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PREDICTION OF THE ANNUAL HEAT LOAD OF AN ARTICULATED ELECTRIC URBAN BUS

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Abstract: This study presents a detailed method to estimate the thermal load of an articulated electric urban bus. Thermal load, which consists of solar, metabolic, ambient and ventilation heat loads, is estimated hourly for a one year period. A mathematical model takes into account the hourly passenger occupancy rate, hourly weather condition of the line and hourly solar loads as input and predicts the heat load accordingly. In order to determine the importance and contribution of each thermal load, all loads are calculated individually. Calculations are made for each hour of a year in order to observe the change of the contribution of each load in time. With the proposed method, the thermal load of an electric bus can be predicted in the system design phase and the HVAC system of the bus can be selected accordingly.
Keywords: Thermal loads, HVAC, Electric bus, Electromobility, Electrification.

BİR ELEKTRİKLİ KÖRÜKLÜ ŞEHİR İÇİ OTOBÜSÜN YILLIK ISI YÜKÜ KESTİRİMİ

Özet: Bu çalışma, körüklü elektrikli şehir içi otobüsünün ısı yükünü tahmin etmek için ayrıntılı bir yöntem sunmaktadır. Güneş, metabolik, ortam ve havalandırma ısı yüklerinden oluşan termal yük, bir yıllık bir süre için saatlik olarak tahmin edilmiştir. Kullanılan model, saatlik yolcu doluluk oranını, hattın bulunduğu bölgedeki saatlik hava durumunu ve saatlik güneş yüklerini girdi olarak alır ve ısı yükünü buna göre tahmin eder. Her termal yükün önemini ve katkısını belirlemek için tüm yükler ayrı ayrı hesaplanmıştır. Her yükün zaman içindeki katkısının değişimini gözlemlemek için yılın her saati için hesaplamalar yapılmıştır. Önerilen yöntemle, bir elektrik otobüsün termal yükü sistem tasarım aşamasında tahmin edilebilir ve elektrikli otobüsün HVAC sistemi buna göre seçilebilir.
Anahtar Kelimeler: Isı yükleri, HVAC, Elektrikli otobüs, Elektromobilite, Elektrifikasyon.

INTRODUCTION

Global warming probably is the one of the biggest environmental problems of today's world. Over the last fifteen years, environmental foundations and organizations have invested hundreds of millions of dollars into combating global warming (Shellenberger et al. 2004). Global warming is partly due to transportation activities which produce substantial amounts of carbon dioxide emissions. In addition, these have side effects like noise pollution and traffic congestion. It is believed that the transportation sector, which consumes the quarter of the world's energy production, should be made more environmentally sensitive for a sustainable growth (Juan et al. 2016). Use of electric buses for public transportation may be an alternative solution in order to decrease urban noise and air pollutions (Pihlatie et al. 2015). As a matter of fact, due to ecological and economic concerns, there is an increasing interest in electric buses. Bus producers are observed to invest increasingly in the production of electric and hybrid buses (Graurs et al. 2015).

As far as fuel and electrical energy consumptions calculated for the SORT 2 (Standardised On-Road Test) driving cycle for 12-meter buses are concerned, it is proven that electrical buses perform better than buses with conventional engines powered with CNG (Compress Natural Gas) or diesel fuel. Energy consumptions per km of CNG, diesel fuel and electric buses are 4.98 kWh, 3.90 kWh and 0.95kWh, respectively (Gis et al. 2017).

For bus operators, it may sometimes make sense to pick up a ready-made electric bus; however, for large bus lines that carry thousands of passengers daily by a populous bus fleet, it may be more efficient to use a bus that is designed especially for the specific line. In order to design an efficient electric bus for a specific line, design requirements should be carefully determined. Special care should be given to requirements imposed by energy consuming sub-systems, such as the HVAC (Heating, ventilation air conditioning) system, which is the main focus of the present study. To design and select elements of a bus HVAC system, it is crucial to accurately determine the heating and cooling loads (Aktacir et al. 2008). Energy consumption of the HVAC system accounts for about 20% of the total energy consumption of a bus (Kamiya 2006). Underestimated HVAC energy consumption may lead to thermal comfort problems, while an overdesigned HVAC system may increase manufacturing and operating costs.

In this paper, we propose a calculation model for the determination of the energy requirements of a HVAC system. More specifically, the HVAC power requirement of a specific route of the Istanbul Metrobus network is calculated in detail by taking into account the annual hourly passenger density, the hourly position of the sun, the location of the bus line and the bus working hours. In the present case, Istanbul Metrobus network is operated by Metrobus vehicles (Figure 1) with internal combustion engines. The present 18-meter articulated ICE (Internal Combustion Engine) driven bus design is taken as a reference to design a HVAC system to be installed in an electrified version of the bus.

Figure 1. A photo of Metrobus

A system design of an electric bus is presented in the study of Göhlich et al. (2018). The researcher concludes that HVAC system is the most energy consuming auxiliary system and therefore must be given special attention in electric bus system design. Additionally, the heating case is considered as the most critical condition if zero-emission operation is required. Exergy analysis was implemented to improve the intercity bus air-conditioning system design by Tosun et al. (2016). In this study hourly cooling capacity was determined in detail. In the study of Ünal (2016), the cooling load of a fully loaded bus at steady state was obtained as 22 kW. Göhlich et al. (2018) assumed a constant 24 kW load for heating in their study. The power required to cool the vehicle in summer at an outdoor temperature of 35 ◦C is assumed to be 24 kW. As a matter of fact these 24kW loads increases vehicle consumption by 1.3 kWh/km for an average velocity 18 km/h and 2 kWh/km for an average velocity of 12 km/h. Javani et al. (2012) studied the role of design parameters on the cooling capacity of hybrid and electric vehicles and energy and exergy analyses were presented. Ružić et al. (2011) investigated thermal interaction between a human body and a tractor cab. In this study a model is developed to calculate solar thermal loads of a tractor cab that has windows on its four surfaces. In the study of Stancato et al. (1992) a mathematical model was developed to simulate cooling loads in a cab and the results of the simulation were compared with experimental results. Fayazbakhsh et al. (2013) developed a model to calculate the head load of an internal combustion engine car. Ding and Zito (2001) presented a differential equation for the cabin that relates the heat transfer coefficient, discharge panel temperature and discharge volumetric air flow to the interior and the solution of the corresponding transient heat transfer differential equation was presented.

In the proposed study, loads are calculated for continuous working condition and heat loads that are required to be reached at first start of the bus are neglected. For calculations and model development, previous work accomplished by Fayazbakhsh et al. (2013) and the ASHRAE standard (2001) are taken into account. In the study of Fayazbakhsh et al. (2013) heat load is calculated for a passenger vehicle with internal combustion engine. In the case presented in our study, the heat generation of the electric motor is negligible compared to the internal combustion engine. Therefore, heat loads of exhaust system and internal combustion are omitted. Moreover, in the case of an urban bus, the number of passenger is large and very changeable. The bus has a very specific line and work schedule compared to a passenger car; and therefore this allows calculating HVAC energy consumption with higher accuracy. Furthermore, the geometry (a bus is very similar to a cuboid) and window/body ratio are different, which increases the solar heat load.

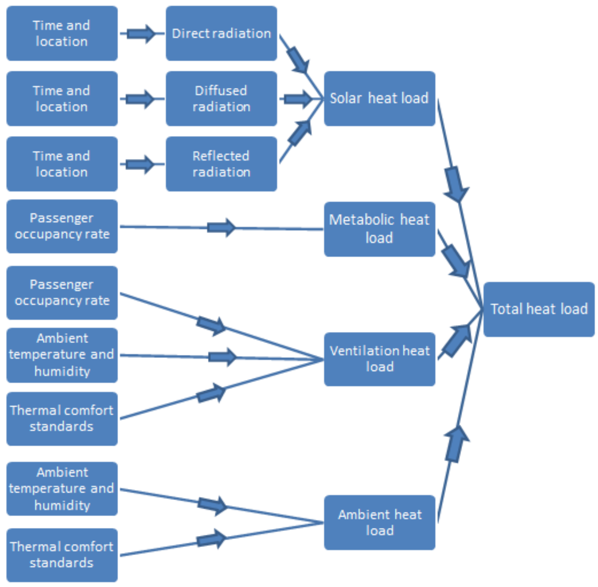

Figure 2. Flow chart of the calculation

In the first section, metabolic, solar, ambient and ventilation heat models are introduced and thermal comfort standards are presented. In the second section, firstly, thermal properties of the vehicle, the location and operation system of the line are presented. After that, the total heat load of the Metrobus for a period of one year is predicted according to the model by considering the weather condition, bus schedule and passenger occupancy rates. Finally the required HVAC system is selected according to the results of the calculations.

The details of the calculations and the flow sequence are summarized in Figure 2.

BACKGROUND

Heat Model

Heat gain refers to the transfer of heat into or out of the bus cabin through a variety of components. There are nine main components of heat gain of an internal combustion bus. These components can support the HVAC system (for example, radiation that heats up the bus in a cold winter day) or can work against the HVAC system (for example, metabolic heat of passenger that heats up the bus in a hot summer day). In order to determine the HVAC requirements of a bus, the heat load is calculated by an inductive method. Here, the model developed by Fayazbakhsh et al. (2013) for a typical internal combustion car cabin is adapted for the electric bus. The heat load model developed for a typical car cabin is given in equation (1).

$$\dot{Q}_{Total} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \dot{Q}_{Amb} + \dot{Q}_{Exh} + \dot{Q}_{Eng} + \dot{Q}_{Ven} + \dot{Q}_{AC} \quad (1)$$

$\dot{Q}_{Total}$ refers to the net total heat flow into bus cabin, $\dot{Q}_{Met}$ is the metabolic heat flow generated by passengers, $\dot{Q}_{Dir}$ is the heat flow of direct radiation, $\dot{Q}_{Dif}$ is the heat flow caused by diffused radiation, $\dot{Q}_{Ref}$ is the heat flow of reflected radiation, $\dot{Q}_{Amb}$ is the heat exchange between ambient and bus cabin, $\dot{Q}_{Exh}$ is the heat flow due to exhaust system, $\dot{Q}_{Eng}$ is the heat flow of the engine, $\dot{Q}_{Ven}$ is the heat flow generated by the air that is blown by HVAC system into bus cabin so as to supply fresh air for passengers and $\dot{Q}_{AC}$ is the thermal load created by the air conditioning system to reach the comfort temperature.
Since, there is no exhaust system, $\dot{Q}_{Exh}$ is omitted and since the heat generation of the electric motor is negligible compared to the internal combustion engine $\dot{Q}_{Eng}$ is also assumed to be zero. Thus, equation (1) becomes;

$$\dot{Q}_{Total} = \dot{Q}_{Met} + \dot{Q}_{Dir} + \dot{Q}_{Dif} + \dot{Q}_{Ref} + \dot{Q}_{Amb} + \dot{Q}_{Ven} + \dot{Q}_{AC} \quad (2)$$

Hence, in order to calculate total heat flow into an electric bus cabin equation (2) can be used.

In Figure 3, thermal loads acting on an articulated bus are illustrated. In the following sections each component is explained in detail.

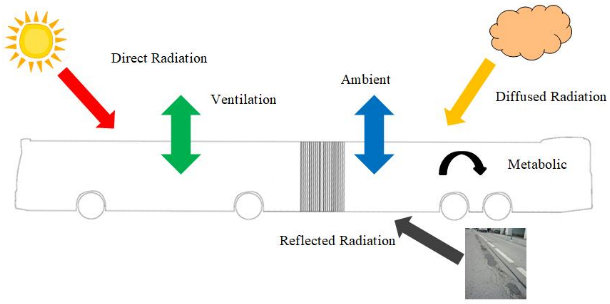

Figure 3. Representation of thermal loads

Metabolic Heat Load

Passengers in the bus cabin create heat load because of their metabolic activities. For the calculation of metabolic load of passengers, they are separated into three different groups; driver, sitting passenger and standing passengers. Therefore, the metabolic load can be calculated from equation (3);

$$\dot{Q}_{Met} = \dot{Q}_{Driver} + n_{sitting}\dot{Q}_{sitting} + n_{Standing}\dot{Q}_{Standing} \quad (3)$$

where $\dot{Q}_{Driver}$ refers to the metabolic load of driver and $\dot{Q}_{sitting}$ and $\dot{Q}_{Standing}$ are the metabolic loads of sitting passengers and standing passengers, respectively. Number of passenger is represented with $n_{sitting}$and number of standing passenger is represented with $n_{Standing}$. Metabolic load of a person can be calculated from equation (4);

$$\dot{Q}_{Met\ one\ Person} = MA_{DU} \quad (4)$$

where metabolic heat production rate, M, of a sitting passenger and standing passenger are defined as 60 W/m^2 and 70 W/m^2, respectively (ASHRAE 2001). Metabolic heat production rate of a heavy vehicle driver is 185 W/m^2 while this value ranges between 60 W/m^{2}and 115 W/m^2 for a car driver so for an articulated bus it is assumed as 120 W/m^2 (ASHRAE 2001).

A_{DU} refers to the DuBois area which estimates the surface area of a person from the mass and height of one person. (5) (Lee et al. 2008)

$$A_{DU} = 0.202\ x\ Weight^{0.425}\ x\ Height^{0.725} \quad (5)$$

$Weight$ and $Height$ refer to mass in kilogram and height in meter of a person, respectively.

Solar Heat Loads

Solar heat load is divided into three components; direct radiation, diffused radiation and reflected radiation. Total solar heat load is the summation of these three components. The solar gain model assumes clear sky conditions for sizing purposes.

Geometry of a bus is assumed to be a rectangular prism. Solar angle of a surface is given in Figure 4 (Goswami et al. 2015).

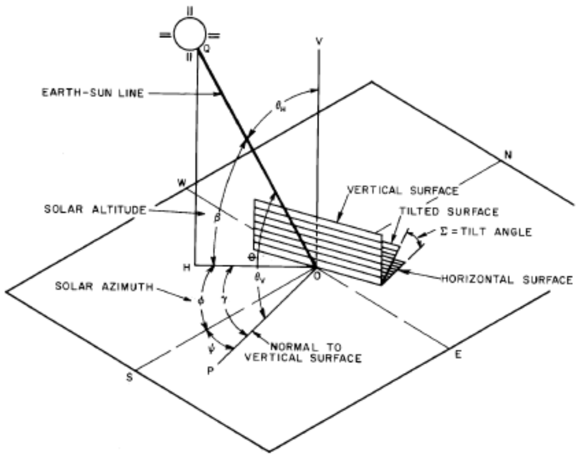

Figure 4. Solar angle of a surface (Goswami et al. 2015)

For any surface, the angle between the surface normal and the position of the sun which is the incident angle, θ, can be found from equation (6).

$$\theta = \cos^{-1}(\sin\beta \times \cos\sum + \cos\gamma \times \cos\beta \times \sin\sum) \qquad (6)$$

where, γ is the azimuth angle, β is the altitude angle and $\sum$ is the surface tilt angle. β can be calculated from equation (7)

$$\beta = \sin^{-1}(\sin L \times \sin\delta + \cos L \times \cos\delta \times \cos H) \qquad (7)$$

where, δ is the solar declination, L is the latitude and H is the hour angle, where

$$\mathrm{H} = 15(Atlantic\ Standard\ Time - 12) \qquad (8)$$

Solar declination, δ, can be calculated from equation (9).

$$\delta = 23.45\sin([360(284+n)]/365) \qquad (9)$$

where, n represents the day of the year.

Direct Radiation

Direct solar radiation (I) is the proportion of the almost rectilinear solar radiation, which reaches the Earth's surface from an angle with a distance of 0.25° to the center of the sun and reaches a normal area, which is oriented perpendicularly to the direction of the radiation.

Direct radiation load can be calculated from equation (10) (Fayazbakhsh et al. 2013);

$$\dot{Q}_{Dir} = S\tau I_{Dir}\cos(\theta) \qquad (10)$$

where, S is the area of the surface, τ is the transmissivity of the surface, I_{Dir} is the direct normal irradiance and $\dot{Q}_{Dir}$ is the heat gain due to direct radiation.

Irradiance of the direct radiation can be calculated from equation (11) (ASHRAE 2001).

$$I_{Dir} = \frac{A}{e^{\left(\frac{B}{\sin(\beta)}\right)}} \qquad (11)$$

A and B are constants that change with monthly. Values of A and B that are obtained from ASHRAE Handbook of Fundamentals (2001) are given in Table 1. Data are for 21st day of each month (ASHRAE 2001). To increase the resolution of calculation A, B and C values are interpolated to their daily values.

Table 1. Extraterrestrial Solar Irradiance and Related Data

Month	A $[\frac{W}{m^2}]$	B $[\frac{1}{airmass}]$	C
January	1230	0.142	0.058
February	1215	0.144	0.060
March	1186	0.156	0.071
April	1136	0.180	0.097
May	1104	0.196	0.121
June	1088	0.205	0.134
July	1085	0.207	0.136
August	1107	0.201	0.122
September	1151	0.177	0.092
October	1192	0.160	0.073
November	1221	0.149	0.063
December	1233	0.142	0.057

Diffused Radiation

Diffused radiation is the portion of solar radiation which arrives on the surface of the Earth after single or repeated dispersion in the atmosphere (Goswami 2015). The diffused radiation heat gain can be calculated from equation (12) (Fayazbakhsh et al. 2013).

$$\dot{Q}_{Dif} = S\tau I_{Dif} \qquad (12)$$

where, I_{Dif} is given in equations (13a) and (13b) (ASHRAE 2001). Equation (13a) is for surfaces other than vertical and equation (13b) is for vertical surfaces.

$$I_{Dif} = CI_{Dir}\frac{1+\cos(\sum)}{2} \qquad (13a)$$

$$I_{Dif} = CI_{Dir}(0.55 + 0.437\cos(\theta) + 0.313\cos^2(\theta)) \qquad (13b)$$

For vertical surfaces of the bus such as front, back, left and right surfaces equation (13b) is utilized.

Reflected Radiation

Reflected radiation refers to the part of radiation heat gain that is reflected from the ground and with strikes the body surfaces of the vehicle (Fayazbakhsh et al. 2013). Heat gain due to reflected radiation can be calculated from equation (14).

$$\dot{Q}_{Ref} = S\tau I_{Ref} \qquad (14)$$

where, I_{Ref} is defined in equation (15) for all surface orientation (ASHRAE 2001).

$$I_{Ref} = I_{Dir}(C + \sin(\beta))\rho_g \frac{1 - \cos(\Sigma)}{2} \quad (15)$$

where, ρ_g is the reflectivity coefficient of the ground.

Ambient Heat Load

There may be difference between outside temperature and inside temperature. This temperature difference causes heat transfer through surfaces from outside into bus cabin on hot days and heat transfer from bus cabin to outside on cold days. The heat transfer can be calculated from equation (16) (ASHRAE 2001).

$$\dot{Q}_{Amb} = SU(T_s - T_i) \quad (16)$$

Here, S is the area of the surface, U is the overall heat transfer coefficient of the surface, T_s and T_i refer to outer surface temperature and cabin temperature of the bus, respectively. U is composed of three parts; conduction of the body, convection of inner and outer surfaces, respectively. U can be calculated from equation (17).

$$\frac{1}{U} = \frac{1}{h_o} + \frac{\lambda}{k} + \frac{1}{h_i} \quad (17)$$

where λ is the thickness of the material, k is the conduction coefficient of the body, h_o and h_i are the convection coefficients of the outer and inner surfaces, respectively. h_i and h_o are assumed only convective and calculated from equation (18a) and equation (18b) (Li et al. 2013).

$$h_i = 9 + 3.5v^{0.66} \quad (18a)$$
$$h_o = 9 + 3.5v^{0.66} \quad (18b)$$

where, v is the relative speed of the air in m/s with respect to corresponding surface. Since the air in the bus cabin assumed to be stationary, from equation (18a) h_i becomes 9 W/m^{2}K and h_o is calculated according to velocity data obtained from the driving cycle test.

Ventilation Heat Load

Passengers in the bus consume oxygen for breathing. Since the volume of the bus cabin is limited for the long-term O_2 needs of passengers, fresh air must be supplied into the bus cabin. According to ASHRAE Standard 62 it is recommended to supply 8L of fresh air per second for one passenger (ASHRAE 1999). The heat load because of air exchange between outside and bus cabin can be calculated from equation (18a) and equation (18b). The first part of the equation is for the calculation of sensible heat load, while the second part is for the calculation of latent heat load.

$$\dot{Q}_{Ven} = \dot{V}_{ven}\rho c_p \Delta T + \dot{V}_{ven}\Delta W(4775 + 1.998\Delta T) \quad (19)$$

where, $\dot{V}_{ven}$ is the air flow rate, ρ is the air density (1.2 kg/m^3), c_p is the specific heat of air (1000 J/(kgK)) and ΔT and ΔW are the temperature difference and humidity ratio difference, respectively.

Thermal Comfort Requirements

Acceptable ranges of operative temperature and humidity for people in typical summer and winter clothing during primarily sedentary activity are defined in Table 2. For the design of the HVAC system of the bus, temperature and relative humidity range that is shown in blue in Figure 5 is selected (ASHRAE 1999).

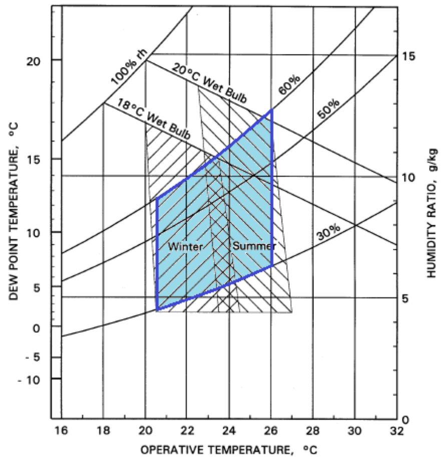

Figure 5. Thermal comfort range of temperature and humidity (adapted from ASHRAE standard 62 (1999))

Temperatures and humidity ratios that are selected as design requirements for HVAC system for different season of year are represented in Table 2.

Table 2. Design temperatures and humidity ratio

Season	Temperature range (°C)	Humidity range (%)
Winter	20.5-23	30-60
Spring	22-24.5	30-60
Summer	23.5-26	30-60
Fall	22-24.5	30-60

DETERMINATION OF HVAC REQUIREMENTS

Location and Weather

In this part of the study, Metrobus Line of Istanbul is analyzed to determine the HVAC heat load requirements of a bus line. Latitude and longitude of Istanbul are 41.015° and 28.979°, respectively. For the calculations, hourly outside temperature and relative humidity data are obtained from Turkish State Meteorological Service. The Metrobus network is mainly established on the East-West axis. Metrobus line is shown in Figure 6.

Figure 6. Map of the Metrobus Line

Operation System of the Bus Line

The Metrobus line has one route. However, on the same route there are separate lines with different start and end stations. In addition, these different lines have different working hours and time tables but, working hours are usually between 06:00 and 24:00. However, there is one line that is only active at night passing through all 44 stations. For HVAC calculation only lines that are active from 06:00 to 24:00 are considered.

Passenger Compartment

Passenger capacity of the 18-m articulated Metrobus is 160. Number of seats is 43+1 (one for driver). Surface areas, S, that are obtained from the technical drawings of the Metrobus and transmissivity, τ, of the surfaces are shown in Table 3. According to the safety regulation, the transmittance of visible light through the window should be at least 70% in the US and 75% in Europe. Transmissivity of the windows of the Metrobus is assumed to be 0.75 according to J.W.Lee et al (2014).

Table 3. Transmissivity and surface area of each bus surface

Surface	Part	Material	Area	Trans-missivity
Front	Wind-shield	Glass	$3.3m^2$	0.75
	Body	Al	$4.2m^2$	0
Rear	Windows and doors	Glass	$1.9m^2$	0.75
	Body	Al	$5.2m^2$	0
Left	Windows	Glass	$17.1m^2$	0.75
	Body	Al	$39.6m^2$	0
Right	Windows	Glass	$17.6m^2$	0.75
	Body	Al	$39.1m^2$	0
Top and bottom	Body	Al	$45.9m^2$	0

In the analysis, It is accepted that surfaces remove the solar radiative heat load that is absorbed through convection because there is always air flow around the body. Thus, it is assumed that there is no heat gain from aluminum surfaces.

Analysis

Since the passenger occupancy rate, which affects the thermal load most, is similar in both directions, HVAC power requirement is calculated for one direction.

Metabolic Heat Load

Hourly passenger density data for one year between each station of the Metrobus line was obtained from the bus service operator. That is an important data to calculate heat loads related to occupancy rate. Heat map of passenger occupancy rate is shown in Figure 7. Horizontal axes shows hours, vertical axis shows the station number and the color indicates the occupancy rate, occupancy rate of 1 means full bus while 0 means empty bus.

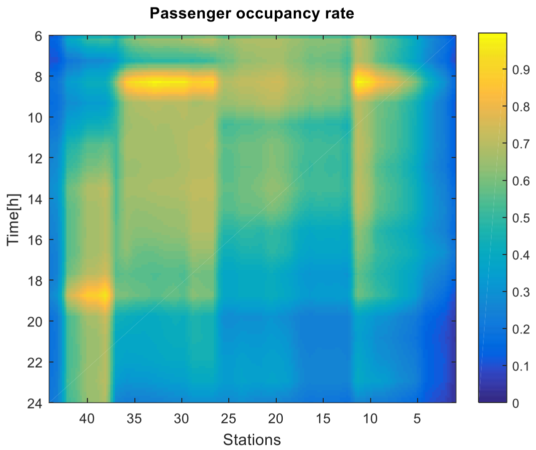

Figure 7. Heat map of passenger occupancy rate

According to Turkish Statistical Institute Health Survey (2010), average height and average weight of Turkish people are 1.67 m and 71.5 kg, respectively. Thus, from equation (20) A_{DU} of an average Turkish person is calculated as follows.

$$A_{DU} = 0.202\ x\ 71.5^{0.425}\ x\ 1.672^{0.725} = 1.80\ m^2 \quad (20)$$

With the help of equation (4) and (20) metabolic heat generation rate of a driver, sitting passenger and a standing passenger is calculated as 216 W, 108 W and 126 W, respectively.

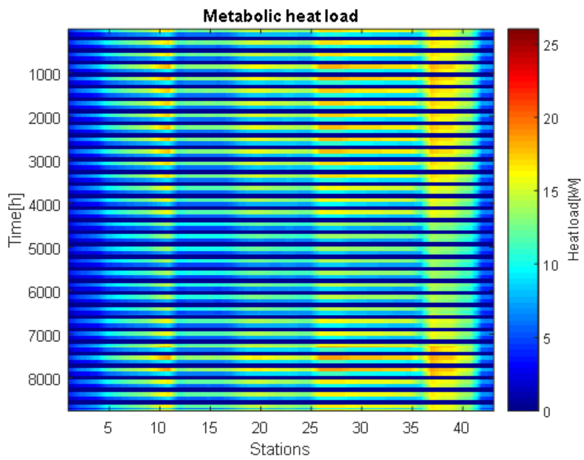

Figure 8. Heat map of metabolic heat load

It is assumed that passengers prefer to sit in the bus when there is empty space. There are only 43 seats in the Metrobus but the passenger capacity is 181. Accordingly, heat map of metabolic heat load is shown in Figure 8.

Solar Heat Loads

As can be seen in Figure 6, the Metrobus line is mainly established on the East-West direction thus, solar heat load is calculated for East-West direction.

Hourly incidence angle of each surface for one year, which are calculated from equation (6), are shown in Figure 9.

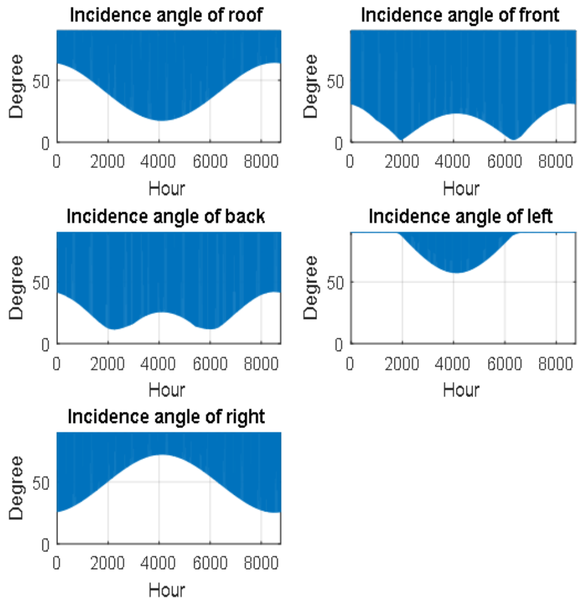

Figure 9. Incidence angle of all surfaces

Hourly altitude angles of all surfaces for a one year period, which are calculated from equation (7), are shown in Figure 10.

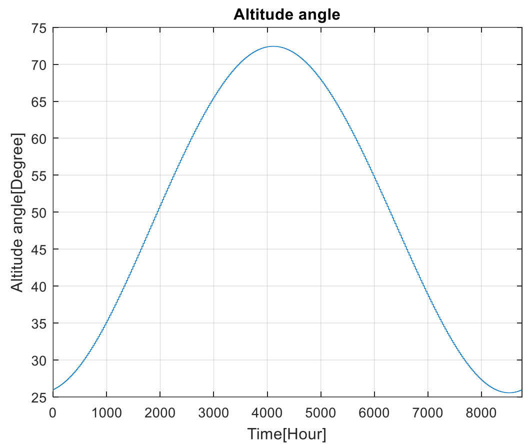

Figure 10. Altitude angles

Sunset and sunrise hours for a one year period are shown in Figure 11.

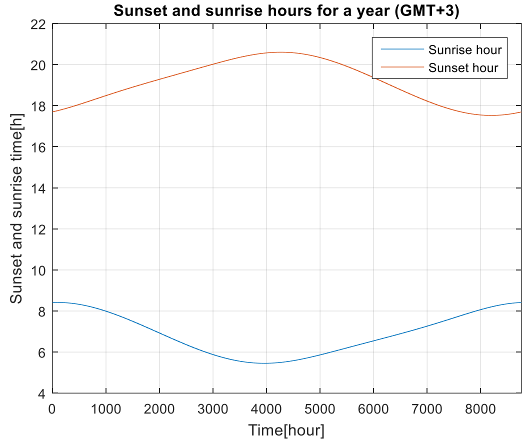

Figure 11. Sunset and sunrise hours

Direct Radiation Heat Load

Direct radiation heat loads for each hour are calculated from equation (10). Transmissivity values and surface areas, which are required in order to calculate direct radiations, are obtained from Table 3. Results of calculation are shown in Figure 12.

Diffused Radiation Heat Load

Heat loads of diffused radiation are obtained from equation (13a) and equation (13b). Heat load that is generated by diffused radiation for each hour of a year are shown in Figure 12.

Reflected Radiation Heat Load

Heat load due to reflected radiation is calculated from equation (14). It is assumed that sunlight is reflected only from the ground (asphalt) and, reflectivity coefficient of the ground (asphalt), ρ_g, is 0.2 according to Tran et al. (2009). Results are shown in Figure 12.

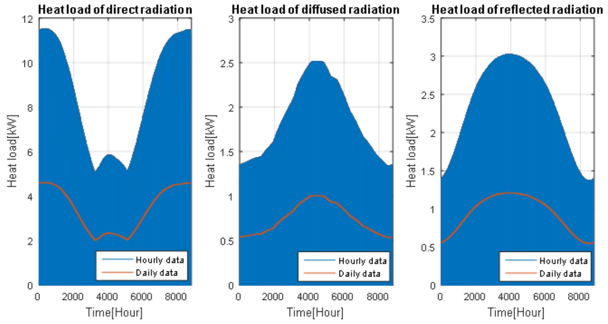

Figure 12. Direct, Diffused and Reflected radiation heat load

Ambient Heat Load

Thickness of the body and glass, λ, may vary from vehicle to vehicle and is easy to measure. In the present case vehicle body thickness is measured as 17mm and glass thickness is measured as 4.8mm. Thermal conductivity of the glass is 1.05 W/mK and thermal conductivity of the body is 0.2 W/mK. (Fayazbakhsh et al. 2013).

For the calculation of ambient heat load, inner temperature of the Metrobus is required. For practical reasons, inner temperature of the bus is assumed to be always in comfortable ranges. Comfortable inner temperature and relative humidity ranges were tabulated in Table 2.

In addition to inner temperature, outer temperatures are also required. Hourly outside temperature data are obtained from Turkish State Meteorological Service.
Temperature difference between outside and inside of the Metrobus in a one year period are given in Figure 13.

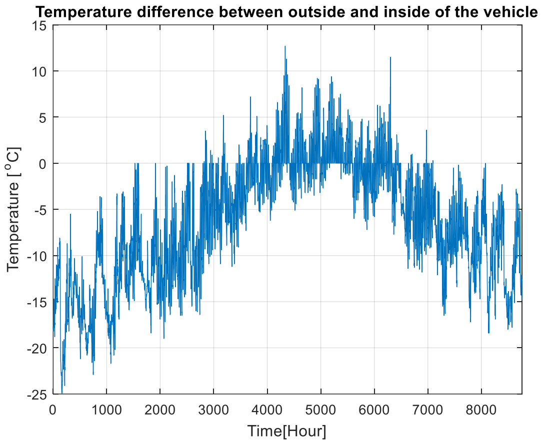

Figure 13. Hourly temperature difference between outside and inside of the vehicle

Convection coefficient of the outer, h_o, and inner, h_i, surfaces are calculated from equation (18a) and equation (18b). Velocity of the inside air is assumed to be zero and the velocity air on the outer surface of the Metrobus is assumed to be 11.33 m/s which is the average velocity of the Metrobus according the driving cycle obtained by tests with zero wind speed assumption.

Ambient heat load due to conductive and convective heat transfer through body and windows for each hour of a year is calculated from equation (17) and result are shown in Figure 14.

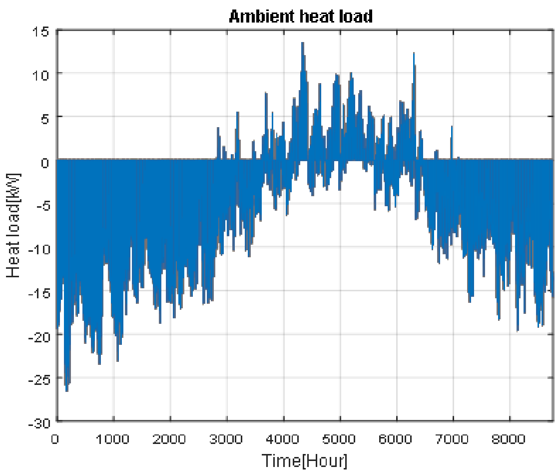

Figure 14. Ambient heat load

Ventilation Heat Load

The temperature and relative humidity must be changed according to thermal comfort requirements so as to keep the interior of the bus comfortable. Target inner temperature and relative humidity ranges are tabulated in Table 2. For the calculation of ventilation load, hourly outside temperature and humidity data that are obtained from Turkish State Meteorological Service are used. In addition to outdoor temperature and target inside temperature, another important data for thermal comfort, target relative humidity and inside relative humidity are shown in Figure 15.

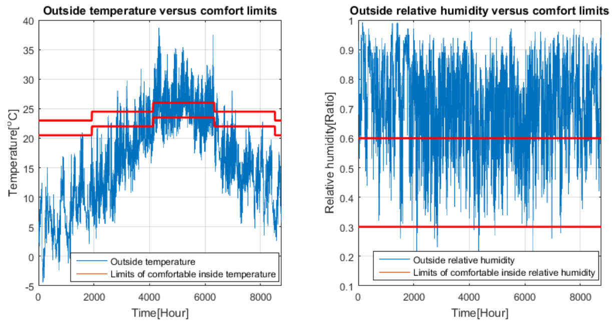

Figure 15. Outside temperature and target temperature (left), Outside relative humidity and target relative humidity (right)

Since the ventilation requirements are directly dependent on the number of passenger, hourly passenger occupancy rate of each station for one year are used in this calculation. Ventilation heat load of the Metrobus are calculated for each hour of the year by using equation (19). Results of the calculation are shown in Figure 16.

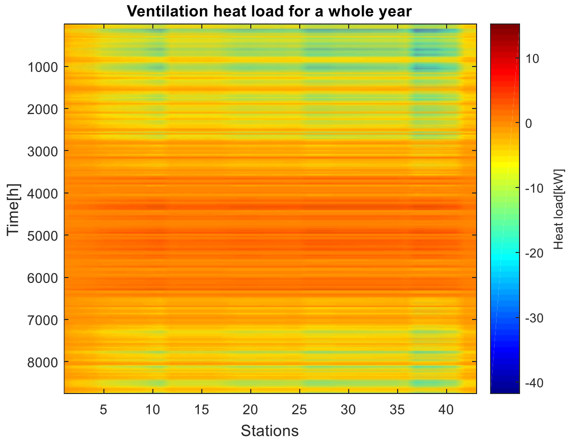

Figure 16. Ventilation heat load

Results

Total heat load of the Metrobus for each hour for one year and for each station is shown in Figure 17.

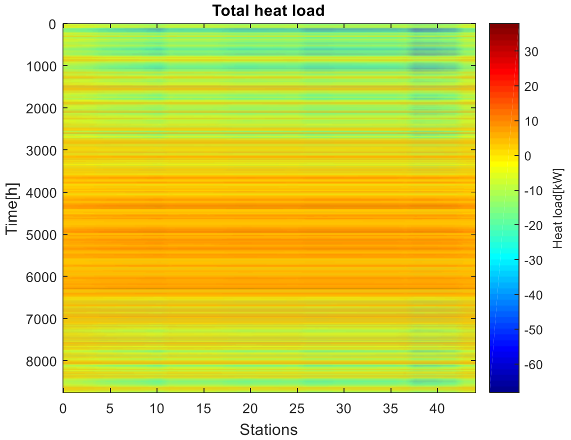

Figure 17. Total heat load

Total heat load of the Metrobus for each service hour for one year is shown in Figure 18. The heat load that is shown in Figure 18 is the average heat load of all stations.

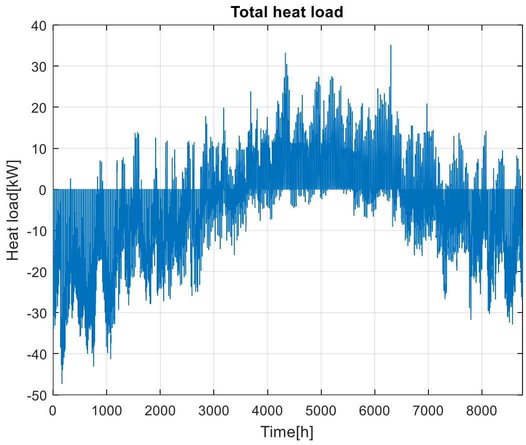

Figure 18. Hourly total heat load

Daily average and monthly average heat load of the Metrobus are shown in Figure 19.

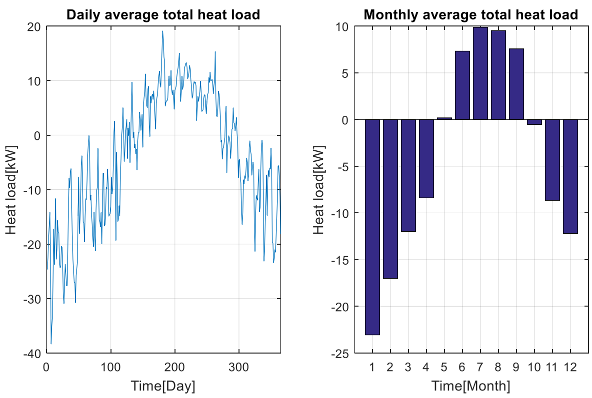

Figure 19. Daily and monthly total heat load

Power requirement of HVAC System

The energy-based efficiency measure of the refrigeration unit, COP (Coefficient of performance) can be defined as follows;

$$COP_{cooling} = \frac{\dot{Q}_{HVAC\ cooling}}{W_{Comp}} \tag{21a}$$

$$COP_{heating} = \frac{\dot{Q}_{HVAC\ heating}}{W_{Comp}} = \frac{\dot{Q}_{HVAC\ cooling} + W_{Comp}}{W_{Comp}} = COP_{cooling} + 1 \tag{21b}$$

$\dot{Q}_{HVAC\ cooling}$ and $\dot{Q}_{HVAC\ heating}$ are the heat loads during cooling and heating, respectively and W_{Comp} is the minimum power that is needed to be supplied to the compressor. COP may vary according to different months, temperature and air mixture ratios. Average coefficient of performance of an air conditioning system of a bus is 3 for cooling and is 4 for heating (Tosun et al. 2016). Thus, according to Figure 19 it can be said that the maximum power needed to heat up the Metrobus is about 12 kW and the maximum power to cool down the Metrobus is about 11 kW. The total energy consumption of the AC system for one year can be calculated from equation (26).

$$W_{average\ day} = \int W dt \tag{22}$$

From equation (21a), (21b) and (22) average HVAC power for one day, $W_{average\ day}$, is calculated. Figure 20 shows the change of average HVAC power by day.

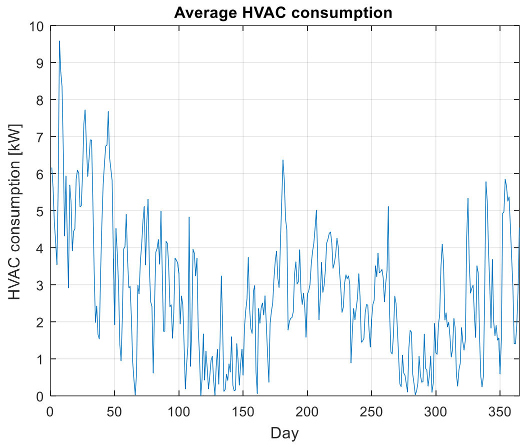

Figure 20. Daily change of HVAC power requirement

CONCLUSION

Accurate calculation of the thermal load of an urban bus in the design phase is crucial for the assessment of the comfort and cost of the bus to be produced. In this study, the thermal load of the Istanbul Metrobus has been estimated by a method that takes into account the annual weather and passenger density with an hourly resolution. Owing to these detailed inputs, short-term fluctuations could be observed and thus an appropriate HVAC system could be selected. Furthermore, average and overall HVAC consumptions have been estimated with higher resolution which is a valuable input for the feasibility of bus electrification projects.

It can be observed that, from the Metrobus line analyzed in this study, the HVAC loads are higher in winter time. The reason for this is that the heating demand is higher due to lower outdoor temperatures. In vehicles with internal combustion engine, heat may be supplied from the waste heat of the internal combustion engine. Meanwhile, in electric vehicles there is no waste heat, and therefore heat must be supplied by the HVAC system.

The ventilation requirement of the vehicle has been observed to depend on the number of passengers in the vehicle; in fact, the number of passenger is the most important input of the HVAC power consumption model. Daily average power consumption of HVAC system varies between 1 kW and 9 kW for HVAC system with COP equal to 3 for cooling and 4 for heating. The yearly average of the HVAC energy consumption is 0.11 kWh/km which represents around 8 % of the total energy consumption of the vehicle, but in winter the average consumption can increase up to 0.27 kWh/km, which is about 25% of the total consumption of the vehicle.

Automatic HVAC controllers used in today's buses activate the HVAC system according to the temperature and humidity of the current air in the vehicle. However, these controllers cannot predict the need of rising thermal loads due to solar radiation, passenger number or outdoor temperature. Finally, as a prospective application, a HVAC controller designed based on the proposed model may be developed to provide necessary pre-heating and pre-cooling in order to keep the bus cabinet in a comfortable temperature range in face of varying thermal load.

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