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Original Research Article

Analysis of the Thermal Comfort Parameters in the Passenger Cabin of a Bus by Finite Element Method (ANSYS)

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Abstract

Uniform temperature distribution in air conditioning systems which is used in the cooling of vehicles cabin with regards to providing thermal comfort to individuals is an important parameter. In this regard, appropriate computer simulations of these systems on the design pharse provide a significant convenience to the success of the design. In this paper, thermal comfort parameters depending on pressure and velocity distribution provided with air condition system of in the bus having 49 passengers including the driver were analysed by using finite elements method (ANSYS). In this analysis, determination of the time required was aimed to reach ≈293 K value with uniformly temperature distribution from initial temperature value of ≈310 K inside the bus. A finite element program is developed considering the thermal load of 49 passengers by using thermal mannequin model. Thus, the internal conditions in bus were stimulated to real conditions. As a result, it was determined that interior temperature of the bus cabin reached to ≈293 K value with uniformly distribution in 15 minutes according to results of obtained in range of 5 minutes with the real condition scenario in ANSYS program.

Keywords: Bus, Air Conditioning, Finite Elements Analysis

Nomenclatures

A	Constant for sited manikin (W/m^{2}K)
A_{CL}	Surface area clothed manikin (m^2)
Ar	Effective radiation area of manikin (m^2)
A_U	Surface area unclothed manikin (m^2)
C,D	Calibration constant for sited manikin and unclothed manikin (W/m^2 K)
CFD	Computational fluid dynamics
ε	Surface emissivity (0.95)
ENDS	Turbulance dissipation rate
ENKE	Turbulent kinetic energy
F_{CL}	Reduction factor for sensible heat exchange of dressed manikin
f_{CL}	Clothing area factor
f_r	Radiation area factor
q''	Body heat gain or loss (W/m^2)
$q''_{T,cal}$	Dry heat loss for the homogenous standard environment (W/m^2)
q''_T	Total heat transfer (W/m^2)
q''_c	Convective heat transfer (W/m^2)
q''_r	Radiative heat transfer (W/m^2)
H	Characteristic length (m)
HVAC	Heating,ventilating and air condition
h_{teq}	Heat transfer calibration coefficient for dressed manikin (W/m^2 K)
h_c	Convective heat transfer cofficient (W/m^{2}K)
h_{cal}	Heat transfer imaginary calibration coefficient for sited manikin (W/m^{2}K)
h_r	Radiative heat transfer (W/m^2)
I_{cl}	Intrinsic clothes insulation
ρ	Fluid density (kg/m^3)
PMV	Pressure maintaining valve
P_{res}	Relative pressure
R_T	Total clothing insulation (m^{2}K/W) with winter/ summer clothes
R_{cl}	Intrinsic clothing insulation for winter/ summer clothes (m^2 K/W)
t	Time (s)
t_a	Ambient air temperature (°C)
t_{eq}	Equivalent temperature (°C)
$t_{\bar{r}}$	Mean radiant temperature (°C)
t_s	Surface temperature (°C)
t_w	Mean ambient temperature (°C)
u_0	Characteristic velocity(m/s)
u_j	Fluid velocity component in direction xj (m/s)
v	Air speed (feet/min)
v_a	Air speed (m/s)
V_x	Velocity in the X direction
V_y	Velocity in the Y direction
σ	Stefan-Boltzmann constant (5,669 * 10^{-8}) (W/m^2 K^4)
Γ	Diffusion coefficient
μ_T	Turbulent (eddy) viscosity (Pa.s)
$S_Ø$	Source term
Ø	Average numerical variables

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1. Introduction

Conditions of comfort provided by the air conditioning system is increased the demand on the vehicles which use air conditioning system. Demands of passengers and drivers for travel with the comfort of air conditioning is increased necessity of vehicles having air-conditioned. There are two main basic parameters as window sizes and power which affect to values of thermal asymmetry of air conditioning system in cabin. Usually, drivers in the sunny summer days are exposed to serious radiation. The surface temperature inside the bus in winter condition is low, and leads to heat loss of the driver and passengers. The heat loss should be compensated to improve comfort by high temperatures in the cabin. High power is required to improve the air quality in the cabin. The thermal comfor in the bus cabin can become very complex in sunny or cold weather conditions, when it is combined with asymmetric radiation load with incoming on drivers and passengers from windows [1,2]. Today, the increased comfort demands inside indoor space and vehicle cabin has been raised attention to optimum energy consumption. As a result, the used parameters in solutions that are found for different climate and ventilation should be predicted to use effect on humans, even in the design phase. Computational fluid models (CFD) and thermal mannequin methods, provide a new approach for the design, evaluation stages and analysis of environmental impacts on humans. Here, the effects of heat exchange of the human body are important factors for the detection of the heat balance and thermal conditions.

Evaluation of the thermal environment in a modern office or a car can create difficulties because of the complex interactions of person and external environmental factors (i.e radiation, air temperature and movement) and the ventilation system. In addition, laboratory measurements include several methodological problems. The evaluation of thermal comfort provided by air-conditioning panels is a good way to learn about the effects of different thermal climatic conditions. Nevertheless, it is a time consuming, expensive and difficult process in terms standardization of air conditioning panels and completion of characteristics. Measurement of parameters related to human is cheaper and repeatable, and so it is widely used. However, different sensors are required for experimental measurements. In a suitable approach, human-shaped thermal mannequins can be used in order to measure temperature changes on the body in terms of compliance with the actual conditions by heating wires and temperature sensors [3,4,5,6]. Heat flow sensors are placed in the thermal mannequin's entire surface and used to measure the local heat flux. The thermal mannequin method has been used for several years in different ways. The new approach in the use of thermal mannequins, is to use a virtual environment simulations along with mathematical models [7].

Burch et.al (1991) are investigated to thermal confor parameters of automobile in very cold winter conditions (≈ -20 °C). A mathematical model have developed for mass and heat transfer between environment and human body which is separated to four sections such as head, body, arms and legs [8,9]. Kayfeci et.al (2009) are investigated use of alternative air condition systems, instead of vapor compression air conditioning system in bus [10]. Ingersoll et. al (1992) are calculated to thermal comfort parameters of bus cabin taking into account parameters such as temperature and relative humidity of circulated air [11]. Chiou et.al (2012), a study based on numerical prediction are performed on the death rates of passengers and drivers which are exposed to particulate substances [12]. In this paper, thermal comfort parameters depending on pressure and velocity distribution which was provided with air condition system on the bus having 49 people including the driver were analysed by using finite elements method (ANSYS) in this paper. In analysis which was performed with ANSYS program by using thermal fmannequin model, it was determined that indoor temperature of the bus reached ≈293 K value with homogenously distribution in about 15 minutes.

2. Materıal and Method

The uniform temperature distribution in terms of thermal comfort in the bus cabin is a difficult phenomenon. Therefore, the internal volume of cabin and the number of passengers is greater significantly according to an automobile. Also, a bus has large glass surface areas although very variable weather conditions. Furthermore, air condition systems is not designed as appropriate to the climatic conditions, whereas climatic conditions should be considered for design. High air temperatures such as desert environment creates high humidity in tropical climates. Air conditioning systems should work in the desired manner under conditions to air temperature reaching up to 50°C. Air conditioning systems designed this way is more functional for eliminating the cooling load in a humid climate, rather than the dry cooling load in a warm climate. In addition, internal car air quality should also be considered.

Generally, air conditioning systems used in buses is tested in small rooms that can be controlled. The bus air conditioning units is tested in larger rooms which represent all volume of buses after the assembly process [13].

2.1. Heating loads

The main factors for design of air conditioning systems in buses are described below;

- ➢ Total inner volume of bus, number of passengers, duration of stay in bus of passengers and distance between stations,
- ➢ Outside air conditions (i.e. temperature, humidity and solar radiation),
- ➢ Size and optical properties of windows,
- ➢ Thermal properties of materials used, dimensions of the bus body and thermal properties of bus (i.e. temperature, humidity and speed of bus),
- ➢ Cooling and heating loads of bus as given in following,
  - Heat flow from the walls (floor, the ceiling and the sides),
  - Heat flow from glass surfaces (side, front and rear glass),
  - Heat differences between internal and external air, heat loads of passengers temperature of engine, heat of air condition system and heat flow through the vents.

Technical specifications of bus which are used in this paper are given in Table 1.

Table 1. Technical specifications of bus

Tecnical specification	Value
Width of bus	2550 mm
Lenght of bus	11857 mm
High of bus	3090 mm
Lenght of seats	580 mm
Width of seats	430 mm
Height from the floor of the bus	350 mm
Heat transfer coefficient of bus body	1.025 W/m^{2}K
Heat transfer coefficient of windows	6.146 W/m^{2}K
Heat transfer coefficient of ceiling	1.489 W/m^{2}K
Heat transfer coefficient of base	2.035 W/m^{2}K

The following values are acceptable for interurban buses [14];

- Number of passengers: 50 people
- Insulation material thickness: 25-40 mm
- Windows colorful and double-decker
- Airflow rate in cabin:190 l/s
- Vehicle speed : 100 km/h
- Internal temperature, dry bulb: 27 C , wet bulb:19.5 C

2.1. The equations used in the analysis

The surface temperature of passenger can be changed as a function of dry heat loss by using a expression which is derived based on Fanger comfort criteria [15].

$$t_s = 36.4 - 0.054.Q_t \tag{1}$$

$$t_{eq} = t_s - 0.155.I_t.Q_t \tag{2}$$

Determination of t_{eq} with equations for convection and radiation; Equations for convective and radiative heat transfer based on clothes situations. For naked situations $F_{CL} = f_{CL} = 1$.

$$R = F_{CL} f_{cl} h_r (t_s - \bar{t_r}) \tag{3}$$

$$C = F_{CL} f_{cl} h_r (t_s - t_a) \quad (4)$$

Equivalent temperature ($t_{eq} = t_r = t_a$) can be calculated by assigning different values as represent to h_r and h_c for homogeneous status having zero air speed. This is same R+C with the actual situation without the no effect. A addictive between h_r and h_c can be considered at different temperature differences and air speeds. Generally, heat transfer coefficients in h_r and h_c in the non-homogeneous environment depend on the difference between surface temperature and teq. Therefore, F_{CL}, f_{cl}, h_r and h_c can be achieved in the same formula. In this case, the total heat loss:

$$R + C = h_r(t_s - \bar{t_r}) + h_c\,(t_s - t_a) \quad (5)$$

By definition as equivalent temperature,

$$R + C = h_r\big(t_s - t_{eq}\big) + h_c\,(t_s - t_a) \quad (6)$$

t_{eq} can be calculated by using Eq.5 and/or 6;

$$t_{eq} = \frac{(hr.\bar{t_r} + hc.t_a)}{(h_r + h_c)} = t_s - \frac{R + C}{(h_r + h_c)} \quad (7)$$

This equation is a fundamental relationship for calculation of teq which is based on the total heat balance equation which as defined by Fanger (1970) [16];

$$PMV = (0.303e^{-0.036M} + 0.028)\{(M - W) - 3.05\,.\,10^{-3}[5733 - 6.99\,(M - W) - p_a = 0.42\,[M - W) - 58.15] - 1.7.\,10^{-5} M(5867 - p_a) - 0.0014M(34 - t_a) - 3.96.\,10^{-8} f_{cl}[(t_{cl} + 273)^4 - (\bar{t_r} + 273)^4] - f_{cl} h_c (t_{cl} - t_a) \quad (8)$$

$$t_{cl} = 35{,}7 - 0{,}028.\,(M - W) - Icl.\{3{,}96.10^{-8}.f_{CL}.[(tcl + 273)^4 - (\bar{t_r} + 273)^4] + f_{cl}.hc.(tcl - t_a)\} = 36.4 - 0.054Q_t \quad (9)$$

Equations developed by Bedford (1936) are achieved, when is include the breathing and heat loss to t_{eg} definition [17].

$$t_{eq} = 0.522t_a + 0.478t_w - 0.0147\sqrt{v}(100 - t_a) \quad (10)$$

When Eq.10 is revised in terms of °F by using global temperature in tg;

$$t_{eq} = 0.522t_a + 0.478t_g + \sqrt{v}(0.0808t_g - 0.0661t_a - 1.474) \quad (11)$$

Vector-valued functions should be calculated in terms of surface integration. The local equivalent temperature $[\bar{t_{eq}}(s)]$ can be calculated as average surface with Eq.12;

$$\bar{t_{eq}}(s)|s| = t_s s \iint q''(\mathrm{r})\,d_s - \frac{1}{h} s \iint q''(\mathrm{r})\,d_s \quad (12)$$

The heat transfer coefficient in homogeneous medium can be calculated by Eq.18;

$$h_{cal} = \frac{q''_{\mathrm{T,cal}}}{(t_s - t_a)} \quad (13)$$

In assessing the environment data;

$$t_{eq} = t_s - \frac{q''_T}{h_{teq}} \quad (14)$$

Where, h_{teg} represent in h_{cal} value. Convective (q''_c) and radiative (q''_r) in the general case where the heat transfer code is present at the same time, the total of the heat flux (q "T) is sum of components in following equation;

$$q''_T = q''_c + q''_r = f_{cl}.h_c.(t_{cl} - t_a) + f_{cl}.h_r.(t_{cl} - \bar{t_r}) = h_{cal}.\big(t_s - t_{eq}\big) \quad (15)$$

Where,

$$f_{cl} = 1 + 1{,}97.R_{cl} = 1 + 0{,}3.I_{cl} \quad (16)$$

Where in the convective heat transfer coefficient between the wall defining the flow temperature environment h_c is given by the following equation;

$$h_c = \frac{q''_c}{f_{cl}(t_{cl} - t_a)} = A.v_a + C(t_{cl} - t_a) + \mathrm{D} \quad (17)$$

Determining the radiative heat flux between walls and environment optimized linear radiation heat transfer coefficient h_r is calculated as follows;

$$h_r = \frac{q''}{f_{cl}(t_{cl} - t^{-}{}_{r})} = 4.\mathcal{E}.\sigma.f_r.\{273.15 + \frac{t_{cl} - t^{-}{}_{r}}{2}\}^3 \tag{18}$$

The heat transfer coefficient in homogeneous medium can be calculated by Eq.18;

$$h_{cal} = \frac{q''_{T,cal}}{(t_s - t_a)} \tag{19}$$

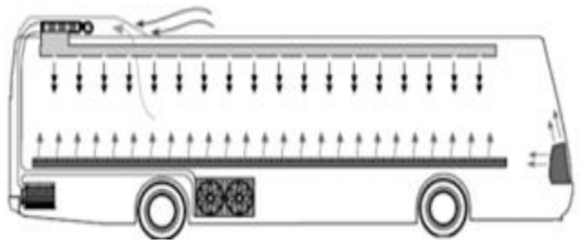
In assessing the environment (value of h_{cal} which is recorded as h_{teq}) data;

$$t_{eq} = t_s - \frac{q''_T}{h_{t_{eq}}} \tag{20}$$

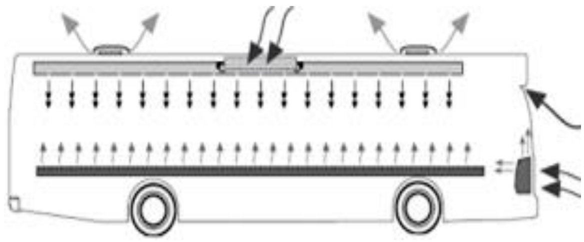
Information in more detail about equations which is used in paper can be seen in the reference [18].

2.3. Design Factors

Fig.1 (a) is shown the mounting locations of components of the air conditioning system on buses traveling between cities. Air conditioning compressor is generally driven by motion received from the motor as shown in figure. Engine and ventilation systems in some buses is mounted as independently from each other. Thus, a constant and stable air conditioning performance can be ensured. But, two independent systems from each other are more complicated and more expensive. This configuration is increase to maintenance costs of bus, and also more space is necessary. Evaporator can be placed to ceiling of bus. Air distribution channels are placed in a secret manner in the interior ceiling. The air pumped by the evaporator enters to the parallel distribution channels and the cooled air enters passenger cabin throughout parallel channels from the lower chamber of the cabin. However, formation of strong air currents coming over passengers should be prevented. The air flow speed coming from these vents should not exceed the speed of 4 - 5 m /s [8]. A ventilation which provides renewal of air in passenger cabin is placed on the propeller region of system and regulates the amount of recycling taken into the cabin. Condenser should not be placed to a region in low air pressure or in high temperature. Temperatures in the inner-city buses is greater than the intercity buses under the same weather conditions when considering the possibility of increase in the number of passengers. Passengers spend less time in inner-city buses. In addition, these passengers agree the conditions that provide less comfort. Bus running in regions having temperate climate has only ventilation and heating systems. Fig. 1 (b) is described to mounting of heating and ventilation system in the in-city buses. Air intake vents can be placed under plate of the bus and on parallel air channel that is in middle of bus windshield. But, this placement is not preferred because the change of mean air flow velocity with bus speed. Propellers can be used to release air from the passenger compartment. When an increase in propeller speed is considered, this time complaints about noise should be eliminated. The noise level should be maximum 65 dB at the level of heads of the passengers. This problem can be solved by reducing of fan speed.



(a)



(b)

Fig.1. Air ventilation system **(a)** of intercity buses **(b)** inner-city buses

3. Discussion and Findings

Configuration of seating arrangement for 49 passengers including the driver prepared in the ANSYS (MAPDL) program is given in Fig.2. This configuration is established according to the main dimensions of a typical inter-city bus for 49 passengers including driver.

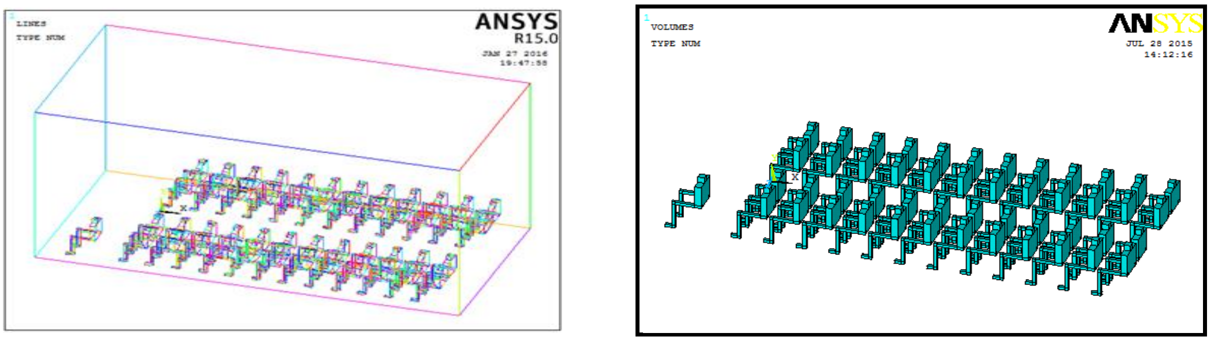
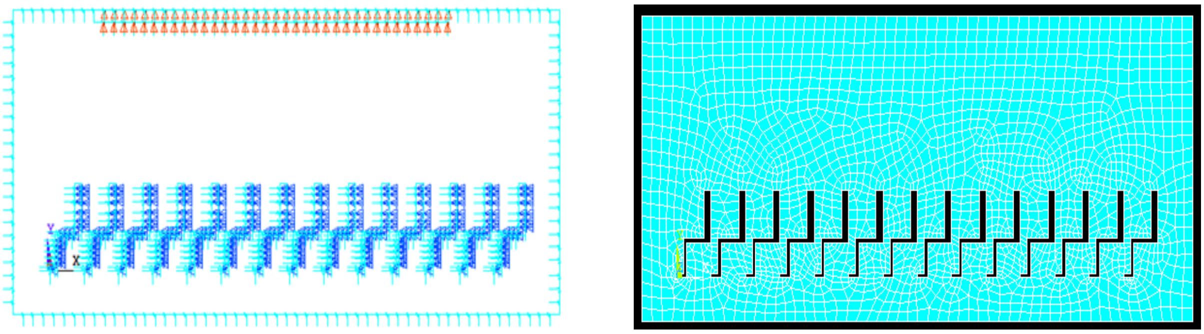


Fig.2. Configuration of seating arrangement for 49 passengers including the driver prepared in the ANSYS (MAPDL) program

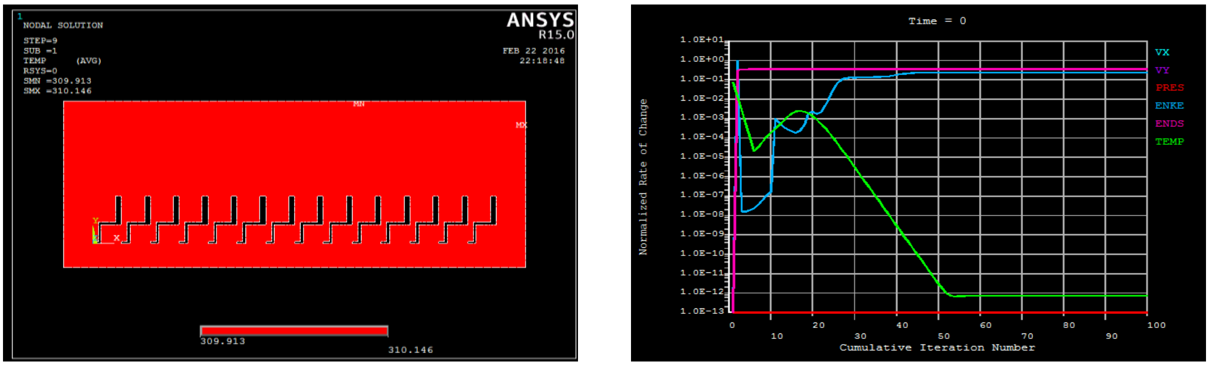
In Fig.3 (a) is given, side view of 3D design of passenger bus, aeration panels and settlement infrastructure designed for 49 passengers including the driver prepared in the ANSYS (MAPDL) program. In the design of the system, there are the one ventilation panels positioned for each passenger seat on top panel of the bus. The panel layout is shown by the red triangle profiles. Passenger seats are shown in dark blue, and passengers created by using dressed mannequins are shown in light blue. Side view of 3D design of passenger bus, mesh structure designed for 49 passengers including the driver prepared in the ANSYS (MAPDL) program is given in Fig.3(b). The mesh structure was created on the basis of detection to the high accuracy of the pressure and temperature distributions in the bus cabin. As seen in the Fig 4(b), number of nodes was increased in the regions where sit passengers in the bus, in order to configure the effect of cold air from air conditioning system clearly. Location of seats were defined with black regions without mesh structure.



(a) **(b)**

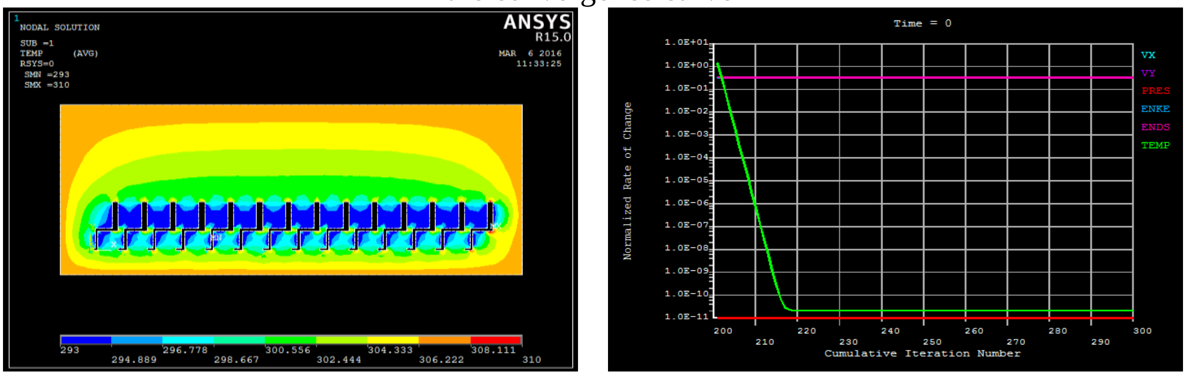
Fig.3 (a) Settlement infrastructure and **(b)** mesh structure formed in ANSYS program in bus cabin

Fig.4(a) shows of the temperature conditions before working the air conditioning system in bus cabin. Temperature profile is distributed as uniform (≈310 K) in all regions of bus cabin as shown in Fig.4(a). Convergence curves of pressure and temperature in bus cabin before working the air conditioning system is given in Fig.4(b).



(a) (b)

Fig.4. Prior to the opening of air conditioning in the bus cabin **(a)** temperature distribution **(b)** the convergence curve



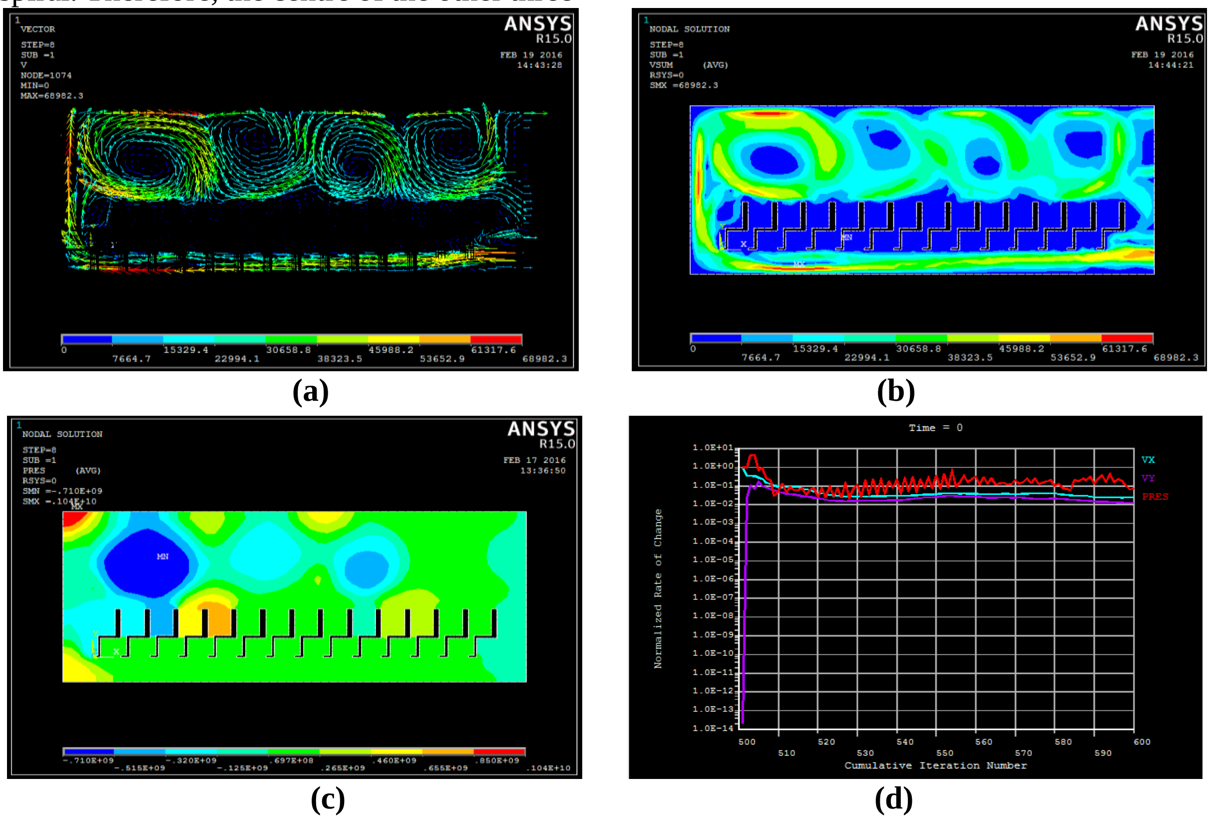
a b

Fig.5. (a) Temperature distribution and **(b)** convergence curves in the bus cabin after working during 5 minutes to the air conditioning system

Temperature distribution and convergence curves in the bus cabin after 5 minutes from working of air conditioning system is given in Fig.5. The highest temperature in ≈306 K (≈33 °C) was observed to the orange band narrowing towards the bottom over the front and rear windows from the ceiling of the bus. A temperature distribution increasing in layers from the passenger section to the ceiling of the bus is realized after working during 5 minutes to the air conditioning system. Among these layers, heat transfer with convection among these layers up to the roof of the bus from passengers section can be observed easily with color scale which is changing from dark blue (≈293 K) to orange (≈306 K). Temperature has increased up to about 296 K (23 °C) within bubbles formed due to breath temperatures of passengers in a limited area. The distribution of temperature in the lower section of seats has not arrived uniform value because of non-uniform temperature incoming from bottom of the bus. It should be noted that, temperature distribution in back seat regions placed of engine of bus is affected from the operation of the engine, and even rear compartment has remained at initial temperature. The effect of cold air flow is seen in the passenger's regions as result of blown cold air from the panels over the passengers as seen in Fig.5. Rise of temperature towards the ceiling and floor of bus from the region of passengers reveals the effect of heat transfer with convection from the external environment inside the bus. Thus, temperature arrived to highest value to regions which contact with the external environment such as the windshield, the rear window, the floor and the ceiling. Then, starting from the opening of air conditioner until the temperature in the bus reaches a uniform distribution, fan speed is increased gradually for increasing of amount of cold air which sent to bus cabin from cooling panel. Thus, the mass of cold air taken into the cabin has been increased over time and has been continued to analysis for uniform temperature distribution in the bus.

Fig.6 is shown convergence curves and distributions of pressure, velocity vector and total vector velocity in the bus cabin after working during 10 minutes to the air conditioning system. Four different turbulence central independently of one another was occurred in region except to passenger zones in the bus cabin as shown in Fig.6. Higher turbulence intensities in the front region extending up to 4th row seats from the windshield of the bus which except in the passenger zones by comparison with other regions is occurred as shown in Fig.6 (a) and (b). This high turbulence has affected to other three turbulence centres, and even it has progressed towards the rear of the bus with these turbulence centres in the form of a spiral. Therefore, the centre of the other three turbulent flow were in position according to the structure of spiral. Furthermore, this high turbulence extends toward rear from under of seats by continuing throughout front window of bus.

It is noted that intensities of velocity vectors increases at intersection zones of turbulence regions. Lower pressure intensities in front region of cabin according to the other region which depends on the high turbulent intensity has emerged as shown in Fig.6(c). Lowest pressure zone formed on the third-row seats. Furthermore, the relatively low pressure zones has occurred in the central portions of the bus. It can be said that ideal zone for air draught are front-bottom region of the bus. Nevertheless, an alternative space for air draught can the point near the middle of the bus's upper surface.

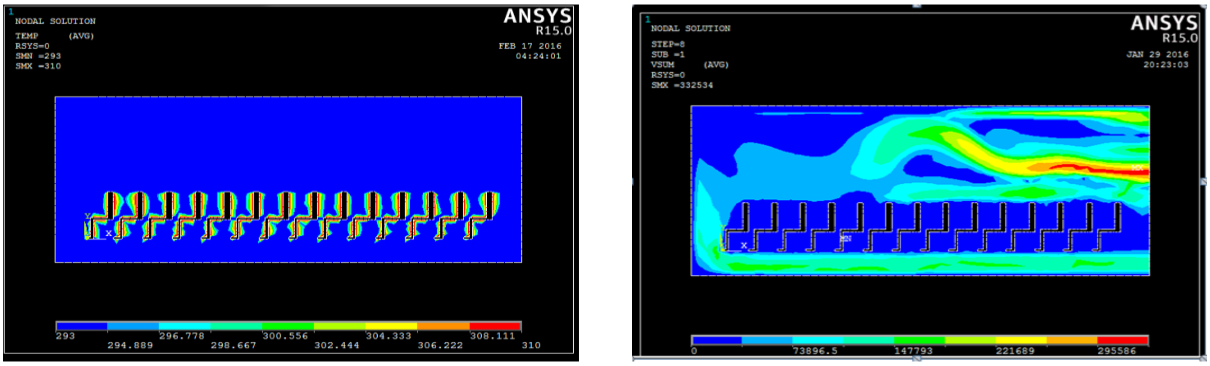


(a) (b) (c) (d)

Fig.6. Distributions of **(a)** velocity vector, **(b)** total vector velocity, **(c)** pressure and **(d)** convergence curves in the bus cabin after working during 10 minutes to the air conditioning system

Distributions of temperature and velocity vector in the bus cabin after working during 15 minutes to the air conditioning system is given in Fig.7. Temperature distribution has reached to targeted temperature value in 293 K (20 ° C) in the bus cabin after 15 minutes from working of air conditioning system as shown in Fig.7 (a). Temperature distribution was increased in the range of 300-308 K due to the heat transfer from body of passengers to cold air in bus cabin. This result has shown that thermal mannequin model which is used for computer simulation of cooling process provided with air conditioning system is effective highly in terms of adaptation to actual conditions.

In addition, intensities of velocity vectors are higher in the rear zone compared with the front zone of bus cabin as shown in Fig.7 (b). Thus, air speed in cabin can be increased somewhat for uniform pressure distribution.



(a) (b)

Fig.7. Distribution of **(a)** temperature and **(b)** velocity vectors after working during 10 minutes to the air conditioning system

4. Conclusion

Thermal comfort parameters depending on pressure and velocity distribution provided with air condition system on the bus having 49 passengers including the driver were analysed by using finite elements method (ANSYS). The conclusions of the study was summarised below;

- Just before the air conditioner is opened, it has been identified that the temperature distribution inside the bus is at equal degrees and the temperature is about 310 K.
- At the end of 5 minutes, a temperature distribution which is increasing in layers from the passenger section to the ceiling of the bus is realised.
- After 10 minutes of operation of the air conditioning system, four different turbulence central independently of one another was occurred in region except to passenger zones in the bus cabin. Higher turbulence intensities has occured in the front region of bus cabin. This high turbulence has affected to other three turbulence centers, and even it has progressed towards the rear of the bus with these turbulence centers in the form of a spiral.
- As a result, it has been detected that temperature distribution has reached to targeted temperature value in ≈ 293 K (≈20 ° C) in the bus cabin after working during 15 minutes to the air conditioning system.
- As a general result of this paper, it can be said that thermal mannequin model which is used for computer simulation with ANSYS (MAPDL) program of cooling process provided with air conditioning system is effective highly in terms of adaptation to actual conditions.

References

[1] Nilsson, H., Holmér, I., Bohm, M. and Norén, O., Effects on thermal comfort using special glazing-comparison of CFD calculations and manikin measurements, Int. ATA Conf., 17-19 Nov, Florence, Italy, 1999A4077, 1999.

[2] Bohm, M., Holmér, I., Nilsson, H., Norén, O., Thermal effect of glazing in driver´s cabs, JTI-rapport, Uppsala, Sweden, ISSN 1401-4963, 2002.

[3] Wyon, D., Tennstedt, C., Lundgren, I., Larsson, S., A new method for the detailed assessment ofhuman heat balance in vehicles - Volvo's thermal

manikin, VOLTMAN, SAE Technical Paper 850042, 1985.
[4] Nilsson, H., Holmér, I. Impact of seat on thermal comfort, In Proceedings of the Sixth International Conference on Indoor Air Quality and Climate (Indoor Air 93) , 127-132, Helsinki, Finland, 1993.
[5] Bohm, M., Comparison of instruments for measurement of equivalent temperature in an experimental cab in a climatic chamber, EQUIV-Report, No: 3, Uppsala, Sweden, 1999.
[6] Murakami, S., Kato, S., Zeng, J., Flow and temperature fields around human body with various room air distribution: CFD study on computational thermal manikin-PartI, ASHRAE Transactions, 103(1), 3-15, 1997.
[7] Murakami, S., Kato, S., Zeng, J., Numerical simulation of contaminant distribution around a modelled human body: CFD study on computational thermal manikin – Part II, ASHRAE Transactions, 104 (2), 226-233, 1998.
[8] Burch, S.D, Pearson J.T., Ramadyhni, S., Experimental study of passenger thermal confor in an automobile under severe winter conditionin, ASHREE Transactions, 97, 239-246, 1991.
[9] Burch, S.D, Ramadyhni, S., Pearson J.T., Analysis of pessenger thermal confor in an automobile under severe winter conditionin, ASHREE Transactions, 97, 247-257, 1991.
[10] Kayfeci, M., Gedik, E., Sağıroğlu, S., Kurt H., Investigation of alternative air conditioning system for vehicles, 5th In ternational Advanced Technologies Symposium, Karabük, Turkey, 2009.
[11] Ingerson, J.G., Kalman, T.G., Maxvell, M.L., Automobile passenger co model-Part II: Human thermal comfort calculation, SAE Technical Paper 920266, 1992.
[12] Chio, C.P, Cheng, Y.H., Ling, M.P. Chen, S.C., Lio, C.M., Quantitative estimitaion of excess mortality for drivers and passengers exposed to particulate matters in long-distance buses, Atmoshperic Environment, 51, 260-267, 2012.
[13] ASHRAE Handbook (2011) - Heating, Ventilating, and Air-Conditioning Applications (I-P Edition), p.136-138, Atlanta, 2011.
[14] Andre, J.C.S., Conceiçao, E.Z.E, Silva, M.C.G.,Viegas, D.X., Integral simulation of air conditioning in passenger buses, 4th International conference on air distribution in rooms, Roomvent '94, Cracow, Poland,1994.
[15] Madsen T, Olesen B., Reid K., New methods for evaluation of the thermal environment in automotive vehicles, ASHRAE Transactions, 92(1B), 38-54, 1986.
[16] Fanger, P.O, Thermal comfort: analysis and applications in environmental engineering, Copenhagen: Danish Technical Pres., Libraries , Australia, 1970.
[17] Bedford, T., The Warmth Factor in Comfort at Work: A Physiological Study of Heating and Ventilation, Medical research council report, H. M. Stationery Office, London. 1936.
[18] Asghar, H.B.A, Designing new model of bus air conditioning system and numerical analysis using finite element methods, Master Thesis, Institute of Science, Süleyman Demirel University, 2016.