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The Effect of Battery Thermal Management System Unit on the Roof of a Bus on Passenger Air-Conditioning Performance

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Abstract

While cooling comfort is important in city buses compared to other vehicles, it is also difficult to keep the cooling performance at a high level. Roof AC units used in commercial vehicles may vary in performance depending on many factors. Therefore, while the design works are in progress, there are some points to be considered while the units are in the packaging phase. These points are that the air used for condenser cooling in the air conditioner suction zone is at low temperature with high flow rate. In this study, it is aimed that the air conditioner and battery cooling unit placed on the roof of a bus developed by ANADOLU ISUZU are not adversely affected by each other. For this reason, in the related study, design and analysis studies were carried out to reduce the negative effects of the hot air coming out of the battery thermal management system (BTMS) in the cooling circuit when the air conditioner is activated. The aim of the study is to ensure that the air-conditioning unit provides thermal comfort in the vehicle as soon as possible and is positioned together on the vehicle roof without losing BTMS performance. The analysis processes of the study were progressed by performing CFD simulations in Ansys FLUENT 2020 software. For simulations, the temperature of the loop is reduced using the RANS equations. The study will be a guide for many new vehicles.

Keywords

BTMS, Battery thermal management system, A/C, Air-conditioning, Commercial vehicle, Class 1, A/C performance, Public transport, Roof top unit, CFD

Introduction

Air-conditioning units in commercial vehicles are mainly divided into conventional and tropical air conditioners. Tropical air-conditioning units are more advanced than conventional units and can continue to operate even at an outdoor temperature of 55°C. In conventional air conditioners, the outdoor temperature reaching over 45°C causes a decrease in air conditioner performance and after a while, the system shuts down to protect itself. This is an important issue to be considered in order to ensure thermal comfort. The quality of thermal comfort is determined by the performance of the air-conditioning system, and it varies depending on many factors such as outdoor temperature, vehicle speed, location of air-conditioning system on the bus, air duct structure and its design, compressor performance, amount of refrigerant gas and its mass flow rate, and performance of condenser and evaporator heat exchangers [1].

Teli et al. [2] wrote an article about the increasing energy consumption of heating, ventilation, and air-conditioning (HVAC) systems in the vehicle. This article focused on reducing energy consumption without loss of comfort. This study also shows that existing HVAC systems can be improved with virtual assistants such as CFD analysis.

Habib et al. [3] carried out a study on homogeneous distribution of passenger air-conditioning unit in the indoor environment of a passenger bus and increasing comfort.

In this study, they expressed sensitivity that air-conditioning unit should not be placed on the vehicle roof in an area where air pressure is low or temperature is high. Air suction quality and compressor play a major role in air-conditioning performance and energy consumption [4].

Lan et al. [5] have studied four design changes to front end by CFD approach with realizable k-ε turbulence model. Their design changes were placement of transmission oil cooler (TOC), sealing design, belly panel design, and cooling fan speed. They have found that placing the TOC in front of the superheat area of the condenser gives much better performance compared to placing it near the sub-cool region and sealing around cooling package and opening belly panel helps to reduce the temperature of the air entering condenser at idle.

Yu et al. [6] have investigated optimal condenser fan control to improve the coefficient of performance of air-cooled chillers. They have used a thermodynamic methodology that determines the optimal condensing temperature set point for condenser fan controls. They indicated that the improved control to an air-cooled centrifugal chiller model showed that the COP could increase by 11.4–237.2%.

Reddy et al. [7] have studied how the suction temperature will decrease by increasing the condenser suction area and what effect it will have on air-conditioning performance. In this study, changes in COP value with decreasing suction temperature are shared. Here, the significant effect of condenser suction temperature on air-conditioning performance is seen.

Especially in an electric vehicle, the aim should be to keep the temperature of the battery, passenger cabin, and driver's cabin at optimum values rather than heating and cooling the passenger cabin. The effects of the mentioned systems on each other should be carefully examined. Studies are being carried out on integrated thermal management systems where these systems can work together [8]. In order for all these systems to operate at optimum levels, it is important to know the components well [9].

The aim of this study is to see the effects of BTMS and air-conditioning unit placed on the bus roof on each other [10]. The study is aimed at removing the hot air evacuated from the BTMS unit from the air conditioner suction area before the vehicle is produced. For this reason, different design suggestions are carried out and they are simulated in Ansys FLUENT R20. Analysis outputs will be shared as no action has been taken and different actions have been taken to eliminate negative effects.

Methodology

In the current study, the flow is considered steady and incompressible. FLUENT solver is set as coupled solver, which solves the governing equations of continuity, momentum, and species transport simultaneously. Governing equations for additional scalars will be solved consecutively. The steady-state RANS equations for the conservation of mass and momentum are given in Equations 1 and 2 [11]:

Continuity:

$$\frac{\partial}{\partial x_i}\left(\rho \bar{u}_i\right) = 0 \qquad \text{Eq. (1)}$$

Momentum:

$$\frac{\partial}{\partial x_i}\left(\rho \bar{u}_i \bar{u}_j\right) = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial \bar{u}_i}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho \overline{u_i' u_j'}\right) \qquad \text{Eq. (2)}$$

Realizable k-ε turbulence model with "enhanced wall treatment" wall function is selected to simulate viscous effects of the flow near the solid body. Heat transfer is neglected, therefore parameters such as air density and air dynamic viscosity are assumed to be constant.

Realizable k-ε turbulence model is a very commonly used model for calculating mean flow characteristics for turbulence conditions. It consists of two transport Equations 3 and 4 [11] which are partial differential equations. The turbulence kinetic energy, k, and its rate of dissipation, ε, are obtained from the following equations:

$$\frac{\partial}{\partial t}\left(\rho k\right) + \frac{\partial}{\partial x_i}\left(\rho k u_j\right) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \qquad \text{Eq. (3)}$$

$$\frac{\partial}{\partial t}\left(\rho \varepsilon\right) + \frac{\partial}{\partial x_j}\left(\rho \varepsilon u_j\right) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b + S_\varepsilon \qquad \text{Eq. (4)}$$

Boundary Conditions

I. The flow is stable (steady state).
II. The realizable k-ε turbulence model and the "enhanced wall treatment" wall function were used to analyze the viscous flow near the wall.
III. The air suction areas of the BTMS module are shown in Figure 1 with numbers 1-2-3. Boundary conditions were taken from the manufacturer for the absorption flow rates from three suction zones. These flow rates were obtained by dividing the cross-sectional area. BTMS air discharge is also shown with number 4 in Figure 2. The flow value here should be equal to the total flow value in the absorption zones (conservation of mass).

FIGURE 1 Representation of BTMS air suction areas.

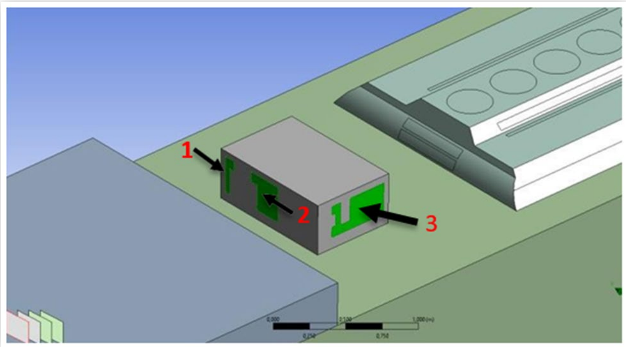
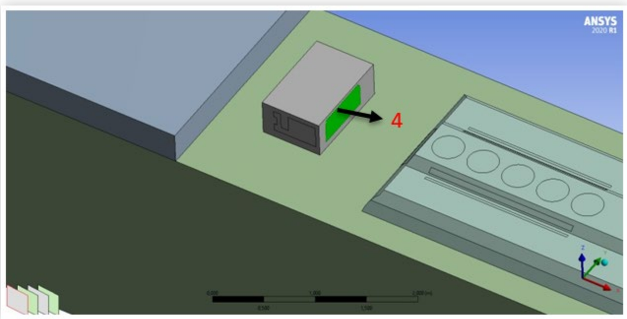


FIGURE 2 Representation of BTMS air discharge area.



The flow rates of the suction and discharge zones marked in Figures 1 and 2 are shared in Table 1. According to these definitions, the flow rates shared by the manufacturer were used in the analysis.

According to the information received from the manufacturer, boundary conditions have been defined such that the ambient temperature will be 45°C and BTMS and AC unit discharges will be constant at 59°C. For this reason, in order to obtain the changes in the air between 45°C and 59°C, it is aimed to obtain the temperature changes by using the parabolic equation or linear interpolation of the physical properties of the air. Density, specific heat, thermal conductivity, and dynamic viscosity of the air are shared in Table 2.

Passenger air conditioner suction and discharge areas are shown in Figures 3 and 4. In the analysis, air inlets and air evacuations were provided into the model from the suction and discharge areas. It is discharged as much as the amount of absorption. Absorptions are at ambient temperature of 45°C and discharges are at 59°C. Therefore, capturing the change in air temperature in case of passenger air conditioner operation is provided by the physical properties of the air summarized above.

TABLE 1 Representation of BTMS air suction and discharge flow rates.

Number	Volumetric flow rate (m^3/s)	Mass flow rate (kg/s) (45°C)
1	0.104	0.116
2	0.446	0.496
3	0.698	0.775
4	1.25	1.387

TABLE 2 Representation of physical properties of air.

#	20°C	40°C	60°C	80°C
Density (kg/m^3)	1.204	1.127	1.06	1
Specific heat (J/kgK)	1006	1007	1009	1010
Thermal conductivity (W/mK)	0.025	0.026	0.028	0.028
Dynamic viscosity × 10^4 (kg/ms)	0.018	0.190	0.199	0.208

FIGURE 3 Representation of AC air suction areas.

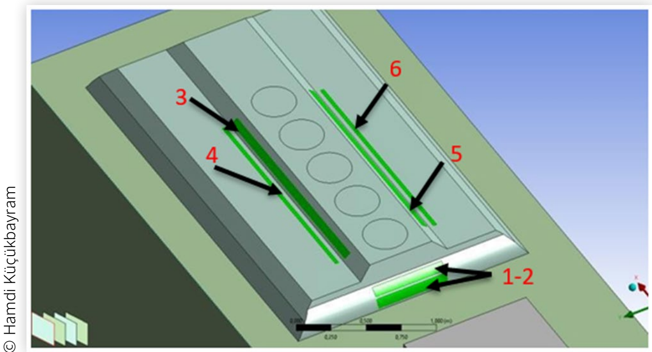
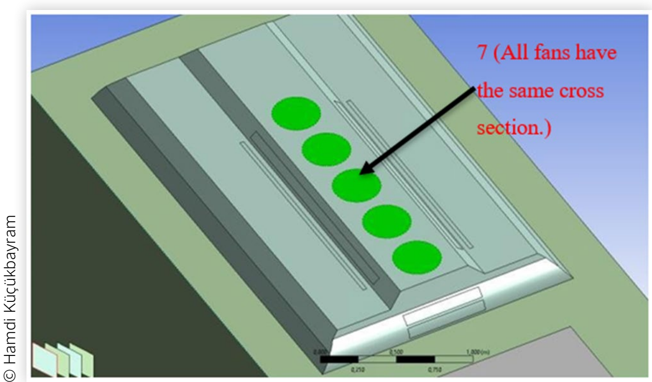


FIGURE 4 Representation of AC air discharge areas.



The suction and discharge flow values (Table 3) of the passenger air conditioner were defined in line with the values received from the air conditioner supplier and used as boundary conditions in the analysis. Flow distribution values in the suction regions were obtained by proportioning the cross-sectional areas.

Design

The vehicle on which this study was carried out is a class 1 electric vehicle with a length of 12 m and is used in urban passenger transportation. The air-conditioning unit in the project is a passenger air conditioner with 30 kW cooling and 17 kW heating (heat pump) capacity using R407C refrigerant. The BTMS in the project is a unit that provides water-type heating and cooling to the batteries with the

TABLE 3 Representation of AC suction and discharge flow rates.

#	Volumetric flow rate (m^3/s)	Mass flow rate (kg/s) (45°C)
1	0.427	0.474
2	0.427	0.474
3	0.965	1.072
4	0.447	0.496
5	0.965	1.072
6	0.447	0.496
7	0.736	0.817

support of a chiller with 6 kW cooling and 8 kW heating capacity. Due to commercial concerns, brand and model information of the components has not been shared. Vehicle dimensions are shown in Figure 5.

Studies have been carried out to increase the cooling performance of the BTMS module and to prevent the performance of the passenger air conditioner from being adversely affected. With the first of these studies, the results of the studies were first examined and then the studies developed step by step were carried out. The studies carried out are shown in Table 4.

Mesh

In all CFD applications, meshing process has a great importance. Meshing needs to be performed with great care to achieve accurate and reliable simulation results. In this study, Ansys FLUENT Meshing (TGrid) is used as the meshing tool. Complete CFD model surface mesh quality is under 0.58 value in skewness with minimum size 1 mm and maximum cell size 1000 mm as the vehicle is available in wind tunnel CFD model. As for volume mesh it covers 30 million cells with five boundary layer thickness for all walls, 1,2 growth rate, and 10 aspect ratios.

Results and Discussion

In this section, outputs of all CFD simulations are presented. The level of the air discharge from BTMS to the passenger air-conditioning suction and the work done as a solution are explained.

In the study specified as Design 1, BTMS discharge affects the passenger air-conditioning pre-suction, and it has been observed that hot air is sucked from the passenger air-conditioning pre-suction zones. Air-conditioning performance will be adversely affected. The air-conditioning suction zone analysis results for Design 1 are shown in Figure 6. BTMS air discharge is shown in Figure 7.

Two design (Design 2 and Design 3) improvement studies were carried out to solve the problem. The separator applied in the study specified as Design 2 is 100 mm away from the BTMS unit, and its effect on the air conditioner suction area is shown in Figure 8. The effect of BTMS air discharge in Design 2 is shown in Figure 9. The separator applied in the study specified as Design 3 is 150 mm away from the BTMS unit, and its effect on the air conditioner suction area is shown in Figure 10. The effect of BTMS air discharge in Design 3 is shown in Figure 11.

The results of the improvement studies are shared below.

1. In Design 2 and Design 3, it has been observed that the separator brackets placed behind the BTMS prevent the entry of hot air into the passenger air-conditioning suction. In both studies, the suction and discharge of BTMS and passenger air conditioner on the roof do not affect each other.
2. Since the brackets applied in Design 2 and Design 3 are located behind the BTMS module, they will create resistance to the discharge side and reduce the airflow value to be pumped. This will adversely affect BTMS cooling performance.

FIGURE 5 Dimensions of the vehicle being studied.

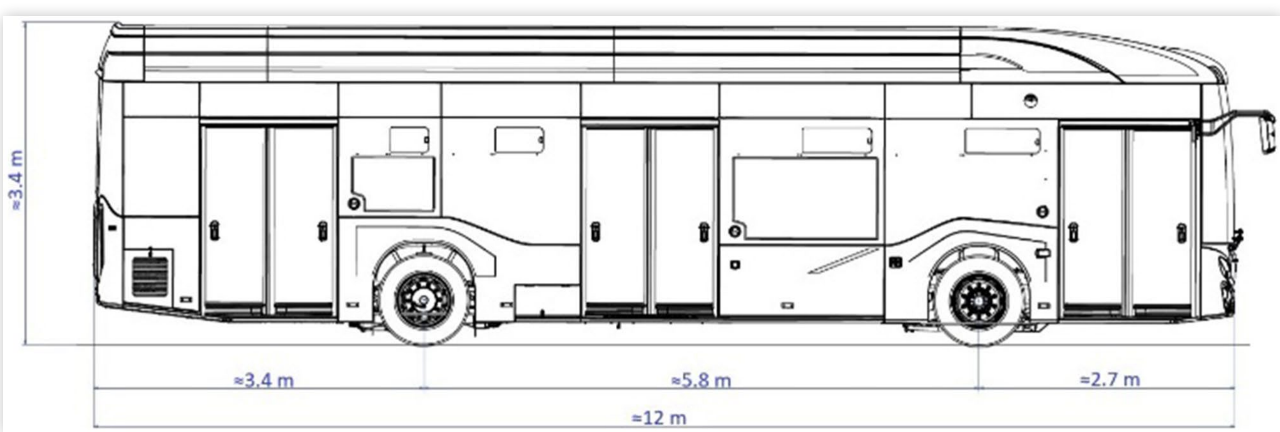


TABLE 4 Representation of design suggestions.

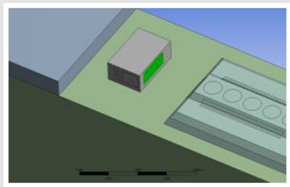
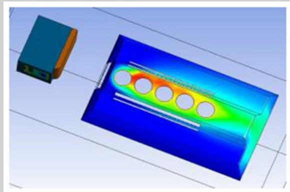
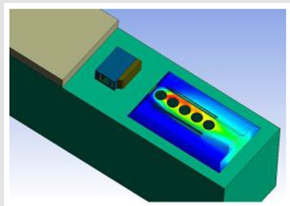
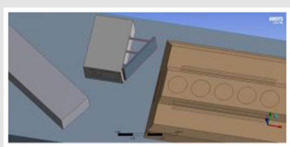
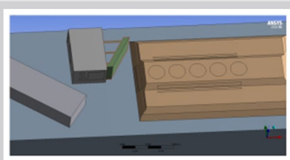
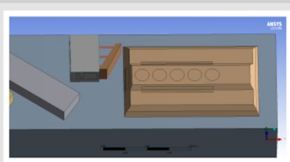
Description	Model
Design 1 is the back-to-back positioning of BTMS and passenger air-conditioning. No separator was used to directly discharge air from the BTMS unit.	
Design 2 is the BTMS and passenger air-conditioning positioned back to back. Additionally, a separator sheet metal cover was applied to the BTMS structure at a distance of 100 mm to directly discharge air from the BTMS.	
Design 3 is the BTMS and passenger air-conditioning positioned back to back. Additionally, a separator sheet metal cover was applied to the BTMS structure at a distance of 150 mm to directly discharge air from the BTMS.	
Design 4 is the cross-positioning of BTMS and passenger air-conditioning. Additionally, a separator sheet metal cover was applied without leaving any distance from the BTMS structure to directly discharge air from the BTMS.	
Design 5 is the cross-positioning of BTMS and passenger air-conditioning. Additionally, a separator sheet metal cover was applied to directly discharge air from the BTMS, leaving a very small distance from the BTMS structure.	
Design 6 is the cross-positioning of BTMS and passenger air-conditioning. Additionally, a separator sheet metal cover was applied to keep a distance of 120 mm from the BTMS structure to directly discharge air from the BTMS.	
Design 7 is the cross-positioning of BTMS and passenger air-conditioning. Additionally, a separator sheet metal cover was applied to keep a distance of 180 mm from the BTMS structure to directly discharge air from the BTMS.	

FIGURE 6 Air-conditioning suction zone analysis results.

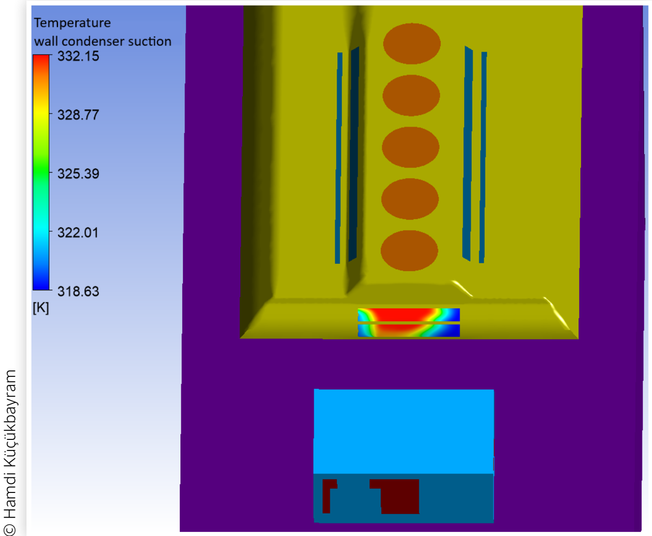


FIGURE 7 Effect of BTMS air discharge.

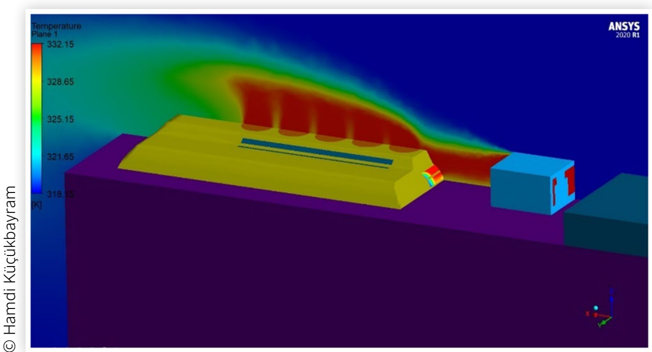


FIGURE 8 Design 2 air-conditioning suction zone analysis results.

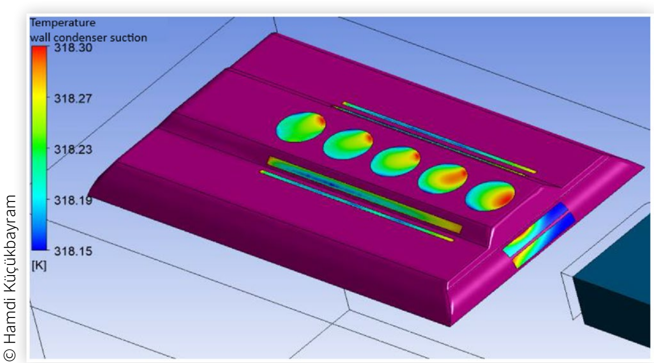


FIGURE 9 Design 2 effect of BTMS air discharge.

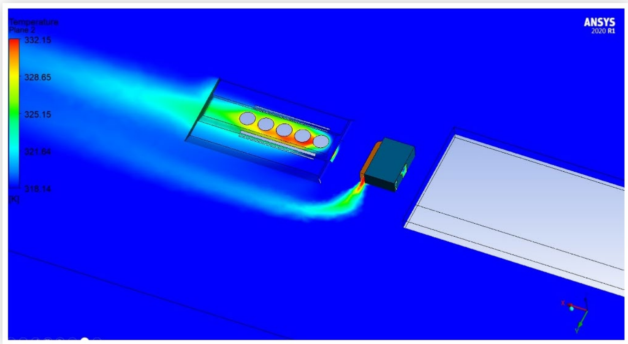


FIGURE 10 Design 3 air-conditioning suction zone analysis results.

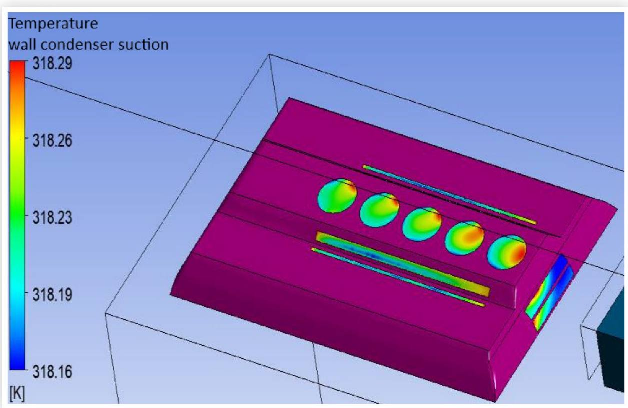
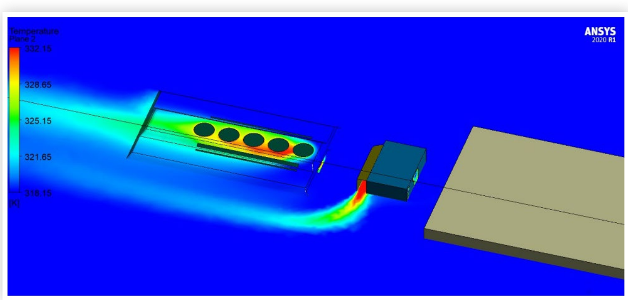


FIGURE 11 Design 3 effect of BTMS air discharge.



For more suitable solutions, it would be more appropriate to arrange BTMS and passenger air conditioners diagonally rather than back to back.

The roof group components of BTMS and the passenger air conditioner are positioned diagonally. Due to the wide and large air-conditioning surface and the side spoilers on the vehicle roof, the BTMS module cannot be placed in a position where it can be completely removed from the air conditioner. For this reason, separator bracket studies have been developed behind BTMS.

First of all, the results of Design 4 and Design 5 studies will be examined. For the Design 4 and Design 5 analysis studies, the final position of the BTMS module and the separator bracket models placed behind it were subject to the updated model. Holes are formed in the upper parts of the separator brackets and relieve the BTMS module in reducing the resistance. The effect of the study specified as Design 4 on the air conditioner suction area is shown in Figure 12. Also the effect of BTMS air discharge is shown in Figure 13.

The effect of the study specified as Design 5 on the air conditioner suction area is shown in Figure 14. Also the effect of BTMS air discharge is shown in Figure 15.

FIGURE 12 Design 4 air-conditioning suction zone analysis results.

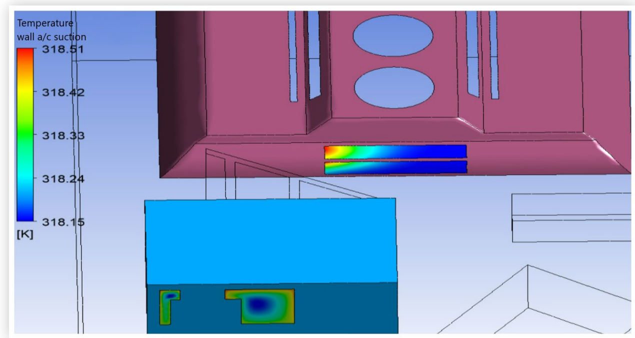


FIGURE 13 Design 4 effect of BTMS air discharge.

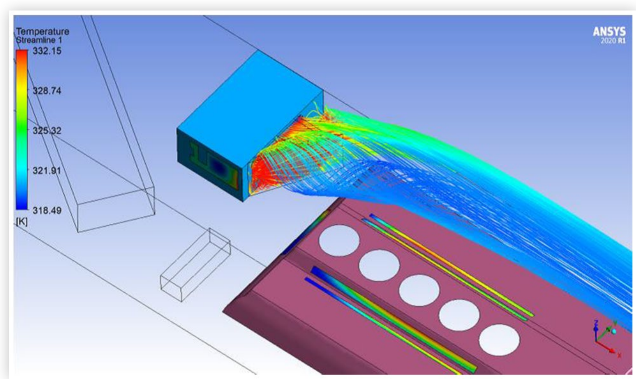


FIGURE 14 Design 5 air-conditioning suction zone analysis results.

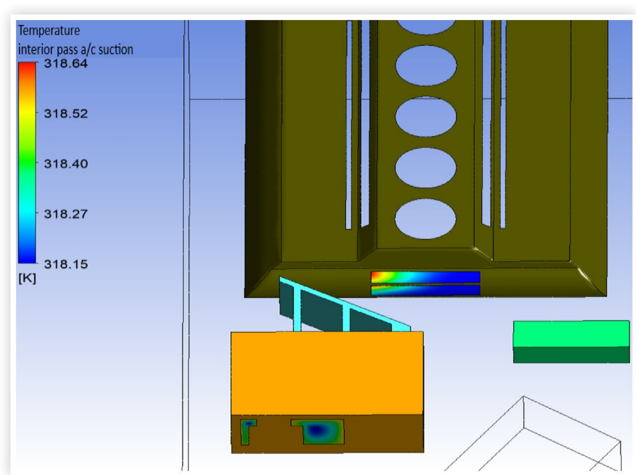
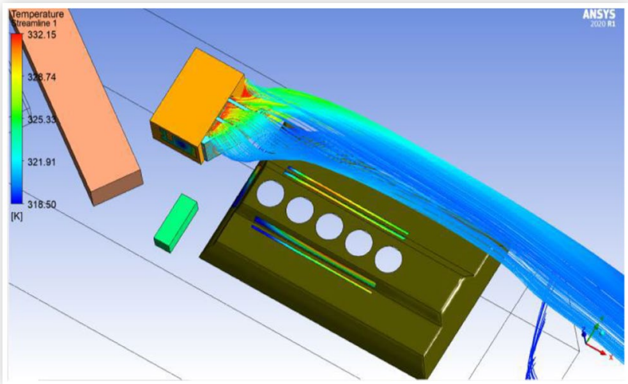


FIGURE 15 Design 5 effect of BTMS air discharge.



The results of the improvement studies are shared below.

1. The results obtained for Design 4 and Design 5 models are shared with visuals. The counter-resistance values on the BTMS module discharge side have been considerably reduced compared to previous studies.
2. It was observed that Design 5 created less resistance than Design 4.
3. It was observed that BTMS discharge in Design 4 and Design 5 studies could not be absorbed by the air conditioner suction zones. Since the ambient temperature behind the bracket placed behind the BTMS only slightly increased, it was observed that the air conditioner front-facing suction areas warmed up to a local maximum of 0.5°C.

The Design 6 and Design 7 models are new models that are further developed on the Design 4 and Design 5 models. In these models, the separator brackets behind the BTMS have been enlarged even more compared to previous studies. Therefore, it is aimed to further improve the results obtained.

The effect of the study specified as Design 6 on the air conditioner suction area is shown in Figure 16. Also, the effect of BTMS air evacuation is shown in Figure 17.

The effect of the study specified as Design 7 on the air conditioner suction area is shown in Figure 18. Also, the effect of BTMS air evacuation is shown in Figure 19.

The results of the improvement studies are shared below.

1. It has been observed that the results obtained in Design 6 and Design 7 models create less resistance than Design 4 and Design 5.
2. In the evaluations of Design 6 and Design 7 models among themselves, Design 7 creates less resistance than Design 6, and the evaluation of this design among all designs further increases the cooling performance on the BTMS side.
3. In the Design 7 study, it was observed that although the bracket added to the back of the BTMS was further enlarged, the counter-resistance could not be significantly reduced.

FIGURE 16 Design 6 air-conditioning suction zone analysis results.

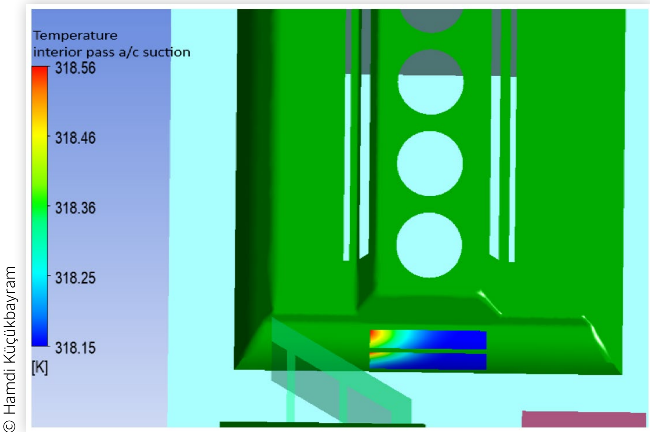


FIGURE 17 Design 6 effect of BTMS air discharge.

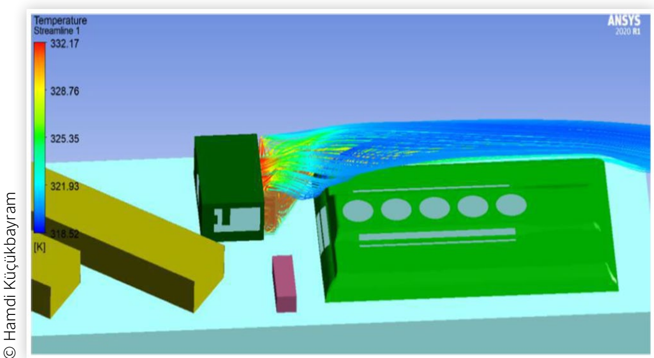
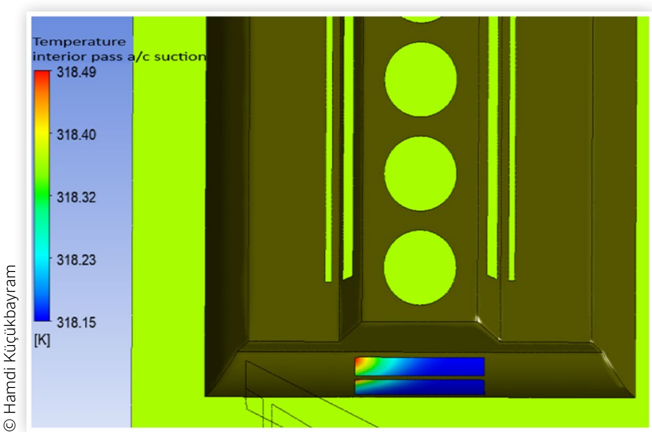


FIGURE 18 Design 7 air-conditioning suction zone analysis results.



As of the last situation, it has been observed that partial positive pressure values are approached to the atmospheric pressure boundary condition in the Design 7 study (Table 5).

4. As with other designs, Design 7 and Design 6 studies prevent hot air from entering the passenger air-conditioning front intake areas.

FIGURE 19 Design 7 effect of BTMS air discharge.

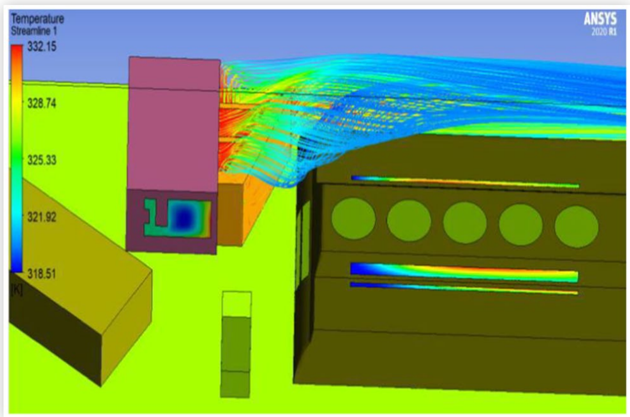


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TABLE 5 Representation of BTMS discharge pressure values.

Models	BTMS resistance to discharge—static pressure (Pa)
Design 4	+49.94
Design 5	+26.01
Design 6	+10.85
Design 7	+9.22

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5. Due to the fact that the ambient air temperature of the front side intake temperatures is slightly warmer (the increase in the ambient temperature behind the BTMS discharge bracket), the maximum ambient temperature of 0.5°C locally increases in the passenger air conditioner front side intakes.

With the Design 7 study, the atmospheric pressure boundary condition was closest to the BTMS discharge. In other words, it shows numerically that this is the operation where the resistance is reduced the most and the cooling performance of the BTMS module will be the highest.

If examined in more detail, as seen in Figure 6, results of Design 1 reach up to 59°C in the air conditioner suction section. This limit does not enable the air conditioner to operate efficiently. With the separator applied in Design 2 and later, the temperature in the air conditioner suction section drops to approximately 45°C. With all improvement applications after Design 1, the temperature in the suction zone was reduced by more than 30%. In short, the separator application and the cross-positioning of the BTMS and the air conditioner have eliminated all concerns about air conditioner performance. However, the increase in BTMS discharge temperature resulting from the addition of a separator has become another problem that needs to be solved.

After the air-conditioning problems are resolved, the main issue we need to focus on is proper air discharge of the BTMS air outlet. A large design area was needed to reduce the BTMS air discharge temperature and reduce resistance pressure. For this reason, air-conditioning and BTMS units are cross-positioned.

The values of the applications that create back pressure at the BTMS air discharge in Design 4 and subsequent design applications are shown in Table 5. The values shared in Table 5 varied depending on the structure of the separator in Design 4 and subsequent design applications. When Table 5 is examined, it is seen that the separator in Design 7 creates the least back pressure. To summarize, when the separator used in Design 7 was compared with the separator used in Design 4, it created approximately 5.5 times less back pressure.

Conclusions

For inner city bus developed by ANADOLU ISUZU, researches and simulations have been carried out to reduce the hot air coming from the BTMS unit discharge to the suction area of the passenger air conditioner. To solve the problem, seven different designs were developed using different positioning studies and separator brackets.

1. With all improvement applications after Design 1, the temperature in the suction zone was reduced by more than 30%.
2. When the separator used in Design 7 was compared with the separator used in Design 4, it created approximately 5.5 times less back pressure.
3. The best option study that will increase BTMS cooling performance and not affect passenger air-conditioning performance is Design 7.
4. The pressure value in the BTMS hot air discharge is closest to the atmospheric pressure boundary condition (the least counter-resistance).
5. The hot air to be discharged from BTMS is not absorbed by the passenger air conditioner.
6. By creating holes on Design 7, both lightning is provided and resistance is reduced.
7. In the analyses, the material of the part placed behind BTMS was preferred as FRP. FRP material retains three times more heat than sheet metal. (Steel: 1500 J/kgK, FRP: 450 J/kg.)

Definitions / Abbreviations

A - Cross-sectional area (m²)

$C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$ - Constant [5]

D_{ω} - Cross-diffusion term [5]

G_b - Generation of turbulence kinetic energy due to the mean velocity gradients [5]

G_k - Generation of turbulence kinetic energy due to the mean velocity gradients [5]

G_{ω} - Generation of ω [5]

k - Turbulence kinetic energy (J/kg)

p_d - Dynamic pressure (Pa)
p_s - Static pressure (Pa)
p_t - Total pressure (Pa)
Q - Volumetric flow rate (m³/s)
S_k, S_ε, S_ω - User-defined source terms [5]
t - Time (s)
u_i - Velocity component of i (i = 1, 2, 3) (m/s)
V - Velocity of flow (m/s)
x_i, x_j - Coordinate components
Y_k - Dissipation of k [5]
Y_M - Contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate (kg/m/s³) [5]
Y_ω - Dissipation of ω [5]
ε - Rate of dissipation (m²/s³)
Γ_k - Effective diffusivity of k (m²/s)
Γ_ω - Effective diffusivity of ω (m²/s)
μ - Dynamic viscosity (kg/m/s)
μ_t - Turbulent viscosity [5]
ν - Kinematic viscosity (m²/s)
ρ - Density (kg/m³)
σ_ε - Turbulent Prandtl numbers for ε
σ_k - Turbulent Prandtl numbers for k
ω - Turbulence frequency (s^{-1})

References

1. Alahmer, A., Abdelhamid, M., and Omar, M., "Design for Thermal Sensation and Comfort States in Vehicles Cabins," *Applied Thermal Engineering* 36 (2012): 126-140.
2. Bhagwat, A.N., Teli, S.N., Gunaki, P., and Majali, V.S., "Review Paper on Energy Efficiency Technologies for Heating, Ventilation and Air Conditioning (HVAC)," *International Journal of Scientific & Engineering Research* 6, no. 12 (2015): 106-116.
3. Gürbüz, H., Akçay, İ.H., Asghar, H., and Ali, Q.A., "Analysis of Bus Air Conditioning System by Finite Elements Method (ANSYS)," *International Journal of Automotive Engineering and Technologies* 5, no. 3 (2016): 115-124, doi:https://doi.org/10.18245/ijaet.287177.
4. Hegar, M., Kolda, M., Kopecka, M., Rajtmajer, V. et al., "Bus HVAC Energy Consumption Test Method Based on HVAC Unit Behavior," *International Journal of Refrigeration* 36, no. 4 (2013): 1254-1262, doi:https://doi.org/10.1016/j.ijrefrig.2012.10.023.
5. Lan, K.T., Xiao, L., Srinivasan, K., and Rahman, S.S., "The Impact of Vehicle Front End Design on AC Performance," SAE Technical Paper 2013-01-0859 (2013), doi:https://doi.org/10.4271/2013-01-0859.
6. Yu, F.W. and Chan, K.T., "Optimizing Condenser Fan Control for Air-Cooled Centrifugal Chillers," *International Journal of Thermal Sciences* 47 (2008): 942-953, doi:https://doi.org/10.1016/j.ijthermalsci.2007.07.018.
7. Kumar Reddy, J., Deopa, S., Sharma, A., and Aggarwal, P., "Impact of Condenser Opening Area on A/C Performance of the Automotive HVAC System," SAE Technical Paper 2015-01-0368 (2015), doi:https://doi.org/10.4271/2015-01-0368.
8. Saraireh, M., "Thermal Management in Electric Vehicles: Modeling and Prospects," *International Journal of Heat and Technology* 41, no. 1 (2023): 103-116, doi:https://doi.org/10.18280/ijht.410111.
9. ASHRAE, "Heating, Ventilating, and Air-Conditioning, Applications," 2019.
10. Hamut, H.S., Dincer, I., and Naterer, G.F., "Performance Assessment of Thermal Management Systems for Electric and Hybrid Electric Vehicles," *International Journal of Energy Research* 37, no. 1 (2013): 1-12, doi:https://doi.org/10.1002/er.1951.
11. FLUENT, "FLUENT 6.1 User's Guide," Lebanon, NH, 2003.

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