KTH ROYAL INSTITUTE
OF TECHNOLOGY

2023

Master's Degree Project in Sustainable Energy Utilization

Second cycle 30 credits

Optimization of comfort-related energy and thermal comfort for commuter trains – A case study of Stockholm commuter trains

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TRITA-ITM-EX 2023:402

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Examination: 8 June 2023

Trita: ITM- EX 2023:402

KTH Royal Institute of Technology

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Stockholm, Sweden

2023

Abstract

Today, energy efficiency is an increasingly important question in which progress is accelerating. One of the high electricity demand energy users is public transport. In addition, passenger satisfaction with thermal comfort is an important parameter. Consequently, it is essential to consider thermal comfort in combination with the energy-saving measures of thermal-related functions. Unfortunately, there have not been a lot of investigations into improving the energy efficiency of the thermal-related functions on commuter trains, where most of the focus has been on traction energy. The first part consisted of a literature study to explore thermal-related functions, energy saving, and thermal comfort of commuter trains. At least three articles have utilized the methodology of evaluating energy measures in IDA ICE; however, none of them has considered door openings that are frequent on commuter trains. The literature study concluded which efficiency measures can be applicable for short-haul distance trains and typical approaches to evaluating thermal comfort.

A commuter train is a complex and transient thermal environment, with passengers entering and leaving the train in short intervals, airflows, temperature fluctuations, and the train's movement. Simplifications of the model were made to simulate the average ambient conditions. To validate and adapt the IDA ICE model, experimental measurements were done during the winter season using a thermal camera, air speed, and temperature measurements. The model was validated through experimental measurements and data analysis. In addition, data analysis was used for evaluating some of the energy measures through available live and history-data of the train fleet. The energy efficiency measures, which are quantifiable, have been quantified using the simulation model in combination with the data-analysis.

Three categories of energy-saving measures are proposed: easily implementable, medium, and measures that require physical changes of the train components. Parking mode has a lot of saving potential of 34 % of annual energy compared to baseline. With a reversible heat pump of 11 kW, heating energy saving of 43 %, and 40 % energy coverage could be obtained, with the potential for up to 100 % energy coverage but being in the category of hard to implement. Door opening reduction with a potential saving of 11 000 kWh per train in annual energy, as compared to the baseline simulated model, being in the category of easy to implement. Balancing temperature heating shutdown could save between 3 100 to 9 500 kWh per train. A setpoint temperature of 18°C could save 16.5 %, and a variable temperature setpoint curve was proposed with similar savings. Ventilation control was among those measures with the highest potential; recirculation, CO_2 and temperature-controlled ventilation heating energy saving of 31 % was simulated. Thermal comfort was improved in the measures affecting thermal conditions. With a setpoint temperature of 18°C during winter and based on clothing values derived from the literature study, an improvement of thermal comfort was observed in the PMV scale for thermal comfort. With combined energy efficiency measures, the simulation results showed a reduction of heating energy of 59 %, and in addition the fan power consumption could be reduced in magnitude of up to 12 000 kWh per train and year.

Finally, further suggestions on research within the area were proposed, mainly to make more long-time measurements and to improve the possibility of energy follow-up by improving the data channels and available information.

Key words: Energy efficiency, thermal comfort, commuter train, ventilation, cooling, heating, energy utilization, case study, temperature, transient

Sammanfattning

Idag är energieffektivitet en viktigare fråga där framstegen har accelererarat. En av energiförbrukarna med hög efterfrågan på elektricitet är kollektivtrafiken. Passagerarnas tillfredsställelse av termisk komfort är en viktig parameter. Följaktligen är det viktigt att analysera termisk komfort i kombination med energibesparingsåtgärder. Det har inte gjorts mycket tidigare forskning inom att förbättra energieffektivitet inom termiska funktionerna på pendeltåg i Sverige, främst har fokus varit på energi för framförandet av fordonen. Den första delen av projektet bestod av en litteraturstudie för att få en uppfattning av vilka typer av energibesparingsåtgärder som är lämpliga för kortdistanståg som också skiljer sig mycket från långdistanståg. Tre artiklar hittades som har använt IDA ICE för att utvärdera energiåtgärder på tåg, men ingen av dem har tagit hänsyn till dörröppningar som är väldigt frekventa på pendeltåg. Flera metoder undersöktes för att bedöma termisk komfort och för att bygga upp en modell och validera den.

Ett pendeltåg är en komplex och transient termisk miljö, där passagerarna går in och lämnar tåget i relativt korta tidsintervall jämfört med långdistanståg. Men det ansågs fortfarande tillräckligt långa tidsintervall för att kunna använda PMV för att bedöma termisk komfort. Jämfört med en byggnad, så sker det stora temperatur-fluktuationer, variationer i antal passagerare, och en kontinuerlig förflyttning av tåget med nya omgivningsförhållanden. För att validera och anpassa IDA ICE-modellen gjordes experimentella mätningar under vintersäsongen med hjälp av värmekamera, lufthastighetsmätning och temperaturmätningar. Modellen kunde valideras med till exempel yttemperaturer, energisignatur från dataanalys, och hur lång tid det tar för modellen att tappa temperatur jämfördes med ett liknande tåg, X61. Dessutom användes dataanalys för att utvärdera några av energiåtgärderna. Energieffektivitetsåtgärdena som är kvantifierbara, har simulerats för hur mycket energibesparingspotential som finns.

Tre kategorier av energibesparingsåtgärder föreslås, lätt genomförbara, medelsvåra, och svåra där fysiska förändringar av tågkomponenterna krävs. Parkeringsläge har stor energibesparingspotential på 34 % av den årliga uppvärmningsenergin jämfört med bas-scenariot. Reversibel värmepump på 11 kW, har värmeenergibesparingspotential på 43 % och 40 % energitäckning, med stor potential för upp till 100 % energitäckning, men i kategorin svår att implementera. Manuellt öppningsbara dörrar på vintern har besparingspotential i värmeenergi på 11 000 kWh per tåg, jämfört med bas-scenariot. Balanstemperatur har besparingspotential på 3100 till 9500 kWh per tåg. Börvärdetemperatur på 18 °C jämfört med 20°C kan spara 16.5 % värmeenergi, samt så föreslogs en variabel temperaturkurva. Med en börvärdetemperatur på 18 °C under vintern, baserat på värden av isolering av kläder från litteraturstudien, observerades en förbättring av termiska komforten i PMV-skalan för termisk komfort. Med kombinerade energieffektivitetsåtgärder visade simuleringsresultateten en minskning av värmeenergi med 59 %, och dessutom kunde fläktarnas energianvändning minskas i storleksordningen om upp till 12 000 kWh per tåg och år.

Slutligen föreslogs ytterligare förslag på forskning inom området, främst att göra fler långtidsmätningar och förbättra möjligheten till energiuppföljning genom att förbättra datakanalerna och tillgänglig information.

Nyckelord: Energieffektivitet, termisk komfort, pendeltåg, ventilation, kyla, värme, energianvändning, fallstudie, temperatur, transient

PREFACE

I am pursuing this as the last part of my master's program Renewable Energy Technology, with the profile of energy utilization. My focus has been energy utilization and thermal comfort in buildings. So, this is a new application of energy utilization and thermal comfort in a more transient environment and into the railway topic. Looking forward to going deep into the topic of energy utilization in thermal functions on commuter trains.

A big thanks to Sidharth Kapoor, researcher at KTH within the railway, and my supervisor Zhendong Liu who assisted me. I appreciated the continues meetings and assistance that I got from them. Thanks to my examiner Jaime Arias for your valuable input.

- Thanks to Joel Forsberg, energy specialist and other colleges at MTR.
- Thanks to Erik Dunkars, energy strategist on Trafikförvaltningen (TF) and the other colleges at TF and SL.
- Thanks to Professor Mats Berg within the railway department at KTH and the other participating colleges in the monthly meetings.

I hope this thesis creates some attention within energy for the thermal comfort of trains and that it may introduce new research topics within this area.

Stockholm, Sweden 2023

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NOMENCLATURE

T99 – Response time of temperature sensor to reach 99 % of end temperature.

PMV – Predicted mean vote.

PPD – Predicted percentage dissatisfied

PD – Percentage dissatisfied

MET- M – Metabolic rate

W – Rate of mechanical work

P_a – water vapor partial pressure

t_a – Air temperature

$\bar{t}_r$ – Mean radiant temperature.

f_{cl} – Clothing

t_{cl} – Clothing surface temperature

h_c – Convective heat transfer coefficient

Clo – Clothing value

C_{res} - Heat exchange by convection in breathing

E_{res} - Evaporative heat exchange in breathing

COP – Coefficient of performance factor

Δt_{CAT} – Allowed temperature range depending on different categories (EN-15251)

S_{CO2} – CO_2 production from a passenger. [L(CO_2)/s]

R - Fraction of recirculated air

C_s - Steady CO_2 concentration (PPM)

In day temperature – Temperature during the door opening time (not during night)

ACH – Air changes per hour

g - Gravitational constant – 9.81 m/s^2

T_{sensor} - Temperature indicated by the temperature sensor

T_1 and T_2 – initial temperature of the air respectively new air temperature (°C)

τ – Thermal time constant (seconds)

t – time (seconds)

Bellow pressure – The pressure caused of the train mass on the suspension – can be used to measure the passenger load.

HVC – High voltage power consumption – Vehicle power consumption [KW]

T_{set} – Temperature setpoint of the train interior [°C]

T_{out} – Temperature outside of the train [°C]

S_{CO_2} – CO_2 production per passenger [L(CO2)/sec]

C_s – Steady state CO_2 concentration [PPM]

R_f – Proportion of recirculated air

R – Recirculated air

F – Fresh air

S – Supplied air

ε_v – Ventilation effectiveness [-]

$\dot{V}_{exh}$ – Exhaust air flow

Φ – Emitted energy from radiation.

σ – Stefan-Boltzmann constant

$t_{\sup ply}$ – Ventilation supply temperature [°C]

M_{air} – Molar mass air

1 INTRODUCTION

1.1 BACKGROUND

Today, energy efficiency is an increasingly important question, with a focus within energy utilization of buildings and public transport. The progress within energy efficiency is accelerating for several reasons, such as high prices and from global and local policies with the aim of improving energy efficiency [1]. The research project Vinnova InfraSweden has a goal of 15 % reduction of electricity for rail traffic, and minimizing operational cost, for more sustainable transport. The research project is financed by Vinnova InfraSweden 2030. The project aims to set up a plan for energy saving in the infrastructure of Sweden.

On commuter trains, thermal comfort is essential to the passengers commuting experience. The Air Handling Unit is the critical component for providing good thermal comfort. The goal of the HVAC is to provide passengers with a high level of thermal comfort [2]. The commuter train in Stockholm has 54 stations with eight lines and a total track length of 241 km. Figure 1 shows the map of the Stockholm commuter train 54 stations, in which the directions are primarily north or south. Approximately 390 000 passengers boarding the trains per working day, and the total number of boarding passengers per year is 20 million [3]. Auxiliary energy on a train can account for 30 % of energy demand [4]. Trains can typically be divided into short- and long-haul distance trains, which differ in thermal comfort criteria and disturbances factors.

A restriction for reducing energy is thermal comfort functions aboard the train. Thus, the passenger’s perception of thermal comfort is important when looking at thermal comfort functions. Therefore, it is an essential part of this project to predict the passengers' thermal comfort depending on certain conditions. For example, on short-haul traveling in commuter trains, many factors influence the perception of thermal comfort, such as the expectation of temperature when entering the train, typical clothing, outdoor temperature, and many others.

Figure 1 - The Stockholm commuter railway map [5]. Updated 2023-02.

Data-analysis methods are generally suitable for analyzing existing objects where big data is available. The energy signature is one example of heating related to the outdoor temperature. However, in the transient conditions of a short-haul train, many factors influence the heating requirement and thermal comfort conditions. As so many fluctuations and different factors are involved, analyzing trains' energy usage and thermal comfort is a complex matter.

Passengers' willingness to pay for different comfort functions, for example, improved air quality and temperature, is around +12 % of the baseline ticket price [6]. The highest passenger valuation for improved train traveling is the travel time, which is up to around 20-25 % in relation to the first mentioned. The cost of a better train travel depends heavily on space utilization; as mentioned, the number of passengers is 390 000 boarding passengers per day, and the average passenger load depending on the hour of day will be presented in the report. In addition, there are many other comfort functions on trains in general, but not all of them are considered in this report, in which only thermal and energy aspects are considered. Examples of comfort related systems that are considered are lighting, door opening, ventilation, air conditioning, heating, air quality, and door operation.

1.2 AIM OF THE PROJECT

This research project aims to investigate the energy-saving possibilities related to thermal-comfort functions on commuter trains and the constraints in relation to thermal comfort. In addition, the goal is to identify the potential of these energy-saving measures. The research questions introduced in the thesis are the following:

- What is the optimum control strategy of the thermal-related functions on the commuter train in Stockholm for optimal energy usage and thermal comfort?
- Which factors influence thermal comfort-related energy consumption the most ?
- Which energy-saving measures have the most potential in commuter trains?
- Does all the energy used for the thermal comfort-related functions today on the Stockholm commuter train create good value for the thermal comfort perception of the passengers?
- What factors influence the perception of thermal comfort on a short-travel train?
- Can thermal comfort be enhanced in combination with energy efficiency measures in commuter trains?
- Which methodology is suitable for evaluating thermal comfort with energy-saving measures on trains?
- What are the differences in possible energy efficiency measures and thermal comfort aspects between short-haul distance trains and long-haul distance trains?

1.3 TASKS

- Literature study

The first part of the literature study investigates previous work on this topic and what energy efficiency measures might suit commuter trains in Stockholm. The second part of the literature study is about suitable methodologies to investigate the proposed energy efficiency measures and thermal comfort aspects, which is combined in the methodology section with the analysis.

- Data collection. Finding the relevant data for comfort-related functions.
- Analyzing the data in terms of energy usage and thermal comfort.
- Experimental measurements in winter conditions.
- Modeling the train, using the experimental measurements as calibration method.
- Sensitivity analysis – Which parameters are essential regarding energy and thermal comfort?
- Quantifying the potential for energy saving, using either the data collected, analytical method, the simulation model, or a combination of them.
- Thesis report writing and presenting the results for the different stakeholders.

2 LITERATURE STUDY

2.1 PROJECTS WITHIN THE TOPIC

A previous study [7] for passenger trains developed an energy model that examined energy usage in the operational cycle. The model was applied to two different train types, one of which was the X61, a shortened version of the model X60. An energy-saving feature is the parking mode which turns off and reduces temperature overnight. However, it was discovered that the parking mode was not used to a large extent. One of the problems described could be caused by humidity and condensing in the trains due to low temperatures. It is suggested for further research to investigate what would happen if a higher temperature setpoint was used instead, such as 15 °C. Auxiliary equipment accounted for 25.8 % of total energy and the HVAC cooling and heating itself accounted for 12.6 % of the total energy of the train. The focus of the study, however, is on energy efficiency and not on thermal comfort of the passengers. Thus, investigations regarding thermal comfort in regard to energy efficiency could be an important aspect to study.

Another study [4] has looked at a dynamic simulation approach and evaluated different measures related to HVAC system energy. The model uses available weather data and the schedule of the train to make accurate estimations. It has estimates of the convective heat transfer coefficient in relation to the speed and the other heat losses through the envelope. The modeling is done using TRNSYS. The model is applied to a case study of a train route in Naples in Italy, evaluating the different measures. The measures proposed and evaluated are enhanced windows glazing (3 % saving), improved thermal insulation (3 % saving), LED-lighting (2 % saving), heat pump for heating and chilling (35 %), CO_2 measure (18 %), free cooling (10 %), and heat recovery exchanger (10 %). For the free cooling, it was discovered that it is important to optimize between CO_2 (ventilation requirement for good air quality) and the free cooling supply, to keep supplying air if cooling is necessary even though the air quality is good. Thus, it is important to look at interactions between the different measures to discover how different measures could enhance or lessen each other. For further research, the smart door opening is suggested to be investigated.

In Switzerland, a study [8] was done on solar gains through the windows. It is described as a source of uncontrolled heat transfer. There are two significant issues with the windows: overheating in the summer due to solar gains, and in the winter, heat losses through the windows. The methodology was measuring the glass properties in a test rig and then combining that with an experimental measurement with an outdoor sensor attached to the train to calculate solar gains. The solar gains were then compared with heating and cooling energy usage estimated in IDA ICE. One coach was simulated in IDA ICE, using passenger load calculated from CO_2 concentration. The average U-value of the envelope was 1.5 W/m^2, and the windows were 1.6 W/m^2. However, they did not account for door openings in the simulations.

In Taiwan, a survey approach study [9] has been done on thermal comfort requirements within short-and long-haul vehicles, both buses and trains of different types. The survey was combined with real-time measurement of the thermal comfort parameters, such as local air speed, humidity, temperature, and operative temperature, using instruments in adherence with ISO7726. One of the limitations of the study is that the study was done in a warm climate. It was shown that there was no significant difference between the perception of thermal comfort on trains compared with buses. Another finding was that there was not a strong relation between indoor temperature and clothing values, which suggests clothing adjustment is not consistently done depending on the indoor temperature, but mainly on the outdoor temperature. However, a relationship was established between outdoor temperature and clothing value for short-haul, in the equation (1.1), in which for long-haul distance, the average clothing value (clo) was 0.2 higher compared to short-distance trains. The clothing values range from 0.95 to 0.55 between outdoor temperatures of 20-32°C. This is approximately between 0.1 to 0.15 clo higher than for typical clothing values in offices. In addition, different adjustments behavior of the passengers is observed to be changing seats (27%), window shading (56%), adjustment of temperature setpoint (31%), air outlet adjustment (64%), drinking (29%) and clothing adjustment (62%) for short-haul traveling.

$$clo = 1.587 - 0.032 \cdot t_{out} \tag{1.1}$$

2.2 EN-STANDARD

The European standard (EU-standard) is standards developed by European Committee for Standardization (CEN), European Committee for Electrotechnical Standardization (CENELEC) and European Telecommunications Standards Institute (ETSI). Several of the standards are for the topic of railway [10]. A report from Shift2Rail [2] addresses state of the art HVAC technologies within railway transport. Different factors that affect the HVAC's energy consumption are comfort requirements, train speed, outdoor temperature, occupancy, and train properties. It also brings up how well the HVAC systems can handle the above-mentioned factors.

Different comfort requirements should be considered as specified in EN standards. One of the standards, EN14750-1:2006, should be regarded while regulating temperature for local transportation. In Figure 2, the indoor temperature setpoint (solid line) profile is shown in relation to the mean outdoor temperature. During operation, the mean interior temperature should be within $\pm$ 2 (dashed lines) from the setpoint temperature (solid line).

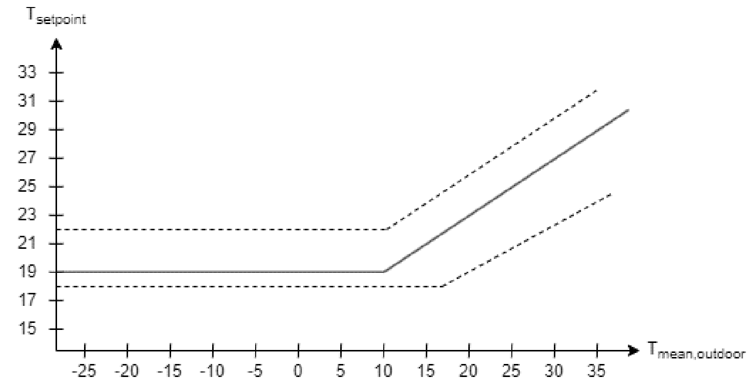


Figure 2 - EN 14750-1:2006 Recommended indoor temperature setpoint for short-haul trains depending on the outdoor temperature. Source: FINE2 [2].

Upon the temperature recommendation in the EN-standard, there are also requirements of air quality (CO_2-level), pressure-comfort, and airspeed of air within the wagon.

Different methodologies are suggested in the article [2] for predicting energy consumption; one is EN-50591. First, the method works as a matrix specifying different operating conditions depending on outdoor temperature, occupancy, and solar conditions. Then three different operating modes are used, in-service with passengers, in-service without passengers, and standby mode. Finally, the total energy consumption for each operating mode is calculated with the operating hours of each point h_i in the matrix with the specified power for each point P_i, according to the equation (1.2) [2].

$$E = \sum_i P_i \cdot h_i \tag{1.2}$$

EN-standard has two different categories of urban trains, category A and B. Category A is for average journeys of more than 20 min and the average time between two stations of more than 3 minutes. Category B has less than 20 min average journey and less than 3 minutes on average between two stations. The table below assumes that X60 in Stockholm is a category A urban rolling stock.

Table 1 - EN-parameters for Urban trains [11]

Explanation	Values
Range of temperature fluctuations from the setpoint temperature in passenger areas	$\pm$ 2°C
Vertical air temperature distribution allowance	4 K
Surfaces (walls and ceilings) temperature in heating mode	10 K less than the mean indoor air temperature
Surfaces (windows) temperature in heating mode	15 K less than the mean indoor air temperature
Air speed at 22 °C	0.25 m/s
Air speed of 27 °C	0.8 m/s

Fresh air supply	15 m^3/person/hour (or 10 m^3/person/hour in extreme conditions)
Minimum overall U-value	2 W/m^2K

Further, EN-standard has testing and evaluating for the thermal parameters, EN-14750-2. The compliance of EN-standard in this test is usually in the production or after production, but there could be further adjustments that are not always evaluated later. However, after changes have been made, the train operator can do non-mandatory tests to assess the thermal parameters.

One example of a long-distance train is the Swedish X2000. For less than 20 °C outdoor temperature, the setpoint is constant at 20°C, and above the outdoor temperature of 20°C, the setpoint temperature follows the equation (1.3) [12]. The control logic of the ventilation system for the X2000 is described in Figure 3.

$$t_{set} = 20 + \frac{t_{out} - 20}{2} \qquad (1.3)$$

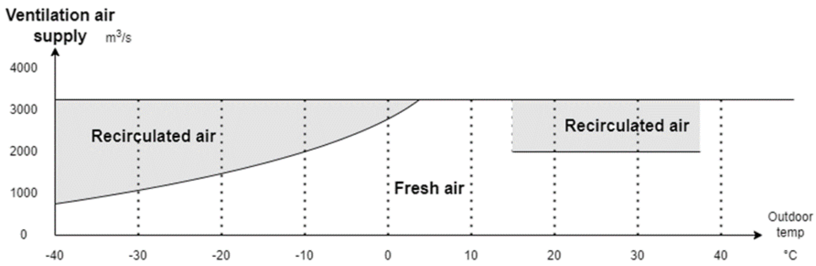


Figure 3 - X2000, one coach ventilation air supply - fresh and recirculated air logics. [12]

The heating of the ventilation air of the X2000 per coach is a maximum of around 16 kW, decreasing from 16 kW to 0 kW between 3°C and 19 °C. In addition, there is radiator heating on the X2000 train, which is a maximum of 20 kW at -40°C, decreasing linearly to 0 kW at 20°C. At -10 °C, total heating power is 26 kW, at 0°C 22 kW; and at 10 °C around 12 kW.

The HVAC system on the X2000 has, in addition to the above, the following characteristics [12]:

- 40 m^3/h ventilation per seat/person with a supply speed of around 0.1 m/s at outlets.
- The radiator position is below the windows.
- Maximum of 80 seats per coach.
- Supply air in both ceiling and some in the bottom of the side walls.
- The air handling unit is equipped with a heater, cooling, and an air filter.

2.3 THERMAL COMFORT

2.3.1 Air speed and draughts

Air velocity is an important factor for evaluating thermal comfort. It is defined as the speed of the air at a certain point with no regard to in which direction it is. Different maximum local airspeed accordingly to EN-standard 14750 are shown in Table 2.

Table 2 - EN-Standard 14750 maximum recommended local airspeed. [12]

Interior temperature (°C)	Maximum recommended local air speed (m/s)
20	0.25
22	0.35
28	1.0
30	2.0

Ventilation flow affects the airspeed within the coach. For example, in summer, low temperatures could introduce draughts if the local airspeed caused by the ventilation flow is too high [12].

Draughts is a local cooling of the human body caused by air movement, which will feel uncomfortable and could be an issue on trains or any other ventilated buildings [13]. A rule of thumb is that in airspeeds below 0.25 m/s, this may not be a problem for thermal comfort, but it is very related to the air temperature of the air movement. In the equation (1.4) [13], the estimated percentage dissatisfied PD [%] can be calculated, where t_a is the air's temperature and the local air's velocity.

$$PD = (34 - t_a) \cdot (V - 0.05)^{0.6223} \cdot (0.3696 \cdot V \cdot 40 + 3.143) \tag{1.4}$$

2.3.2 Clothing

Clothing is a factor that affects heat loss from the body. The unit clo expresses the heat resistance in the unit m^2K/W and represents the thermal insulation the clothing provides. Clothing items can be added mathematically by addition (in unit clo). Clothing values are highly dependent on each passenger, who has different personal preferences for clothing. However, some factors influence the average, typical clothing values, mainly the outdoor temperature, activity level, and wind speed.

A parallel study that is ongoing at the same time as this study, investigated the thermal comfort and temperature setpoint in buses through surveying the thermal comfort perception of the bus passengers. The experiments were done on two different days, 1 March 2023 and 3 April 2023, with 8 °C respectively 3.8 °C outdoor temperatures. At outdoor temperature of 8.0°C the clothing value was on average 1.0, and at outdoor temperature of 3.8°C the average clothing factor was 1.28. However, the spread of the clothing values was large in respectively day, accounting for in which 50 % of the closest average values is ranging from $\pm$ 1.1 Clo in the first case and the second case down to 1.05 Clo, with a high degree of outliers for the rest 50 % outside of the boxplot. The considerable variation in clothing values could be of significant individual passenger dependency on Clo-values or error sources. The study was done within Stockholms län in the public transport, on buses. It is assumed that similar clothing will be worn by passengers on the trains at these outdoor temperatures, for further details of the study, see [14].

Using the equation (1.1) and the data from [14], the following graph, in Figure 4, with clothing value depending on outdoor temperature, is used to predict the clothing values depending on the outdoor temperature.

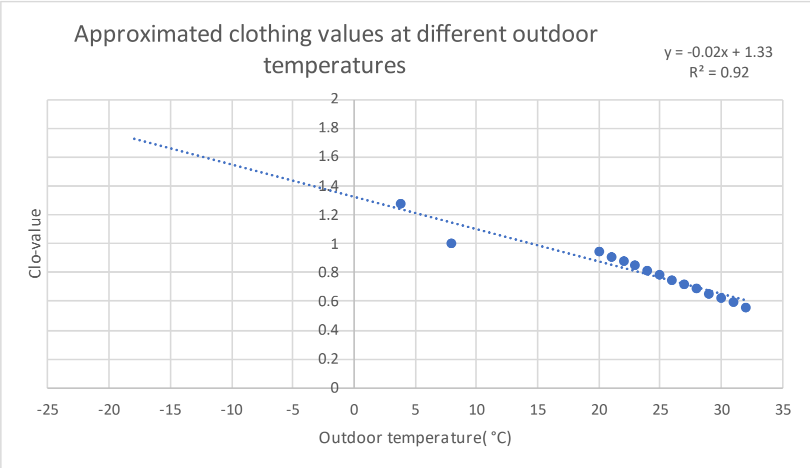


Figure 4 - Clo-value depending on outdoor temperature.

2.3.3 Operative, air, and radiant temperature

Operative temperature consists of air temperature and the mean radiant temperature, and it is used for describing the heat exchange between humans and surroundings. Operative temperature is the average value of mean radiant temperature and the air temperature.

In the report of Thermal Comfort in Rail Vehicles, based on EN standards, it is stated that radiant temperature is highly relevant for the thermal comfort of the passengers [11], thus the surface temperatures are considered as well as air temperature. Figure 23 shows the importance of the operative temperature in relation to the air temperature. Therefore, the passengers on the trains are affected by the air temperature and the radiant temperature from the surrounding surfaces.

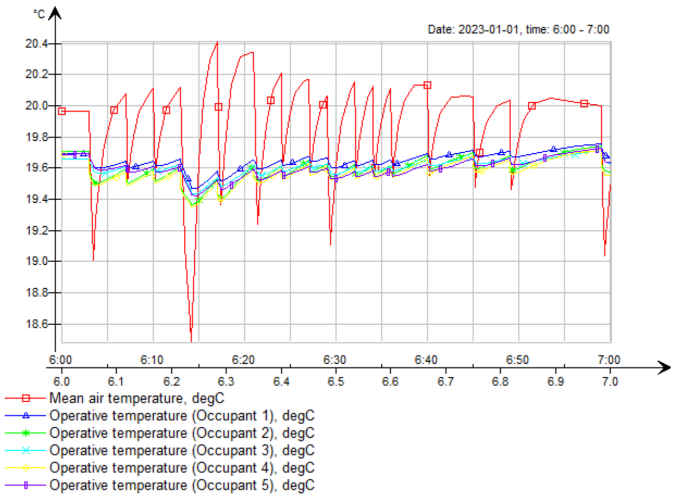


Figure 5 - How operative temperature varies from mean air temperature with door openings. From IDA ICE simulation of the train coach.

The mean radiant heat loss for a person (K), described in the equation (1.5) depends on the emissivity of surrounding surfaces, the temperature of the clothing surface (K), and the mean radiant temperature (K) [15].

$$T_{mrt,person} = \varepsilon \cdot \sigma \cdot \left(T_{cl}{}^4 - T_{mrt}{}^4\right) \tag{1.5}$$

The mean radiant temperature depends on the surface temperatures and view factors ($F_{p->i}$) according to the equation (1.6).

$$T_{mrt}{}^4 = \sum_{i=1}^{n} \left(T_n{}^4 \cdot F_{p->i}\right) \tag{1.6}$$

2.3.4 Perception of thermal comfort

Predicted mean vote (PMV) estimates the perception of thermal comfort for the people, including the factors air velocity, temperature, humidity, activity level, and clothing. The PMV-scale corresponds to the explanations in in Table 3 [16].

Table 3 - PMV-seven-point scale [16]

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

The PMV can be calculated using the equation (1.7).

$$PMV = (0.303 \cdot e^{-0.036 \cdot M} + 0.028) \cdot L \tag{1.7}$$

where L is thermal load. The thermal load is the heat balance in the body which is the difference between internal heat production of the body and the heat losses from the body. The thermal load is described in a heat balance of the body in equation (1.8). In the equation of the thermal load, metabolic rate (M), mechanical power (W), sensible heat losses, evaporative heat losses from skin (E_{evap}), and the respiratory convection respectively evaporative heat losses (C_{res} and E_{res}) [17].

$$L = (M - W) - H - E_{evap} - C_{res} - E_{res} \quad (1.8)$$

The full equation with L inserted the PMV equation becomes as following in equation (1.9).

$$PMV = (0.303 \cdot e^{-0.036 \cdot M} + 0.028) \cdot [(M - W) - 3.05 \cdot 10^{3} \left(5733 - 6.99(M - W) - P_a\right) - 0.42((M - W) - 58.15) - 1.7 \cdot 10^{-5} M(5967 - P_a) - 0.0014M(34 - t_a) - 3.96 \cdot 10^{-8} f_{cl}((t_{cl} + 273)^4 - (\bar{t}_r + 273)^4) - f_{cl} h_c (t_{cl} - t_a)) \quad (1.9)$$

The equation (1.9) is complex and requires iterations. The relation between PMV and PPD is as shown in Figure 6.

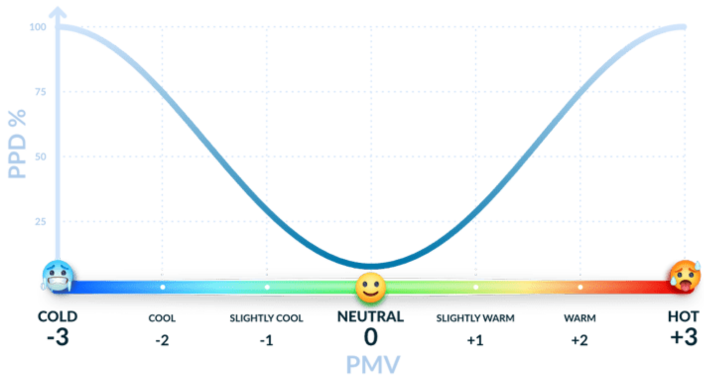


Figure 6 - Relation between PPD and PMV. [16]

The predicted percentage of dissatisfied can be calculated from the PMV using the analytical equation (1.10).

$$PPD = 100 - 95 \cdot e^{-(0.03353 \cdot PMV^4 + 0.2179 \cdot PMV^2)} \quad (1.10)$$

The acceptable PMV-level according to ASHRAE 55 recommendations is a PMV between -0.5 to 0.5, and the ISO7730 recommends between -0.7 and 0.7 for existing buildings.

With the following assumptions, 30 % humidity (based on Figure 46), metabolic equivalent (MET) value of 1.5, local air speed of 0.2 m/s (measured), same mean radiant temperature and air temperature, and 20 % dissatisfaction, the comfortable temperatures are shown in Figure 7 depending on clothing values.

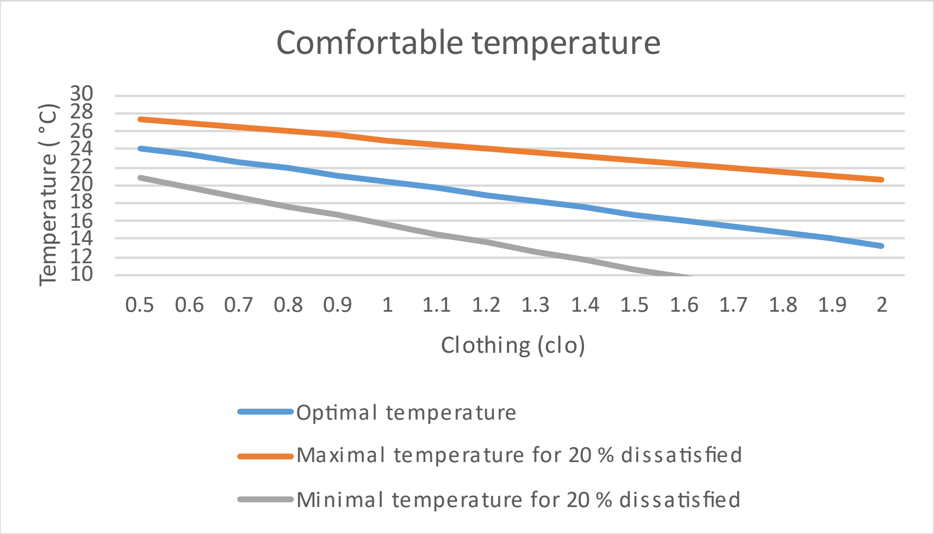


Figure 7 – Comfortable temperature depending on clothing value.

However, there are a few limitations to the PMV method of assessing thermal comfort. The main limit is that it does not account for transient responses, such as a change in temperature while entering the train. Other problems are limited experimental data for developing the standard of assessing thermal comfort, and the assessment methods have been developed in well-controlled environments and for a small group of people [13]. Observations have also shown that the temperature acceptance for thermal comfort correlates with outdoor temperature more than the clothing factor. The idea is described in standard EN-15251: "If a change occurs such as to produce discomfort, people react in ways that tend to restore their comfort." [13] Some of the mentioned factors are an expectation of indoor climate, previous thermal conditions, and outdoor temperature. An interval of good temperature ranges can be accepted through the transient responses to the temperature of the human body and psychological perception.

Transient conditions for thermal comfort might be necessary considering commuter trains typically mean short commuting times, and thus at least two changes of thermal conditions are experienced by the commuters. According to an experiment from a study done at Loughborough University, when the experiment participants boarded the train, there was a peak in thermal sensation until it reached a steadier state. However, it is unclear under which circumstances the rise in thermal sense occurs, as there is a lack of studies with enough sampling rate to catch the peak in thermal sensation. In addition, the study also shows how skin temperature changes rapidly after entering the train [18].

Survey-based research to extend the driving range on electric buses by adjusting setpoint temperature has been done in the Netherlands for two days in November with an average outdoor temperature of 13.4°C. The thermal sensation (TS) was mainly dependent on age, thermal comfort (TC), bus temperature (T_{bus}), and clothing, where thermal comfort is evaluated on a 9-point scale question ranging from very cold (-4) to very hot (4). The thermal sensation is between 0 and 4, 0 being comfortable and +4 extremely uncomfortable. The multiple regression equation (1.11) in which the thermal sensation depends on the above mentioned parameters. [19]

$$TS = -6.797 - 0.008 \cdot age + 0.918 \cdot TC + 0.310 \cdot T_{bus} + 0.527 \cdot Clo \tag{1.11}$$

Assuming the average age of the passengers of the commuter trains in Stockholm are 38 years old (estimated from [20]), thermal comfort (TC) of neutral (0), and different clothing values. However, it is uncertain how applicable this is for lower temperatures, as the survey was conducted at 13°C outdoor temperature.

The equation (1.12) describes the optimum indoor bus temperature setpoint derived from the analysis.

$$T_{bus} = \frac{6.797 + TS + 0.008 \cdot age - 0.918 \cdot |TS| - 0.527 \cdot clo}{0.310} \tag{1.12}$$

The regression results from the study [19] shown in the equation (1.12) can be compared with optimum thermal comfort models, such as PMV. The main conclusion in the article is that a setpoint temperature of 15°C would equal slightly cool and give 8% less energy usage. Applying equation (1.12) to Stockholm trains, with the average passenger age of 38 years old thermal comfort and clothing value adapted accordingly to Figure 4. Assuming the similar conditions as in the study being autumn and spring. Since the study was based on short-haul buses, it may be good to catch the factor of ΔT between outdoor and indoor temperature and passengers' expectations when entering the bus.

Table 4 - Optimum bus temperature derived from [19].

Optimum bus temperature (°C)								
Outdoor temperature (°C) -->	-15	-10	-5	0	5	10	15	20
Thermal sensation 1	23.3	23.5	23.7	23.9	24.1	24.3	24.5	24.6
Thermal sensation 0	20.1	20.3	20.5	20.7	20.8	21.0	21.2	21.4
Thermal sensation -1	13.9	14.1	14.3	14.5	14.7	14.8	15.0	15.2

2.3.5 Air quality

One of the main governing constraints for the ventilation system is the CO_2 concentration. CO_2 is also one of the governing parameters for recirculation, as the CO_2 level is not affected by filters in the AHU [15].

The CO_2 production of the passengers can be described with the following equation (1.13) [15].

$$S_{CO_2} = 4.653 \cdot 10^{-3} \cdot MET \tag{1.13}$$

Which leads to 0.0065 L (CO_2)/s per passenger. The steady state CO_2 concentration can be calculated through equation (1.14).

$$C_s = \frac{1-\varepsilon_V}{\varepsilon_V} \cdot \frac{N \cdot S_{CO_2}}{\dot{V}_F} \cdot (1-R_f) + \frac{(1-R_f) \cdot \frac{N \cdot S_{CO_2}}{\dot{V}_F} + C_{OA}}{1-R_f} \tag{1.14}$$

, where R_f is the proportion of recirculated air. R is recirculated air, F is fresh air and S is supplied air.

$$R_f = \frac{R}{R+F} = \frac{R}{S} \tag{1.15}$$

R is the recirculated air and F the fresh air. S is supplied air.

The fresh air can be expressed as an equation (1.16). Only the fresh air flows affect the CO_2-concentration, which are; the ventilation caused by the door openings, air infiltration, and fresh ventilation air supply.

$$\sum \dot{V}_F = \dot{V}_{door} + \dot{V}_{\inf} + \dot{V}_S - \dot{V}_R \tag{1.16}$$

The ventilation effectiveness typical on commuter trains is unknown, and depends on factors such as obstructions, passenger density, door openings and ventilation diffusor's location and ventilation flow. Because of the door openings and high ventilation airflow compared to buildings, a ventilation effectiveness of 0.9 was assumed. This is however highly uncertain since there are not many studies within ventilation effectiveness of trains. The other assumptions are fully seated (64 passengers in one coach), MET 1.4 , 1 ACH (infiltration/exfiltration), 2100 m3/h ventilation, 10 % door opening time (of one hour = 360 s, averaged out in one hour, gives 90 L/s average), and zero recirculation. It yields a steady-state concentration of 1059 PPM. With the assumptions made, the steady state calculated CO_2 concentration depends on the number of passengers as according to Figure 8.

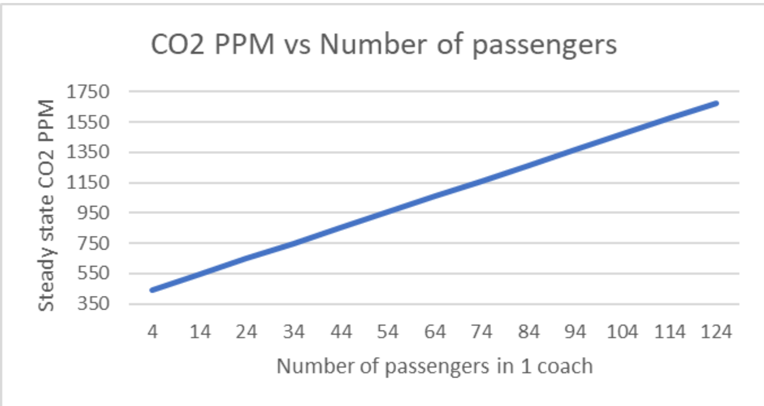


Figure 8 – CO_2 PPM depending on number of passengers

The importance of the CO_2 concentration can be quantified in a percentage of dissatisfied, PD_{CO2} (%), depending on the CO_2-concentration according to equation (1.17) [15].

$$PD_{CO2} = 395 \cdot e^{-15.15 \cdot (C_{CO2} - 400)^{-0.25}} \tag{1.17}$$

2.3.6 Vertical temperature differences

In Figure 9 the acceptance PD_{vert} (%) of vertical temperature differences between head and ankles are shown to maintain good thermal comfort. Due to the ventilation outlets at the bottom of the coach, the vertical temperature differences are large. Temperature fluctuations close to the floors are widespread, as seen in the measurements, depending on the current heating requirement of the train.

$$PD_{vert} = \frac{100}{1+e^{5.76-0.856\cdot\Delta t_p}} \tag{1.18}$$

Where Δt_p is the temperature difference between the ankle and head.

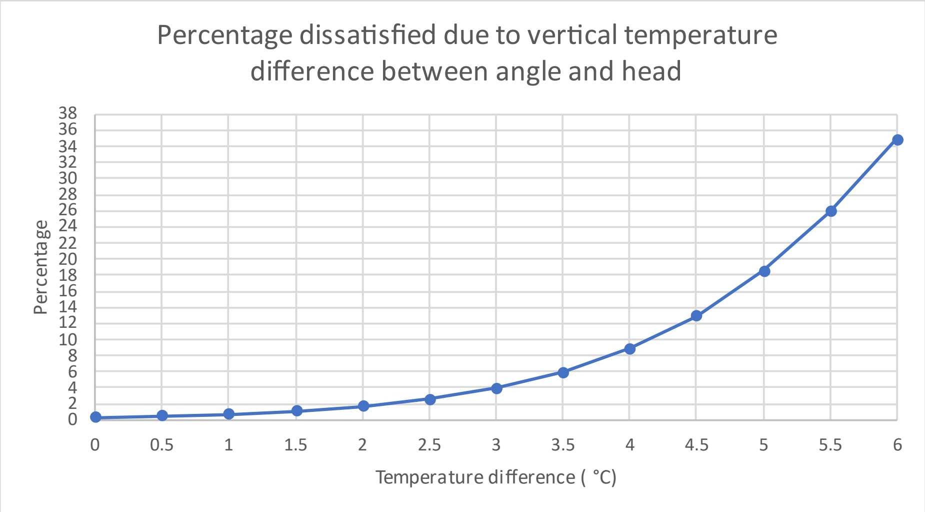


Figure 9 - Percentage dissatisfaction depending on the vertical temperature difference between head and ankles [13], based on ISO7730.

However, it is unknown how this function concerns different clothing and how it can apply to short-haul train travel. The definition of the ISO7730 is where the indoor thermal comfort is within the desirable range; slight variations are within desirable. But it is also stated that it can also be applied to other circumstances [21]. However, it can estimate how vertical temperature differences can affect thermal comfort.

2.4 METHODOLOGIES FOR EVALUATING THERMAL COMFORT

Different disturbances of the thermal stability of the interior of the train are described in [22]; these are:

- Solar radiation (direct and diffuse) and solar incidence angle.
- The heat emission from passengers (latent and sensible) – Internal gains
- Exterior temperature
- Relative humidity
- Wind speed
- Air infiltration/exfiltration
- Heat transfer through envelope
- Ventilation air supply (recirculated and fresh air)
- Lighting
- Clothing

The thermal comfort of vehicles can be evaluated using various approaches. Two different methodologies are described in a study on thermal comfort in vehicles [12]. These are thermal comfort tests and numerical simulations. In addition, the study investigated differences between buses, railway, and cars. The first methodology for thermal comfort evaluation is the thermal comfort test, analyzing different parameters in a vehicular environment experimentally. However, it is described that a vehicular environment thermal comfort evaluation is complex, with many parameters fluctuating quickly. The thermal comfort evaluation test includes tools such as manikin and empirical models. Then the empirical model and the results of the manikin experiment can be gathered to be able to predict thermal comfort. The second method for thermal comfort evaluation mentioned is the numerical method, which could be less costly than the first methodology since it does not require experimental testing. With the numerical method for example, a ventilation system can be designed to maximize thermal comfort [23]. However, in this case with X60, which is not a ventilation design change or a complete re-design of the train, it might be more feasible to use data analysis and models of predicting the thermal comfort, to evaluate different measures and their effect on thermal comfort.

One methodology relevant for this type of research is the methodology applied in the following researc paper. [24]. The research paper aimed to quantify suitable measures to save energy on thermal comfort-related functions on trains. A model simulating the HVAC system of a train was developed. The model calibrated with experimental measurements of the train, in addition, input data was used from onboard data collection from previous years. The measurement done was thermal mass, air tightness, and thermographic imagery. The program used for modeling was IDA ICE. The IDA ICE model could be validated with the thermal camera and other measurements. It is unclear how door openings were considered in the model.

ADSE company consulting has a similar methodology; tests data collected to build a prediction model of the thermal-related system, using the system data and meteorological information as input [25]. For example, they studied a double-decker train in a test tunnel, conducting experimental measurements. Different measures were investigated, including demand-controlled fresh air supply and exhaust heat recovery. The measures were evaluated by comparing the power consumption with and without the measures at different occupancy rates. Demand-controlled fresh air supply was evaluated at different occupancy rates, and heat recovery was assessed at different exterior temperatures, which are the main factor affecting the effect of each measure. It suggested the occupancy rate can either be measured with CO_2 concentration level or by relating bellow pressure to passenger load. The potential of the two measures evaluated was 8.8 % for demand-controlled fresh air, 13.8 % for air heat recovery and 16.9% combined in relation to the total annual electrical consumption of the HVAC system. [22]

In another research paper [26], evaluation methods for thermal comfort is adressed and investigated on the impact of the thermal environment on humans. The human perception of thermal comfort depends on human temperature regulation. Most of the heat produced by the human is through the metabolism, which is quantified in the measuring unit Met (metabolic rate). The human body adapts to the surrounding thermal environment to remain at a body temperature of around 37°C. However, the thermal disturbances on trains can be significant, and the time intervals at which passengers enter and exit the trains are short, with significant temperature differences. In addition to the internal body adaptation for thermal stability, clothes adjustments can be made, but that is not typical on short-haul trains. The factors which affect the human thermal response are operative temperature (air and radiant temperature), air speed, humidity, metabolic activity, and clothing. But in addition to those factors, it is also mentioned in the paper that exposure time is also a factor among these mentioned. PMV (predicted mean vote) and PPD (predicted percentage of dissatisfied) are among the most common approaches for evaluating thermal comfort. PMV is evaluated at a 7-step scale from -3 (cold) to 3 (hot) and is connected to the deviation from the neutral balance of the human body. However, a problem that might arise with applying this to a transient environment is that the model is primary for steady-state conditions. Slow fluctuations can be acceptable in the PMV model [26]. More about PMV and PPD is described in section 2.3.4.

A way of evaluating thermal comfort is to use PMV and PPD; there are at least 7 studies that used that methodology for evaluating thermal comfort in vehicles (bus) [27]. It is a common method for evaluating vehicle comfort. Thus, it is assumed to apply to short-haul trains as well considering that the average traveling time is in average for commuting in Stockholm 41 minutes (door to door) [28]. Most of the 7 studies evaluated indoor temperature, air velocity, mean radiant temperature, clothing values (clo), and metabolic rate.

In buildings, there are standard procedures to measure thermal comfort, characterized by being relatively stable compared to the transient conditions on trains. One example of such a procedure for measurement is suggested by Socialstyrelsen in Sweden [29]. In the report, there is a clear relationship between the humans' thermal comfort perception and what can be measured in terms of temperatures (air and operative temperature), airspeed, and mean radiant temperature. These are also the preliminary parameters to measure in case of complaints from, e.g., a tenant living in one apartment. Floor temperature and air movements are measured. Generally, the difference between operative temperature and mean air temperature is rarely more than 1-2 degrees Celsius [29], still but it can sometimes be more due to solar gains from the roof or cold floor. The operative temperature is essential for the human perception of temperature and the thermos mechanism in the body; however, there can be other important factors causing discomfort as vertical temperature differences or air speed/draught in the head/neck area. In buildings there is a distance from the wall in which is considered the occupancy zone. However, in trains, these are different since the seats are placed next to the windows. The ceiling area is equivalent in both, where there are no people present, but this zone is still very relevant as this is where the temperature sensors are located on trains in general.

2.5 QUANTIFYING SAVING FOR MANUAL DOOR OPENINGS

In a study on a retail store, the energy loss from having the door open while running air conditioning was investigated. The methodology used was to measure infiltration rate, indoor/outdoor temperatures, surface temperatures, and power consumption. Air conditioning was kept at the same energy use per day. The door evaluated in that case had dimensions of 3m height and 1.2 m width in a store of 36 m^2. Two days were measured, one day with the door open and the other with the door closed, with similar outdoor conditions and surface temperatures (confirmed with a thermal camera). The infiltration rate was measured using tracer gas, keeping the same target value of 10 ppm, measuring the tracer gas supplied and trying to keep a uniform concentration. The infiltration rate was 21 times higher with the doors open (1 400 m^3/h vs. 66 m^3/h. The AC was running in full capacity in both cases, but in the open-door case, the indoor temperature was around 5 °C higher compared with the closed-door. This was also verified with the Pham & Oliver equation (1.19) [30].; however, due to wind pressure not being considered in that equation, the infiltration rate was in real slightly higher. [30].

$$\dot{V} = 0.226A(gH)^{0.5}\left[\frac{\rho_i - \rho_o}{\rho_i}\right]^{0.5}\left[\frac{2}{1+\left(\rho_o / \rho_i\right)^{0.333}}\right]^{1.5} \tag{1.19}$$

With a door height of around 1.95 m and width of 1.35 m for the X60-train, the densities are calculated as the equation (1.37) for an outdoor temperature of 0°C and an indoor temperature of 20°C. Gravitational constant, g of 9.81 m/s^2. The following Table 5 is obtained.

Table 5 - Pham and Oliver equation for different temperature setpoint and outdoor temperatures Derived from [30] using equation (1.19)

Pham and Oliver steady					L/s
Liter air infiltration and exfiltration/second/coach due to door opening					
Outdoor temperature down, indoor temperature to the right	**18 °C**	**19 °C**	**20 °C**	**21 °C**	**22 °C**
10 °C	869	921	970	1016	1060
5 °C	1112	1153	1193	1231	1268
0 °C	1315	1350	1383	1416	1448
-5 °C	1493	1524	1554	1583	1612
-10 °C	1655	1683	1710	1737	1763
-15 °C	1805	1831	1856	1880	1905

Assuming only the air will be cooled down (or exchanged) [31], since the doors are only open for short periods and disregarding the change of surface temperatures and thermal mass of the train, then there it is only necessary to consider the temperature drop of the air.

To estimate the air infiltration due to the door opening, the temperature drops inside the passenger seating area, and air speeds were measured. The wet air density in winter has been assumed to be 1.2 kg/m^3. The door height is 1.95 m, and the width is 1.35 m. The temperature drops observed in the train were during winter at around 0 °C +/-3 outdoor temperature of 1.31 °C (based on 30 measurements) within the passenger seating area. Dimensions of each coach are approximately 16.5 m, width 2.7 m, and height 2.3 m. The total volume of the interior is approximately 103 m^3. The specific heat of air at 0°C is 1005 J/(kg*K). In this relation, the humidity does not greatly affect the specific heat of the air.

$$E = (V_1 \cdot \rho_{air} \cdot c_p \cdot \Delta T_{1,drop}) \cdot 2.778 \cdot 10^{-7} \cdot factor \tag{1.20}$$

- V_1 is the total volume of the train and $\Delta T_{1,drop}$ is the temperature drop, and E is the energy loss in kWh due to the temperature drops.
- Where the factor is an upscaling factor since temperature drops vary throughout the coach and have been observed to be larger at some measuring locations. The thermometer will not catch the full extent of the temperature drop due to the time constant of the thermometer; the t99 response time is 60 seconds. In the simulation, the temperature drops caused by the door opening, from base temperature to minimum temperature, takes 1 minute to reach.

It takes approximately 4-time constants to reach 99 % of the end temperature (the definition of t99). Thus, with a response time t99 of 60 seconds, the time constant is approximately 15 seconds. The sensor temperature after each timestep t, is calculated through equation (1.21).

$$T_{sensor} = (T_2 - T_1) \cdot \left(1 - e^{\frac{-t}{\tau}}\right) + T_1 \tag{1.21}$$

Where, T_2 is the simulated or real temperature at the next timestep, T_1 is the previous timestep simulated or real temperature, τ time constant of sensor, and t timestep.

2.6 SUMMARY LITERATURE STUDY

Several different energy efficiency measures were found which are suitable for trains, which need to be investigated whether they are suitable for the specific case of Stockholm commuter train. Three examples of IDA ICE modeling as methodology for analyzing train thermal behavior exist. Thermal comfort on trains depends on short- or long-haul trains and should be separated. However, there were similarities for buses compared to short-haul trains, which suggests that research done on buses also applies to commuter trains. A typical approach was experimental measurements with a simulation model to investigate energy and thermal comfort. In comparison to buildings, trains have more thermal disturbances, which makes the internal environment more transitory. In the next step, different methodologies are compared to evaluate energy efficiency measures in combination with thermal comfort.

3 METHODOLOGY

The project starts with a literature review, looking at previously made studies on thermal comfort and energy efficiency measures. Then it continues with understanding the system's current state on the train through technical manuals. In addition, site visits will be done to do measurements to simulate the thermal comforts energy functions accurately. Finally, communicating the findings in presentations and a report with the results from both the simulation model and the data analysis.

3.1 DATA COLLECTION AND ANALYSIS

1. Data channel lists are analyzed, and dataset selection lists are made in R2M. R2M is the data platform used for X60 trains.

The train's electrical schemes are analyzed to understand the different channels and what systems different power meters would include.

2. Relevant data collection from R2M and other sources using a macro for big volume data downloads.
3. Data filtering. Filtering faulty data. Identifying faulty data.
4. Analyzing the data.
5. Interpreting the results, answering research questions, and connecting it to the simulation model.

3.2 EXPERIMENTAL MEASUREMENT

The experimental measurements are mostly to calibrate the simulation model and understand the interior conditions' behavior.

The measurement equipment used is the following:

- FLIR thermal camera – Flir-E6390
- Testo 440 dP with air probe temperature sensor. Accuracy +-0.4°C, response time t99 60 seconds.
- Anemometer – Testo 405-Vt

Testo 405-Vt is a hot wire anemometer in which the flow affects the convective heat transfer of the wire, and the electrical power is measured. In addition, it measures the air speed at which the volume flow can be derived.

3.3 SIMULATION MODEL

Based on the experimental measurements and in-data from technical manuals and MTR input, the simulation model will be built in IDA ICE. The model will be used to evaluate energy efficiency measures and to simulate the baseline of how energy is used on the train.

3.4 IDA ICE

IDA ICE is software for modeling the energy usage of buildings. The energy performance of the building simulates throughout a defined period with certain ambient conditions retrieved from weather data based on selected weather station. In this project, expert versions are used in versions 4.8 and 5 Beta. Version 5 has more advanced functions for modeling the indoors more accurately. The software can be adapted for specific user or model requirements. Energy balances can be modeled for one year with a selected resolution, which makes it possible to model door openings. The software has built-in functions for evaluation of thermal comfort. The air handling unit can be adapted to match the ventilation and combined heating on the train. As output, the IDA ICE model produces graphs and energy data on a preselected period or the graphs for the resolution selected. Examples of output graphs are supplied ventilation flow, airflows within the zone, Fanger's comfort indices, surface temperatures, ventilation supply temperature, etc.

3.5 Limitations

Since the project will be ongoing in the colder part of the year, no measurements are done in the summertime. Thus, this project will focus mainly on the colder outdoor conditions when no cooling demand for experimental and modeling.

The experimental measurements will be limited in scope and will be done for a certain number of days at different conditions. However, no long-time measurements will be done.

This will be an investigating project for optimal energy performance and thermal comfort, so the focus will not be to comply with already-stated standards within this topic. Still, the results will be compared to the standards available.

Does not consider driving cabs. Because these have different standards and regulations compared to the coaches, the driver cabin can be seen as a long-haul traveling compartment as the driver will be there for extended periods and will most likely not have as much clothing during cold days as the passengers. The individual temperature range adjustment is larger for the driver cabin, allowing a comfortable temperature for each driver and surrounding circumstances.

3.6 Ethical considerations

There are not any significant ethical concerns related to this project. The data collected does not contain personal information. Possibly travel habits and such can be read from the data, but it is generalized and cannot be connected to individuals.

If there is any sensitive data that cannot be published publicly, data numbers will be converted into percentages or normalized numbers that cannot be linked to actual quantities.

4 STOCKHOLM COMMUTER TRAINS

In this section the Stockholm commuter train X60 is described.

Table 6 - Explanation of "coach" and "1 train". In this figure 1 train is shown (consisting of 6 coaches).

4.1 HVAC

Each coach has an air handling unit, and each driver's cabin has an air handling unit. The air handling units are placed on the roof. Each coach air handling unit consists of a supply and return air fan, an air filter, a resistive heater, and an air conditioning system. The cooling system consists of a simple air conditioning system, which removes the hot air before being supplied through ventilation. While the refrigerant evaporates, the cooling coils are colder than the air, thus cooling the airflow through the HVAC system. The supply air flow rate is higher for the cooling case than for the heating case, as seen in Table 7 and Table 8.

The main parts of the air conditioning unit consist of a compressor, condensing unit, expansion valve, and evaporator. Each air handling unit has two cooling circuits with a total cooling power of 23 kW per air handling unit and coach.

Table 7 – HVAC specifications for cooling case.

Explanation	Value
Cooling capacity	23 kW
Supply airflow (while in cooling mode)	3 300 m^3/h
Outdoor airflow	1 100-2 000 m^3/h
Exhaust airflow	2 200 m^3/h
Refrigerant	R134a 2×5.3 kg

Table 8 - HVAC specifications for heating case.

Explanation	Value
Heating capacity	48 kW
Supply airflow (while in heating mode)	2 200 m^3/h
Outdoor airflow	1 100-2 000 m^3/h
Exhaust airflow	2 200 m^3/h
Type of heating	Resistive heating

The COP of the air conditioner is assumed to be the same as the assumption made in [7], a COP factor of 2. The data of the air conditioner power consumption is shown in Table 9.

Table 9 - Air conditioner data of the AHU.

Air conditioner	Value
Compressor air conditioning system	9.2 kVA
Condenser fan power for air conditioning system	1.5 kVA
➔ Assumed COP value = 2	

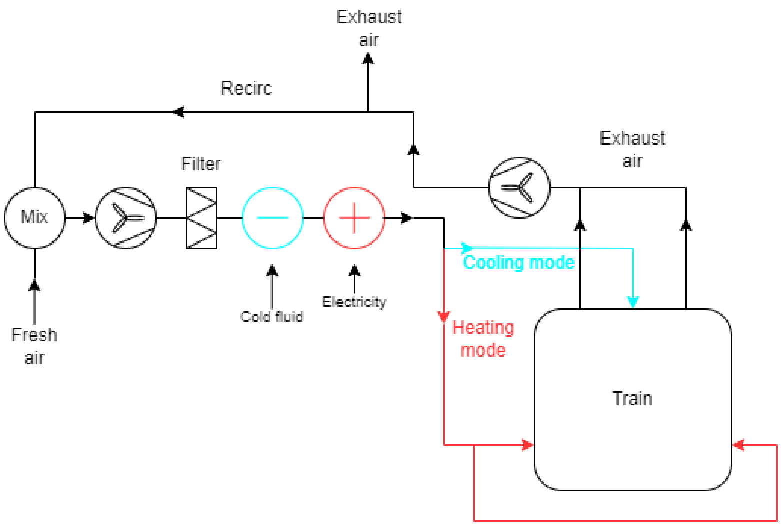


Figure 10 - HVAC flowchart system on the train.

The air conditioning unit is located within the HVAC of each coach. The specification of the air handling unit is shown in Table 7. It works through latent heat of vaporization in the evaporator absorbing the heat from the supply air and ejecting it in the condenser. The relative humidity is decreased through the air conditioning system through the air moisture at the cold surfaces of the evaporator [32]. Each air handling unit has air conditioning circuits, in which one circuit has a flowchart that looks as Figure 11.

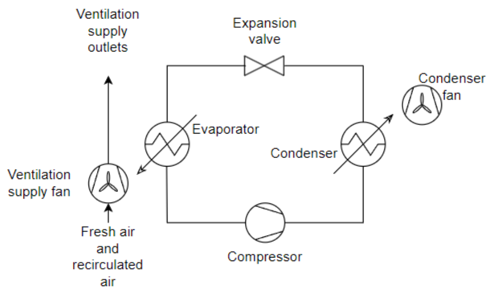


Figure 11 - Air conditioning system flowchart.

The refrigerant used is R134a, in a two-circuit system in each HVAC of the coaches. Condenser fan uses 1.5 KVA.

The setpoint temperature is accordingly to EN14750-1:2600, Category B, zone 3, which is shown in Figure 12.

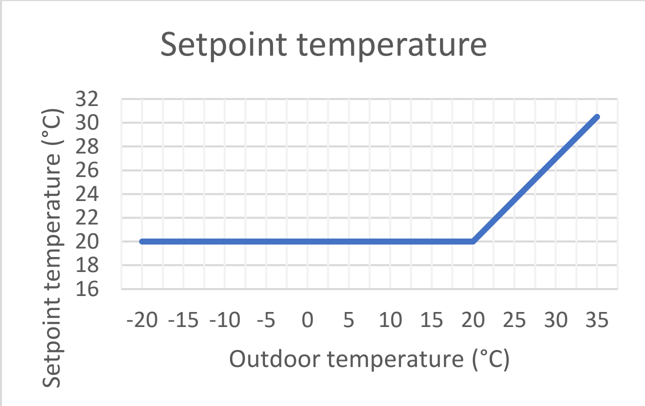


Figure 12 - Setpoint temperature depending on outdoor temperature.

Ventilation flow is a mix of both recirculated air and fresh air. One of the main constraints for how much can be recirculated is the CO_2 concentration [12], which is not reduced over the air filter. The humidity is another constraint for how small the ventilation flow can be [12]. The partial recirculation of air is only active at certain outdoor temperature conditions, above 26°C and below -5°C. In between -5°C to 26°C there is 100 percent fresh air flow of 2 200 cubic meters per hour and coach.

The ventilation system, combined heating and cooling coils is controlled through the temperature sensor in the train and the outdoor temperature. The measurement of the interior temperature is proceeded to a controller, which compares the setpoint to the current state. The controller then controls the ventilation mode (heating or cooling), and the supply temperature. Because of the time constant of the temperature sensors, it acts as a control filter against fast fluctuations.

4.2 Lighting in the coaches

In the SL trains, the lighting is always turned-on during operation. In each coach, there are approximately 24 fluorescent tubes. Each of the fluorescent tubes, assuming they are the same throughout the coach, has the following specification:

- Master TL5 HE – 28W/840

This lighting corresponds to a total internal gain of 670 W per coach, assuming it is similar throughout the coaches. In winter, the lighting provides heat with the equivalent efficiency to the heating system. However, in the summer, the internal gains add need to be cooled. Based on the assumed COP-factor of 2 for the cooling system, the total power needed for lighting and the equivalent cooling need to cool of the heat generated from the lighting would be 6 kW for one train.

4.3 HVAC Fans

In the ventilation system, there are a supply fan and an exhaust fan. The power consumption for those depends on the fan efficiency, pressure drops and the airflow. For the X60 train the power supply for the fans is approximately 2.4 kVA (including power factor) for one coach.

4.4 PASSENGER LOAD

Passenger load is an essential factor considering thermal-related energy. This is because humans provide sensible and latent heat of approximately 115 Watt per person [33]. The average passenger kilometer for various periods is shown in Table 10 [28]. The statistics refer to an average working day for 2019 in the months of January-April, September-December; the winter holidays in December and January are not included. The average trip distance is almost the same in all the time ranges, averaging 18.2 km (max deviation of 2.5 km). However, depending on the number of trains in operation on a given route and whether they are short or long trains, the passenger density in the coaches may vary. Thus, looking at the passenger load average of the trains at different times of the day is essential.

Table 10 – Passenger-km statistics. Stockholm [28]. Storstockholms Lokaltrafik 2019.

Time ranges from	06:00	09:00	15:00	18:00	21:00	
Time ranges to	09:00	15:00	18:00	21:00	06:00	Total
Person-km	1915	1962	2036	717	693	7322
(Person-km)/hour	638	327	679	239	77	1960
Percentage (Person/km)	33%	17%	35%	12%	4%	100%

Further, to get the passenger density of the trains, observations with corresponding bellow pressures were to relate passenger load with bellow pressure. In total, 33 observations, 15 for X60A and 18 for X60B-version of the commuter train. Two linear regressions were used for estimating the passenger load in Figure 13.

Table 11 - Estimation of passenger load.

Scale	Explanation
1	very few passengers
2	few passengers
3	seats 1/3 taken
4	seats 50 % taken
5	almost all seats taken
6	all seats taken
7	all seats taken, and standing passengers
8	almost fully crowded
9	fully crowded
>=10	maximum crowded

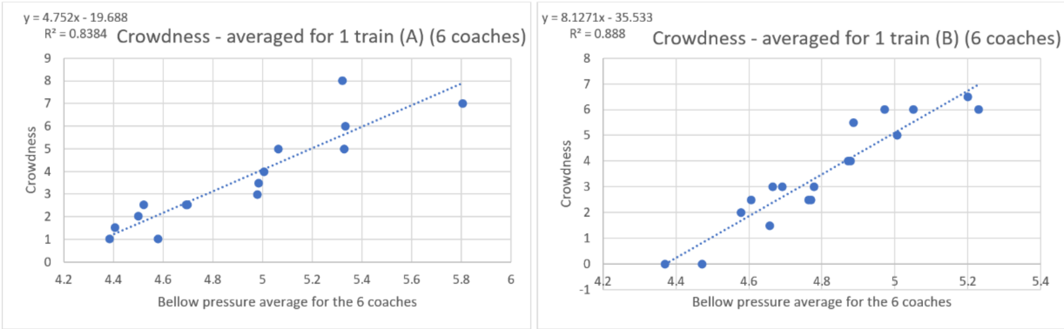


Figure 13 - Passenger load relation with bellow pressure in train version A respectively B.

4.5 Outdoor temperatures

From SMHI, average outdoor hourly temperatures between 2016 and 2022 are used for estimating the clothing values. First, the average monthly temperature values are divided into average daily temperatures between 9 am-3 pm, and morning and afternoon/evening temperatures are merged. This is done to account for the higher temperatures, especially in the summer during the day compared with morning and evening.

Table 12 - Outdoor temperatures between 9am-3pm. [34]

Month	Outdoor temp (°C) 9-am-3pm	Outdoor temp (°C) 3pm-11pm and 6am-9am
Jan	-0	-1
Feb	1	-1
Mar	4	2
Apr	9	6
May	15	11
Jun	20	17
Jul	21	18
Aug	20	17
Sep	16	13
Oct	10	8
Nov	5	4
Dec	2	1

In Figure 14, a duration chart is shown, used for assessing the balancing temperature.

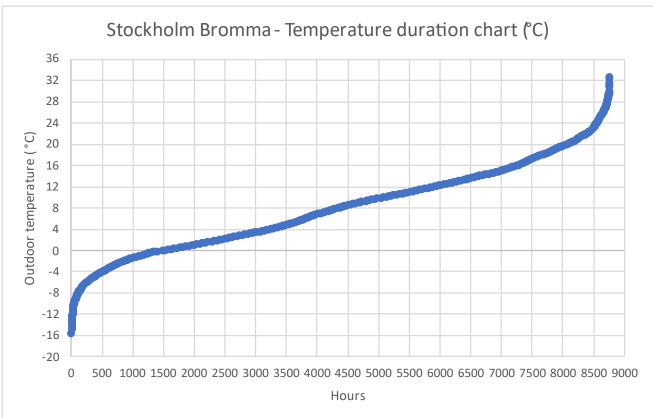


Figure 14 - Temperature duration chart for Bromma Stockholm. Derived from SMHI 2022 hourly weather data.

4.6 DOOR OPENINGS

In the figure below, the doors are open for 12.6 % of the 16-hour in-service on that specific day, however, not all those 12.6 % door opening share are at outdoor stations. The door openings have been retrieved from the R2M data platform. All doors of one side of the train open simultaneously, with orange and blue representing the two different sides of the train. This example day is the input of the IDA ICE simulation model.

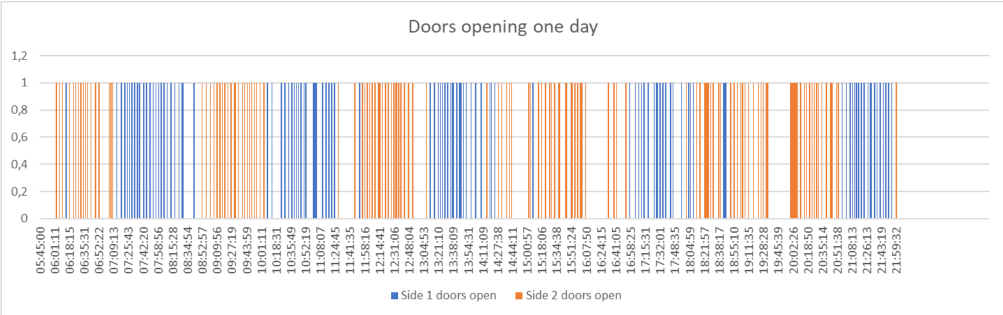


Figure 15 - Door openings for one shift (16 hours) of a random train. Blue and orange represent each side of doors.

4.7 DATA COLLECTION PLATFORM

The data collection platform is the NEXALA R2M, a real-time monitoring of different data channels from the trains. An example of the layout of the platform are shown in Appendix G. Different channels are available such as braking system, HVAC, auxiliary, doors, high voltage, cab, battery, etc. Most channels are digital, providing information if systems are on or off, indicating whether doors are open or closed, and faults detections. Analog channels offer more information, mainly sensor data such as electricity consumption, speed, wind, temperatures, pressures, etc. However, it can be hard to interpret the channels due to difficulties understanding the data channels' exact meaning.

4.8 ENERGY EFFICIENCY MEASURES

Based on the literature study and an overlook of the train systems and observations throughout the project, the following energy efficiency measures were proposed.

- Savings through having manual door openings during wintertime.
- Ventilation strategy – Recirculation, free cooling, and ventilation flow – Speed regulation of ventilation fan and recirculation damper control.
- Temperature setpoint for optimal performance (and thermal comfort).
- Optimum parking mode.
- Extra window layer – Both increasing U-value and minimizing the solar gains (especially during summers) – Not included in this report.
- Dead-zone temperature ranges – Turning off heating/cooling completely at certain conditions – Until certain minimum/maximum temperature levels of indoor or/and outdoor temperatures have been reached.
- Heat recovery/Reversed air conditioning – This thesis will analyze the potential.
- Energy saving of having optimal setpoint temperature for thermal comfort.

5 MODELING

5.1 HEAT TRANSFER

Heat transfer is divided into three categories, radiation, convection, and conduction.

According to the equation (1.22), radiation heat transfer emitted by a body depends on the surface's temperature and emissivity. The emissivity ε depends on the surface; for example, white paint can have around 0.95, and black paint also 0.95. On the other hand, white paint only has an absorptivity of around 0.2, while black paint has an absorptivity of around 0.95 [35].

$$\Phi = \varepsilon \cdot \sigma \cdot A \cdot T^4 \tag{1.22}$$

Conduction is the heat transfer through solid objects, with the heat flow going from the warmer to the colder. λ - is the thermal conductivity (W/mK), A is the surface area, the temperature difference between the two surfaces, and L is the distance between the surfaces.

$$Q = \lambda \cdot A \cdot \frac{\triangle T_{surfaces}}{L} \tag{1.23}$$

In addition to the normal U-values input to the IDA ICE model, there will be convection due to the train's speed. Based on [7], the convection heat transfer from the train speed can increase by up to 20 %. Stanton number quantifies heat transfer in relation to forced convection. The Stanton number can be used to estimate the convection heat transfer coefficient, particularly if the skin friction is well-defined. However, many factors affect skin friction, such as the surface of the side envelopes, the material, the surface's condition, and the air's viscosity surrounding the train. The skin friction is directly proportional to the surface area [36], and thus more miniature-scale trains can be used and are used in experiments done in studies within this area. A problem though might be that most studies are done for high-speed trains, in which the skin friction can account for approximately 50 % at speeds of 250 km/h [37], and the skin friction increases with the square of velocity [36].

The total U value of each surface is calculated through the equation (1.24).

$$\frac{1}{U_{tot}} = \frac{1}{\alpha_{\text{in}}} + \sum_n \left(\frac{\delta}{\lambda}\right)_n + \frac{1}{\alpha_{ext}} \tag{1.24}$$

The average U-value of the envelope is assumed to be 1.1 W/m^{2}K which is typical for trains in the Swedish climate in a standstill [7]. To account for the convection due to the train speed, equation (1.25) are compared with an estimation of the forced convection through Stanton number.

$$\alpha_{ext} = 3.5 \cdot v_t^{0.66} + 9 \tag{1.25}$$

v_t the train's running velocity is equal to the train's average distance divided by the time [38].

For a given overall U-value, the extra convection due to the train speed can be added as described in equation (1.26).

$$U_{tot} = \left(\frac{1}{U} + \frac{1}{\alpha_{ext,speed}} \right)^{-1} \tag{1.26}$$

To evaluate the convection heat transfer coefficient using the skin friction, the range of skin friction coefficient was interpreted from [39], approximated to be in the magnitude of $4 \cdot 10^{-3}$. However, due to the uncertainty in the skin friction coefficient for the X60 commuter train, a sensitivity analysis is conducted for different skin friction coefficients with first approximations obtained from [39]. Further, the Stanton number can be calculated directly from the skin friction coefficient using the equation (1.27) [40].

$$St = \frac{C_f}{2} \tag{1.27}$$

In the equation (1.28) the convection heat transfer coefficient can be obtained from the equation based on Stanton number, density of air, specific heat coefficient and the velocity of the surrounding air. A simplification was made that the airspeed is equal to the speed of the train.

$$\alpha = St \cdot \rho \cdot V \cdot c_{p,air} \tag{1.28}$$

The following convection heat transfer coefficients were obtained depending on the speed, as seen in Table 13.

Table 13 - Heat transfer coefficients at different train speeds.

Speed of train (km/h)	20	40	60	80	100	120	140	160
Convection heat transfer coefficient – using Stanton	14	28	43	57	71	85	99	114
Convection heat transfer coefficient – Using Equation (1.25)	34	49	61	72	82	91	100	108

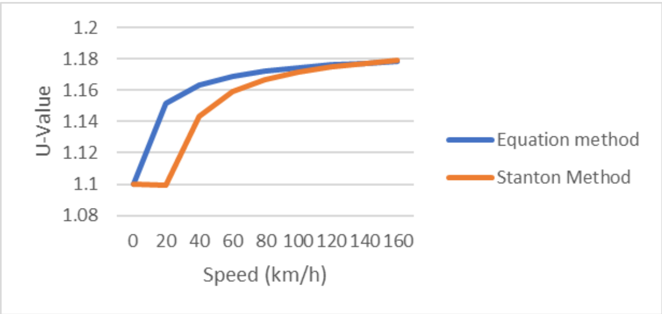


Figure 16 - Comparing overall U value derived from the equation (1.25) respectively *Stanton method.*

The average of a train in operation for a 12-hour period in one day was 30 km/h, which results in a U-value of approximately 1.14.

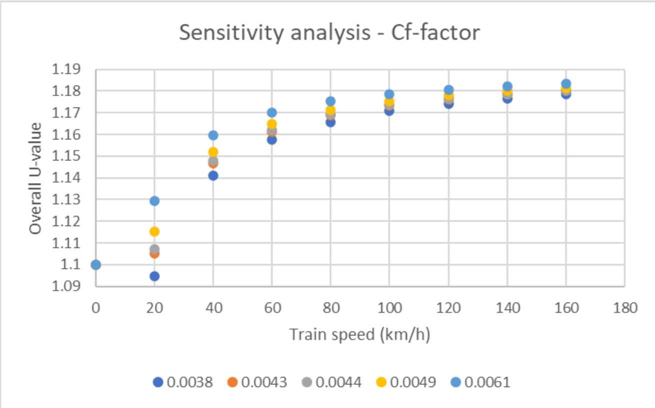


Figure 17 - Sensitivity analysis of skin drags coefficient effect on the overall U-value.

The energy supplied through the ventilation is calculated through the enthalpy difference, including the sensible and latent energy in equation (1.29).

$$\dot{Q}_{vent} = \dot{m}_{fresh,vent} \cdot (h_S - h_{out}) + \dot{m}_{reci,vent} \cdot (h_S - h_{in}) \tag{1.29}$$

5.1.1 Heat balance

The magnitudes of the heat transfer will depend on the heat balance within the coaches. The heat balance depends on solar radiation (diffuse and direct), heat transmission through envelope, air exchange, different internal heat gains (sensible and latent), supply and exhaust air and heat recovery.

5.1.2 Cooling

Mainly during the warm period of the year, when the outdoor temperature is higher than the setpoint temperature, the cooling system needs to cool with a cooling load equal to the heat gains. Several factors affect the cooling load, including outdoor temperature, high humidity, solar radiation, internal gains, and solar gains [32].

5.1.3 Heating

In contrast to cooling, heating is needed when the outdoor temperature is less than the setpoint temperature and internal gains not enough to maintain indoor temperature; the heating system needs to compensate for the heat losses from the coach for the heat balance. Important factors are low outdoor temperature and infiltration of air [32].

The heating is distributed through the ventilation system. Thus, the heating of the air can be described with the approximative equation (1.30), assuming constant air density and specific heat.

$$Q = \dot{V} \cdot \rho \cdot c_{p,air} \cdot \Delta T \tag{1.30}$$

However, in summer the cooling consists of sensible and latent cooling, and can be calculated through the enthalpy difference and air mass flow in equation (1.31)

$$\dot{Q}_C = \Delta h \cdot \dot{m}_{air} \tag{1.31}$$

Based on the dimensioning temperature of the train, down to -35°C [41] the supply temperature is calculated, assuming 48 kW at -35°C outdoor temperature. Heating is divided into two components, outdoor air to supply temperature and recirculated air to supply temperature. Assuming the temperature of the recirculated is equal to setpoint temperature $t_{out} <= t_{set} <= t_{\sup ply}$, the supply temperature can be described in equation (1.32). Air density is assumed $1.246 kg / m^3$, c_p is assumed $1.004 kJ / kg \cdot K$. P is the heating power supplied to the air.

$$t_{\sup ply} = \frac{\frac{P}{\rho \cdot c_p} + \dot{V}_R \cdot t_{set} + \dot{V}_F \cdot t_{out}}{\dot{V}_R + \dot{V}_F} \tag{1.32}$$

Equation (1.32) leads to a maximum supply temperature below 55°C at an outdoor temperature of -35 °C and a setpoint temperature of 20°C. At -20°C outdoor temp 45°C supply temp, at -10°C outdoor temp 39°C supply temp and at 0°C outdoor 33°C supply temp. It is assumed to be a linear relation of the supply temperature between outdoor temperature of -35 °C and 20 °C (equal outdoor and indoor temperature). According to the equation, the heating consumption at different outdoor temperatures is with the given supply temperature from the previous calculation (1.33). With the same assumption for the driver cabins, assuming the two driver cabins equals 40 % of 1 coach in heating need.

$$P = \rho \cdot c_p \cdot \left[\dot{V}_F \cdot (t_S - t_{out}) + \dot{V}_R \cdot (t_S - t_{set}) \right] \tag{1.33}$$

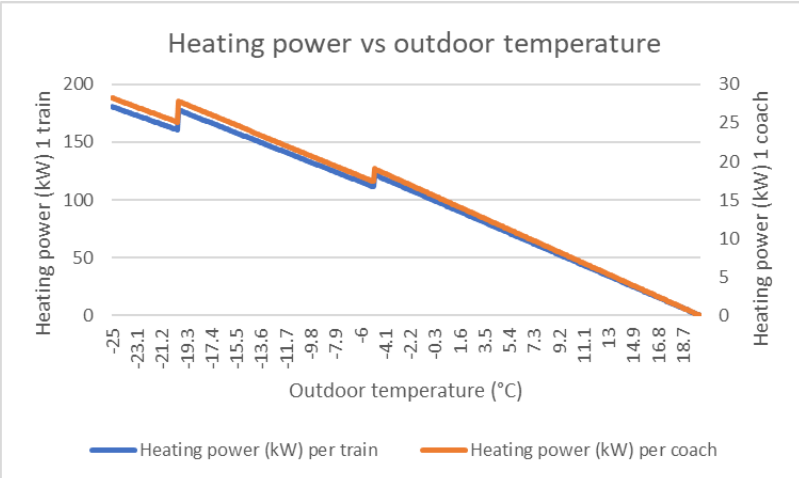


Figure 18 - Heating power vs outdoor temperature - Estimated.

5.1.4 Passenger heat gains

Metabolic activity can be measured in the unit MET, metabolic equivalent. Metabolic equivalent varies depending on the activity and person. There are two main criteria, the difference between body core temperature and skin temperature should be optimal, and the second is an energy balance of the body heat transfer [13]. In the equation (1.34) heat balance of a body is described. Metabolic rate, M is person dependent. Mechanical power, which the person performs, is neglected in this situation for trains since the person is not conducting any physical work on the train. H is separated into convection, radiation, and conduction in dry heat transfer. Then there is evaporative heat transfer.

$$M - W = H_{conv} + H_{rad} + H_{cond} + E_{evap} + C_{res} + E_{res} \tag{1.34}$$

5.1.5 Interior heat mass

Internal heat mass has been approximated based on the seatings and other equipment on the train, and then validated in section 5.7.

5.1.6 Ambient conditions

The ambiance conditions can vary depending on which train is considered on a particular day. Depending on the train's route, the train will be exposed to different ambient conditions, such as tunnels, surrounding shadings, length between stations, and temperature differences. Mostly it is solar radiation which is the main factor depending on where the train is located. Outdoor temperature will be similar since all the routes are relatively close to Stockholm. A study in China [38] used the methodology which considers the temperatures and solar gains at each station. One methodology which could be applied is to measure the conditions in the morning, day, and afternoon for the different routes to get an overview of how it affects the trains thermally. However, in this simulation, the ambient conditions are extracted from the Bromma airport weather station, and shadows are adapted to resemble the average surrounding.

Depending on the season, the solar gains will be from different angles of the train. For example, in January, the sun from sunrise to sunset is 6 hours, with the solar coming from the south, thus small solar gains. In autumn and spring, the sun is in the morning and afternoon, facing partly towards the train's windows.

5.2 IDA ICE – Input and modeling

The driver cabins have been simplified to simulate the heat exchange through the internal, with a heating capacity of 9 kW, cooling capacity of 3.9 kW, and supply flow constant. The actual flowrates for driver cabin are accordingly to Table 14. The driver cabin’s heating, cooling, and thermal comfort have been separated from the coach’s energy and thermal comfort.

Table 14 - Driver cabin ventilation flowrate

Supply air	1000 m^3/h
Fresh air	100 m^3/h
Recirculated air	900 m^3/h

Based on [38] the following assumptions was done for the IDA ICE model.

- Absorptance factor for the outer envelope surface – 0.7

Considering the door openings, a critical parameter in IDA ICE is the smoothing factor, which can be set between 0-5, with 0 being zero smoothing. As seen in Figure 19, with a degree of smoothing 5, the bar charts are smoothed to an almost constant door opening but with a smaller continuous flow, compared to Figure 50. In addition, smoothing is set to zero, which allows to see the temperature drops of the short door openings.

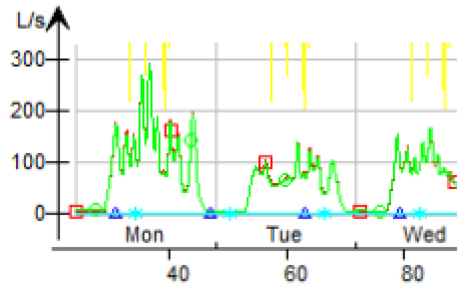


Figure 19 - Example of door opening air flow with degree of smoothing 5.

Thermal bridges among trains are common [42], and thus the thermal bridges are set to “poor”, which gives an average of approximately 0.35 W/K (m joint).

The passenger load has been adapted based on the data analysis of the average passenger load and increased to almost the maximum passenger load during rush hour on weekdays. In Figure 20, the distribution of the passengers is shown. For seating passengers, the center of mass has been assumed to be 0.6 meters above the floor, and for standing passengers near the doors, 1.5 above the floor. In Figure 21, the assumed number of passengers in each coach is shown.

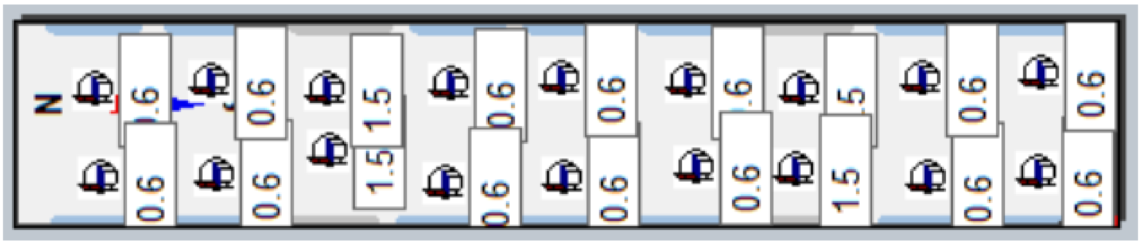


Figure 20 - Distribution of passengers in the coach.

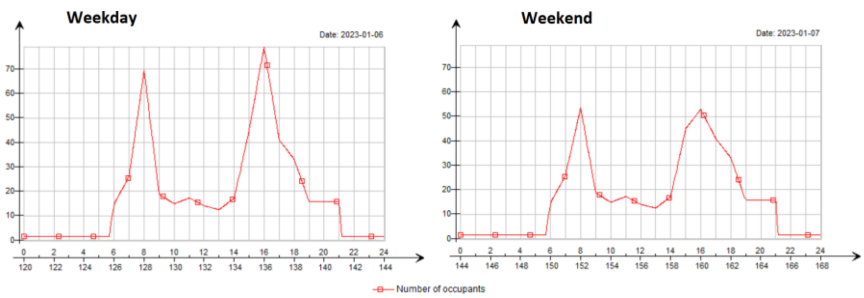


Figure 21 - Simulated number of passengers per coach.

The emissivity depends on the color of the surfaces, roughness, and material. For this case, it has been assumed to be a white color surface.

- Train windows U-values.
- Weather station: Bromma airport.
- Schedule for door opening.
- Ventilation supply temperature control.
- Clothing-values adjustments.
- Recirculation.
- The time constant of temperature control.
- Met values has been assumed between 1.4-1.6.

To have a smooth supply temperature that does not react momentarily against the temperature drops, but instead gives a somewhat constant heating supply to account for all the disturbing energy balance factors. Time constants of temperature sensors.

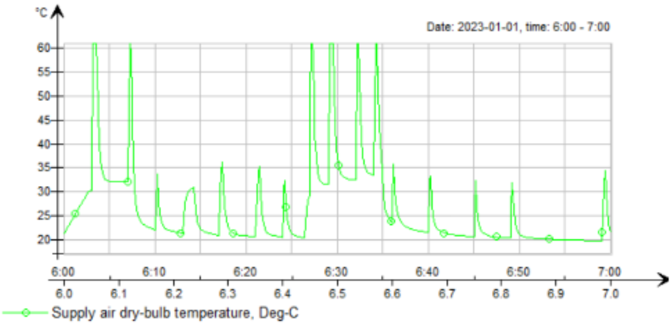


Figure 22 - Example of ventilation supply temperature without temperature time constant.

The clothing value has been set to 1.1 ± 0.7 and is automatically adapted for a neutral PMV, with a PMV of -1 for maximum clothing and a PMV of 1 for minimum clothing, with linear adaption in between.

5.3 Limitations in IDA ICE model

- The position of the train varies depending on the journey, and the climatic data which is used is from the weather station in Bromma airport.
- Shadows from the surroundings only depend on the solar path, but the obstructions in relation to the building are constant. This would be very dependent on whether the train is in an open landscape or below obstacles from the train stations. To at least account for the tunnels in Stockholm city center, a sensitivity analysis is provided for more obstructions from sun and how it affects the energy balance.
- In the simulation model in IDA ICE, there are limitations that apply to how well it can simulate the airflow caused by the door opening, especially for a train. One example is the wind dependency of the infiltration through the door openings, which are highly dependent on which station is considered and the wind shield from the surrounding. A simplification in the simulation model is made for wind speed at given height. However, this can be adjusted by adapting the Cd factor to

what has been observed in experiments or from data analysis [43]. In IDA ICE, the pressure contribution caused by wind is calculated at each of the external surfaces of the building in the height of the roofs. Then, the wind load is assumed to be constant over the surfaces. It considers the wind directions and faces azimuths.

$$U_{wind} = U_{measured} \cdot coeff \cdot \left(\frac{H}{H_{ref}} \right)^{AEXP} \tag{1.35}$$

- Another simplification, at least in IDA ICE 4.8, is the temperature within the zone is constant at all locations. Thus, the temperature close to the doors can be different, giving another airflow as infiltration airflow is temperature dependent.
- IDA ICE 5 (expert-version), is a Beta-version, so the knowledge network around the functions of this version is not fully developed regarding forums etc.
- IDA ICE 5 simulations are only being used in this project for the short-time analysis of chosen days and hours. For the yearly energy analysis, the IDA ICE 4.8 is used. With the detailed analysis, the IDA ICE 5 model is too large to run for yearly simulations.
- The air speed for the thermal comfort evaluation is set to a constant within the zone of 0.2 m/s.

5.4 Door opening

In general, door openings cause infiltration and exfiltration, an uncontrolled ventilation flow rate. Infiltration and exfiltration, not only from doors but from leakages, cause the energy demand to increase in the cooling or heating season. In addition, it can cause cold air draughts through the coach. Therefore, the ventilation system is controlled in flow, supply temperature, and filter, ensuring thermal comfort and good air quality [44].

A study [45] to verify analytical and CFD predictions with an experimental method investigates short-time periods of the door opening, which is more like train conditions of the door opening. The testing procedure was conducted by first increasing the CO_2 concentration to 5000 ppm; then, the door was opened for 10-40 seconds intervals. Then CO_2 level was measured immediately after the door was closed again, and the infiltration was calculated through the equation (1.36). The temperatures were outdoor 20 °C and indoor -20 °C and the door opening sizes were three different 1.36 m x 3.2 m, 1.0 m x 3.2 m, and 0.43 x 0.69 m. A linear relationship between the infiltration and the door opening time, with a correlation coefficient (R2) of 0.99 for the time ranges investigated. The main reason in that case for the linear relation is the air temperature difference between outdoor and indoor remaining approximately the same (in magnitude), assuming short door opening periods.

$$L = \frac{V}{t} \ln\left(\frac{C_1}{C_2} \right) \tag{1.36}$$

Having frequent door openings can also cause temperature fluctuations which affect thermal comfort. A CFD simulation was done in a study of how door opening affects the thermal environment in a vehicle. The study included the effect of the actual movement of the door, but it was concluded that that did not have a significant impact on the air movement; it was rather the temperature difference that was the driver of air exchange, therefore, the study could be applicable for trains as well. It was observed in the simulations in the study that large temperature differences occurred vertically after the door had been open. The assumptions in the simulations were an indoor temperature of 25 °C and an outdoor temperature of -5 °C. However, this study is not considering wind speed. The airspeed was 1 m/s in the upper part of the opening area and 0.6 in the lower part of the opening area. In 5 seconds after the door was closed, the internal air speed was 0.6 m/s, seen in 0, Figure 73 [46].

In Figure 72 in Appendix D, the airflow through the door is divided into different vertical sections, inflow in the upper region, boundary layer in the middle layer, and outflow in the lower vertical region. For the upper region in Figure 23, the air flows out of the coach, in the middle there is a boundary layer between outflow and inflow, and in the lower region the air flows into the coach. If neglecting the pressure difference and density differences the inflow of air and outflow of air will be equal.

There are various causes of air infiltration caused by the door opening: temperature difference, pressure difference, and wind load. Natural convection is caused by density differences, which are dependent upon the air temperatures of the indoor and outdoor volume [47]. The air density of air is calculated accordingly to the equation (1.37).

$$\rho_{air} = \frac{P_{air} \cdot M_{air}}{8314.5 \cdot (t + 273.15)} \tag{1.37}$$

The model for airflow in large openings used in IDA ICE is the CELVO model [44], which can calculate the flow bidirectionally. The model takes the pressure differences at different vertical heights of the door, and depending on the pressure difference, different conditions will be fulfilled which decides the flow direction at different heights. The total pressure difference across the door depends on the wind-induced pressure, stack effect pressure, and pressure difference caused by the ventilation system [44]. The pressure caused by the wind are calculated through the equation (1.38).

$$P_{wind} = \frac{1}{2} \cdot C_p \cdot \rho \cdot U_z^{\ 2} \tag{1.38}$$

The wind pressure coefficient C_p depends on the facade orientation, wind direction. V_z is the wind speed at the height z as described in equation (1.35)

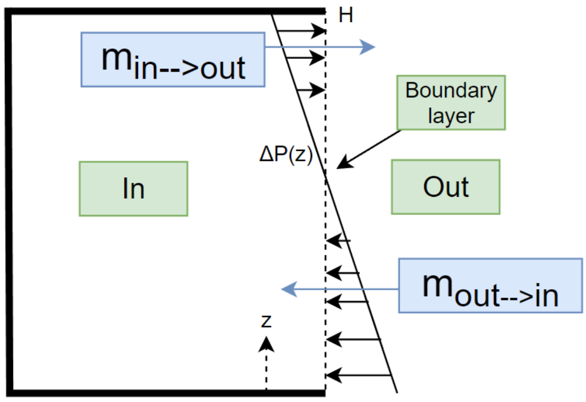


Figure 23 - Large door opening airflow. Pressure difference.

Pressure differences at the top (z=H) and the bottom (z=0) are calculated accordingly:

$$\Delta P_{z=0} = (P_{in} - \rho_{in} \cdot g \cdot z_{0,in}) - (P_{out} - \rho_{out} \cdot g \cdot z_{0,out}) \tag{1.39}$$

$$\Delta P_H = \Delta P_{z=0} - \left(\rho_{in} \cdot g \cdot z_H + \rho_{out} \cdot g \cdot z_H\right) \tag{1.40}$$

5.5 Balancing temperature

A balance point temperature of a building is the temperature that is typically used as a reference to calculate the degree days of a building and the heating demand. It is the point where there is a balance of heat gains and heat losses in the building, and the temperature indoors remains the same without the heating system on. The internal gains of a building can consist of lighting, equipment, solar gains etc. [48]. In a train, this balance point temperature is more complex due to many factors affecting the heating demand, but mainly the passenger load (internal gains), which has large fluctuations, and the external conditions depending on location etc. In the autumn and spring, temperatures can be in-between the need for cooling and heating. With a more detailed definition of the balance temperature for trains, such as conditions of both outdoor temperature and indoor temperature range. In combination with a dead-zone temperature range between 18°C and 23°C.

For the flow scheme of the air handling unit for balancing temperature, some adaptations have been made compared to the baseline model. In addition, the criteria for the indoor and outdoor temperature for the electric heating unit have been changed.

The control for the control of electrical heating unit has two constraints; if the interior temperature is above 18°C and the outdoor temperature is above a certain temperature, then the heating unit is turned off.

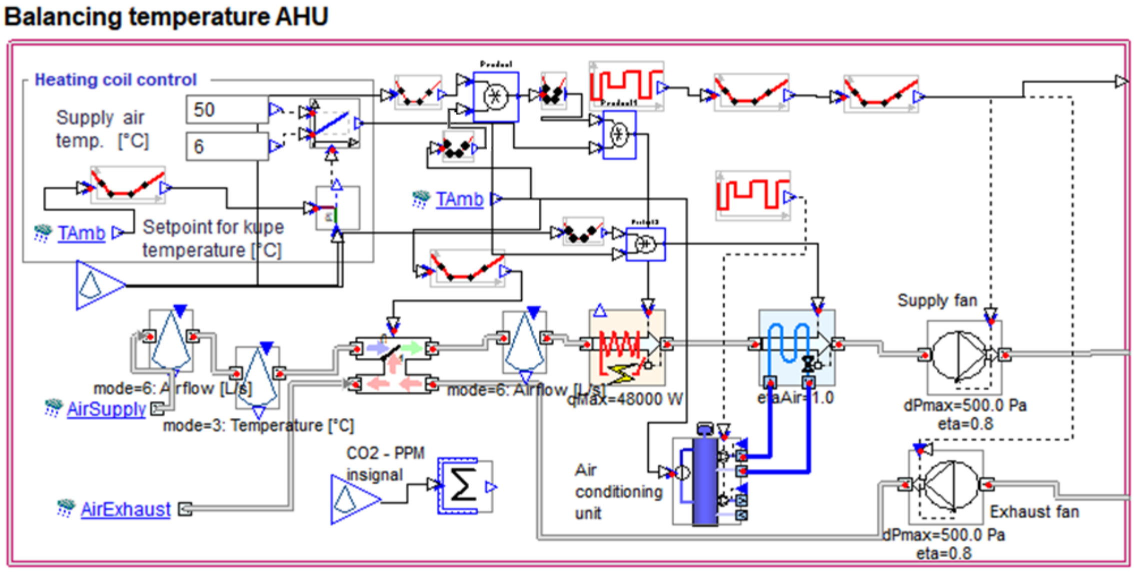


Figure 24 - AHU adaption for dead-zone temperature range between 18°C to 23 °C.

5.6 CO_2 AND TEMPERATURE-CONTROLLED VENTILATION FLOWRATE

The recirculation control is presumed to be the same as the baseline scenario. The ventilation flow rate is controlled through 400 PPM minimum ventilation flowrate to 1100 PPM maximum flowrate and through heating/cooling demand. For the minimum ventilation flow rate, it is based on the ventilation requirement of 15 m3 per person and hour [11] and a passenger number of 62 people (fully seated). In Figure 25 the CO_2 concentration using different ventilation control strategies are shown. These concentrations can be adjusted with the different control strategies input.

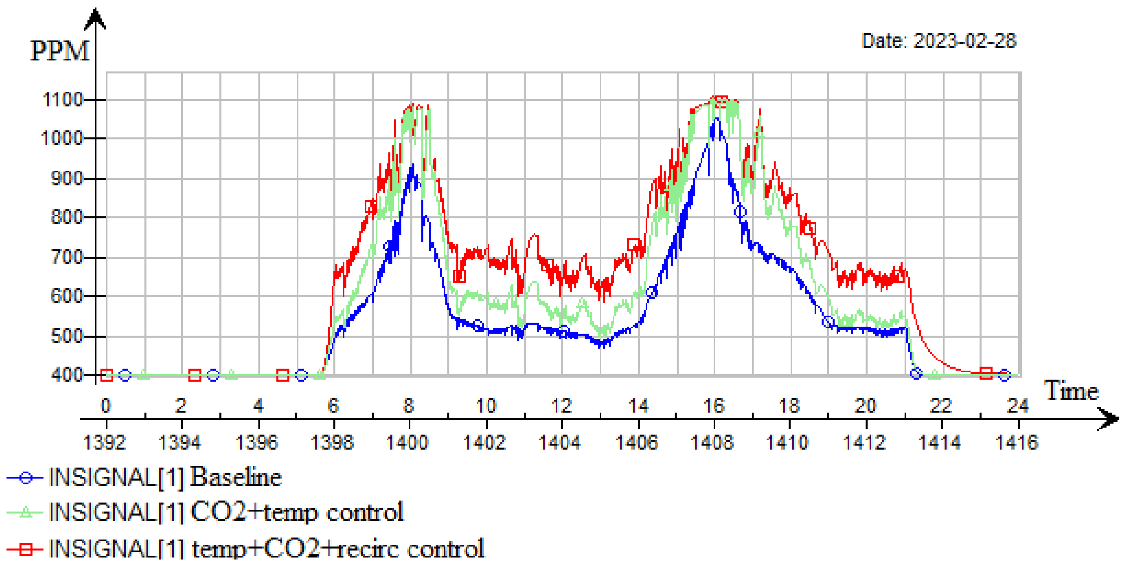


Figure 25 - Comparison CO_2 PPM different ventilation control strategies.

5.7 VALIDATION

In Figure 26 the surface temperature of the inside of the doors is shown, at an outdoor temperature of 8°C. The black line indicates the surface temperature of the door in which it has been closed and then not opened again until 7:30. The blue line indicates the door, which was at an almost steady state, and then started opening and closing cycles at a rate of around 1 opening (20 seconds) every 3 minutes. This figure verifies how the IDA ICE model considers the door opening and how it affects the surface temperature at the door. These results are reasonable in the sense of the time constant of the door and how the repeated openings affect the surface temperature. The surface temperature affects the radiation according to the equation (1.22), and should thereby be considered for thermal comfort.

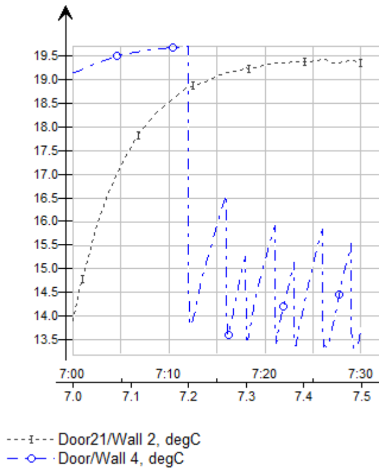


Figure 26 - Doors interior surface temperatures of west respectively east side.

Figure 27 displays the energy signature for the IDA ICE model verifying with the heating derived from data analysis and predicted heating demand at various outdoor temperatures. The IDA ICE heating values have been derived based on monthly temperatures and the monthly average heating from the model.

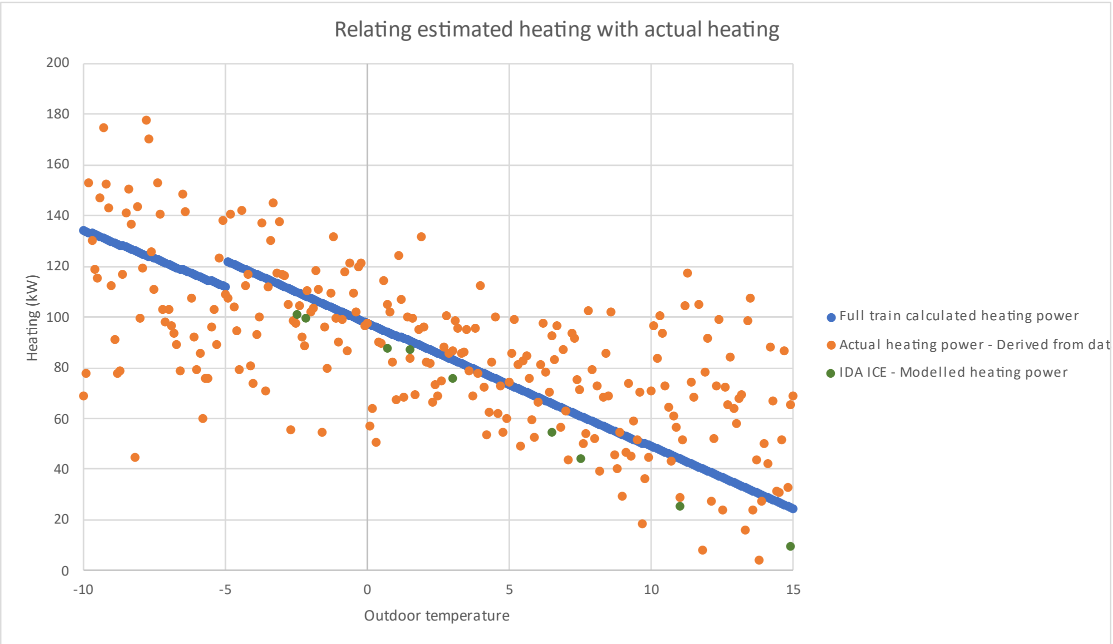


Figure 27 - Energy signature in heating- Comparing Data, Calculated and IDA ICE model.

To analyze how well the time constant of the inertia is to the literature study, a comparison was made. On this day chosen for comparison, there are almost no solar gains, especially during the night so the only heat transfer will be the heat transfer through the envelope. At 10 hours of a shutdown of all systems onboard (no heating), the temperature in the simulation model reaches 11.6°C. In the calculation based on heat transfer coefficients and heat capacities numbers from [7], X61-model, the temperature reaches 11.4°C with an outdoor temperature of 0°C (see for how the calculation was done, using Euler method). In Appendix F, further details about the Euler method calculation are shown.

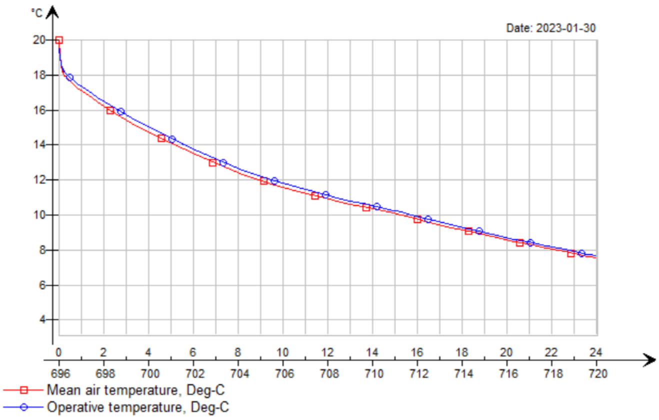


Figure 28 - Temperature drop for 24-hour period, with ventilation off, outdoor temperature 0°C. No occupancy/equipment/lighting.

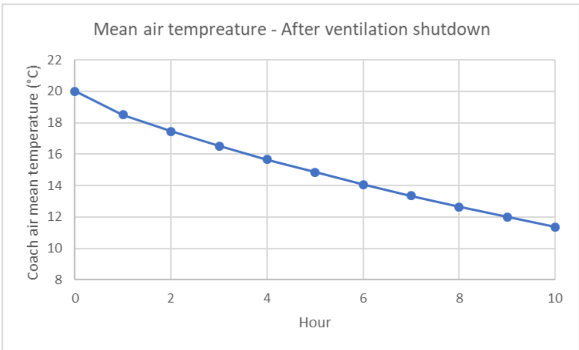


Figure 29 - Temperature drop using Euler-method - Based on heat transfer coefficients and heat capacities from [7]

In the Table 15 below, the differences in measured surface temperatures with thermal camera and simulated values are compared. The highest difference temperature difference is for the ceiling, which may be because of the local temperature difference due to door openings or could be an indication for an actual higher U-value in reality of the roof since the actual U-value is not known (but estimated from other sources). For the wall, the measured values were divided into vertical sections while measuring, in which the average exterior wall temperature was calculated.

Table 15 - Validation of interior coach surface temperatures.

Location	Measured	Simulated	Absolute difference	% difference
Ceiling	21.6	19.2	2,35	12%
Floor	18.1	17.9	0.2	1%
Lower wall	21.0	-	-	-
Middle wall	16.8	-	-	-
Upper wall	17.7	-	-	-
Average exterior wall	18.5	18.1	0.41	2%
Interior door to driver cabin	20.6	19.9	0.75	4%
Doors	8.6	8.3	0.3	4%

During a day of 0°C, a random door opening was chosen as input to IDA ICE. Assuming a linear decrease, as indicates in chapter 5.4, of the temperature from 20.83°C to 18.4°C, corresponding to a drop of 0.1345 °C per second, then an increment by the observations from IDA ICE simulation, a comparison with the temperature sensor can be approximated. Calculating the new sensor temperature after every timestep of 1 second, it can be calculated how much of the temperature can be detected by the temperature sensor, using equation (1.21). However, this is based on the simulation, and it is difficult to estimate the actual rate of temperature drop.

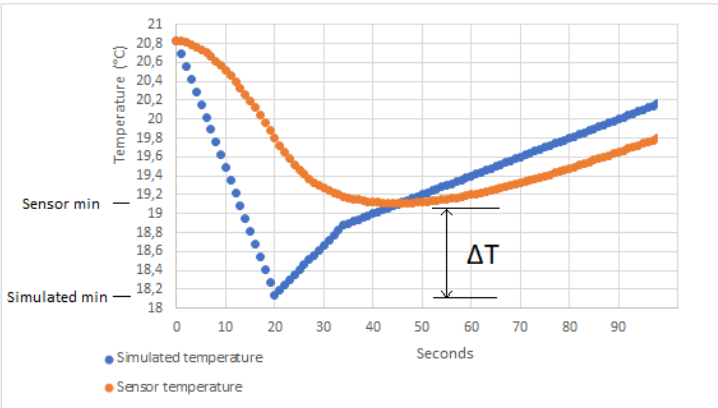


Figure 30 - Explanation of sensor temperature vs real temperature (or simulated).

6 Results and discussion

In this section the experimental measurements results, data analysis and simulation results are shown in combination with the discussion.

6.1 Experimental measurements

6.1.1 Winter measurements

In Figure 31, different airspeeds can be noticed depending on whether the measurement is in the bottom, middle, or upper vertical section.

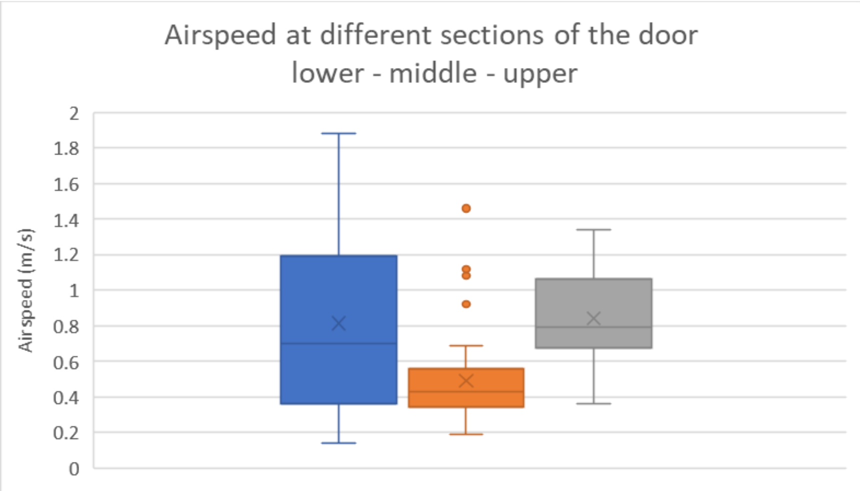


Figure 31 - Air speeds at different sections of the door. Blue is the upper door section, orange is middle and the gray is the lower section of the door. Measured between outdoor temperatures of -3 to 4 °C outdoor temperature. 81 Measurements.

Table 16 - Air speed measurements and temperature.

Speed perpendicular to door 1 m away (m/s)	0.42	0.43	0.56	0.56						
Vertical air speed at seat (m/s) Doors closed	0.26	0.11	0.24	0.12	0.2	0.24	0.25	0.19	0.22	0.2
Horisontal air speed at seat (m/s) Doors open	0.17	0.16								
Horizontal air speed at seat (m/s) Doors closed	0.11	0.09	0.2	0.1	0.06	0.03	0.07			
Indoor temperature at seats ("C)	20.9	21	21.8	21.9						
Data channel temperature (°C)	19.68	19.74	19.98							
Temperature drop (air probe) at seat after doors open (°C)	2	0.7								

Table 17 – Thermal camera measurements

Surface temperature (°C)	(number of measurements)		Similar conditions - 3 measurements periods - Around -4°C to 0°C		
Lower wall (3)	Seat (7)	Interior surface plate (3)	Ceiling (6)	Grip handle (1)	Floor (9)
21.03	20.31	19.33	21.55	20.00	18.08
Shoes (2)	Jeans (3)	Thick jacket (3)	Outer door (4)	Outdoor surface (5)	Outer window surface (1)
25.40	32.23	24.40	2.18	-0.78	-0.20
Between coach outer surface (1)	Lights (3)	Upper wall (1)	Middle wall (1)	Vent. supply outlet (3)	Internal door surface (1)
-0.40	25.13	17.70	16.80	28.03	8.60

Based on Table 17, the overall surface temperatures are relatively high in comparison to the air temperature.

6.1.2 Spring measurement

For the spring season, one day of typical spring conditions was used for measurement, the time was around 2 pm with outdoor temperature is around 11°C, wind condition 4 (7) m/s, and relative humidity 31 % (weather data). Passenger load was between the measurement time increasing from 1.5 to 2.5 on the scale described in Table 11.

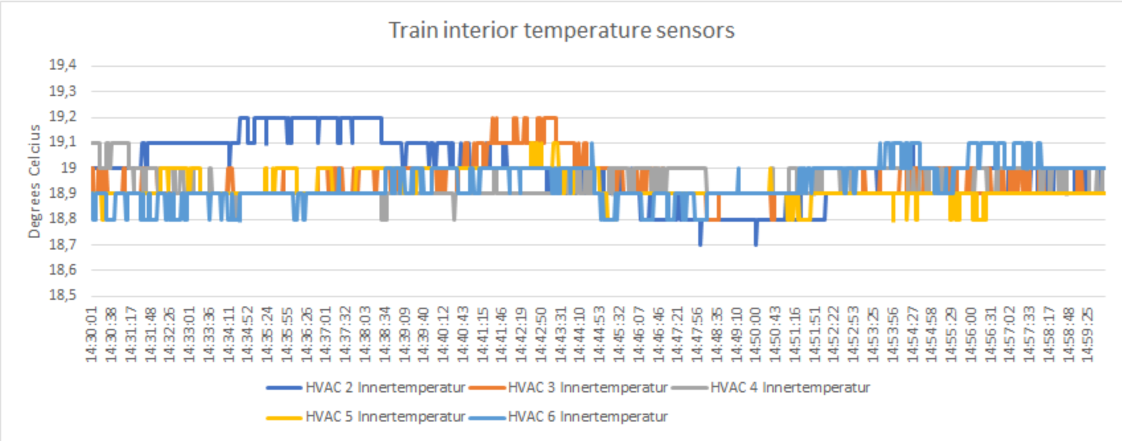


Figure 32 - Temperature sensors of the train interior temperature at ceiling height.

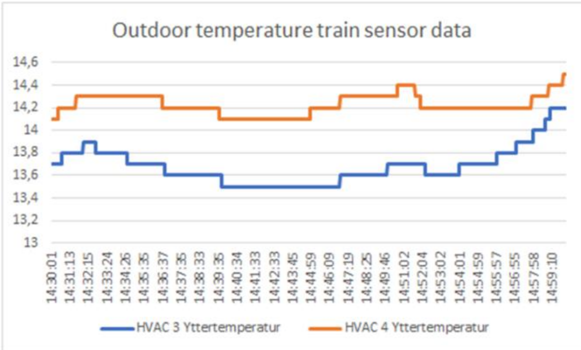


Figure 33 - Train outdoor temperature sensor data - Outdoor temperature.

6.2 ANALYSIS OF SIMULATION

In Figure 34, one morning rush with an occupancy load of up to 80 passengers per coach is simulated. The internal passenger gains (filled green) are increasingly lowering the ventilation air's supply temperature (green graph). As a result, CO_2 concentration (red) reaches 1000 PPM, and after this peak, the passenger load will decrease according to the average occupancy schedule. The fluctuations in the CO_2 concentration are due to the door openings.

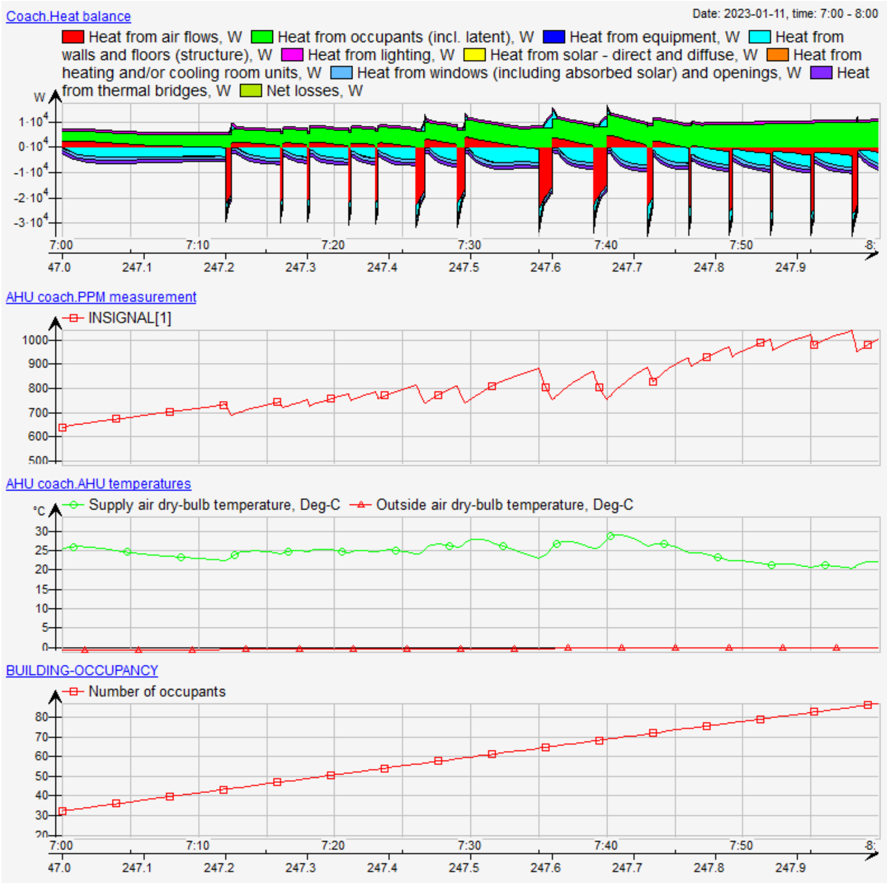


Figure 34 - Passenger load, AHU temperatures, PPM, and heat balance of one coach.

Comparing Figure 34 with Figure 35, in which Figure 35 indicates the hour before where the passenger load is low. In these two figures the heat balances are heavily dependent on the passenger load.

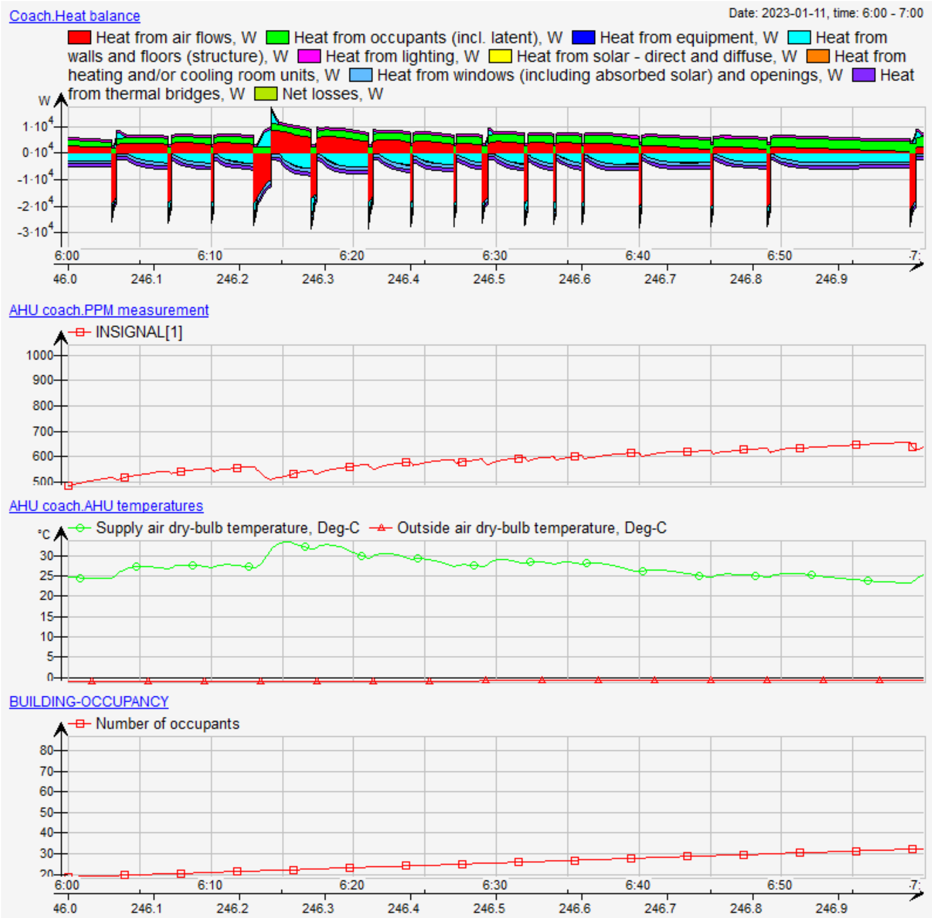


Figure 35 Passenger load, AHU temperatures, PPM, and heat balance of one coach.

In Figure 36, the airflows are caused by the door opening, infiltration, exfiltration, and ventilation flow. The red bars are the exfiltration through the door openings of one coach, green is the infiltration through the door openings, and yellow is the ventilation supply and exhaust. The ventilation flow rates are decreased slightly in the simulation schedule while the doors are open as a function of the ventilation control.

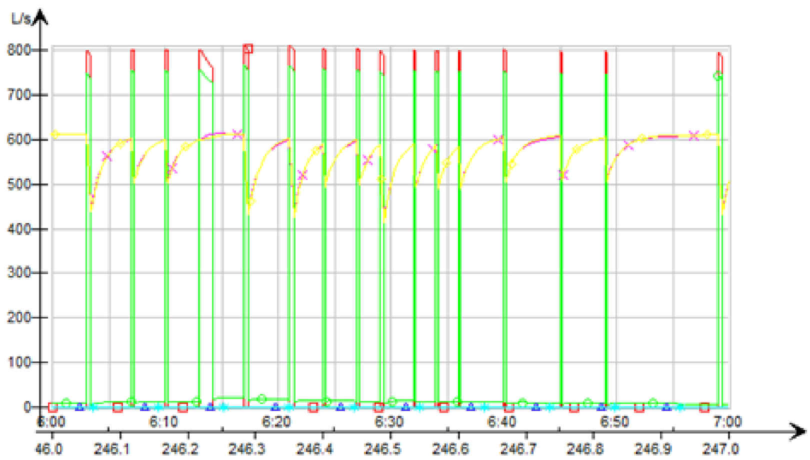


Figure 36 - Airflow due to door openings and ventilation. Yellow lines are the constant ventilation flow, which has setback of ventilation at door openings. The green and red graphs are supply respectively exhaust air flow.

In Figure 37 Figure 38 the PPD and PMV values are shown for different passengers with different MET-values, which indicates a high dependency of PPD and PMV in relation to MET-values.

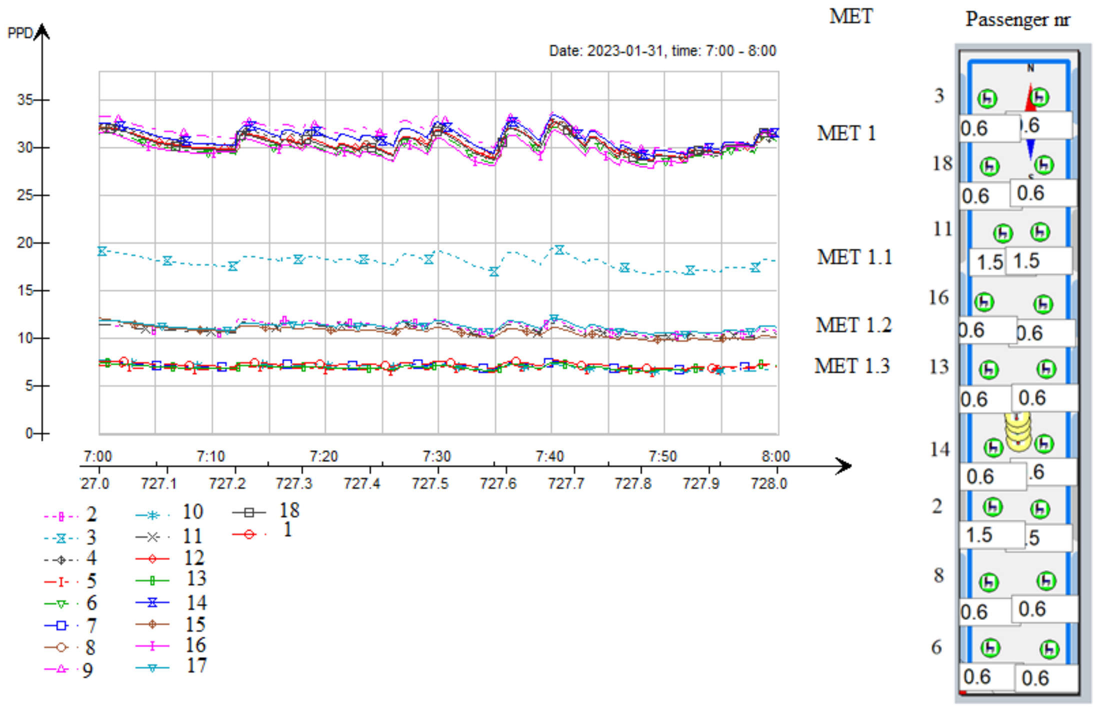


Figure 37 - PPD for different occupants and different MET & location.

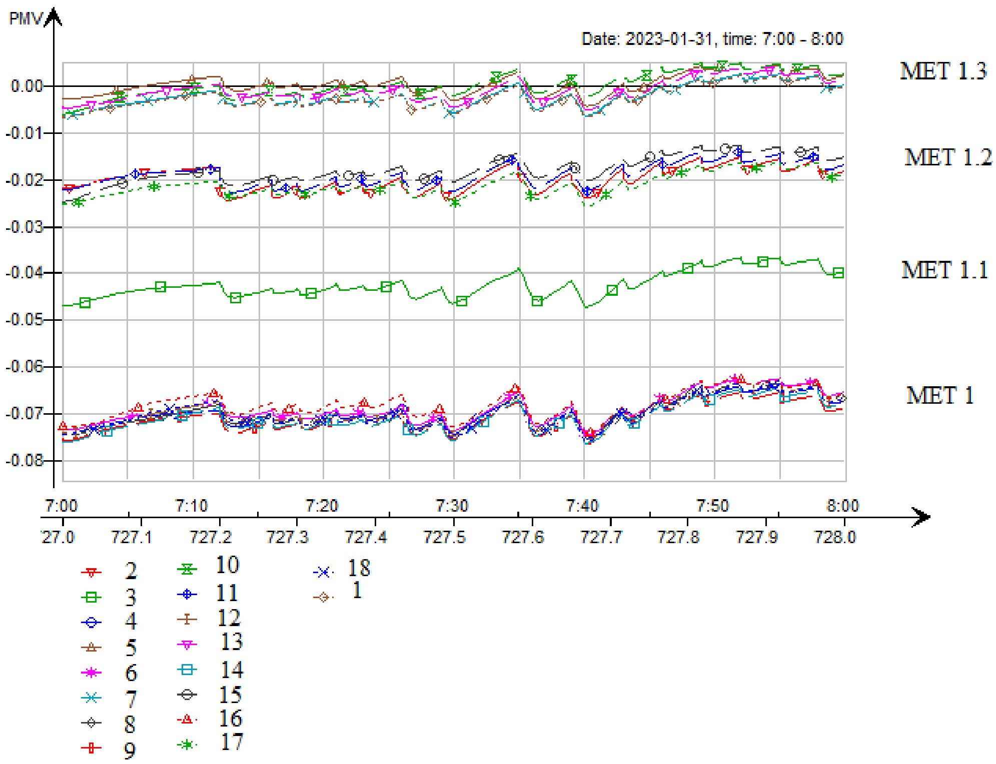
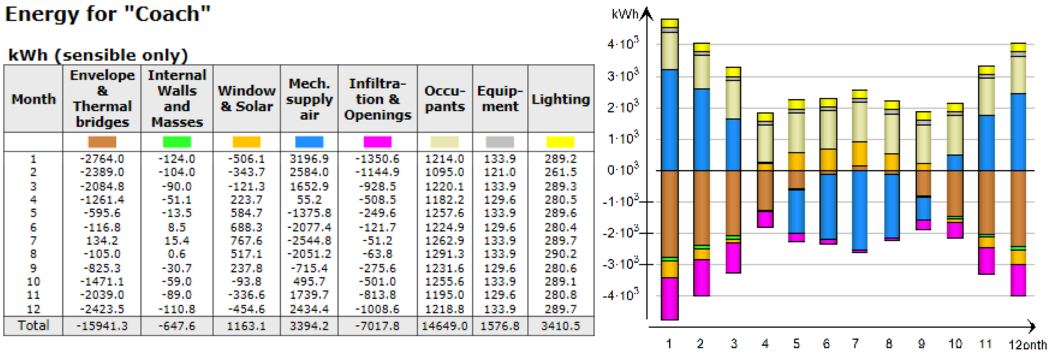


Figure 38 – Continuation of previous figure. PMV-values.

In Figure 40 and Figure 39, the heat monthly heat balances are shown for the baseline case of one coach. However, for the summer the energy has not been evaluated or validated, so only the winter months should be looked at in these two figures.

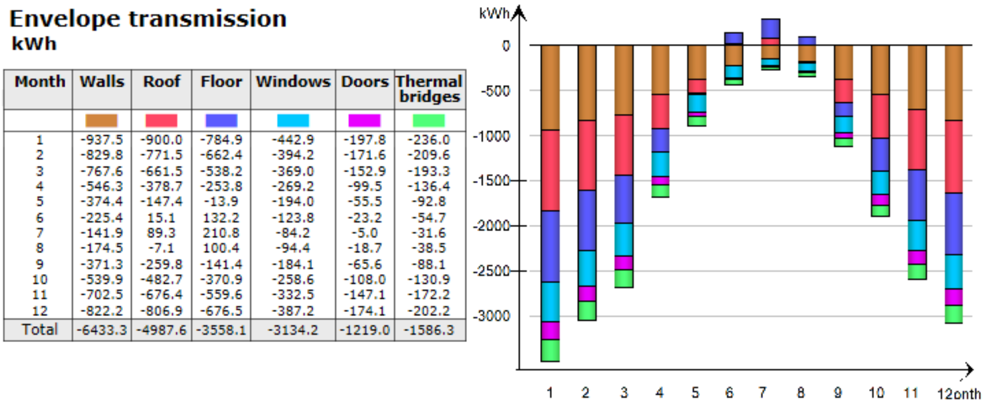


Energy for "Coach"

kWh (sensible only)

Month	Envelope & Thermal bridges	Internal Walls and Masses	Window & Solar	Mech. supply air	Infiltration & Openings	Occupants	Equipment	Lighting
1	-2764.0	-124.0	-506.1	3196.9	-1350.6	1214.0	133.9	289.2
2	-2389.0	-104.0	-343.7	2584.0	-1144.9	1095.0	121.0	261.5
3	-2084.8	-90.0	-121.3	1652.9	-928.5	1220.1	133.9	289.3
4	-1261.4	-51.1	223.7	55.2	-508.5	1182.2	129.6	280.5
5	-595.6	-13.5	584.7	-1375.8	-249.6	1257.6	133.9	289.6
6	-116.8	8.5	688.3	-2077.4	-121.7	1224.9	129.6	280.4
7	134.2	15.4	767.6	-2544.8	-51.2	1262.9	133.9	289.7
8	-105.0	0.6	517.1	-2051.2	-63.8	1291.3	133.9	290.2
9	-825.3	-30.7	237.8	-715.4	-275.6	1231.6	129.6	280.6
10	-1471.1	-59.0	-93.8	495.7	-501.0	1255.6	133.9	289.1
11	-2039.0	-89.0	-336.6	1739.7	-813.8	1195.0	129.6	280.8
12	-2423.5	-110.8	-454.6	2434.4	-1008.6	1218.8	133.9	289.7
Total	-15941.3	-647.6	1163.1	3394.2	-7017.8	14649.0	1576.8	3410.5

Figure 39 - Heat balance.



Envelope transmission

kWh

Month	Walls	Roof	Floor	Windows	Doors	Thermal bridges
1	-937.5	-900.0	-784.9	-442.9	-197.8	-236.0
2	-829.8	-771.5	-662.4	-394.2	-171.6	-209.6
3	-767.6	-661.5	-538.2	-369.0	-152.9	-193.3
4	-546.3	-378.7	-253.8	-269.2	-99.5	-136.4
5	-374.4	-147.4	-13.9	-194.0	-55.5	-92.8
6	-225.4	15.1	132.2	-123.8	-23.2	-54.7
7	-141.9	89.3	210.8	-84.2	-5.0	-31.6
8	-174.5	-7.1	100.4	-94.4	-18.7	-38.5
9	-371.3	-259.8	-141.4	-184.1	-65.6	-88.1
10	-539.9	-482.7	-370.9	-258.6	-108.0	-130.9
11	-702.5	-676.4	-559.6	-332.5	-147.1	-172.2
12	-822.2	-806.9	-676.5	-387.2	-174.1	-202.2
Total	-6433.3	-4987.6	-3558.1	-3134.2	-1219.0	-1586.3

Figure 40 - Heat transmission through envelope.

Further, the monthly heat recovery from the recirculation can be seen in Table 18.

Table 18 – Simulated recirculation, and kWh heating saved through the recirculation

	Heating (kWh)	AHU heat recovery recirculation (kWh)
Jan	14 142	1 113
Feb	12 445	831
Mar	11 493	98
Apr	7 180	0
Maj	3 858	0
Jun	2 039	579
Jul	1 223	2 112
Aug	1 452	530
Sep	3 929	104
Okt	7 104	0
Nov	10 495	71
Dec	12 457	388
Total	87 819	5 830

In Figure 41, the simulation made in IDA ICE 5, the energy balance of the coach changes with the passenger load. The door openings are the bar peaks in the energy balance, affecting the CO_2 concentrations (the fluctuations in the PPM in-signal).

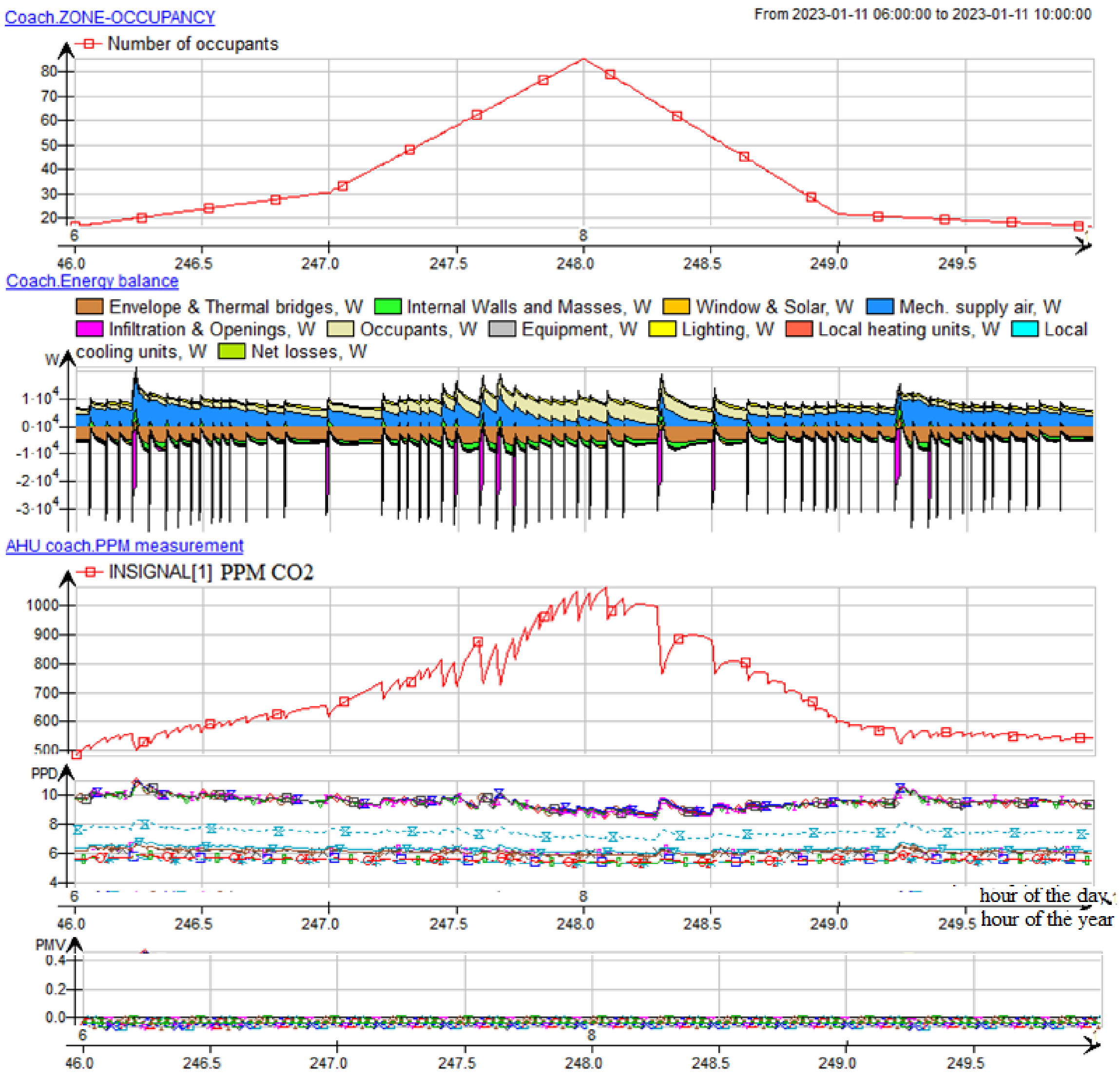


Figure 41 - Baseline thermal comfort analysis, at outdoor temperature of 0°C. Setpoint 20°C.

6.3 Measures

In this chapter the different measures are discussed regarding their potential and simulation results.

6.3.1 Parking mode

Only with a temperature setback

In this measure, everything is remained same as baseline except that the temperature setpoint is set to 15°C at night, as seen in Figure 42. A decrease per coach of 11 884 kWh or 71 300 kWh per train in heating per year, corresponding to a 14.3 % decrease in annual energy.

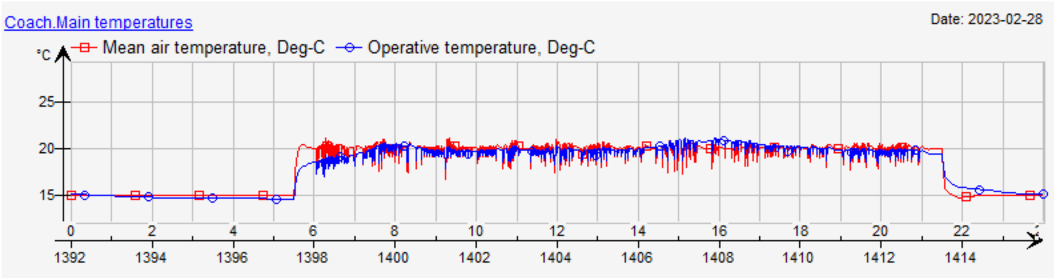


Figure 42 - Night temperature setback to 15°C. Outdoor temperature 2-3°C

Since the ventilation system is running continuously in this case, the temperature increases and decreases rapidly. In the next measure, the ventilation fans will be turned off for a few hours after parking. The heating goes entirely on, and free cooling is used to cool down rapidly according to the ventilation control logic seen in Figure 42.

Table 19 - Simulated annual saving of energy compared with no parking mode. Only temperature setback.

Setback temperature (°C)	Annual heating energy saved
15	13.2%
12	17.2%
9	19.2%
5	19.5%

As shown in Figure 43 below, the heating demand and supply temperature are reduced with the temperature setback. This figure includes the heating demand of the two driver cabins. However, in the morning the heating demand is increased and the supply temperature is increased before the train is taken into service at 6 am for this specific case and is then decreasing from the peak until around 8 am.

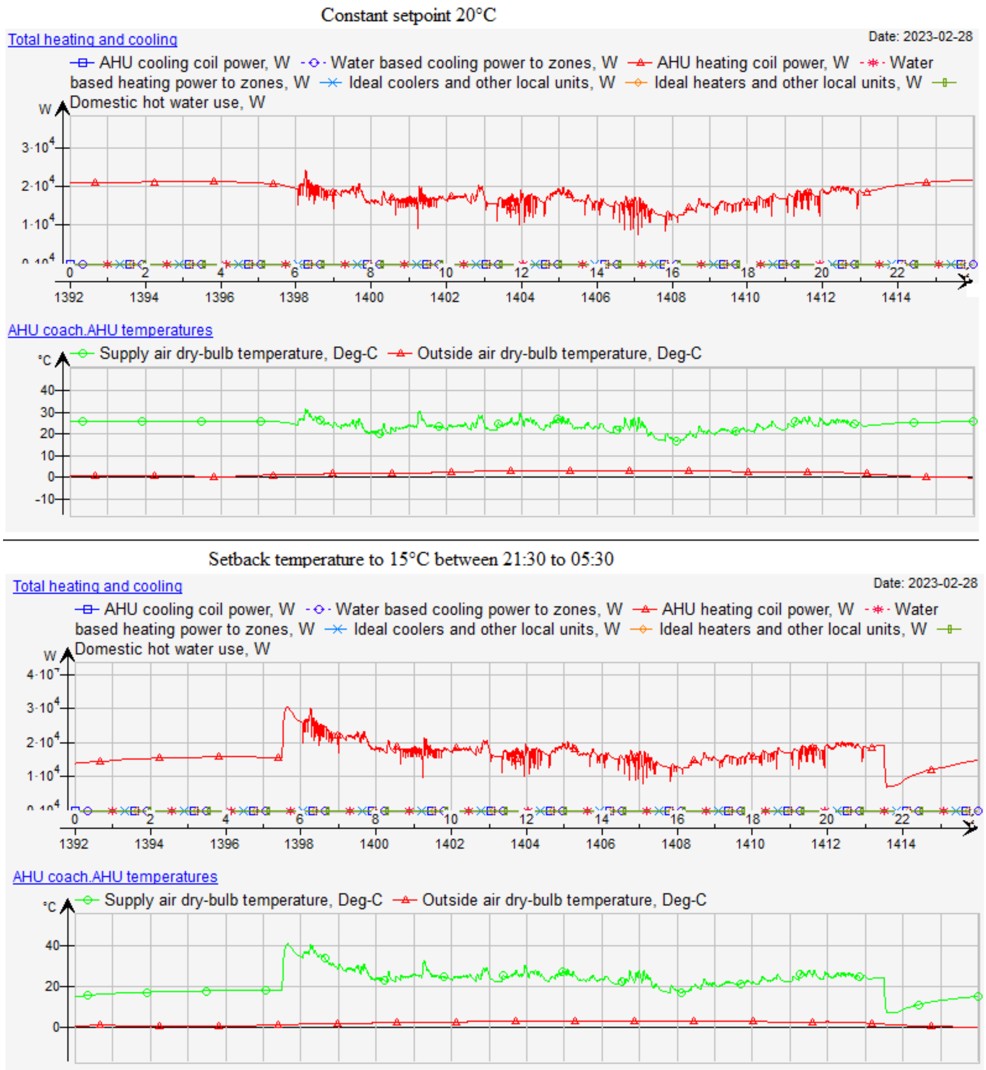


Figure 43 - Comparison without and with temperature setback at nighttime.

For the parking modes, different temperature setpoints are not reached at all outdoor temperatures, as it takes time for the coach to cool down. This is observed in Figure 44 where different interior setpoint temperatures are used at an outdoor temperature of 0°C. In the first case, at an outdoor temperature of 0°C in Figure 44, 15°C, 12°C and almost 9°C are reached; thus the 9°C graph and 5°C are the same in the left figure. For the simulation, the cooling function (air conditioning) has been turned off for the winter, to avoid the cooling to be turned on to cool down the train to setback temperature setpoint. The day before and after had a similar outdoor temperature of 0°C, respectively -8°C. From this perspective, a setback temperature of 5°C is rarely reached.

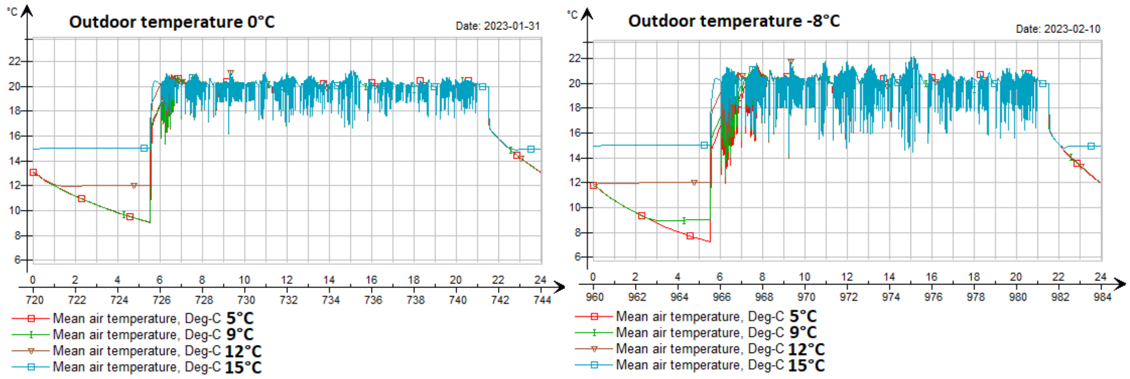


Figure 44 - Comparing temperature setpoints of parking mode temperature setback.

The optimum setpoint temperature for the parking mode depends on the train operator's operating constraints, such as the heat up-time for the train to go back into service and the safe temperature to avoid condensation and other potential problems. Because of the thermal mass of the train, it takes time to reach the setback temperature, and thus it is not a linear saving about the setback temperature. Already at 15°C setback temperature 13.2 % of annual energy is saved, as in Table 19, increasing to 17.2 % for 12 °C, but only to 19.2% saving for 9°C. At 15 °C, the temperature is close to in-service temperature, which could arguably be relevant for the reliability of the train system returning to in-service quickly. In the worst case, having the train in-service at 15°C might also be possible if a malfunction happens.

Temperature setback, recirculation, and ventilation off (few hours)

In this evaluation, the temperature setpoint is set to 15°C at night, combined with recirculation and ventilation turned off in the first few hours. Ventilation was turned off until reaching 15 °C to simulate the natural cool-down through the envelope instead of forcing it to 15°C with the ventilation air. The air quality is recovered to the background CO_2 level quickly as the ventilation is kept on for half an hour after operation in the simulation model.

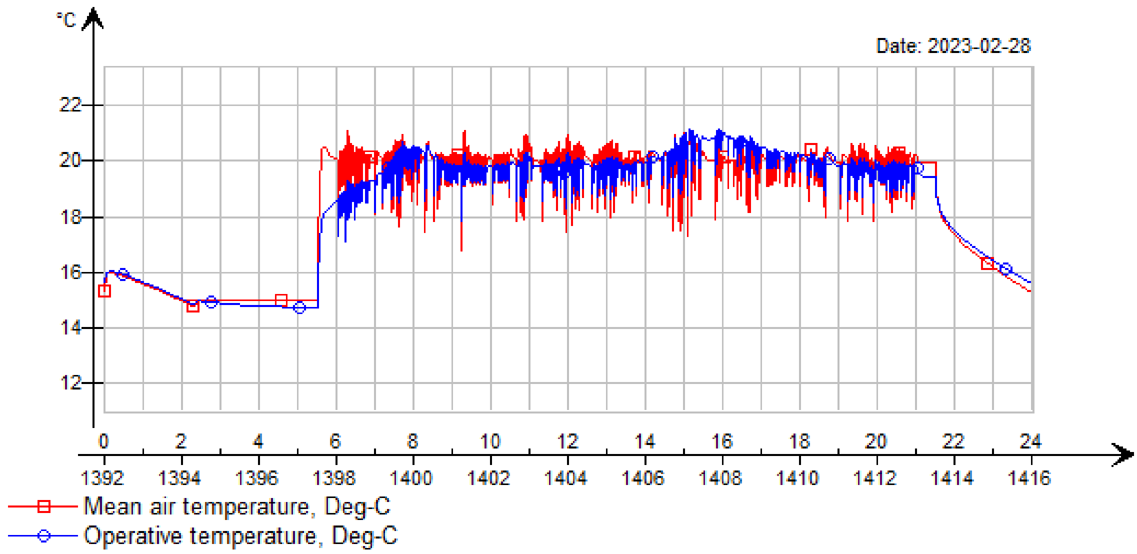


Figure 45 - Temperature setback and recirculation during night

28 300 kWh saving in heating energy per coach or 169 000 kWh per train and year, corresponding to 34 % reduction of heating energy per year.

Since it would be large costs if a train cannot go in-service for some technical problem, the system's operational reliability is of utmost importance. This is one of the reasons efficient parking modes are not in use, as they can cause problems with returning to service. From this perspective, it might also be arguable that the parking modes are in-between the most energy-efficient and in-service modes, so it is simple to take it back to service. Another important parameter to consider is if the temperature setback in parking mode is too low, below the dew point, it will start to condense water in the interior of the train. Figure 65 shows the monthly variation in relative humidity; in the winter season, the simulated relative humidity is at most around 50 %. For relative humidity of 30 % at 20 °C air temperature, the dew point is 2°C, and thus no condensation will form with the significantly higher parking temperature than 2°C in winter.

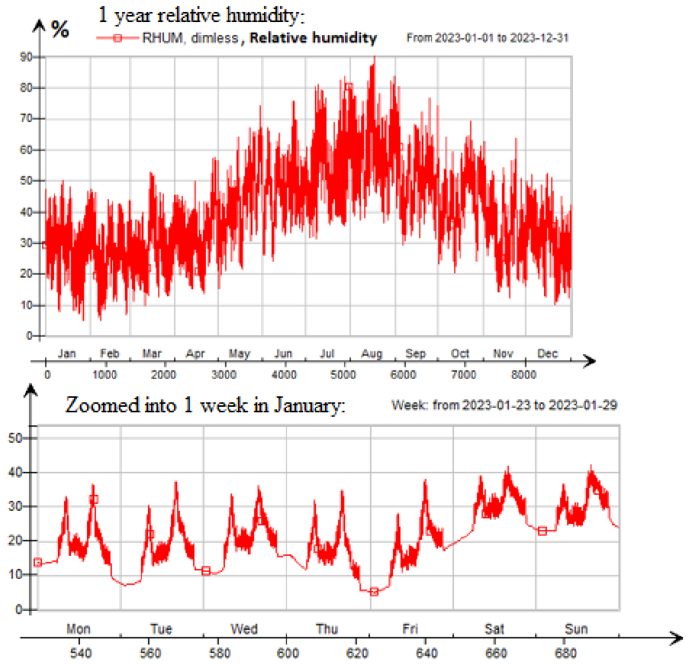


Figure 46 - Monthly variation in relative humidity in IDA ICE simulation model within the coach.

6.3.2 Reversed air conditioning

With a heat pump capacity of 23 kW, the savings are simulated to be 60 % of the heating energy (in which electric heating is needed 16 % of the time for the colder days). With 11 kW heat pump capacity per coach, the energy saved simulated in heating is 43 %, with direct electric heating being 60 % of the total electricity heating. The choice of 23 kW respectively 11 kW, is due to the cooling capacity of one cooling circuit being 11 kW, then assuming the same COP value of 2 for the heating. The COP-value assumption of 2 for the heating case, could arguable be in the lower region of performance for R134a in heating, but the performance of the heat exchangers is not known and thus this assumption considers that the optimal performance may not be reached using the existing components. Also, during winter conditions, defrosting of condenser may be required.

For the reversible heat pump, a four-way valve is introduced to the system, in which it can switch direction between evaporator and condenser. In addition, an extra expansion valve for the heating case in combination with check valves needs to be added to the system [49]. Potential flowcharts of the reversible air conditioner is shown in Figure 47 and Figure 48.

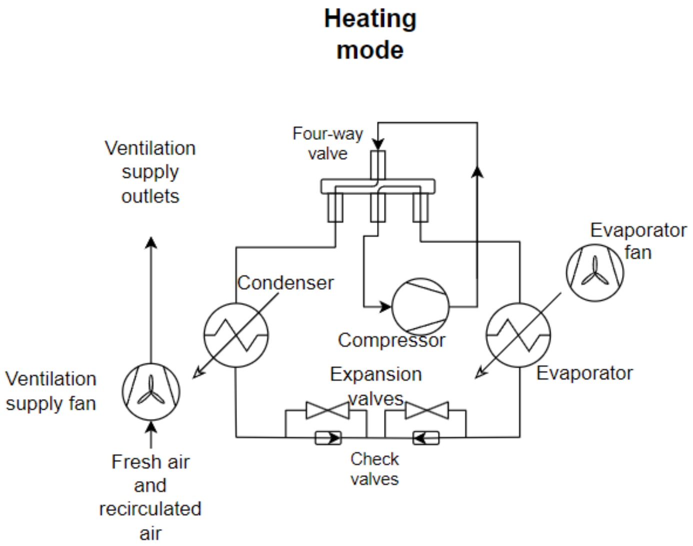


Figure 47 - Reversible air conditioner/heat pump in heating mode. [49]

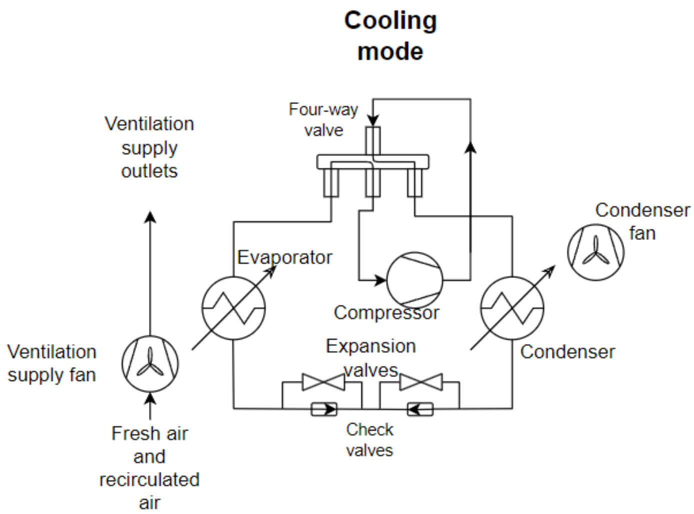


Figure 48 - Reversible air conditioner/heat pump in cooling mode. [49]

6.3.3 Door opening reduction

In the modeling based on a Cd factor of 0.43, a 60 % reduction of door openings between 9 am-3 pm indicated a saving of 8 000 kWh yearly saving per train. In addition, a slight increase in thermal comfort was observed in the simplified IDA ICE 4.8 model of less than 1 % regarding PDH (hours) and max PPD.

Table 20 - Saving through reducing door opening time.

3500 seconds of outdoor station door opening/day						
Reduction of door opening:	10%	20%	30%	40%	50%	60%
kWh saving/train/day	19	38	56	75	94	113
kWh saving/4 months total/80 trains	180 000	360 000	540 000	720 000	900 000	1 080 000

With an extension of the door opening reduction time, from 9 am to 3 pm respectively 6 pm to 10 pm, the savings is approximately 11 000 kWh per train of 6 coaches.

In Figure 50, during rush hour between 16-17 pm, the door openings positively affect the PPM level. In Figure 41, the same behavior can be seen, and in addition, the energy balance of the coach can be seen to be highly affected by the passenger load. It can therefore be argued that the door openings during high passenger load can contribute to better thermal comfort regarding air quality. During the highest passenger load in the morning, at 8 am, the heating requirement is low, as seen in the energy balance (in blue) in Figure 41. However, when the passenger load decreases, the door openings add on extra heating need, especially seen at 9:15 am in Figure 41 (increment of blue supply air heating power in energy balance), between hour 249 and 249.5. In the case of 9:15 am, the PPM level is already at a low level at less than 600 PPM.

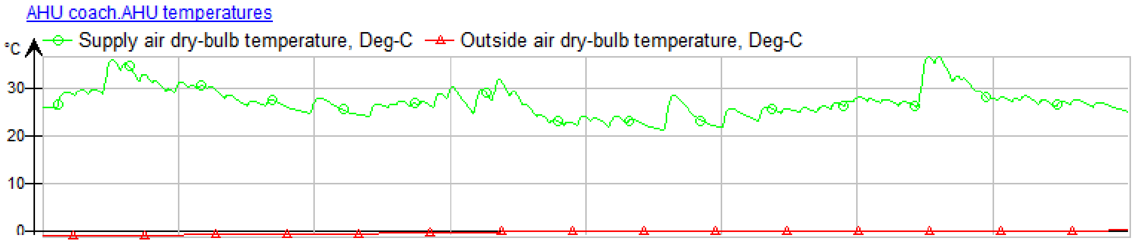


Figure 49 - In addition to Figure 41. Supply and outdoor temperature in morning. IDA ICE 5 model.

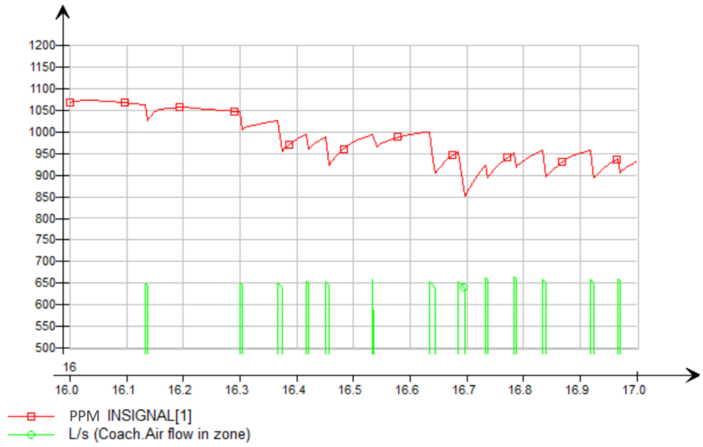


Figure 50 - Door openings ventilation and CO_2 PPM between 16-17 pm rush hour.

To make estimations of the energy loss from door opening through temperature drop, more temperature sensors are needed to get an accurate overview of the average temperature drop of the interior air. Further, based on experimental measurements at around 0°C, since this temperature drops measured by the air probe sensor in Appendix I does not indicate the actual temperature drop, as explained in Figure 30. This methodology of measuring temperature drops was a difficult method for evaluating the energy loss accurately. However, it can still give an idea of the minimum energy loss.

Seen in Figure 5 is the operative temperature fluctuations, they are not significant concerning the air temperature fluctuations, indicating door opening does not affect thermal comfort significantly.

Airspeed was measured both vertically and horizontally, to experimentally estimate the door opening air infiltration. An omnidirectional anemometer would have been preferred to get more accurate air velocity measurements within the train coach.

Data channels for power consumption and door openings were considered for specific outdoor temperature ranges to analyze how the door openings affect the train's energy consumption. First, two-hour in-service train intervals were downloaded from R2M data platform using macro. Then a relation could be obtained through seconds of door opening (of each 2-hour sample) and the average consumption, seen in Figure 53. In Figure 51, each second of door opening increases the average power consumption of a train by 0.0405 kWh/train.

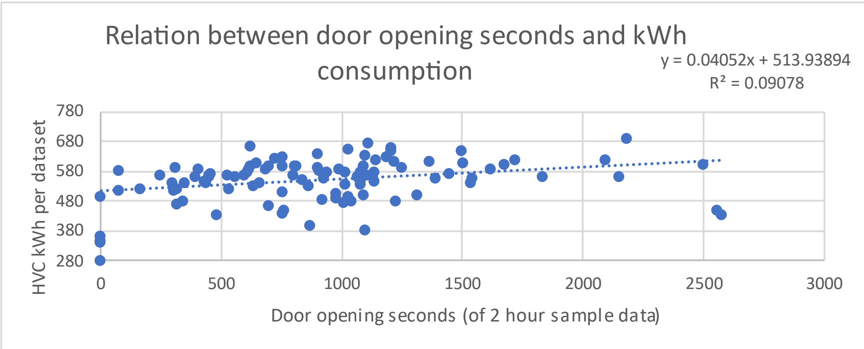


Figure 51 - Relation between door opening seconds and kWh consumption.

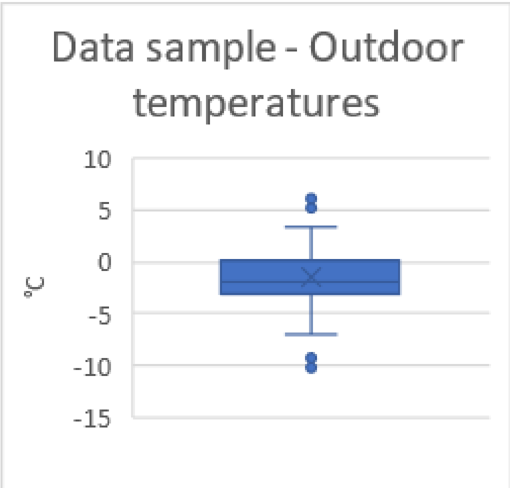


Figure 52 - Sample data - Outdoor temperatures – For Figure 53

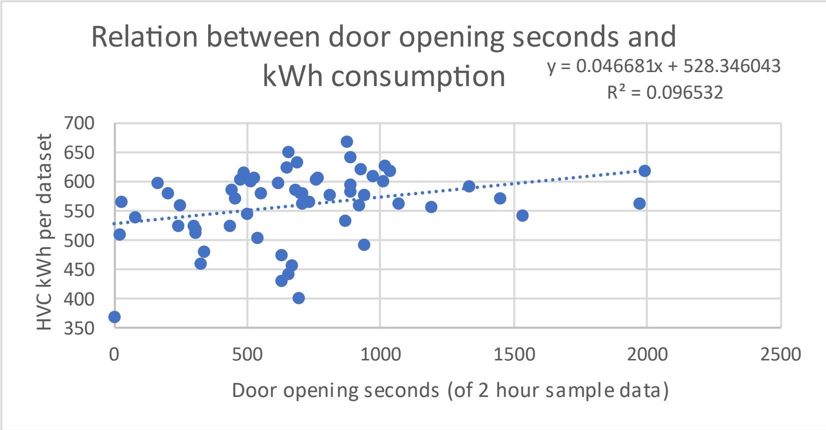


Figure 53 - Relation between door opening seconds and kWh consumption. Temperatures between -5 to 0°C.

Per coach, this extra consumption, due to door opening, can be converted into estimated air infiltration. Taking the average -1° outdoor temperature from Figure 50, assuming air density of 1.25 kg/m3 and Cp of 1003J/(kg·K), the indoor temperature setpoint of 20°C corresponds to 1 080 L/s/coach.

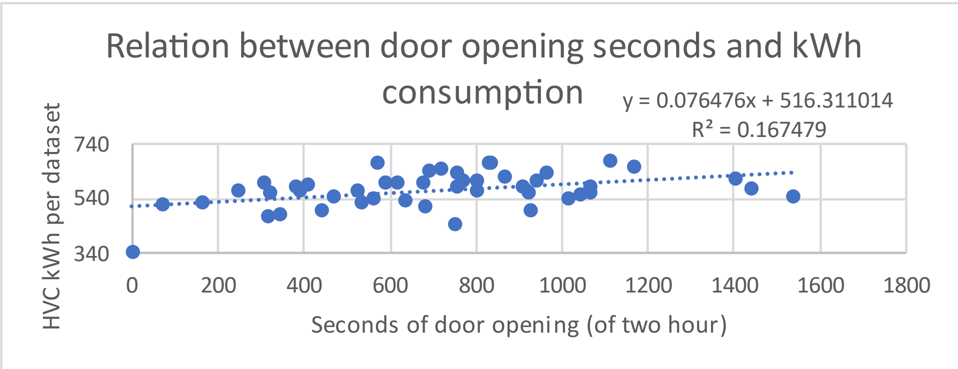


Figure 54 – For passenger load between 1.5-4.5 (on the defined scale). Temperatures between -2 to -9°C.

Factors considered for improving accuracy is the passenger load, which is observed to have a significant effect on the heat balance of the coach. Factors not included are at which stations the doors are opening, which has variations in door opening time, passenger flow, and wind conditions.

Table 21 - Comparison between data-analysis cases of different data for door opening.

Passenger load	1.5-9	1.5-5	1.5-4.5	See Table 11
Outdoor temperatures	-10°C to 7°C	-5°C to 0°C	-9°C to -2°C	°C
Energy loss per second of door opening and train	0.040523	0.04668	0.07648	kWh/second/train
Energy loss per second of door opening per coach (2 doors)	0.006754	0.00778	0.01275	kWh/second/coach
L/s of air exchange	923	1064	1743	L/s /coach

6.3.4 CO_2 and temperature-controlled ventilation

In this measure, everything is kept the same as in baseline except for the ventilation control, which depends on the CO2 concentration and to keep the setpoint temperature same as baseline. Simulated saving of 24 % heating energy. The minimum set ventilation flow is 5.27 L/s/m^2 or 260 L/s and coach compared with 610 L/s for the baseline. In addition, there will be energy saving from reduced fan power of supply and exhaust ventilation fans. However, the relation between fan power and volume flow is not proportional and depends on several factors such as fan efficiency curve. An aspect important to take into consideration with reduced ventilation flow, is the ability to provide good ventilation with high overall ventilation effectiveness, which is the ventilation system ability to provide ventilation to every location within the train.

6.3.5 Recirculation and CO_2+temperature control

Recirculation control, fresh air proportion, 300 PPM – 0.4, 600 PPM – 0.7, 800 PPM – 0.9, 1000 PPM – 1, and linear regression in-between. Combined with the measure of CO_2 and temperature-controlled ventilation, the simulated savings in heating energy is 31 %.

Ideally, the ventilation of fresh air flow rate would be controlled through the number of passengers, with the requirement of 15 m3/passenger/hour [11]. This could be sensed through the bellow pressure and related, as in Figure 13, or through the CO_2 concentration. The supply air to the coach can be controlled through the temperature setpoint, and the recirculation can be controlled through the fresh-air requirement.

6.3.6 Heat recovery

A full-year simulation indicated that a heat recovery of 40 % temperature efficiency could save 34.1 % of the heating energy for the train coaches, with the recirculation device removed and a heat recovery system implemented. Thermal comfort is not affected for the cold season as the heating capacity is still enough without heat recovery.

Table 22 - Savings with heat recovery per coach depending on heat recovery percentage.

Heat recovery percentage	Percentage saving
0% (not including recirc)	Baseline
10%	6.3%
15%	10.9%
25%	20.3%
35%	29.5%
40%	34.1%

There are several potential ways in which heat recovery could be implemented, for example, run-around-coil heat exchangers. With a run-around-coil heat exchanger, it is possible to have the exhaust in a particular position of the supply air but still exchange heat in between. There are several innovations within this topic of heat recovery in HVAC ongoing [50], but however no research papers within retrofit of heat recovery system in trains HVAC could be found.

6.3.7 Setpoint temperature

The temperature sensor of the train is in the ceiling, in the middle door section. In Figure 55, the temperature fluctuations measured with experimental measuring, and the train temperature sensor is shown. NTC is the measurement signal of the experimental temperature measurement.

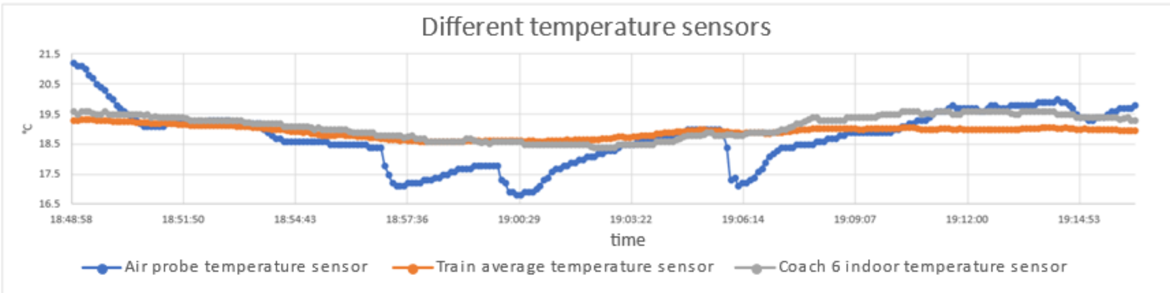


Figure 55 - Comparing temperature sensor of the train with measured temperature.

With a setpoint temperature of 18°C instead of 20°C, the annual heating consumption is reduced by 16.5 %. Hours of dissatisfied decreased by 10 % for the annual simulation (with assumption of 1.1 Clo +-0.7). In Table 23, a summary of the main parameters regarding to the setpoint temperature is shown. As can be seen, the maximum supplied heat is reduced by 35 %.

Table 23 - Comparing thermal comfort in the two different setpoints temperatures 18° and 20°C, at outdoor temperature 0°C. Clothing 1.33. MET 1.5+-0.1. IDA ICE 5 model.

Setpoint temperature (°C)	Max supplied heat W/m2	Max heat removed W/m2	Min rel %	Max rel %	Max CO_2 ppm	Max PPD %	Max age of air (hours)	Occ hours	PDH %
18	42	185	31	48	1090	5.3	0.053	157	8.3
20	65	170	28	43	1140	7.2	0.054	157	10.6

Further in Figure 56, the setpoint temperature of 20°C respectively 18°C is compared in regard to PMV and different MET values. Only small differences in PMV could be observed changing the setpoint temperatures as seen in the Figure 56.

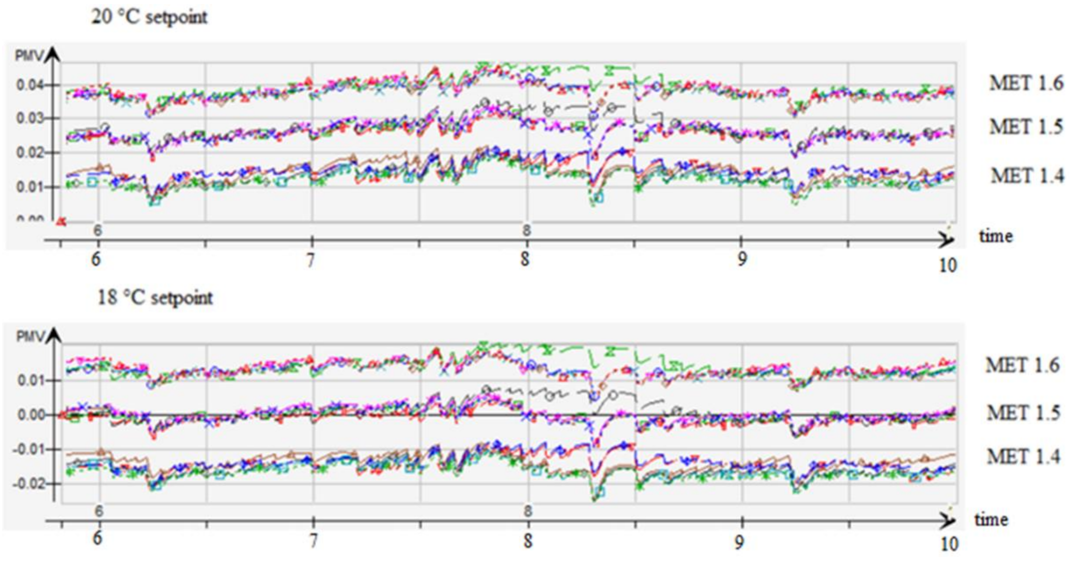


Figure 56 - Comparison of PMV at outdoor temperature 0°C at setpoint temperature 20° respectively 18°C. Clothing 1.33. MET 1.5+-0.1.

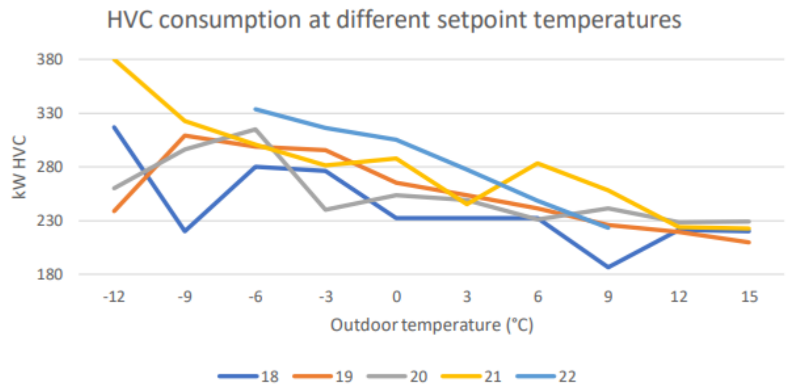


Figure 57 - High voltage meter consumption at different setpoint temperatures, filtered.

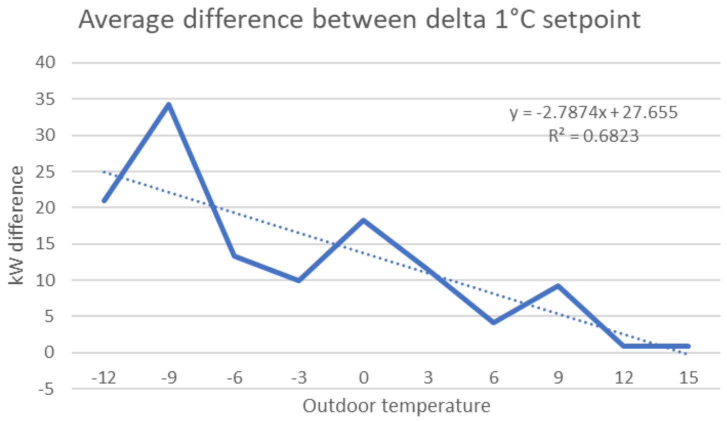


Figure 58 - Between -6 to 9 °C. 1 °C delta setpoint impact on electricity consumption.

The average electricity consumption (kW) different per Δ1°C in setpoint temperature is approximately 11 kW/°C/train at 0°C outdoor temperature. With the input of how many hours each temperature occurs in one year, based on Figure 14, the kWh saving could be calculated derived from the data analysis in Figure 58. The data analysis then led to a saving of 61 000 kWh/year/train per 1° of setpoint lowering. Thus, lowering from 20°C to 18°C provides savings of 120 000 kWh/year/train, equal to approximately 24 %. The results of 16.5% heating energy saving from the simulation are thus lower than the results of the data-analysis. The main reason for that could be that the data-analysis numbers are based on in-train-service hours, and thus the actual heating energy saving can be lower than the results derived from data-analysis. Another aspect why this data-analysis can overestimate the saving, is due to other temperature dependent loads can be included in Figure 58. To have the data analysis more precisely, the heating value would need to be explicitly displayed as a data-channel.

On this day below, in Figure 59, the outdoor temperature is around 4°C and the indoor setpoint is 20 °C. The temperature sensor can be seen to fluctuate between 19.4°C and 20.8°. The actual temperature fluctuations are larger, as this temperature sensor misses the fast variations in temperature, as seen in the experimental measurements. Figure 59 indicates the temperature is between 20.6°C and 19.4°C for extended periods, which is important to consider in thermal comfort.

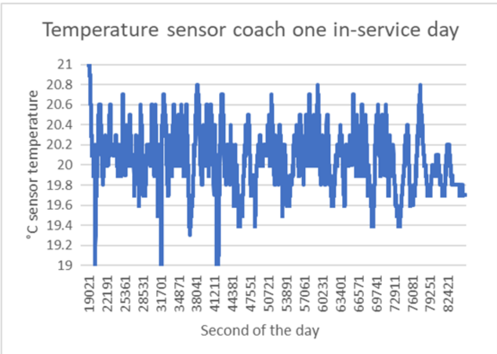


Figure 59 - Temperature sensor of the train coach temperature (train sensor). Fluctuations between setpoint of 20°. 16 January.

6.3.8 Balancing temperature

The definition of the balancing temperature in this case is the outdoor temperature at which an appropriate indoor temperature can be held without the need of heating. Compared with the baseline of setpoint temperature 20°C, the energy decreased with 16.5 % using 18°C setpoint. When the dead-zone temperature for heating and cooling was further implemented, the energy decreased to 71 984 kWh/year/coach. As the dead-zone temperature interval is set to 18-23 °C, it is a more reasonable comparison to compare with a baseline of 18°C. In that sense, the energy saving is 1 312 kWh/coach/year, which is equal to 7 872 kWh/train/year.

In Table 24, the different balance temperatures are tested; between 10-12°C, there are reasonable temperatures at which heating can be turned off. However, further constraints need to be introduced, as if the passenger load is low, the internal gains will decrease, and it may require heating to secure a minimum interior temperature. As can be seen in Table 24, the minimum-air temperature, with the baseline assumption of passenger load, reaches 17.7 °C if the heating is turned off at 12 °C and above with an overall total energy reduction of 0.6 %. For the Table 24, the balancing temperature heating turnoff is only active during in-service hours.

Table 24 - Annual simulation of heating system turned off above pre-defined temperatures.

Temperature at which heating turns off	8°C	10°C	12°C	Baseline
Min temperature (°C)	13.8	15.6	17.7	19.2
Min operative temperature (°C)	15.5	17.4	19.3	19.5
PDH (hours)	8595	8667	8741	8765
Nr of hours between 8°C/10°C/12°C < hr > 20C	3 650	2 900	1 950	-
Energy reduction for 1 train compared to baseline (kWh)	20 478	9 558	3 150	-
Heating energy % reduction	3.9 %	1.8 %	0.6 %	

The duration chart in Figure 14 shows the time duration for certain temperature intervals. For example, in Figure 60, on 22 April, the blue indicates the heating of the interior coach temperature, which would be decreased if larger temperature fluctuations were allowed through turning off the heating above the balancing outdoor temperature.

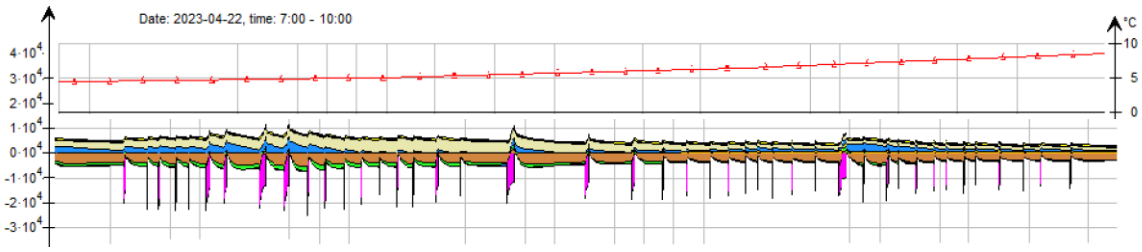


Figure 60 - Fluctuations and balance temperature.

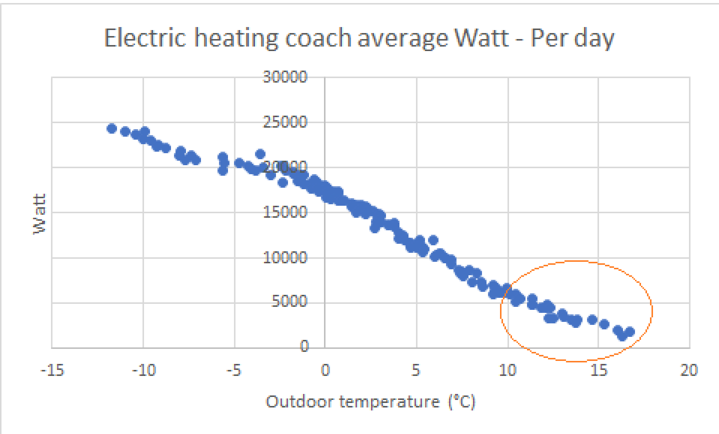


Figure 61 - Potential region for balance point temperature.

In this region which is pointed out in Figure 61, is the balancing point, at which the heating can be turned off only decreasing the indoor temperature slightly. The balance point temperature was found in Table 24 to be between 10-12°C for the baseline case.

However, this is based on the average passenger load derived from the data, which has also increased slightly in peak hours to simulate the peaks in the comparison between measures. Especially before peak hours, the case currently is that the train is heated to have precisely the setpoint temperature. The temperature has exceeded the setpoint due to internal gains from passengers. With this strategy of turning off the heating at a specific outdoor temperature, the temperature will be slightly smaller than the setpoint before the peak, allowing the internal gains to heat the train instead, thus saving energy. Since as shown in the duration chart, Figure 14, the hours between 10-12°C and 20°C are extensive, which means that these operating conditions occur frequently.

However, in the Table 24 above, the balancing temperature was not used during the nights. Balancing temperature throughout one full day could benefit in the summer, as there will be a lower initial temperature in the morning, reducing the need for cooling. This also reduces the heating during summer nights.

6.3.9 Optimum setpoint temperature

Compared to X2000 in Figure 3, the recirculation is not used in the most common outdoor temperatures but only at low and high outdoor temperatures. That is very similar to the recirculation control logic of the X60 train, as in Table 18. It gives a wide interval where no heat recovery of the exhaust air through the recirculation.

Based on Figure 4 (estimated clothing value depending on outdoor temperature) and Figure 7 (comfortable temperature depending on clothing value), an optimum setpoint temperature could be derived. Based on the clothing values projection line in Figure 4, airspeed 0.2 m/s, MET values of 1.5 ±0.1 and relative humidity 30 %, the optimum setpoint temperature could be calculated for PMV-value of 0 (neutral), shown in Figure 62.

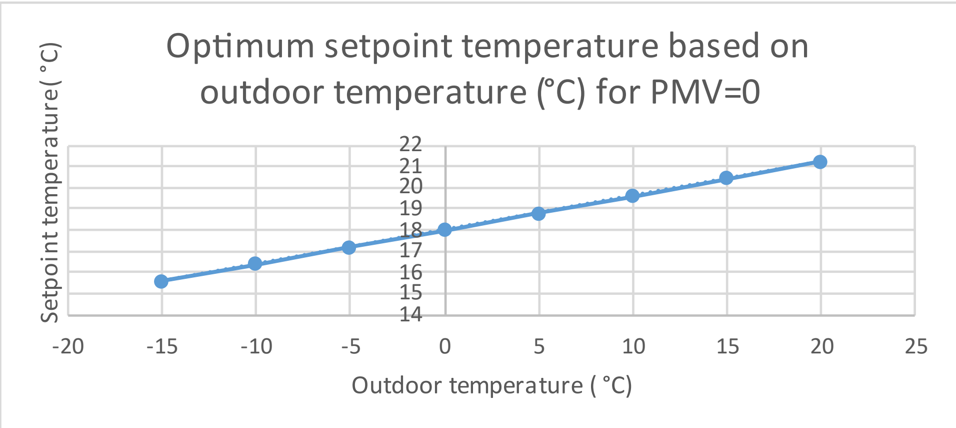


Figure 62 - Optimum setpoint temperature based on outdoor temperature (°C) for PMV=0

The following input temperature setpoint was used:

Table 25 - Setpoints used for optimum setpoint temperature and for baseline comparison.

Outdoor temperature	Baseline setpoint temperature (°C)	Optimum thermal comfort setpoint (°C)
-30	20	16
-15	20	16
0	20	17.5
10	20	19
20	20	20
40	34	34
	With linear regression in-between	

Through the setpoint temperature of optimum thermal comfort in Table 25, an energy saving in heating of 15 % was simulated. With the optimum setpoint temperature according to the PMV and the assumptions made, the results in regards to minimum temperature, max heat supplied and PDH are shown in Table 26.

Table 26 - Thermal comfort evaluation of optimum setpoint temperature. Clothing 1.1+-0.7.

	Min temp (°C)	Min op temp (°C)	Max heat supplied (W/m2)	PDH (hr)
Setpoint curve	15.35	15.72	131.7	8484
Baseline	19.22	19.48	157.3	8767

6.3.10 Energy signature

In Figure 63, the energy signature of the X60 train is shown, based on data-analysis. What can be observed is the heating energy being dominant in electricity power need. In combination with the temperature duration chart in Figure 14, the total energy need can be calculated. In the zoomed in figure on Figure 63, seen in Figure 64, a clear relation of cooling power with the outdoor temperature can be seen. However, in relation to the heating energy mostly seen in Figure 63, the cooling power is much lower. The conclusion based on this, is that heating requires much more electricity power than the cooling, and thus the focus should be on reducing the heating as of the higher potential of electricity power reduction and energy reduction (based on the energy signature and temperature duration chart). In Figure 63, all power consumers of the train are taken into account, except for the electric motor power, however, in Figure 64 the constant power consumption observed has been removed and thus filtered to only cooling or temperature dependent loads.

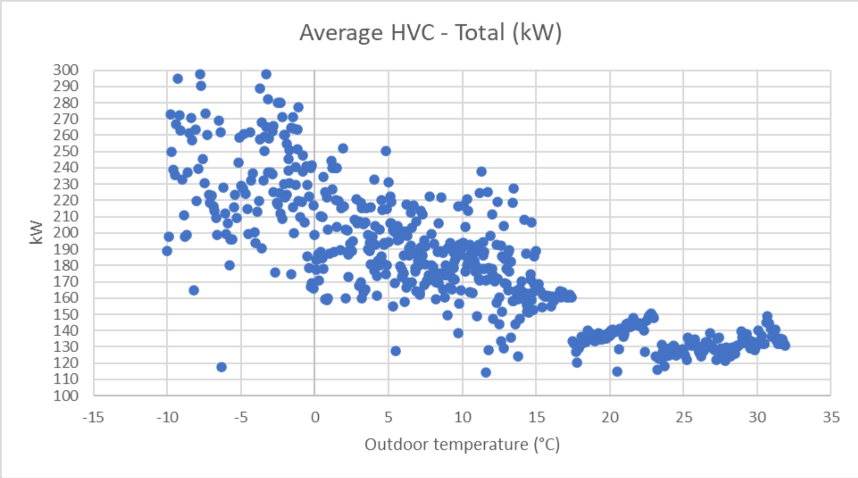


Figure 63 - Energy signature with total HVC - From -10°C to 32 °C

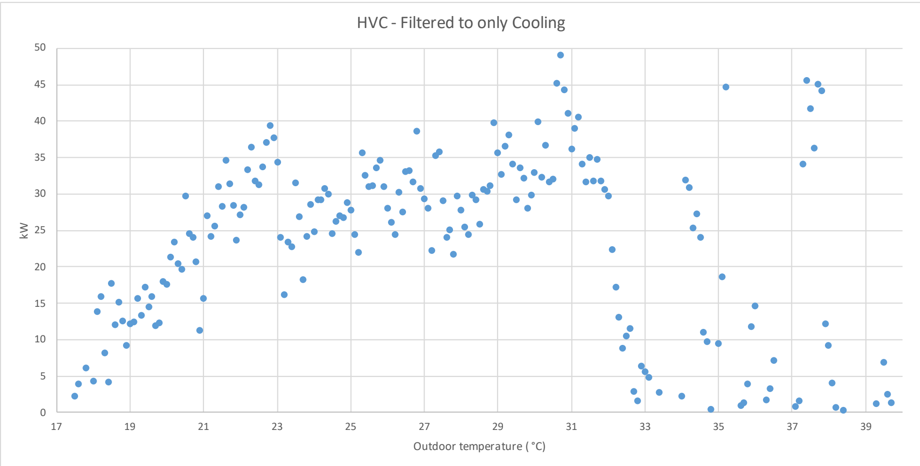


Figure 64 - Energy signature during summer.

6.3.11 Window layer

The benefits of an extra window layer need to be considered depending on the season, in summer it will be the most beneficial if the solar gains can be reduced. However, solar gains could be beneficial in winter, especially since the heating has a COP factor of 1. Therefore, the extra heat might be more valuable compared to the extra cooling needed in summer with a COP factor of 2. In wintertime, though, the solar hours are much less, and the solar position is more from the south, affecting the solar gains to being minimal. Therefore, a seasonal analysis must be done [4]. The window layer measure will be further investigated by a company providing window films.

6.3.12 Combined measures

For combined measures, some of the easy-to-medium measures to implement was considered:

- Setpoint temperature of 18° (and variable in summer as before).
- Ventilation temperature and CO_2 controlled ventilation. Minimum ventilation set to 270 L/s, which is equal to 15 m^3 per hour and person at fully seated. Thus, the ventilation flow will be way above the recommended by the EN-standard (15 m^3

fresh air per hour and person. The maximum ventilation is set to same as to the summer ventilation flowrate (in maximum). Thus, the ventilation flowrate during rush-hours can be even higher than in the baseline. As in the baseline the flowrate is only dependent on outdoor temperature.

- Parking mode of 14°C and recirculation in parking mode of 80 %.
- Heating turnoff above 11.5°C outdoor temperature.

These measures together provide a heating energy saving of 59 % compared to the baseline model. In addition, there are energy savings from the fan-power reduction and reduced cooling need because of lower initial temperature in morning. The additional saving from the fan power can be in magnitude of 12 000 kWh per train and year, as from the simulation results. However, for fan power, the defaults settings in IDA ICE have been used, which may impact the results regarding fan power.

The key thermal comfort aspects for this are:

- Minimum temperature 13.8 °C (parking mode)
- Minimum operative temperature 15.6 °C (parking mode)
- Maximum CO_2 concentration 1545 PPM (compared with 1560 PPM in baseline).
- PDH of 7973 hours (compared to 8768 hours in baseline). However, since the clothing values has been selected to be variable between 1.1 ± 0.1 Clo and adaption to min and max at PMV of -1 respectively 1.

7 SENSITIVITY ANALYSIS

7.1 U-VALUE EXTERIOR WALL AND ROOF/FLOOR

Table 26 compares different U-values regarding the yearly energy demand for heating. The same U-value was considered for the floor and roof and a separate U-value for the exterior walls.

Table 27 - Comparison of U-values - Sensitivity analysis.

Average U-value	U-value wall	U-value floor/roof	Envelope transmission wall (kWh)	Envelope transmission roof (kWh)	Envelope transmission floor (kWh)	Heating demand (kWh)
1.13	0.875	0.78	5 333	4 087	2 916	-
1.24	1.05	0.944	6 432	4 984	3 554	+4.4 %
1.38	1.28	1.156	7 812	6 167	4 396	+8.2 %

For the final simulation version in this thesis, the U-value was slightly less than 1.1, as the thermal bridges were reduced in the simulation.

7.2 THERMAL BRIDGES

Table 28 - Thermal bridges - Sensitivity analysis.

Thermal bridges setting	Thermal bridges kWh/year/coach	% of heating demand
Typical	3 213	4%
Poor	6 794	8%

For the model, thermal bridges setting smaller than typical was chosen.

7.1 CD FACTOR DOOR OPENING

In Table 29 sensitivity analysis of the Cd factor for the door openings are tested in one week of January, between 23rd and 29th of January, with outdoor temperatures of around -10 °C to 0 °C (in day temperature).

Table 29 - Sensitivity analysis of Cd factor for door opening.

Cd	Inflow/outflow from door opening (1 coach) average 23/1-29/1-2023
0.28	700 L/s
0.32	810 L/s
0.34	900 L/s
0.4	1050 L/s
0.6	1500 L/s

8 DISCUSSION

It was discovered to be challenging to gather the right data for analysis since the data channel names and the explanation for them were relatively cryptical. With a lack of information of which power consumption channels go to which systems, it was hard to distinguish the different consumers.

To ensure a successful retrofit, one should consider the new system's reliability under the worst conditions [51]. The trains must be designed following what conditions will be the regular operation of the train [12]. In that sense, it is essential to be able to implement energy measures on the trains, that they are reliable and tested before, especially if changes are needed within the control or physical changes.

The setpoint temperature is static in the EN standard except for the summer. As in Figure 2, the recommended setpoint is 19 °C, down to 18°C within range. These could be improved to fit Swedish climate conditions, ranging from very cold to warm summers and in-between. The deviations in temperature of maximum +-2°C can be hard to maintain since the heating supplied from below is close to the passengers. But with energy efficiency measures, the supply temperature can be decreased as the heating demand decreases, which could lower vertical temperature differences.

It is also important to emphasize the importance of good thermal comfort for the passenger; the valuation for air conditioning is high, as mentioned in the introduction. There is a risk of decreased ridership of the trains due to bad thermal comfort. Thus, the importance of thermal comfort analysis in combination with energy-saving measures. Still, it was shown that most of the energy-saving measures go in line with thermal comfort as well. Heat recovery and reversible air conditioners have a neglectable effect on thermal comfort but substantial energy-saving potential.

As in Appendix I, the seat position temperature can vary broadly. In the lower graph, the supply temperature is 20°C. The temperature variances in vertical level are due to the heated air coming from the sides close to the ankles, which makes the temperature higher in this vertical region because the setpoint is measured in the ceiling in both cases 20°C.

Setpoint temperature and reduced door openings require no system change and is easier to implement, with policy procedures for using setpoint temperature and manual door opening. Reversible air conditioning and heat recovery system of exhaust air, requires physical system changes, and has a cost to implementation in comparison to the other two measures mentioned. To be able to investigate the reversible air conditioning, details need to be gathered about the condenser, evaporator, and compressor. A four-way valve would be needed to switch in-between cooling and heating mode of the heat pump. And as mentioned, to ensure the retrofit is successful, the system needs to have high reliability, which needs to be carefully experimented with and tested before large-scale implementation. With experimental testing, in combination with simulation of a reversible air conditioner, it can be seen how the system behaves in all possible conditions. For heat recovery – different technologies can be compared to investigate the possibility of retrofitting the air handling unit which are installed on the X60-models. From the first look (site-visit), the air handling unit looks spacious with the possibility of adding extra components.

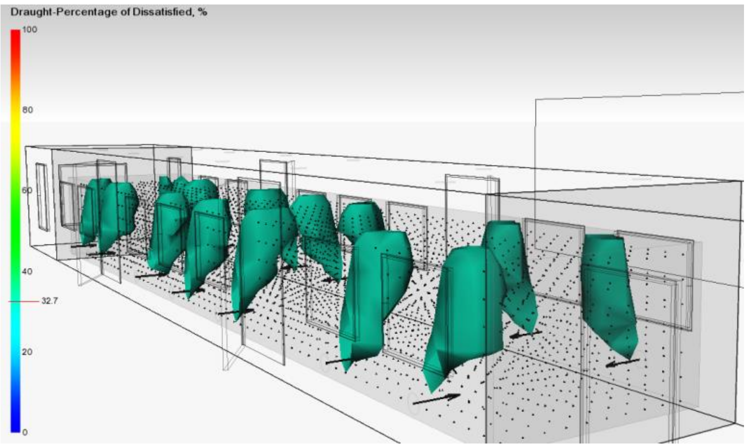


Figure 65 - Figure of draughts (% dissatisfied) - at around 30 % dissatisfied.

As shown in Figure 65, the ventilation outlets have been simplified compared to the real train, which causes airspeed to be wrong at the locations close to the outlets. Thus, looking at draughts in the simulation model is not feasible. Draughts can be occasional in the summers. The dissatisfaction from draughts occurs mainly during the summers, when the cold air comes from the top through the ventilation.

The lighting of the coaches is efficient with light fixtures that distribute the lighting efficiently. However, in the summer with the high amount of sunlight during days, it could be feasible to have a smart control of lighting depending on the ambient lighting condition. The lighting was approximately 700 W per coach, but as explained in the section about lighting, this increases the cooling need equivalently. Especially in the summer, lowering the stress on the cooling system can be an important aspect to consider, especially as that had most complaints of too high interior temperatures during warm days.

In IDA ICE, the model captures the door openings and the transient conditions of the train's thermal environment. However, whether the transient conditions are well-captured in the thermal comfort indicators cannot be verified. A simplification was made with the clothing value adaptations depending on outdoor temperature due to the lack of data in the lower temperature regions. In evaluating the setpoint temperature, the clothing value was set to 1.33 for a day with an outdoor temperature of 0°C, indicating better thermal comfort with a setpoint of 18°C instead of 20°C.

Air infiltration of the door openings is very wind dependent. Also, in the first seconds of door opening, the speed profile of the air infiltration through the door will be different compared to the profile described in Figure 23. For a temperature drop sensor to relate the air infiltration caused by door openings, the sensor must catch the quick temperature changes. In [52], moving the sensor at a speed of 2 m/s is recommended to quickly reach equilibrium with the air temperature, which was not done during the measurements. The sensor was kept in the same place during the measurements, with the probe free of surrounding objects.

It can be argued that not all the thermal-related energy is used efficiently. Significantly since almost none of the heat is recovered through the recirculation (only used in very low or high temperatures), as seen in Table 18. Another aspect that was occurring that the temperature was higher than the setpoint temperature in the seating area, seen in first graph in Appendix I. And through the simulation model showed that a lower setpoint temperature of 18°C would provide better thermal comfort, as seen in Table 23. As mentioned, the heating would not always be required if some temperature fluctuations were allowed, which a balancing temperature strategy can help with. During nights, the energy is not used efficiently since it is common for the systems to keep operating, with a setpoint of 20 °C and ventilation exhaust heat losses. During the night, the value of the energy should be only to maintain proper setback temperature, and from the functionality point of view of going quickly back to service.

Another important aspect that causes uncertainty is the simplification in IDA ICE of measuring the mean volume air temperature. It needs to be further investigated in how it affects the model and what the relation is between sensor temperature and temperature at the seating position vertically.

The measures are divided into different categories depending on the simplicity of implementing them. Some of the hard measures require physical changes of the train components with experimental testing. Medium is more system control strategies in which the logic in control system must be adjusted. Easy – means it is only behavior that must be changed, to control the system in a different way, where it is already possible to do. Parking mode must be adapted or simplified, and is thus categorized into medium, but can also be easy if, for example, setting temperature to 18°C during night instead of 20°C.

The baseline has been adapted without any parking mode and measures. Some of the trains have 18°C, but most of the trains have 20°C setpoint during the winter. To evaluate the total energy saving, the whole fleet must consider how often trains are parked outdoors and in-service hours. To evaluate the savings from reduced door openings, the current state of door openings needs to be quantified of the whole fleet and then preferably be compared to summer (as manual door openings are used in summer). Thus, the savings in energy simulated need to be compared to the actual situation on fleet level. The measures can still be scaled up to the fleet level after considering how many trains are in the baseline situation.

Short-haul and long-haul distance trains are different in how thermal comfort should be considered, affecting the possibilities for energy-saving measures. For example, for CO_2 – concentration, it can be more fluctuating in short-haul distances due to large varieties of passenger loads; meanwhile, for the long-haul, it is common not to be more passengers than fully seated.

As a tool to easier identify and evaluate energy efficiency measures in future, one potential change of the data collection system is to adjust it for energy monitoring. Energy monitoring would be a tool to easily overview the energy consumption of the thermal functions of the train, and what happens with different implementations or conditions. In energy monitoring data software, it would be possible to see the current energy consumption of the different train systems on both fleet and train levels. In that way, it would be easy to find irregularities and take corrective action against overconsumption of energy. A possibility is also to implement monthly benchmarking reports to follow up on energy-saving measures continuously. The live data is already available in R2M data platform, but more detailed data channels would be required to distinguish the consumption of different systems.

The research can significantly impact energy saving in large quantities, reducing the burden of the electricity grid during the coldest days. Furthermore, from a sustainable point of view, reducing electricity usage will decrease CO_2 emissions. The potential for CO_2 equivalents reduction is directly proportional to the electricity usage by 90 grams/kWh of electricity, the CO_2 equivalents stated by Swedish Environmental Protection Agency [53].

9 CONCLUSION

To the mid-life upgrade of the trains, a significant potential can be seen in energy saving through a reversed air conditioner or heat recovery. It needs to be investigated and experimentally tested how the system can be retrofitted with a reversed air-conditioner for pre-heating before the electric heating elements.

Optimal parking mode was found to be around 12-15°C for simplicity of returning to service and for energy saving, which is not dramatically better with lower temperatures. Parking modes have a lot of energy-saving potential without influencing passengers' thermal comfort but only maintain proper setback temperature. The current systems are not used in efficient ways. Recirculation, for example, provides an almost neglectable amount of heat recovery, as it is rarely used.

During rush hours, there is not a lot of potential for energy saving, and also, the air quality is increased through door openings during rush hours regarding CO_2 concentration.

The data analysis – is very sensitive to selected filters. Heating and cooling have been derived from total consumption data channel. For an accurate data analysis, it is essential to have the data channels separated into different electrical consumers. Further, the data analysis methodology for the correlation between door openings and consumption could be observed in the data analysis, but many influencing parameters are hard to filter away.

An X60 Stockholm commuter train model was modeled in IDA ICE, to evaluate different energy-saving measures and thermal comfort. It was concluded that this model is suitable for assessing energy measures with the possibility of making adaptations to the input or adjusting in code. Rapid fluctuations in temperature could be evaluated in regard to operative temperature. The flexibility of the simulation model in IDA ICE makes it applicable for simulating a train and understanding what happens through all the graphs that can be obtained. And through the validation, the model was confirmed with the correct behavior and output results. Vertical temperatures are too uncertain for the simulation, with high dependency on the resolution, and the CFD modeling in IDA ICE 5 has not yet been released. Nevertheless, the behavior of the simulation model seems reasonable, as validated in section 5.7.

The simulation concludes that improving the setpoint temperature can decrease energy demand and increase thermal comfort. Just by clarifying the directions of 18° during winter, thermal comfort and energy will benefit from it. In the simulation model, the PMV level was mostly above 0 (neutral), indicating it is too warm for thermal comfort. This is one of the easier measures to implement since the setting is already available on the train. The potential simulated saving is 16.5 % in heating energy compared to the baseline. To go further with the setpoint temperature, a first proposal of optimum setpoint depending on outdoor temperature was suggested, with potential savings in heating energy of 15 % and increasing thermal comfort. High temperatures above setpoint temperature were observed in the seating area during the cold outdoor and high heat supply temperatures. Therefore, there is much potential in improving thermal comfort, but further verification is required with survey input. Metabolism (MET) values have a large effect on the PPD level, and thus it is very individual of the thermal comfort perception, a compromise must be made. With survey input, more data can be gathered about the thermal comfort perception of short-haul traveling in Sweden, and more accurate predictions of thermal comfort perception can be made.

To evaluate the potential energy saving on fleet level, the baseline has to be adapted for the whole fleet, as different trains are parked indoors or outdoors, and other modes of the trains are used. However, this report quantifies the potential savings on train level and can be scaled up to fleet level. The potential savings align with most of the previous studies within the area, with new input of thermal comfort aspects and for this specific case of the X60 model.

The potential of energy saving compared to baseline are:

- Parking mode:
    - Setback temperature parking mode: to 15 °C 13.2 % of annual heating energy, and for 12°C, the potential simulated saving is 17.2 %.
    - Temperature setback and recirculation: 34 % reduction of heating energy per year.
- Physical changes of AHU:
    - Heat recovery: With 15 % heat recovery, 10.9 % of annual heating energy can be saved.
    - Reversible heat pump: With 11 kW capacity, a heating energy saving of 43 %, 60 % of the heating energy is from electric heating and the rest is covered by the heat pump. Up to 23 kW, the energy coverage by the heat pump increases significantly.
- Door opening reduction. With a 60 % percent reduction of door opening between 09 am and 15 pm, an annual heating energy saving of 8 000 kWh per train was simulated compared to the baseline door opening schedule. With an extension of reduction for door opening, from 9 am to 3 pm respectively, 6 pm to 10 pm (last door opening simulated in the baseline), the savings increased to 11 000 kWh per train of annual heating energy.
- Ventilation control:
    - CO_2 and temperature-controlled ventilation: Simulated saving of 24 % of annual heating energy, with a minimum ventilation flow rate of 260 L/s per coach.
    - Recirculation + CO_2 + temperature-controlled ventilation: Simulated savings in heating energy of 31 %.
- Balancing temperature – Heating shutdown above a certain outdoor temperature: between 3 100 to 12 500 kWh saving per train and year in simulated heating energy reduction.
- Setpoint temperature:
    - Static setpoint temperature: With a setpoint temperature of 18 °C instead of 20°C, an energy saving in heating of 16.5 % was simulated.

    - With a variable setpoint temperature depending on outdoor temperature during winter: a heating energy saving of 15 % was simulated.
- Combined measures: Setpoint 18°C, temperature+CO_2 controlled ventilation, parking mode 14°C, balancing temperature 11.5°C. The total saving compared to baseline combining these mentioned measures, can provide a heating energy saving of 59 %. In addition, the reduction of fan power can be in the order of magnitude of 12 000 kWh per train and year.

9.1 FUTURE WORK

Adapt thermal comfort models for the Swedish climate for short-haul traveling, for example, considering cold winters, in-between temperatures in spring/autumn, and warm temperatures during the summer, preferably through a survey approach. The best solution to get the passenger's perception of the thermal comfort aspects of the train is to do a survey analysis in combination with real-time measurements.

Long-time measurements on the trains for higher accuracy to see a variance with different parameters. Temperature sensors at different placements on the train and in the vertical passenger positions. In addition to temperature sensors, long-term CO_2 levels can be measured to validate the simulation model and to verify the air quality regarding changing the ventilation strategy.

Further adaptations of the IDA ICE model can be done, one example is clothing values depending on the outdoor temperature. As for short-haul trains, it was found in the literature study that most people keep their clothing the same as outdoors; however, for long-haul distance trains, the situation is different.

To make the simulation more accurate, further details are needed in U-values of the train envelope which was not obtained in this project.

To improve energy follow-up in general, monitoring of the HVAC system is needed in deeper detail as compared to now for more detailed data channels. Further validation of the simulation model using more detailed data channels in data analysis of the operating trains. Then further validations and calibration of the simulation model can be done to make the model more accurate in reflecting reality.

Further savings evaluations through manual door openings can be done on a fleet level, considering how many doors would not be opening in manual door opening mode.

For the measures with the category of hard to implement, pre-studies can be done to see if retrofit is possible, such as some sort of heat recovery of exhaust air, or reversible heat pump. Common with both is that experimental studies must be done, and the importance of validating the reliability of the system.

Further analysis can be done in IDA ICE, varying different variables, and comparing the outcome in terms of energy and thermal comfort. The different parameters can be varied and tested for different seasons and for different conditions such as passenger load.

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Appendix

Appendix A Thermal images

Figure 66 - Outdoor upper part of door surfaces temperatures (around -1°C outdoor air temperature)

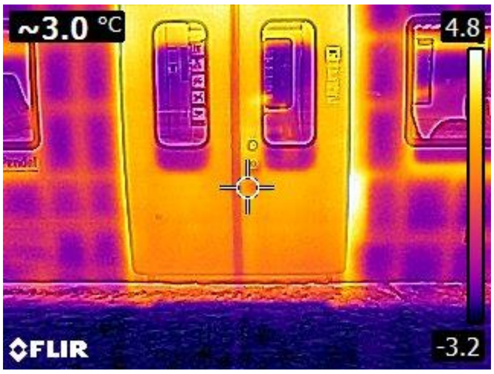
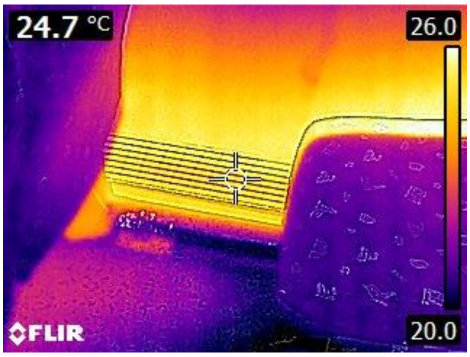


Figure 67 – Outdoor lower part of door surfaces temperatures (around -1 °C outside air temperature)



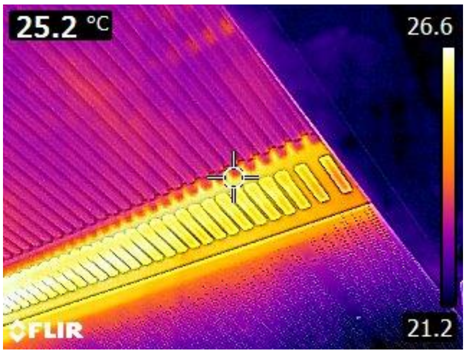


Figure 68 - Ventilation supply outlet in heating mode and lighting temperature.

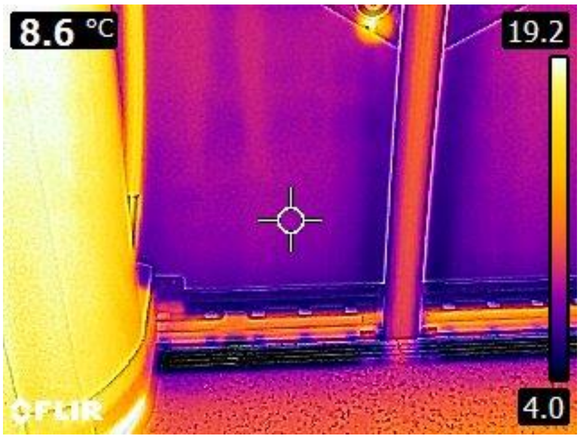


Figure 69 - Indoor side of doors surface temperatures (around -1°C outdoor temperature)

Additional thermal images are available and have been used for validation but has not been attached in this report.

Appendix B Measurement tools

Figure 70 - The measurements tools used.

Appendix C Experimental measurement

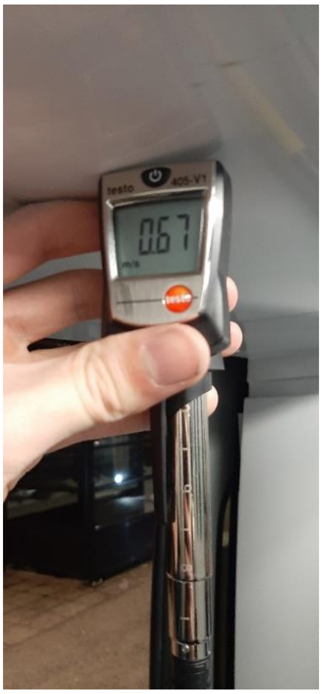


Figure 71 - Measurement at upper vertical section of the train door. Outdoor temperature around 0°C.

Appendix D Door opening figures

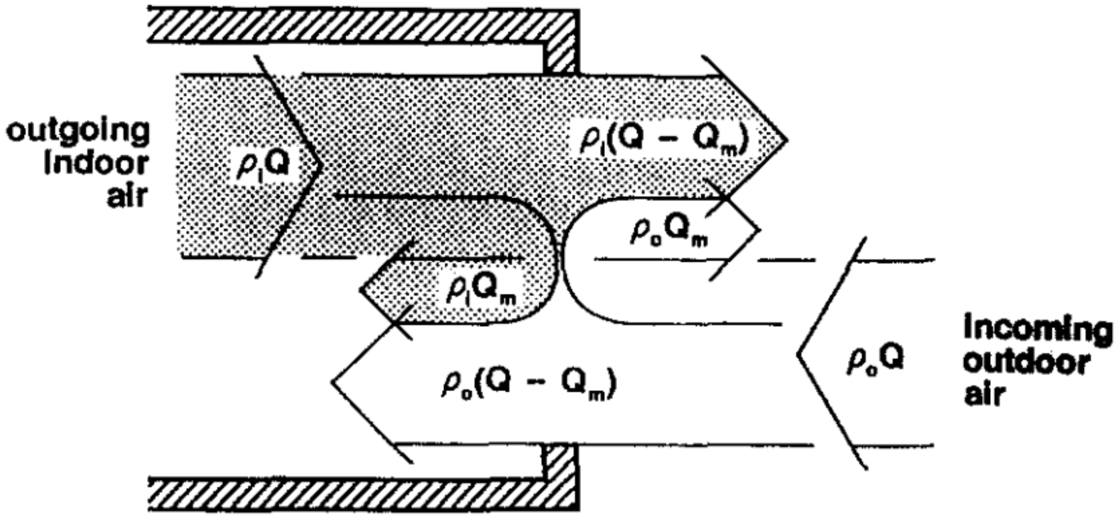


Figure 72 - Mixing factor explaination. Large door opening airflow modelling. [54]

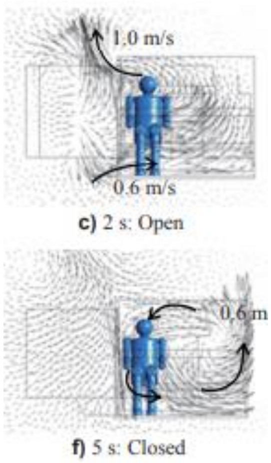


Figure 73 - Air speed after door opening and after door closed [46].

Appendix E Proposed survey questions

As suggested in future work, a survey approach is relevant to adapt thermal comfort models of how the people in Sweden percept the thermal comfort on trains.

What parameters should be recorded along with the survey:

- CO_2-concentration
- Relative humidity
- Operative temperature (globe measurement)
- Train temperature setpoint.
- Air temperature at standing position (1.5 meters) and sitting position (0.6 meters).
- Outdoor temperature.

What questions can be appropriate to include in such a survey depends on the season and type of train (short-haul or long-haul), but for winter and short-haul trains, the following questions can be relevant: Preferably, the scale of the answers should be generalized so it can easily be analyzed. Preferably the type of answer should be generalized on a scale between a specific range or in a type that can easily be analyzed for large data quantities, except for individual input texts, which may be required for some of the questions.

1. Thermal sensation just when they entered the train.
2. Date, time, and train number (to compare with weather data and sensor data in the train).
3. If they were waiting at the station (outdoors), or if they just arrived to the station (to understand if they are warmer due to walking/running to the train or have been standing outside in cold weather).
4. If they are standing or sitting.
5. If the air quality feels good.
6. How long time they have been on the train.
7. How they percept the door openings in terms of thermal comfort.
8. If they feel the current temperature on the train is suitable for thermal comfort.
9. If they feel a difference in temperature in floor and seating/standing area vertical area.
10. Preference: if they want to keep the same temperature or have slightly warmer or colder temperatures.
11. How they are affected by the solar from the windows.
12. If they generally feel warm or cold.
13. Description of clothing (to get clo-value).
14. Extra note: if any complaints in the thermal aspects.
15. Suggestions for better thermal comfort.
16. Age.

Appendix F Thermal time constant of train calculation

Euler's method was applied to differential equations. In the heat balance equation of the coach, all the systems were turned off, and a day with almost no solar gain was selected. Thus, those terms in the equations could be neglected. The only remaining term is the heat transfer through the envelope and first and second vehicle thermal system. First and second vehicle thermal system are described in [7], page 47. The following input was used as from [7].

$$\sum \dot{Q}_{uc} = \dot{Q}_{She} + \cancel{\dot{Q}_V} + \cancel{\dot{Q}_{Sun}} + \cancel{\dot{Q}_{Win}} + \cancel{\dot{Q}_{Aux}} + \cancel{\dot{Q}_{Pass}} + \dot{Q}_{S2}$$

$$C_i \frac{dT_i}{dt} = \sum \dot{Q}_{uc} + \cancel{\dot{Q}_{HVAC}}$$

$$C_{veh} \frac{dT_{veh}}{dt} = \dot{Q}_{S2}$$

$$\dot{Q}_{She} = k_{She} \cdot A_{She} \cdot (T_u - T_i)$$

$$\dot{Q}_{S2} = K_{S2} \cdot (T_{S2} - T_i)$$

Paramter .... *Notation*	**X55**	**X61**	**Unit**
Shell heat transfer coef. .... k_{She}	0.98	1.1	W/m²K
Second system heat tranfer coef. K_{S2}	$1.73 \cdot 10^4$	$5.17 \cdot 10^3$	W/K
Interior heat capacity .... C_i	$4.12 \cdot 10^7$	$2.88 \cdot 10^7$	J/K
Second system heat capacity ... C_{veh}	$3.24 \cdot 10^7$	$2.24 \cdot 10^7$	J/K
Outer shell area .... A_{She}	1100	710	m²

Figure 74 - Input and equations from [7].

Where t_1 and t_2 are the temperatures of first and second system. And t_1 is mean temperature inside the train. With initial conditions assumed to be $t_1 = t_2 = 20°C$, and outdoor temperature constant 0°.

$$T_1(t + \Delta t) = t_1(t) + t_1'(t) \cdot \Delta t \tag{1.41}$$

$$T_2(t + \Delta t) = t_2(t) + t_2'(t) \cdot \Delta t \tag{1.42}$$

, where t is time.

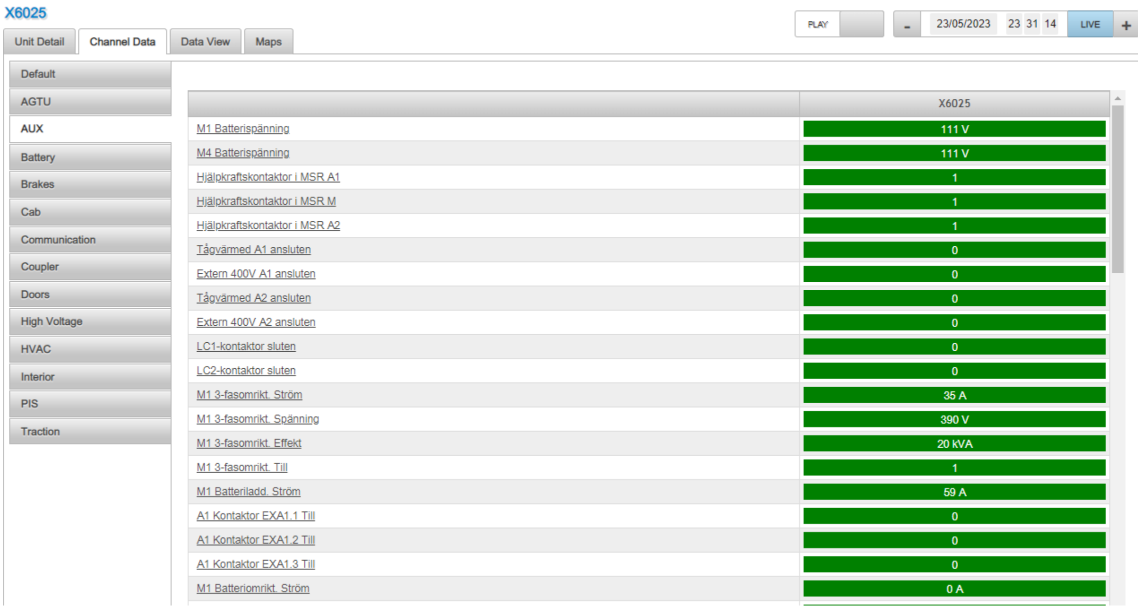
After each timestep the new derivative was calculated accordingly:

$$T_1' = \frac{h_1 \cdot A \cdot (0 - T_1) + h_2 \cdot (T_2 - T_1)}{c_1} \tag{1.43}$$

$$T_2' = \frac{h_2 \cdot (T_1 - T_2)}{c_2} \tag{1.44}$$

With timesteps of 1 minute (60 seconds) in the Euler's method, the Figure 29 could be obtained, which is compared to the IDA ICE simulation model.

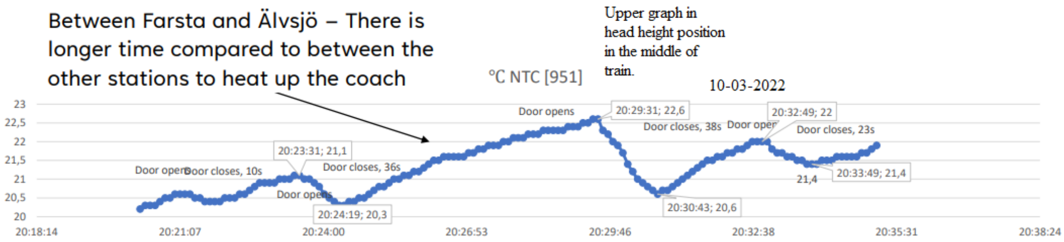
Appendix G R2M Data Platform

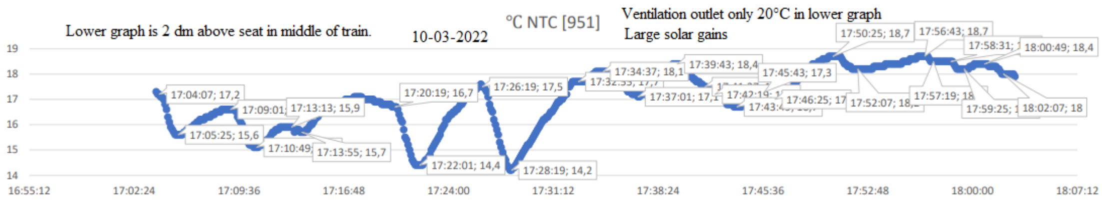


Appendix H Optimum temperature calculation

ϕ RH (%)	30	MET	Tair will be optimum setpoint temperature to get PMV=0				
v (m/s)	0.2	1.5	data goalseek setpoint temp Tair to get PMV of 0				
outdoor temp give a certain clo temp (from Clothing excel)					data - whatIfAnalysis - goalseek		
Outdoor temp	Clo	M (W/m^2)	T_air (°C)	T_rad (°C)	clo	PMV	PPD
-15	1.661415	87.3	15.59571	15.59571	1.661415	0.00066	5.000009017
-10	1.54939	87.3	16.3868	16.3868	1.54939	0.000275	5.000001564
-5	1.437365	87.3	17.18039	17.18039	1.437365	0.000346	5.000002483
0	1.32534	87.3	17.97596	17.97596	1.32534	-0.00097	5.000019663
5	1.213315	87.3	18.78914	18.78914	1.213315	0.000225	5.000001051
10	1.10129	87.3	19.59623	19.59623	1.10129	-0.0004	5.000003361
15	0.989265	87.3	20.41764	20.41764	0.989265	2.83E-05	5.000000017
20	0.87724	87.3	21.24182	21.24182	0.87724	-5.3E-05	5.000000059
		PMV calculated with Visual basic code					

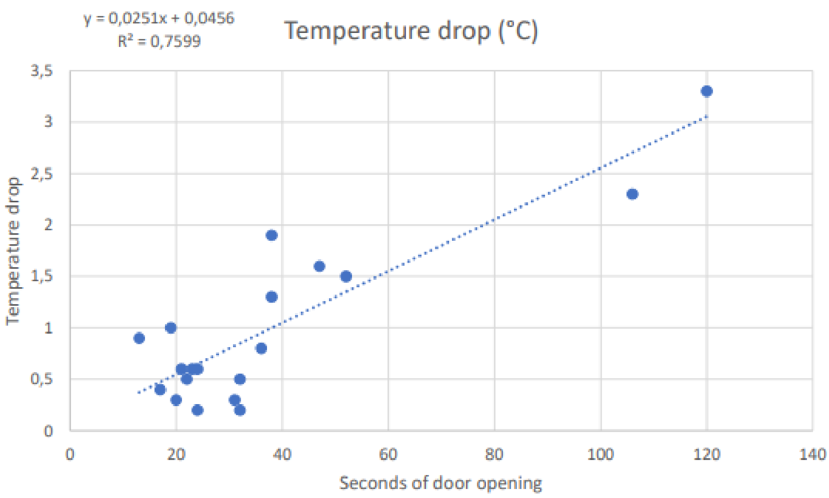
Appendix I Experimental measurements





In the lower graph, the supply temperature of the ventilation air was only 20°C. In the upper graph, the supply temperature was above 30°C. These are experimental measurements from 10-03-2022.

Based on several measurements, and analysis of the measurements, the following correlation could be obtained with door opening seconds and temperature drop catched by the experimental temperature measurement.



Additional experimental measurements and analysis are available and have been used for validation but have not been attached in this report.

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