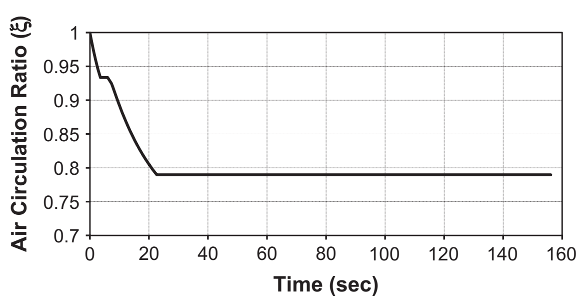

Fig. 15. Percentage of circulated air versus time of simulation (low pollution).

Second case: the pollution level of the surrounding ambient was high: in this case, the fresh air percent point was at the minimum level (for prevention of dust and other pollutants from entering the cabin) and consequently the proportion of circulated air is at its maximum level. The variation of circulated air percent point is shown in Fig. 16.

Due to high rate of pollution, controller keeps the amount of circulated air at the high level and the vents for fresh air were approximately closed. (Pollution factor in this case is 0.95, about 96% of air is circulated and 4% is fresh air). At this situation the cabin mean temperature reduces as a result of more [illegible]ulated air entered to cabin.

To clarify the role of fuzzy con[illegible]er, the va[illegible]ion of air temperature inside the cabin was inv[illegible]ated witho[illegible] using a fuzzy controller. Initial conditions [illegible] either [illegible]s were t[illegible] same (Tehran climate, velocity of car: 4[illegible]m/h, cool[illegible] capa[illegible] 2.623 KW and without passengers). In [illegible] case w[illegible]hout th[illegible] of a fuzzy controller, the air volumetri[illegible] rate [illegible] the blower outlet and circulated air ratio were adj[illegible]ted [illegible] m^3/s a[illegible].5 respectively. The Result is shown i[illegible]g. 17. It i[illegible]own i[illegible]s figure as well as Table 2 that the am[illegible]f time for c[illegible]e cabin air has decreased by 140 s wh[illegible]sin[illegible]e controller.

The advantages [illegible]plying a fuzzy controller in this simulation are:

(1) Results showed that for applying a fuzzy controller, less time was ne[illegible]d for the system to regulate the temperature of the cab[illegible] to the desired one (23 °C). This decreases the [illegible] duration of air conditioner which causes a [illegible]stantial effect on the automobile fuel consumption rate.
(2) The exit velocity of the blower can be adjusted automatically [illegible]o minimize noise and storm effects.
(3) A more uniform variation of air temperature can be achieved and the controller can reduce the difference between the inside air temperature and the desired temperature of passengers by regulating the power and fresh air vent in each moment.

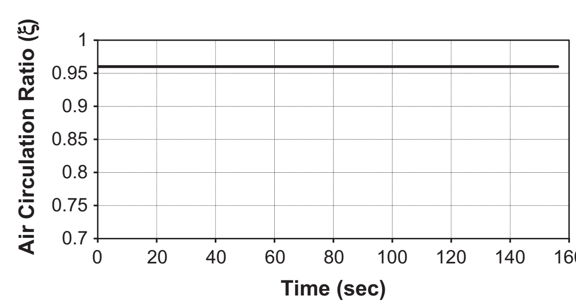

Fig. 16. Circulated air percent point versus time of simulation (high pollution).

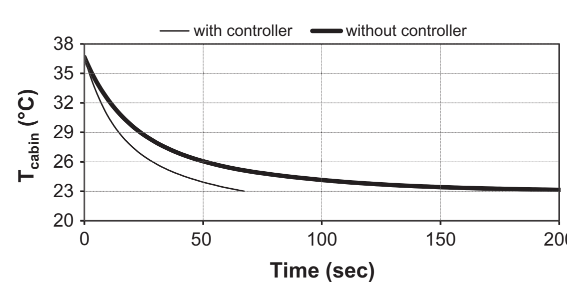

Fig. 17. Comparison of temperature variation of air inside car's cabin with and without controller.

Table 2
Comparison of an automobile air conditioning system with and without controller for time needed to regulate the temperature of a cabin to a comfortable temperature (Tehran climate, 40 km/h and without passenger).

Type of air conditioning system	Time needed to regulate temperature to desired 1 pa (23 °C)
Without controller	200 s
With controller	60 s

(4) The thermal comfort of passengers was provided at the steady state situation.

7. Conclusions

Thermal comfort is one of the most important comfort factors. Automobile cabin thermal environment is complex and continually varies during time. In this paper thermal comfort is studied for a cabin. First, modeling of air handling system, thermal loads model inside the cabin and cabin modeling were introduced in order to obtain temperature and relative humidity of air inside the cabin. Then, a fuzzy controller with varying two effective parameters (blower outgoing air velocity and the circulated air percentage) was applied to the cabin model for controlling the inside cabin air temperature. Results showed that through applying a fuzzy controller, less time was needed for the system to regulate the temperature of the cabin to the desired one (20 °C). This decreases the operation duration of the air conditioner which causes a substantial effect on the automobile fuel consumption rate.

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