ABSTRACT

SANKARAN, SRIRAM. Retrofitting an Effective Air Conditioning System for a Hybrid Vehicle Conversion. (Under the direction of Dr. Eric Klang).

In times when customers look for safe, comfortable, fuel efficient and eco-friendly automobiles, hybrid vehicles are one of the definite way forwards for the automobile industry. Hybrid vehicles have risen in popularity over the last 2 decades. But, the increase in popularity has not mirrored into a huge demand, primarily, due to the high costs associated with hybrid vehicles. Consequently, the interest in converting existing IC engine powered vehicles into hybrid vehicles has grown considerably. There are number of companies specializing in this field offering their services to interested parties. This is the motif of "EcoCAR", an advanced vehicle technology engineering competition series, where collegiate teams are challenged to convert conventional IC engine models into vehicles run on alternate vehicle technology. Teams are expected to model and build such a vehicle while maintaining market acceptable performance, comfort and safety standards. One of the automotive systems responsible for safety and comfort is the air conditioning system. It is the single largest parasite of the vehicle power source. This results in an adverse impact on the vehicle performance by reducing fuel economy, acceleration and increasing tail pipe emissions. Research has shown that this impact is higher on high fuel economy vehicles like hybrid vehicles. Hence, it could be stated that AC systems adversely impact the very advantages which bolster the cause of hybrid vehicles. "*EcoCAR2: Plugging In to the Future*", the second edition of the series, requires a fully functioning air conditioning system that can cool down the vehicle cabin below the ambient temperature by at least 5.5°C in 2-5 minutes under specific conditions. This research deals with creating a step-by-step procedure to model an effective air conditioning system for a vehicle that has been converted into a hybrid. It was modeled for EcoCAR2 developed by a team of NCSU students. The step by step procedure involved a heat load analysis, a performance analysis, heat load reduction and finally, selection of components. A peak vehicle cooling/heat load of 2175 W was calculated that resulted in an accessory electrical load of 3590 W. The performance of the vehicle running on a UDDS drive cycle with the above electrical load was performed to enumerate its impact. 3500 W of electrical load resulted in a 12.1 mpg drop in the fuel economy. The cooling load was finalized after considering various methods to reduce the vehicle heat load. With the finalized cooling load, a compressor of 18 cc/rev was selected along with a condenser of 0.20 m^2 and an evaporator of 0.054 m^2.

Retrofitting an Effective Air Conditioning System for a Hybrid Vehicle Conversion

by
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DEDICATION

To அம்மா. (My Mom)

BIOGRAPHY

Sriram Sankaran was born in Chennai, a south-eastern city in the state of Tamil Nadu, India to Sankaran Subramanian and Srimathi Sankaran. He has an older sibling, Kousalya Sankaran. His father, an electrical engineer, was responsible for his curiosity and interest in understanding objects, motion of objects, etc from an early age. Keen interest in Physics and Mathematics coupled with a passion for automobiles led to a Bachelors degree in Mechanical Engineering at Rajalakshmi engineering college, an Anna University affiliated institution. SAE MINI Baja, India played an important role in his decision to pursue a Masters degree in Mechanical Engineering at North Carolina State University. After graduation, he will be taking up an offer from Cummins Inc., Rocky Mount to begin his career as a ‘Rapid problem solving’ team project lead.

ACKNOWLEDGMENTS

Given an opportunity, I would rename this page “gratitude”. More than acknowledging people, I would like to express my heartfelt gratitude to a number of people who have helped me in getting to where I am today.

My parents have always had and will always have a positive influence on me. My mother wanted me to have the best and highest level of education, right from childhood. My father, my greatest source of inspiration, has stood by me during my mistakes (quite a few big ones) and made sure I got back into the right path. My sister was my earliest challenger and has always believed in me, more than I do. These people are the foundation on which I have built my life and I cannot repay them enough in this lifetime.

I have always been blessed with the best of friends and they have been instrumental in building my personality and confidence. Right from Arun Prasadh who kindled my passion for automobiles to Rajmohan Selvaraj who stood forth for my place in SAE MINI BAJA, I would like to thank each and every single friend who has supported me through thick and thin. It would take at least 2 pages to mention every single one of them. But, they would personally know who all I am referring to (Outlaws & Lalli). As always with any international student, it would have been tough without my friends in Raleigh who made it a home away from home.

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Last, but definitely the biggest ones, there are 2 important people without whom none of this would have been possible. One is my advisor, Dr. Eric Klang. His support and his encouragement during the bad days are definitely, unparalleled. His effort in helping me build a thesis, in a field which is not is part of his expertise, deserves more than a note of gratitude. I feel blessed to have worked under such a wonderful person. The other person is my ‘better half-to-be’, Vidya Seshadri. From my high school days to this moment, she is responsible for my every single success.

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LIST OF SYMBOLS & ABBREVIATIONS

1. CO_2: Carbon dioxide
2. VOC: Volatile organic compounds
3. AC: Air conditioning
4. IC: Internal Combustion
5. TR: Ton of Refrigeration
6. USEPA: United States Environmental Protection Agency
7. ASHRAE: American Society of Heating, Refrigerating and Air Conditioning Engineers
8. CFM – Cubic feet per minute
9. V: volume, m^3
10. HV: High voltage
11. ESS: Energy Storage system
12. ADVISOR: Advanced vehicle simulator
13: NREL: National renewable energy laboratory
14. m: meters
15. CLTD: Cooling load temperature difference
16. SCL: Solar cooling load
17: CLF: Cooling load factor
18. Q: Heat gain, W
19. A: Area, m^2
20. U: Overall heat transfer coefficient, W/m^2. K
21: T_{cab}: Vehicle cabin temperature, °C
22: Tamb: Ambient temperature, °C
23: $\dot{m}$: Mass flow rate, kg/s
24. CFH: Cubic feet per hour
25. α: absorptivity
26. τ: transmissivity
27. I: specific enthalpy, J/kg.K
28. γ: angle of incidence, degrees
29. ρ: reflectance
30. $\sum$: Surface tilt angle to horizontal, degrees
31. HEV: Hybrid electric vehicle
32. rpm: rotation per minute
33. mpg: miles per gallon
34. kpl: kilometers per liter
35. bhp: brake horse power

1. INTRODUCTION

Automobile vehicle technology has been steadily moving towards alternate vehicle technologies due to a variety of reasons ranging from rapid depletion of non-renewable resources to global warming issues. The advantages are aplenty and the passenger car manufacturers have understood that hybrid vehicles are here to stay. There has been a steady increase in the number of hybrid or fully electric passenger car models in the last decade. Many conventional IC engine models have hybrid variants as well. Still hybrid vehicle technology is primarily viewed as a vehicle propulsion technology. The vast reserve of hybrid/electrical technology could also be tapped to energize vehicle accessory systems instead of relying on the usual 12V auxiliary battery and/or the power derived from engine. The air conditioning system is one such system and is also the accessory which extracts the largest quantity of power from the engine. In the following sections, the motivation behind the entire thesis is explained by reasoning out the needs of an effective automobile air conditioning system.

1.1. Importance of automobile AC systems and its adverse impact on vehicle performance

An automobile air conditioning system is an integral member of automotive systems because it is not only responsible for the thermal comfort of a passenger but also the safety of the passenger, to an extent. The primary purpose of an automobile air conditioning system is to maintain the vehicle cabin temperature and humidity at comfortable levels for a passenger.

It is also responsible for recirculation of air inside the cabin and preventing stagnation of stale air which will consist of CO_2 from the passengers, VOCs and other particulate contaminants.

Modeling of automobile air conditioning system varies from an ordinary AC system of buildings due to various reasons. Dynamicity of the ambient conditions, more occupancy per unit area, more glass windows per unit volume are few of the reasons which complicate of the modeling of automobile air conditioning systems. Once the initial modeling issues of adapting HVAC design of buildings to automobile AC systems is circumvented, focus shifts to finalizing the optimum amount of air conditioning required to provide cooling. Passenger vehicles manufacturers are, generally, accused to have over sized AC systems. Conventional automobile AC systems draw their power from the engine in the form of belt driven compressors. Hence, oversized systems tend to draw more power from the engine resulting in lesser fuel economy and also higher emissions. Research suggests that for vehicles with higher fuel economy, air conditioning systems at peak load will reduce fuel economy by 50% and increase emissions considerably (No_x by nearly 80% and CO by 70%). Farrington and Ruth record in their paper that the power to drive the compressor could be as high as the power to drive the vehicle at a constant speed of 35mph[1]. They also observe that a vehicle might use 612 gallons of gasoline on an average per year. More gasoline means more emissions. Vehicle owners are spending a lot more on fuel and spare parts unknowingly and governments are fighting adverse environmental after effects of automobile air conditioning. This could be ‘nipped in the bud’ with effective modeling. Further evidence of adverse impact will be discussed in the “Literature review” section.

1.1.1. Significance of effective modeling of automobile AC systems

Modeling of automobile AC system is one section which has received the least attention from automobile manufacturers. There have been minimal changes to the fundamental design of AC systems over the past 5 decades. The term “effective” is included in the title because the objective is to shed light on the areas that need improvement in automobile AC modeling. There are 2 important sections which suffer due to ill advised modeling. One is, as explained in the previous section, vehicle performance and emissions. Other is the comfort of the passengers.

Though AC systems affect fuel economy and emissions, vehicle performance testing is usually done without the AC system. In 1995, a technical report of the U.S.E.P.A detailed the needs to include an emissions test which will test the vehicle emissions with the AC system running [2]. This resulted in the introduction of Supplemental Federal Test Procedure, SFTP. SFTP had a phased introduction starting from 2001 with 25% of cars being subjected to SFTP in the starting year. By 2005, all cars were subjected to SFTP [1]. This is a significant step in AC modeling improvement because it will prevent over sizing AC systems. At the same time, it will lead to improved research on heat load calculations for automobiles which determine the size of an AC system. This will, in turn, result in finding ways to reduce heat load of a vehicle.

Improper modeling compromises the thermal comfort and safety of a passenger in a number of ways. Oversized AC systems tend to provide uncomfortable amount of cooling and might also result in a drier cab condition than necessary. Undersized AC systems results in the obvious opposite effects and can also result in the system being run for longer than necessary duty cycle. Also, oversized systems will cause condensation and fogging of glasses. Fogging is a safety issue. Even blowers used in AC systems need to be properly sized because very high velocity of air through the vents tends to cause uneasiness [3]. Thus, sufficient evidence has been provided to support the need for effective modeling.

1.1.2. Introduction to EcoCAR

EcoCAR, as explained in its website, is an advanced vehicle technology engineering competition established by the United States Department of Energy and General Motors. The first edition was conducted by Argonne National Laboratory. The second edition of the series, "*EcoCAR2: Plugging In to the Future*", is the competition concerned with this thesis. Students of 16 North American universities have to design and incorporate "advance vehicle technology solutions" into GM donated vehicles. The solution should be able to reduce the fuel emissions, improve fuel economy, vehicle energy efficiency while maintaining acceptable standards of performance, utility and safety on the basis of consumer experience. A properly working AC system is also part of the requirement and carries 25 points. 25 points will be awarded if the AC system is able to cool down the cabin air temperature by at least a temperature of <5.5°C (10°F) under ambient conditions within 2 to 5 minutes.
The advanced vehicle technology adopted by NC State University EcoCAR team is series powertrain architecture with a smaller engine in place of the engine in the GM donated car. The car donated for EcoCAR2 is a stock Chevrolet Malibu. The competition, at this stage in 2012, is about to have the "Year One" competition. This end of the first year competition is conducted to evaluate the team's design and analysis. The team does not get the vehicle till early parts of the second year. Hence, any design related calculation and analysis done up to

this stage and in this thesis, are all based on CAD files provided by GM and various other sources.

Figure 1. 1. GM donated Chevrolet Malibu

1.1.3. Introduction to High Voltage Automotive AC systems

The quintessential hybrid passenger vehicle, Toyota Prius, is an excellent advocate to the success of HV-AC systems. As explained earlier, conventional automobile AC compressors are belt driven off the engine. The compressor creates very low pressure in its inlet and sucks in the saturated vapor from the evaporator. The vapor is compressed into a high pressure superheated vapor which will drop the heat gained across the condenser. Saturated liquid flows out of the condenser and into the expansion valve/orifice tube where it expands and further cools down before picking up heat across the evaporator.

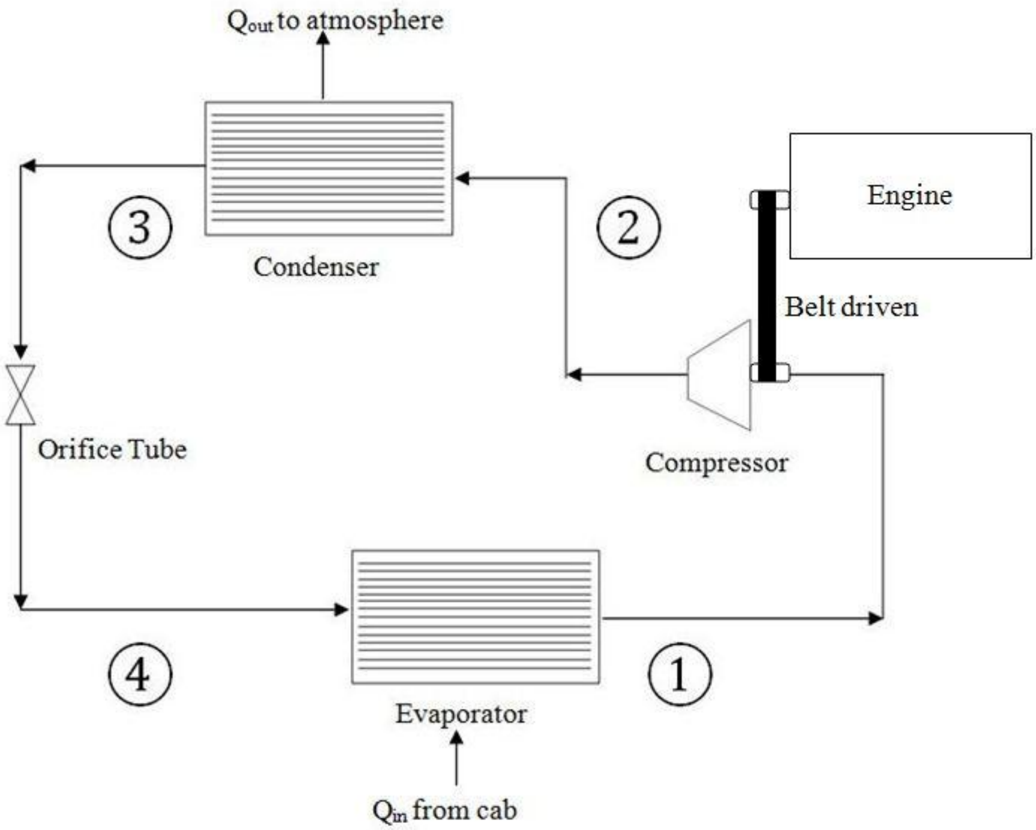

Figure 1. 2. Belt driven Automotive AC system

HV-AC system is similar except for the fact that, the compressor is driven by a motor powered by the energy storage system (ESS) on board. These are usually hermetic compressors. The motor of the compressor is driven using the AC power supplied through a 3 phase AC inverter. Caution is required while working with these HV systems as they could prove to be fatal. There are numerous advantages to such a system which will be explained thoroughly in upcoming chapters.

1.1.4. Market demand on hybrid conversions

With increasing fuel prices and high level of consumer awareness on emission issues, the interest in hybrid vehicles has risen steadily in the past decade. As mentioned earlier, high profile and established automobile manufacturers like GM, Toyota, Honda, Ford, BMW, etc. have been developing hybrid models or creating hybrid variants from conventional models. The sports utility vehicle from Ford, *Escape* is an example of hybrid variants of existing IC engine models. Hybrid models have gradually increased to a market share of around 3% of all vehicles sold in the later part of last decade. Though the increase in number of models has provided the consumers with a wide variety of options, there has never been a dramatic increase in the demand. Experts feel that the main reason is cost. When buyers can opt for an IC engine which offers more value for money at the same cost, the competition becomes one-sided [4].

An indirect consequence of this issue is people trying to convert their old vehicles into hybrids. A testimony to this fact could be increasing number of online forums where people discuss and get opinions to convert. A number of companies are trying to use this opportunity to build a stable market out of such conversions. Some of these companies did their first conversion even before Prius was announced. *Evtransportal.org* provides a list of such conversion companies [5]. Most of these companies have fleet vehicle conversions as their major focus, passenger vehicle conversions will be their target. A recent interaction with an executive from one such company confirmed their desire to foray into passenger vehicle segment. AC systems will not need much attention in case of fleet vehicles but it is an important accessory of passenger vehicles. It is significant because AC systems tend to affect the fuel economy more in case of vehicles with more fuel economy. Simulations and tests conducted on hybrid and conventional gasoline vehicles prove the fact [6].

1.2. Statement of Intent

The objective of the previous sections in this chapter is to build a reasonable foundation to explain the intention behind this thesis. The purpose is to model, select components and retrofit an AC system for a hybrid vehicle-convert. This exercise will be attempted on EcoCAR2. The constraints are the team's design targets and competition requirements. The team's design targets are primarily higher fuel economy, lesser emissions, better passenger comfort and vehicle performance. Competition requirement has already been outlined in the "Introduction to EcoCAR" section. The numbers behind the team's design targets will be outlined in the following sections.

1.3. Approach

Any air conditioning system design process starts with the calculation of cooling needs. The cooling load calculation inputs include details about geographical location of vehicle, orientation of vehicle, hour of the day, vehicle interior & exterior specifications, vehicle condition and ambient conditions. Once the required tonnage has been calculated, the performance of the vehicle with AC as the accessory load will be tested using ADVISOR, a vehicle simulation tool developed by NREL. The simulation results will be analyzed to determine the adverse impact on the vehicle performance and determine the accessory load threshold. Once the threshold has been determined, methods to reduce the heat load will be applied, in order to size the AC system appropriately. The reviewed cooling load will be used to determine the compressor mass flow rate, cubic capacity and the rest of the components. The compressor physical size will be chosen to suit the space available under the hood and vehicle weight limitations (Team target). Further recommendations on other components will also be provided.

A term to note in the rest of the text will be ‘*Hybrid-convert*’. *Hybrid-convert* is a term used to define a vehicle which has been converted from an IC engine to a HEV.

2. Literature Review

2.1. Previous literature on automobile AC systems design, modeling and passenger comfort

AC systems modeling for automobiles vary considerably from AC systems modeling for building. There are numerous books and papers available for AC systems modeling. However, automotive AC systems are given less to no attention at all in most of these books. *ASHRAE handbook- HVAC applications, 2007* gives a brief and excellent introduction to Automotive AC systems, important variables to consider while modeling it and its components[3]. For further understanding of automotive air conditioning systems, Boyce H.Dwiggins's *Automotive air conditioning* and Steven Daly's *Automotive air-conditioning and climate control systems* provide ample information on the subject[7][8]. Still, even these books do not provide a detailed understanding about the modeling steps and methods for these systems.

Chapter 9 of *The CRC Handbook of Mechanical Engineering, Second Edition* offers quick and precise information on AC systems, the different types of AC systems, air conditioning processes and cycles, components involved, design criteria, load calculations and various other basics involved with air conditioning[9]. *Alternative Technologies for Automobile Air Conditioning*, a paper by B. Multerer and R. L. Burton is very comprehensive. It explains about AC sizing and heat load reduction. The paper also compares and analyzes in detail, the different types of available automotive air conditioning systems [10].Chung-Lun Li *et al* describe the modeling of the vehicle's AC systems equipped with automatic control systems. They model AC systems in several ways which differ in control strategies and present a virtual model of vehicle compartment and the variations of many state properties including vehicle compartment temperature and humidity. The paper helps us understand the intricacy of automotive AC system modeling and also its relative complexity when compared to an residential AC system modeling [11].

Evaluation of Advanced Automotive Seats to Improve Thermal Comfort and Fuel Economy by Jason Lustbader, explains about the importance of modeling seats to provide better thermal comfort and thereby aid effective AC modeling. In a specially developed laboratory by NREL, tests were conducted on the effect of improved seating. The results proved that a reduction of around 4°C for the contact temperature was possible with an advanced seat and a low mesh mass back seat.[12]. The author uses ADVISOR to enumerate an increase of 4.5% on an EPA city cycle due to these changes. Similarly, the advantages and the impact of effective AC modeling on the thermal comfort of the passengers are elaborated in Julian Weber's *Automotive development processes: processes for successful customer oriented vehicle development* [13]. Eugene Talley's *Hybrid Air Conditioning Systems Overview* deals with hybrid and electric air conditioning systems. The paper provides an overview of AC systems in hybrid models in the market, electric compressors in AC systems and hybrid compressors[14]. A web article by Dave Hobbs, *Mastering Hybrid HVAC Systems, Pt 1 — Toyota Prius*, is also a fantastic source to learn about High voltage AC systems[15].

2.2. Previous literature on automobile AC components

Alternative Technologies for Automobile Air Conditioning, gives a good introduction to hermetically sealed electric compressors and scroll compressors (most common compressor in automotive AC systems nowadays)[10]. Eugene Talley's paper and Dave Hobbs's article, provide good exposure to electrical compressors, compressor manufacturers and their functioning[14] [15].

For information on conventional AC components, *HVAC handbook* by Robert.C.Rosaler and *Heating and cooling of buildings* by J.F.Kreider *et al* discuss in detail about these components, their functioning and design in entirety[16][17].

2.3. Previous Literature on heat load calculations for automobiles

Heat load calculation for buildings is an area which has been thoroughly researched and the information available on this topic is widespread. But heat load for automobiles has not received as much attention as it warrants. Heat load calculation of buildings and automobiles vary in numerous ways. One of the earliest papers concerning heat load calculations of automobiles is *Analysis on Air-Conditioning Heat Load of a Passenger Vehicle.* It is a pioneering paper which covers the topic completely and helps to understand the methodology behind heat load calculation of automobiles[18]. Different kind of contributing factors to heat load and impact of vehicle conditions are explained clearly. Shimizu *et al* also write about the significance of the hour of the day and the corresponding solar radiation which are major factors in determining the peak heat load.

In the efforts to build an automobile passenger comfort model, Ingersoll *et al*, explain the vehicle cab cool down calculation. It is another excellent resource for cooling cold calculation procedure. They built a virtual thermal comfort model which solves to predict the thermal comfort of a passenger considering inputs ranging from clothing of the passenger to the speed of the vehicle[19]. Zheng e*t al* provide a step by step procedure to calculate the vehicle heat load using Microsoft Excel. They use the CLTD/CLF/SCL method to calculate the heat gain through the vehicle body and glazing, heat loss through convection, heat gain through infiltration, utilities and occupants. Once the vehicle heat load was calculated theoretically, vehicle testing was done to compare the values under specific conditions similar to the theoretical calculation [20]. Previously mentioned *Alternative Technologies for Automobile Air Conditioning* also summarizes a brief vehicle cooling load calculation values without explaining the calculation methodology. Malik and Bullard give an account about cooling load contributing factors without providing any calculations. The paper is specifically for hybrid electric vehicles and explains about AC load during intermittent stops in a duty cycle (signal stops) They calculate the vehicle heat load using a technique similar to above mentioned papers and employ ADVISOR to compare heating/cooling options based on energy efficiency. They conclude that an effective AC system on a HEV should consume only about 3-4 gallons/year during traffic stops [21].

Heat load through the glass is a major contributor to the vehicle cooling load due to the high percentage of glass in vehicle cab surface area. Above mentioned papers in this section provide brief information about calculating the glass heat load. For a more focused explanation, *Heating, ventilating, and air conditioning: analysis and design* by Mcquiston, Parker and Spitler is a preferred book among experts. ASHRAE handbook cites the book as a reference. The book explains methods to calculate solar irradiation depending on hour of the day, incidence angle, material, etc. and provides examples to elaborate the same[22]. Another book which offers excellent material on the same subject is *Solar engineering of thermal processes* by Duffie *et al.* It talks in depth about solar radiation through glass and its effects on the heat load [23].

2.4. Previous literature on adverse impact of air conditioning on vehicle performance

NREL has published considerable research papers on this topic. *Impact of Vehicle Air-Conditioning on Fuel Economy, Tailpipe Emissions, and Electric Vehicle Range,* by Farrington and Rugh highlights the problem with the help of simulations run on ADVISOR. The paper elaborates on the different tests run, the drive cycles used and the required vehicle specifications. Using ADVSIOR, they compared a conventional IC engine powered vehicle, an electrical vehicle and a hybrid vehicle to prove that the impact of AC systems on fuel economy and emissions is worse. The tests are run on 4 different duty cycles and all four cycle results are compared to form the above conclusion [1].

V.H. Johnson, in her paper, explains the impact of air conditioning on vehicle fuel economy and approaches the problem through thermal comfort. Obtaining TMY data from the NSRDB for a particular region for a specific time and day, she models a passenger thermal comfort which could predict passenger’s comfort and thereby, produce an approximate value for the time for which AC is switched on. This time is used to predict the amount of fuel being used by the air conditioning [24].

2.5. Previous literature on reduction of vehicle cooling load

Farrington and Rugh outline the methods briefly in their paper and explain opportunities to reduce vehicle cooling load. They tested the impact of an advanced UV and IR reflectance glass and advantages of using recirculated air to condition the vehicle compartment. Results of these enumerate the reduction possible in the heat load. *An Overview of Vehicle Test and Analysis from NREL's A/C Fuel Use Reduction Research,* a report presented at VTMS-8 provides a comprehensive view of the subject. NREL researchers conducted tests by changing the insulation, ventilating a parked vehicle, using solar reflective glazing and providing instrument panel cooling to reduce the vehicle heat load. The results of their test prove that a considerable reduction is possible [25].

2.6. Conclusion

Automotive air conditioning has been around since the late 1930's and has not changed a lot in terms of operations and components. ASHRAE handbook and other HVAC handbooks provide the required amount of data to understand the various types of AC systems and components. As mentioned earlier, it was not until recent, that the adverse impacts of AC systems on performance have received the required amount of attention. NREL has been a leader in this field and it is evident from the number of papers produced by researchers associated with NREL.

The areas, pertaining to this thesis, that require more literature or focused research work, are methods to reduce vehicle cooling load, modeling an automotive AC system and automotive heat load analysis. The 3 papers mentioned in heat load analysis literature review section are informative but this is a significant area which requires much more attention in order to improve AC modeling and performance. Papers which detail the ill effects of over sizing air conditioning systems for automobiles are required. Instruction or approach to model an automotive air conditioning system is another area which lacks material. Books, journal articles and magazine papers explain design of components and selection criteria. But none of them explain design specifications to cater to a specific cooling load or ways to pick components based on cooling load. This is a very important topic because wrong component selection could lead to disastrous performance both on the vehicle front and the AC front. Also, concentrated research on methods to reduce vehicle cooling load could benefit automotive AC system and automobiles on the whole as they will reduce fuel consumption, emissions and improve passenger thermal comfort.

3. Vehicle cooling loads calculation

3.1. Introduction

The first step in retrofitting an automotive AC system to the *Hybrid-convert* is calculating the vehicle cooling load. Steps to do the calculation for a generalized vehicle will be explained in the following section. The cooling load for the EcoCAR2 vehicle has been calculated as an example to aid the explanation and also, because, EcoCAR2 is the prime focus of this research endeavor.

3.2. Calculation approach

Traditional approach to vehicle cooling load/heat load calculation involves picking a time of the day, geographical location, ambient conditions and vehicle state (stationary or moving) for the calculation. Then information on vehicle physical specifications and various materials in the car (seats, dashboards, etc.) are gathered from the vehicle. Since the vehicle was not available, most of these values had to be derived from the CAD files available with the EcoCAR2 team. Some information like materials of seats, windows and dashboards was assumed. To compensate for assumptions involved in the vehicle calculation, the traditional approach was altered slightly to include heat gain due to parking in an open garage under direct sunlight. This will take care of any shortfalls in the heat load calculation. To check if the adjustment is reasonable, at the end of the calculation, comparison with heat load of a similar vehicle calculated using the traditional approach was done.

3.3. Data procurement for calculation

For a vehicle cooling load calculation, the inputs required were split into six categories.

1. Vehicle physical characteristics
2. Ambient conditions
3. Glazing surfaces
4. Solar Flux
5. Vehicle state and occupants
6. Conditioned air characteristics

3.3.1. Vehicle physical characteristics:

1. Type of vehicle: Type of vehicle is the input which will specify if it is a 2 seater or a 4 seater, a passenger vehicle or a fleet vehicle. The significance behind the passenger or fleet vehicle specification is the difference in air conditioning requirements for different vehicles. The vehicle used in the calculation was a 4 seater passenger vehicle.

2. Vehicle geometry: Vehicle geometry is obtained from the CAD file obtained by the EcoCAR2 team from GM. The required vehicle geometry inputs are

1. Length of vehicle cabin at floor level: 1.566 m

2. Width of vehicle cabin at floor level: 1.465 m

3. Height of vehicle cabin from floor level: 0.97 m

4. Dashboard material: No information was available and hence, ABS plastic the most common dashboard plastic material was chosen for the calculations. The dashboard area was approximated using common knowledge and the width of the car. ABS material properties were obtained from internet [26].

5. Seat material: Since the vehicle is not available, seat material of the vehicle base model was obtained from the vehicle website [27]. Seat material chosen was leather. Properties for leather were obtained from internet[28].

3.3.2. Ambient conditions:

A very important step in the cooling load calculation process is selection of the ambient conditions.

1. Geographical location: Usual expectation among vehicle users/designers is that the thermal comfort should not be compromised during Arizona summers or the Canadian winters. The one extreme that concerns this thesis is the Arizona summer. Added to that, the EcoCAR AC testing will be conducted at Mesa, Arizona which is close to Phoenix. Hence the city chosen was Phoenix, Arizona. The geographical location of Phoenix, Arizona is

33° 26' 53" N / 112° 4' 23" W.

2. Day and time: July 21 was chosen because it is the only date for which ASHRAE handbook has the required data for calculation. Time of the day was chosen by a trial and error method which will be elaborated upon later.

3. Orientation of passenger compartment: 0° East.

3.3.3. Glazing:

As mentioned earlier, heat load through the glass contributes to a major portion of the vehicle heat load. Hence, glass properties require proper attention during the calculation.

1. Glass surface area: The area covered by glass was calculated from the CAD files.

Front door side window: 0.11 sq.m.

Rear door side window: 0.25 sq.m.

Windshield: 0.88 sq.m.

Rear window: 0.856 sq.m.

2. Thickness of the glass: 3-5 mm.

3. Type of glass: Tinted single pane glass. The values for the glass was obtained from [22].

3.3.4. Solar irradiation:

Values pertaining to the specific time and day are obtained from NSRDB[29]. NSRDB has solar radiation data which can be accessed by the public. Typical meteorological year (TMY) data is acquired from 1961-1990 and 1991-2005 National Solar Radiation Data Base (NSRDB) archives. TMYs are hourly data sets of solar radiation and meteorological data over a 1 year period. They are averages and do not represent extremes. Hence, these data are not suitable for calculating worst case scenarios. The data is available for 1020 locations and Phoenix, Arizona is one of them. The data will be provided in the following calculation section.

3.3.5. Vehicle state & Occupants:

Vehicle was considered to be in motion at a steady 35 mph (56.32 kph) moving in the same direction as it was parked (0° East). A driver and a passenger in the front seat with summer clothing were considered to be inside the vehicle. The level of activity for the driver and passenger are driving and sitting respectively. The respective heat gains was obtained from [18].

3.3.6. Conditioned air characteristics:

1. Volume rate: ASHRAE standard 62, *Ventilation for Acceptable Indoor Air Quality*, provides data on required occupancy and air requirements. A table dedicated to outdoor air requirements for ventilation of vehicles provided in the standards was used to obtain the required numbers[30].

2. Percentage of Outside air: AC systems could condition 100% outside air, 100 % recirculated air or a mixture of both. Depending on the difference in the ambient and cab conditions, the load on AC could get higher or lower. For EcoCAR, the rules prescribe 100% recirculated air at full fan speed.

3. Cab conditions: Cab condition inputs include cab temperature and relative humidity. The cab temperature was estimated from a calculation shown in the following section. Relative humidity was not considered because of the very dry climate of phoenix and humidifiers are not part of the heat load calculations.

3.4. Cab temperature estimation at end of 2 hours of parking

The cab temperature estimation is a bulk modeling analysis conducted to estimate the average temperature of the cab materials (including air). The bulk modeling clubs together the seats, the cabin air, the dashboards, glass surface and the floor carpets. This simplifies the calculation while maintain reasonability because when considered individually each of the cab components differ in the temperature. The dashboard and steering wheel could easily be, 80-100 °C and cabin air temperature could go as high as up to 60 °C [31]. Hence, the bulk modeling was a justified choice.

3.4.1. Solar irradiation data

The TMY hourly data was obtained for July 21for Phoenix, Arizona from TMY 3 data files. The file includes an enormous amount of data. Of which, the ones required for the calculation has been listed out in the appendix section. The first and foremost information required is the ambient temperature on July 21. To obtain a better resolution for the calculation, interpolation of data was done to acquire temperatures for every 15 minutes. This was done using MS Excel and a snapshot of the Excel graph has been shown below.

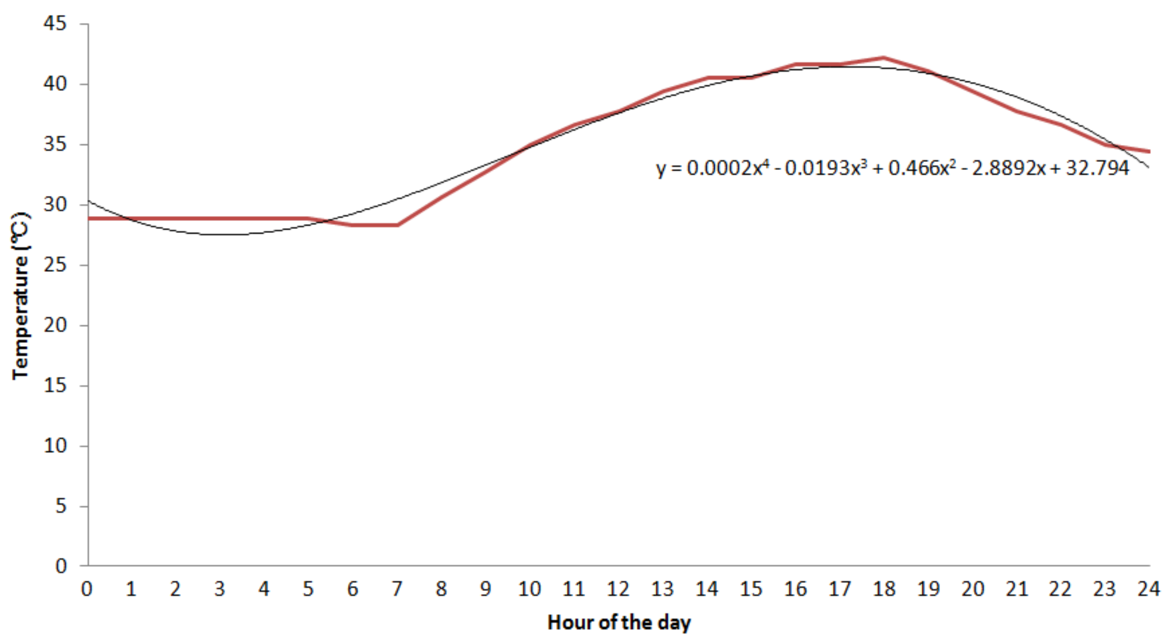

Figure 3. 1. Hour Vs Temperature

The time of the day chosen was 4 – 6 PM and the car was assumed to be parked between the hours under the sun in an open garage. The temperatures between 4- 6 PM obtained after the interpolation was

Table 1. Time and temperature - (4 PM - 6 PM)

Time (PM)	Temp (°C)
4	41.2
4.15	41.3
4.30	41.4
4.45	41.5
5	41.5
5.15	41.6
5.30	41.6
5.45	41.6
6	41.5

These temperatures were adjusted to make sure the mean temperature of the actual and interpolated data remain the same. The mean temperature of the actual data was 35.3°C and the interpolated data was 33.9°C. The mean temperature was then brought up to 35.3°C.

3.4.2. Heat gain calculation

The heat gain/loss during the 2 hours of parking was split into different categories as mentioned below:

1. Heat load through the roof, doors

2. Heat load through the glass

4. Heat load due to radiation

5. Heat load due to convection

Heat load calculation method: There are, basically, 3 methods of heat load calculation. They are

1. Total equivalent temperature differential method

2. Transfer function method

3. CLTD/SCL/CLF method

CLTD/SCL/CLF method is the popular method of all the 3, as mentioned in *ASHRAE handbook Fundamentals* [32]. It is also the preferred method amongst heat load calculation experts[19][20][9]. Since the calculation is largely based on the above citations, CLTD was the method chosen. The method has been explained in the following section.

CLTD/SCL/CLF Method:

CLTD is a method which considers the impact of temperature difference between the ambient conditions and the vehicle compartment conditions, the range of the day's temperature, solar irradiation and the heat stored in the vehicle mass to calculate a theoretical temperature difference. It accounts for the walls, roofs, floors, and glass with the help of factors. The time of the day, the day of the year, the orientation of the material and the glass will also be accounted for.

There is a time lag between the solar irradiation entering the AC controlled space and the radiant energy becoming a cooling load. CLF accounts for this important fact. SCL accounts for heat transmission by the glass surfaces.

1. Heat load through the roof, doors:

Heat load through the roof and doors (excluding the glass) of the vehicle cabin is calculated by using the following formula.

$$Q_{roof} = A_{roof}*U_{roof}*CLTD \text{ -- } (1)$$

Similarly, $Q_{door} = A_{door}*U_{door}*CLTD$

(i) The roof and door area were obtained from the CAD files.

A_{door} = 1.54 sq.m. (On each side); A_{roof} = 1.70 sq.m.

(ii)The coefficient of convection, U was determined using a formula prescribed in [18].

$$U_{roof} = 2.63*(T_{cab} - T_{amb})^{0.25;}\ U_{door} = 1.98*(T_{cab} - T_{amb})^{0.25}$$

(iii) The CLTD values were obtained from ASHRAE handbook fundamentals [32]. CLTD was corrected, as per ASHRAE instructions, before being used in the formula.

The calculation was performed for every 15 minutes between 4 and 6 PM. The values are available in the appendix.

2. Heat load through the glass

The heat load calculation through the glass can be split into 2 categories:

1. Heat load calculated through CLTD.

2. Heat load calculated through SCL.

(i) Heat load through CLTD: This is similar to the calculation in the previous section. The convection coefficient U was found out using,

$$U_{glass} = 1.98[(T_{cab}-T_{amb})*\cos(\theta - 90)]^{0.25}$$

(ii) Heat load through SCL: Since glass is a transparent object and allows solar radiation through to heat the interior, Solar cooling load factor (SCL) has to be calculated. SCL is calculated using the following formula,

$$Q_{glass,\ SCL} = A*SC*SCL$$

SC and SCL values were obtained from [32]. The glass, as mentioned before, was assumed to be a single pane glass. The windshield is considered to be 0% tinted and the other glasses (windows, rear glass) are considered to be 60% tinted.

3. Heat gain through convection

Convection through the glass, doors and windows can be calculated using the formula,

$$Q_{conv} = U*A*(T_{cab} - T_{amb})$$

The calculation was performed in MS excel and the values are listed in appendix.

4. Heat loss through radiation

Heat loss through radiation occurs from the body and the glass. The body's radiation capacity depends on the color of the body.

Radiation loss was calculated using,

$$Q_{rad} = \varepsilon*\sigma*(Tcab^4-Tamb^4)$$

The calculated values have been listed in appendix.

5. Net heat gain and temperature rise

(i) Heat loads from the previous sections were summed up to obtain the net heat gain.

$$Q_{net} = Q_{roof} + Q_{door} + Q_{glass} + Q_{glass,\ SCL} + Q_{conv} + Q_{rad}$$

The Qnet values between 4 – 6 PM are tabulated below.

Table 2. Net Heat gain between 4 PM and 6PM

Time (PM)	Q_{net}, W
4.00	1405.66
4.15	-737.57
4.30	-154.32
4.45	-38.61
5.00	-70.35
5.15	-153.67
5.30	-47.51
5.45	-15.59
6.00	-144.23

` The values, shown above, were obtained when the initial cab temperature, Tcab was assumed at 50°C.

(ii) Temperature rise calculation: The corresponding temperature rise was calculated using the formula.

$$Q = \dot{m} * c_p * \Delta t$$

Since this is a bulk modeling analysis, $\dot{m} * c_p$ is actually $\sum [(\dot{m}_{air} * c_p air) + (\dot{m}_{glass} * c_p glass) + (\dot{m}_{body} * c_p glass) + (\dot{m}_{interiors} * c_p interiors)]$. The Δt and the respective Tcab calculated corresponding to Table 2 are provided below.

Table 3. Temperature difference and Cab temperature

Time	Δt, °C	T_{cab}, °C
4.15	14.96	64.96
4.30	-7.85	57.11
4.45	-1.64	44.46
5.00	-0.41	55.05
5.15	-0.75	54.30
5.30	-1.64	52.67
5.45	-0.51	52.16
6.00	-0.17	52.00

To verify the calculations and that the assumptions made were not too radical, the entire calculation was repeated for initial T_{cab} temperature values from 35 – 52 °C.

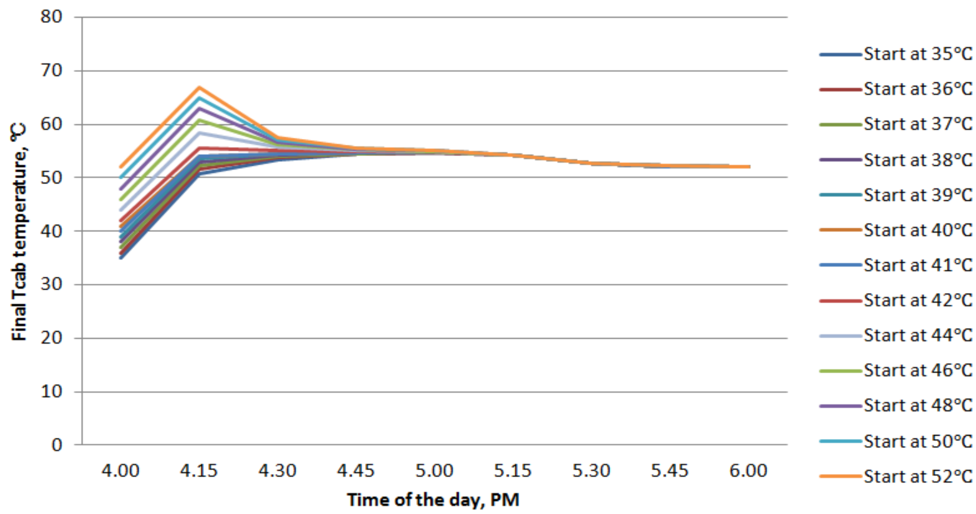

Figure 3. 2. Time of the day Vs Final Cab Temperature (4 – 6 PM)

The above graph was formed using values calculated for every 15 minutes between 4 – 6 PM with the initial cab temperatures ranging from 35 °C - 52 °C. The graph depicts that irrespective of the initial cab temperature, the cab temperature tends to reach an equilibrium temperature around 52°C at 6 PM. This suggests that the assumptions are proper and also, that the initial cab temperature selection does not impact the heat load calculation.

Armed with the cabin temperature rise value, a preliminary vehicle cooling load calculation was conducted to determine if the final cab temperature was reasonable. The vehicle cooling load was found to be around 1 KW. Comparing this to common peak heat load values calculated in other papers for sedans, it was found to be at least 50% less than commonly obtained values. The reason behind the disparity was that the time period chosen for the calculation was not the most suitable for a heat load calculation. Though 4 – 6 PM was the time period with highest temperatures of the day, the solar radiation peak actually occurs during noon. Hence, the entire calculation was repeated for time period between 11AM – 1 PM. The net gain and Δt values for a initial cab temperature of 50°C corresponding to 11 AM – 1 PM are tabulated below. Also, a graph of temperature variation over the time period similar to the one in Fig# has been provided.

Table 4. Temperature difference and Final Cab temperature (11 AM - 1 PM)

Time (PM)	Net Gain, W	Δt, °C	T_{cab}, °C
11.00	-25.45	-	50
11.15	-131.44	-0.27	49.73
11.30	-30.77	-1.40	48.33
11.45	4.74	-0.33	48.00
12.00	14.51	0.05	48.05
12.15	86.06	0.15	48.20
12.30	42.63	0.92	49.12
12.45	26.39	0.45	49.58
1	87.28	0.28	49.86

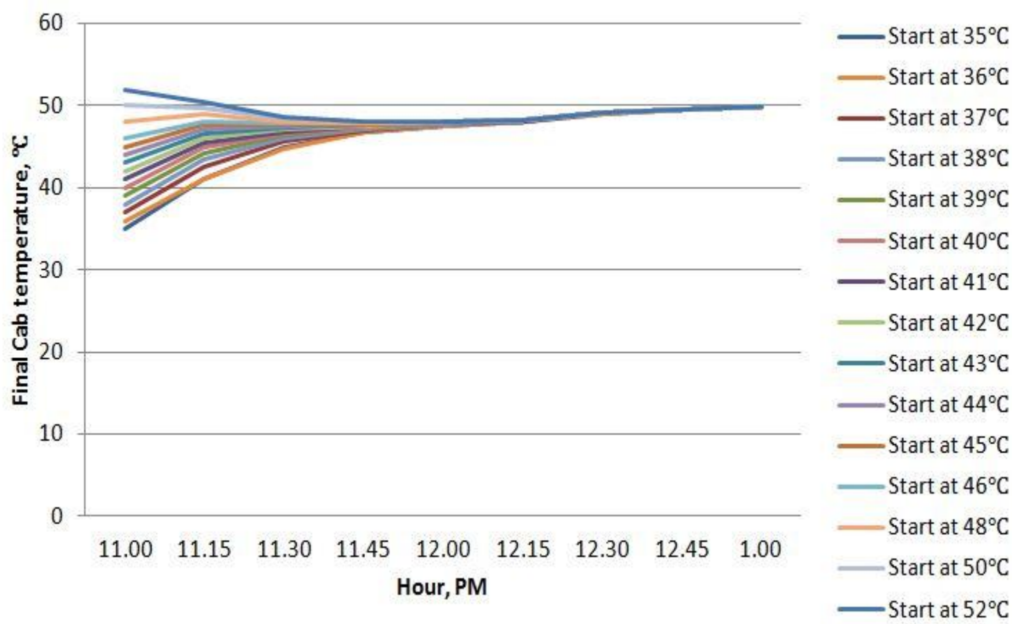

Figure 3. 3. Time of the day Vs Final Cab Temperature (11 AM - 1 PM)

The heat load calculation for 11 AM – 1 PM was done and the calculation is shown in the following section.

3.5. Vehicle heat load

The initial conditions used for the vehicle heat load calculation were:

1. Location: Phoenix, Arizona

2. Time & day: 1 PM, July 21st

3. Ambient temperature, T_{amb}: 38.60 °C

4. Vehicle cabin temperature, T_{cab}: 49.85 °C

5. Vehicle speed, v = 35 mph (56.32 kph)

The vehicle heat load calculation is very similar to the calculation done to find the temperature rise. The heat load calculation was split, as before, into different categories:

i. Heat load through roof, doors, glass:

The heat load was calculated using the CLTD method and equation 1 was used to determine the heat load. As before, solar cooling load factor of glass is also calculated. The calculated values were

Q_{roof} = **1838.90 W**; Q_{doors} = 140.34 W; $Q_{windshield}$ = **-350.60 W**

$Q_{rear\ glass}$= **-418.42 W**; $Q_{side\ windows}$ = **-545.72 W**;

Solar cooling load factor: $Q_{windshield}$ = **101.16 W**; $Q_{rear\ glass}$ = **149.69 W**

$Q_{side\ windows}$ = **80.55 W**

ii. Heat load due to occupants:

The occupants assumed were a driver and a passenger in the front seat. A driver while driving contributes **220 Watts** and an idle passenger contributes **102 Watts.**

iii. Heat load due to infiltration:

Vehicle cabin is not a controlled volume and ambient air infiltrates the cabin through various gaps. The heat load due to infiltration was calculated using the following equation[20],

$$Q_{infiltration} = \dot{m}*(I_{cab} - I_{amb})$$

Mass flow rate is estimated using [20],

$$\dot{m} = 1.5 * 40 \text{ cfh}$$

Specific enthalpies, I, could be obtained from [33]

Substituting the values in, $Q_{infiltration}$ = **4.43 W**

iv. Heat load due to absorption by interiors:

The solar radiation which passes through the glass will be absorbed by the vehicle interiors. The absorbed heat will add to the vehicle heat load as it cannot be radiated back to the atmosphere through the glass.

$$Q_a = A*\alpha*I*\tau$$

(a) Calculation of solar radiation, I: It can be split into direct radiation and diffuse radiation.

Direct radiation, $I_D = I_{ND}* \cos \gamma$ (horizontal surface)

$$I_{ND} = A/ [\exp (B/\sin \beta)];\ \sin \beta = (\cos l* \cos d* \cos h) + (\sin l* \sin d)$$

For a vertical or inclined surface,

$$I_v = I_D*\rho*[(1- \cos \sum)/2]$$

The calculations are provided in the appendix section and the values obtained were,

$Q_{a,\ windshield}$ = **161.61 W**; $Q_{a,\ rear\ glass}$ = **83.12 W**; $Q_{a,\ side\ windows}$ = **22.04 W**

v. Heat load due to utilities:

The component which contributes to the heat load is the blower motor which is present inside the dashboard. The heat gain value for a blower motor was obtained from [32].

Q_{blower} = **439 W.**

vi. Air conditioning load:

The load on the air conditioning system to cool the cabin was also taken into account. The formula used to calculate the load was,

$$Q_{ac} = m_{air} * c_p air * \Delta t_{ac}$$

Where m_{air} – mass of the air in the cab, kg

$c_p air$ – specific heat capacity of air in the cab, J/kg.K

Δt_{ac} – required rate of temperature decrease, °C /s.

The rate of temperature decrease was set at 14 °C in 4 minutes to satisfy to EcoCAR2 competition requirements.

$$Q_{ac} = 2.5 * 1008 * .06$$

$$Q_{ac} = \mathbf{147\ W}$$

vii. Net heat gain:

$$Q_{net} = Q_{roof,\ CLTD} + Q_{doors,\ CLTD} + Q_{glass,\ CLTD} + Q_{SCL,\ glass} + Q_{occupants} + Q_{infiltration} + Q_{interiors\ absorption} + Q_{utilites} + Q_{ac\ load}$$

Substituting values from previous sections,

$$\mathbf{Q_{net} = 2175.2\ W}$$

The above value is the amount of cooling that AC system has to provide to reduce the temperature of the cab by at least 6°C in less than 5 minutes.

3. 6. Conclusion

The vehicle cooling load, found above, was compared to cooling load values calculated for similar vehicle [18] and then deemed reasonable. Noticeable exclusions from the vehicle heat load calculation are latent heat load, heat load through radiation and heat load through convection. Latent heat load was not considered because Phoenix is not a humid place. Calculations of heat load through radiation were found to be negligible. Heat load through convection causes depreciation of the net heat load until the vehicle cab temperature reduces below the ambient temperature.

It could also be argued that, for a humid place like Miami, the precautionary peak heat load calculation will compensate for the extra latent load. Similarly when convection contribution to heat load reaches a significant level, other loads such as interior absorption load and air conditioning load are low.

4. Performance analysis using ADVISOR

The adverse impact of air conditioning systems on vehicle performance has been well documented in the initial sections of this thesis. This is, mainly, due to the amount of power drawn from vehicle power sources. Air conditioning systems constitute a major portion of the accessory power load on the engine, which could be as high as 7-8 bhp.

The first step in reducing the accessory power load on the engine is an efficient heat load analysis. If the heat load analysis is not efficient, the vehicle will either have an over sized or under sized AC system. Once the heat load analysis is done, the next step is calculating the required amount of mechanical or electrical power to supply the necessary cooling for the vehicle. In case of a conventional AC system, the load is mechanical. For hybrid electric vehicles like a Toyota Prius, the load is electrical.

The objective of this chapter is to explain the benefits of a performance analysis for a *Hybrid-convert* running with the accessory load of the air conditioning system. The analysis was done for the EcoCAR2 vehicle and the results will be discussed below.

4.1. Introduction to ADVISOR

The performance analysis of the vehicle was done using ADVISOR, *ADvanced Vehicle SimulatOR.* ADVISOR was developed in 1994 by NREL to develop, simulate, analyze and understand hybrid electric vehicles. ADVISOR is, primarily, a tool that could analyze vehicle powertrains and provide details on power flow among components. ADVISOR is user friendly and is based on MATLAB/Simulink[34].

ADVISOR was chosen for the analysis purpose because of various advantages. Previous experience with the software and amount of ADVISOR expertise available were major deciding factors. ADVISOR is very flexible and user friendly. Unlike most commercially available component simulation and performance analysis software, ADVISOR offers the user to work in the background, meaning, with the codes instead of the graphical user interface.

The vehicle model in ADVISOR is made up of a number of .m files (Matlab extension), each containing specific data on the vehicle. For instance, a snapshot from one of the Matlab files which was used in the analysis, "Vehicle" data file is shown below.

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% VEHICLE PARAMETERS
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Note on vehicle mass:
%       The actual curb weight of a 1998 Japanese PRIUS is 2783 pounds(full tank).
%       If you wish to accurately set your totalvehicle mass
%       to this value in the A2 GUI, you should use the mass override
%       checkbox and enter in the value 1398, which is (2783+300)/2.205 = 1398 kg, which comes from
%       adding on 300 lbs of EPA test mass, and then converting pounds to kilograms.
veh_glider_mass=950; % (kg), vehicle mass w/o propulsion system (fuel converter,
                     % exhaust aftertreatment, drivetrain, motor, ESS, generator)
veh_CD=0.3;  % (--), coefficient of aerodynamic drag, 0.3 from toyota press release
veh_FA=2.3;    % (m^2), frontal area, 1.746 from Unique Mobility calculation
% for the eq'n:  rolling_drag=mass*gravity*(veh_1st_rrc+veh_2nd_rrc*v)
%veh_1st_rrc=0.009;  % (--) not sure about this yet!
%veh_2nd_rrc=0;     % (s/m)  not sure about this yet!
% fraction of vehicle weight on front axle when standing still
veh_front_wt_frac=0.45;  % (--) cg is 1.542 from rear axle on empty PRIUS
% height of vehicle center-of-gravity above the road
veh_cg_height=0.569;    % (m), .569 for PRIUS_JPN from Unique Mobility testing
% vehicle wheelbase, from center of front tire patch to center of rear patch
veh_wheelbase=2.738;    % (m), 2.55 m for PRIUS_JPN

veh_cargo_mass=140; %kg  cargo mass
```

Figure 4. 1. ADVISOR - Matlab File snapshot

The base file, originally, belonged to a Toyota Prius model. The numerical values were changed to accommodate it to the EcoCAR2 *hybrid-convert*. The values in the above picture belong to the EcoCAR2 model developed for this analysis. Hence, flexibility was a major advantage. Added to all these advantages, ADVISOR was a free software developed by NREL, a government funded research lab.

4.2. Performance Analysis

Before developing the vehicle model in ADVISOR, the amount of electrical load on the vehicle was determined. This was found out so that the performance could be analyzed in incremental steps of electrical power load.

4.2.1. Determination of accessory electrical load

The electrical load could be determined from the amount of power required to run the compressor. The compressor is the "heart" of any air conditioning system. The mass flow rate of the refrigerant is an important factor in the performance of AC system and it is dictated by the compressor. To determine the mass flow rate of the refrigerant,

$$Q_{in} = m_{R134A}*(h_1-h_4)$$

Where

Q_{in}: heat gain across evaporator, W.

R134A: refrigerant

Temperatures T_1 and T_4 could be understand from the picture below.

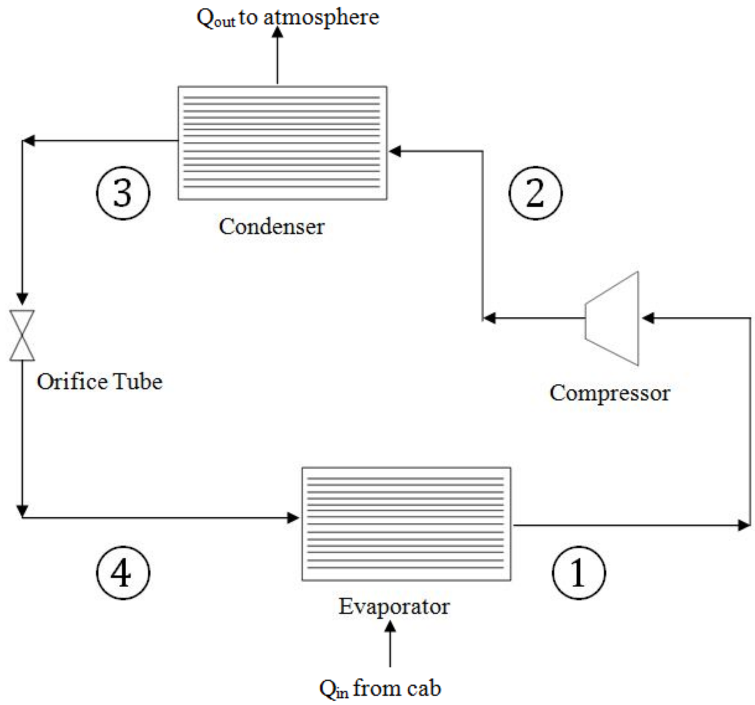

Figure 4. 2. HV-AC system

Using the Figure 4.2,

State 1: Saturated R134A Vapor from evaporator to compressor

Temperature, T_1; Specific enthalpy h_1; Specific volume, V_1;

State 2: Compressor super heated, high pressure vapor to condenser

Temperature, T_2; Specific enthalpy h_2; Specific volume, V_2;

State 3: High pressure saturated liquid to orifice tube

Temperature, T_3; Specific enthalpy h_3; Specific volume, V_3;

State 4: Low pressure, low temperature liquid to evaporator

Temperature, T_4; Specific enthalpy h_4; Specific volume, V_4;

Ideally,

Process 1: State 1 – State 2 (constant volume process)

Process 2: State 2 – State 3 (Isothermal process)

Process 3: State 3 – State 4 (Isenthalpic process)

Process 4: State 4 – State 1 (Isothermal process)

Assumptions for the calculation include:

1. Processes are ideal.

2. Temperature of refrigerant at evaporator, T_1 = 12 °C.

3. Temperature of refrigerant at condenser, T_3 = 100 °C.

Temperatures T_3 and T_1 assumptions are valid because it is common practice to have a difference of 10- 15 °F (6- 8 °C) between the refrigerant and the air which passes though the heat exchanger. Since a temperature of at least 20 °C is desired at the outlet of the evaporator and 80-90 °C is a common temperature under the hood[3], the assumptions were deemed realistic.

Based on the assumptions and knowledge about the process,

$T_2=T_3$; $T_4=T_1$; $h_3=h_4$.

Also, $V_1=V_2$. But since volumetric efficiency of scroll compressors is, typically, around 90%,

$V_2=V_1*.90$.

ASHRAE fundamentals handbook 1997 [32] was used to obtain the different state values. The values can be found below.

Table 5. State Values of the AC system

State	T_{suffix}, °C	Specific enthalpy, kJ/kg	Specific volume, m^3/kg	Density, kg/ m^3
1	12	405.51	.04636	-
2	100	453.20	-	150.47
3	92	374.02	-	646.7
4	12	374.02	-	1253.3

Using the values in the above table, mass flow rate of refrigerant can be determined.

Since Q_{in} = 2175 W (cooling load),

$$2175 = m_{R134A}*(405.51 - 374.02) *10^3$$

$$\mathbf{m_{R134A} = 0.07\ kg/s}$$

With the mass flow rate m_{R134A},

Electrical power required to run the compressor,

$$E = m_{R134A}*(h_2-h_1) \quad + \text{heat loss}$$

Heat loss is, typically, 5-7% of the electrical load.

$$(1-.07)\ E = 0.07\ (453.20-405.51)*10^3$$

$$\mathbf{E = 3590\ W}$$

This electrical load is the amount of power required to run the compressor to provide the required amount of cooling at the specified rate. This load was used in the performance analysis with accessory load being added in incremental steps of 700 W. The five step analysis and the results are provided in the next section.

4.2.2. Vehicle Modeling

The vehicle model was built in ADVISOR using data procured from the EcoCAR2 team. The vehicle specifications used in the analysis are provided below:

Vehicle data:

Base weight: 950 kg

Frontal area: 2.3 m^2; Coefficient of drag: 0.3;

Weight fraction: 0.45 front/0.55 rear

Wheelbase: 2.738 m

Center of gravity: 0.569 m

Cargo mass: 140 kg

Engine Data:

A Kubota 1.5 L diesel engine is the engine chosen by the team. Due to the non-availability of the engine physically, a 1.5L diesel engine, already available in ADVISOR, was sized and modeled to suit the purpose.

Size: 1.5 L

Power: 44.2 bhp at 3000 rpm

Fuel used: Biodiesel.

Weight: 114 kg

Changes were made to the Matlab file to use biodiesel in the analysis. A snapshot of the change is shown below.

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
fc_fuel_den=0.856*1000; % (g/l), density of the fuel
fc_fuel_lhv=37.6*1000; % (J/g), lower heating value of the fuel
```

Figure 4. 3. Biodiesel values in Matlab file

Driveline mode: Series

Battery Specifications: Li-ion battery with 6 modules.

Weight: 151 kg

Nominal Voltage: 340V DC

Peak Discharge: 177 kW

Minimum pack energy: 18.9 kW-hr

Generator Specifications:

Continuous power: 37 kW

Generator Specifications (Contd):

Peak power: 80 kW

Voltage: 220 – 400 V DC

Weight: 33.5 kg

Motor Specifications:

Continuous power: 45 kW

Peak power: 103 kW

Voltage: 250 – 403 V DC

Weight: 66 kg

Wheels specifications:

Radius: 0.34 m

Weight: 4 x 35 kg.

With the above specifications, the vehicle was built in ADVISOR. A picture of the vehicle model built in ADVISOR is shown in the next page. Points to note with respect to vehicle model are:

1. The number of battery packs in the model is 26, instead of the original 6. This was done because of the specification of Li-ion batteries available in ADVISOR. Since the battery model of A123 Li-ion batteries was not available, the available battery model was altered to suit the requirement. The number of modules had to be increased to 26 to reach the nominal voltage of 340 V DC.

2. ADVISOR selected the generator as per the series model developed and the generator was sized to meet the requirement.

3. The transmission in the EcoCAR2 vehicle is automatic. But, ADVISOR does not have automatic transmission options for series driveline option.

4. The vehicle gross weight was over ridden to reach 2182 kg which is the calculated weight of the vehicle.

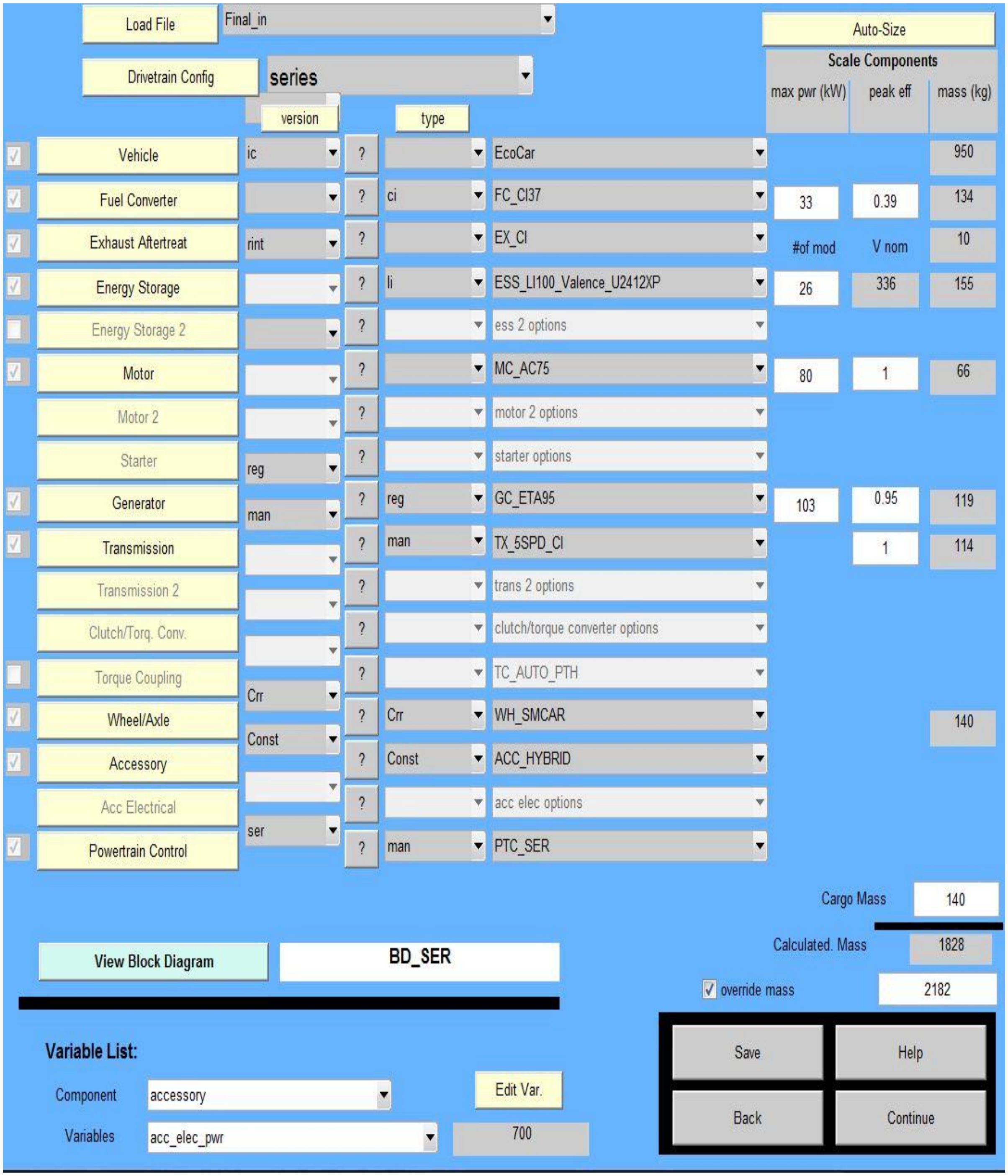

Figure 4. 4. Vehicle model – ADVISOR

4.2.3. Setup

The most important step of the simulation setup is the selection of drive cycle. The drive cycle chosen to represent the conditions is the Urban Dynamometer Driving Schedule (UDDS). UDDS is commonly known as “LA4” or “the city test”[35]. This cycle is the perfect choice because it is used to test light duty vehicle testing. Also, EcoCAR2 team runs their simulations with UDDS cycle. Other settings during the setup include allowing a linear correction of state of charge, SOC. A snapshot of the simulation setup with the UDDS cycles has been provided below.

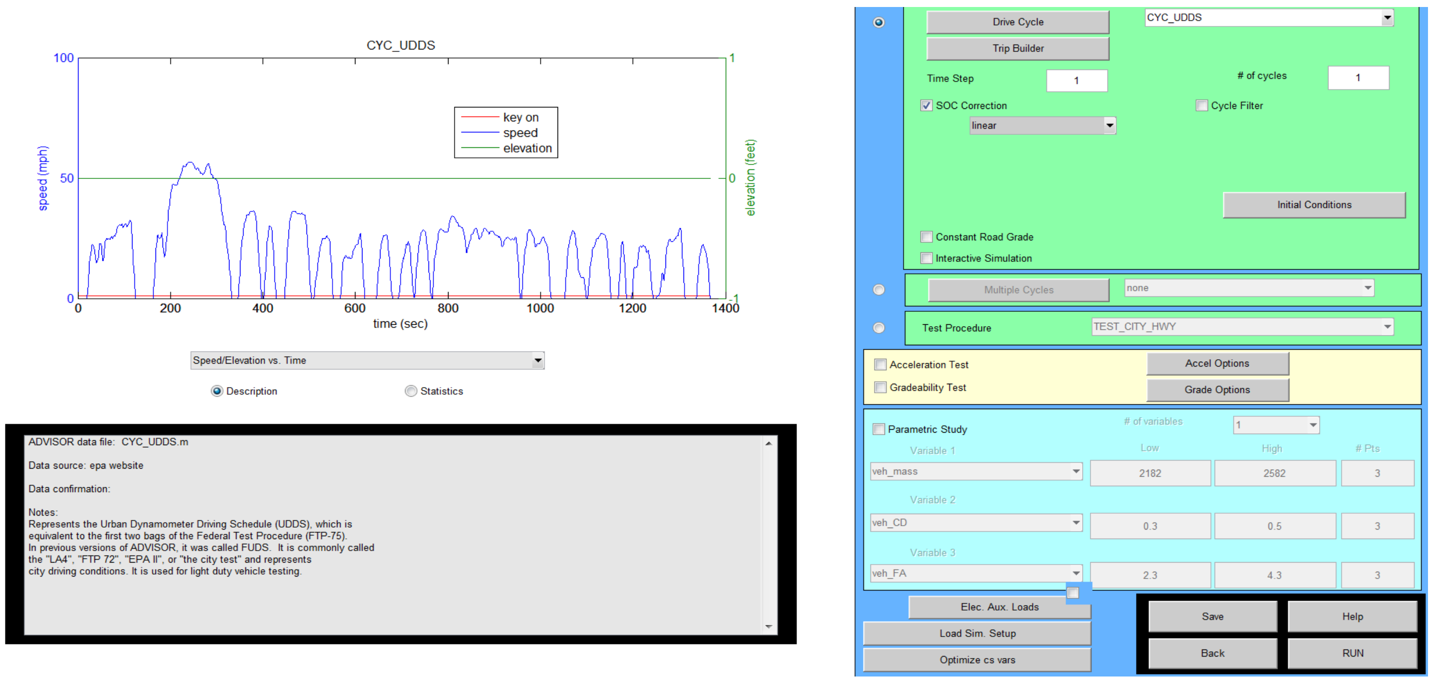

Figure 4. 5. UDDS Cycle

4.2.4. Simulation & Results

Once the vehicle model was finished and the simulation was setup, the simulation was run six times with incremental steps of 700 W from 0 W to 3500 W**.** Running the vehicle model over only one cycle of UDDS resulted in the vehicle running in pure electric mode. Hence, the simulation was run twice over the UDDS cycle to get significant results. The result was judged based on 2 factors. They are

1. Fuel Economy
2. History of SOC

Fuel economy is a very common term and does not require explanation. SOC, state of charge is the value which indicates the percentage of charge remaining. 0% SOC means no charge and 100% SOC means fully charged. Both these values where compared for each of the six simulations.

Simulation results of 2 of these tests will be analyzed and discussed below to understand the adverse impact of the accessory load.

1. Benchmark (No accessory load)

Fuel economy tests of commercial passenger vehicles are run without switching on their air conditioning system. The EcoCAR team also determines the fuel economy running a no accessory load test. The simulation result of the 0 Watt test is given below.

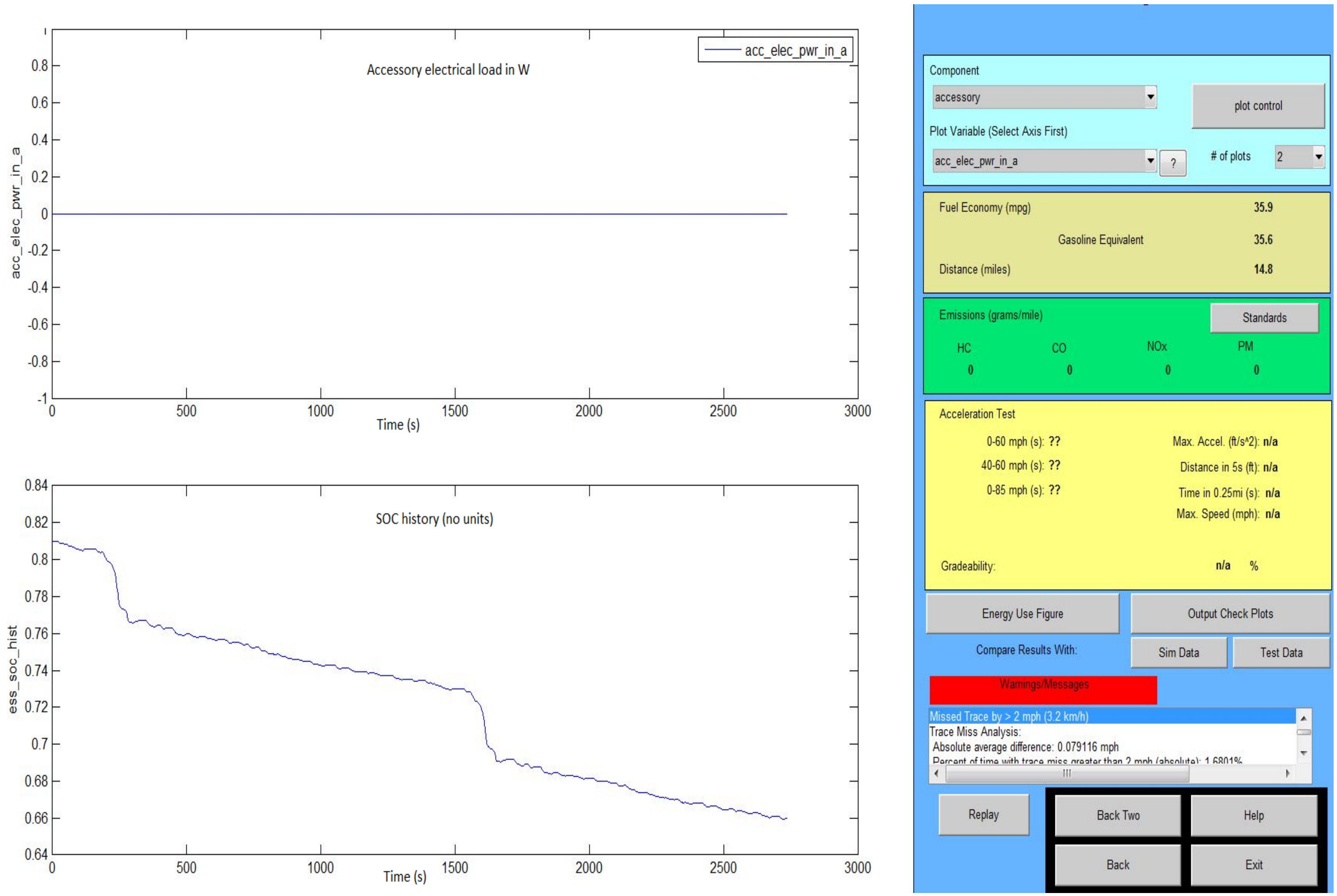

Figure 4. 6. 0 W Accessory electrical load

Tested by EPA, the Chevrolet Malibu stock vehicle's fuel economy inside the city is 25mpg (10.6 kpl) [36]. The vehicle designed by the EcoCAR2 team and modeled for this analysis has an improved fuel economy of almost 36mpg (15.3 kpl). The SOC has reduced 15 % to 0.66 in 46 minutes (2 UDDS cycles).

2. Simulation Results

The results obtained from the rest of the 5 simulations are shown below. At the end of the results, a comparison table of the results and a graph to depict the comparison is included.

(i) 700 W accessory load

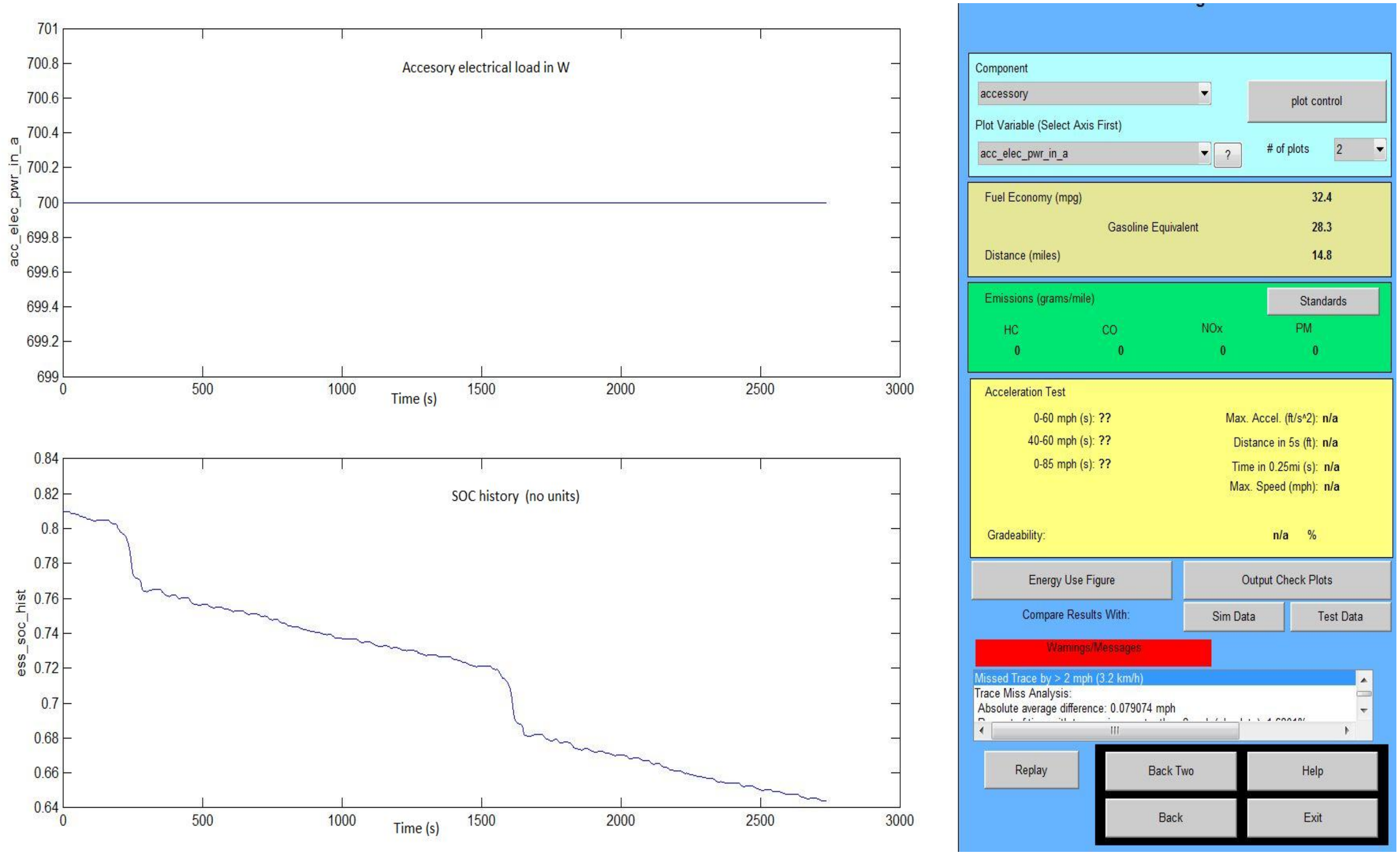

Figure 4. 7. 700 W Accessory electrical load

(ii) 1400 W accessory load

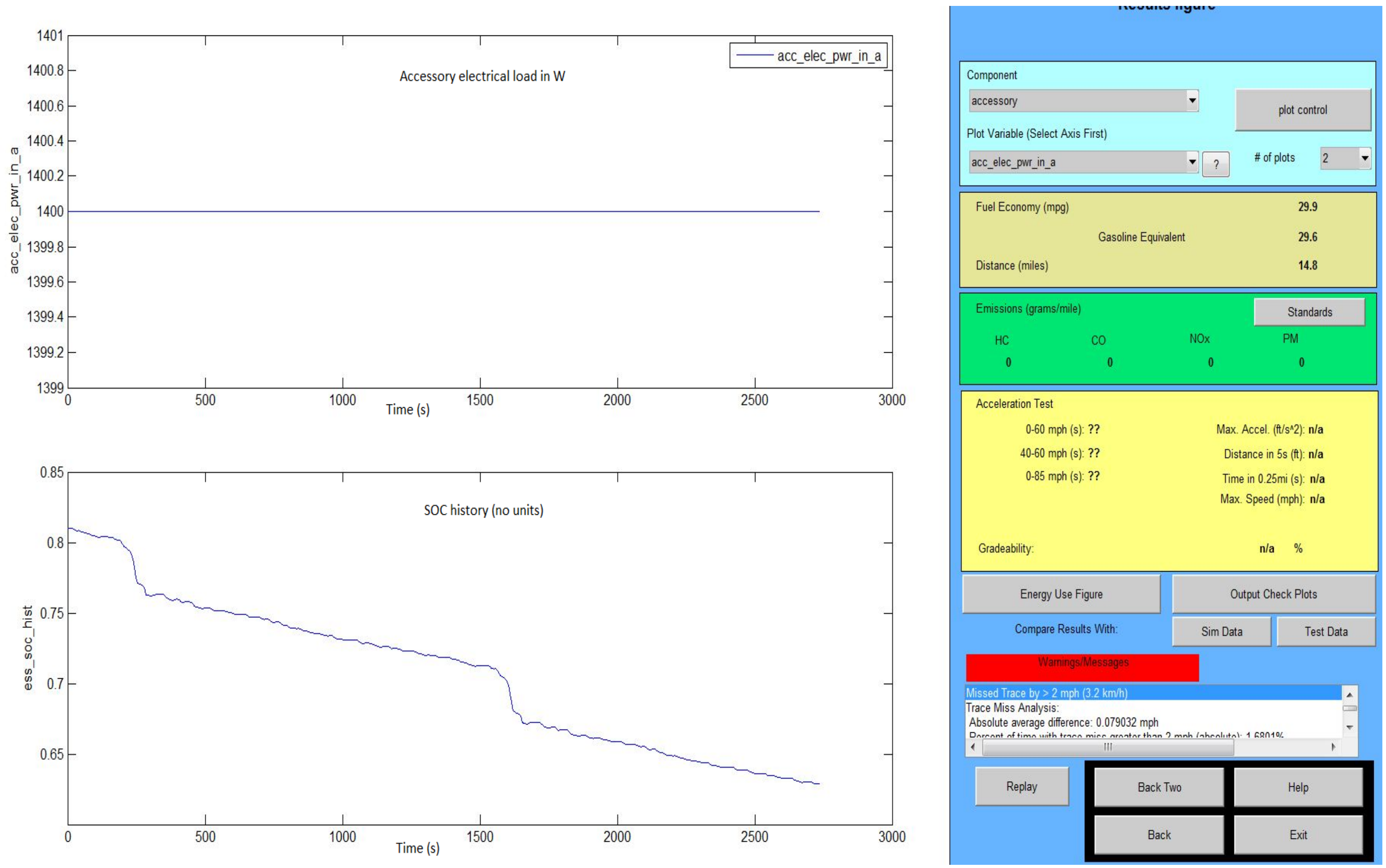

Figure 4. 8. 1400 W Accessory electrical load

(iii) 2100 W accessory load

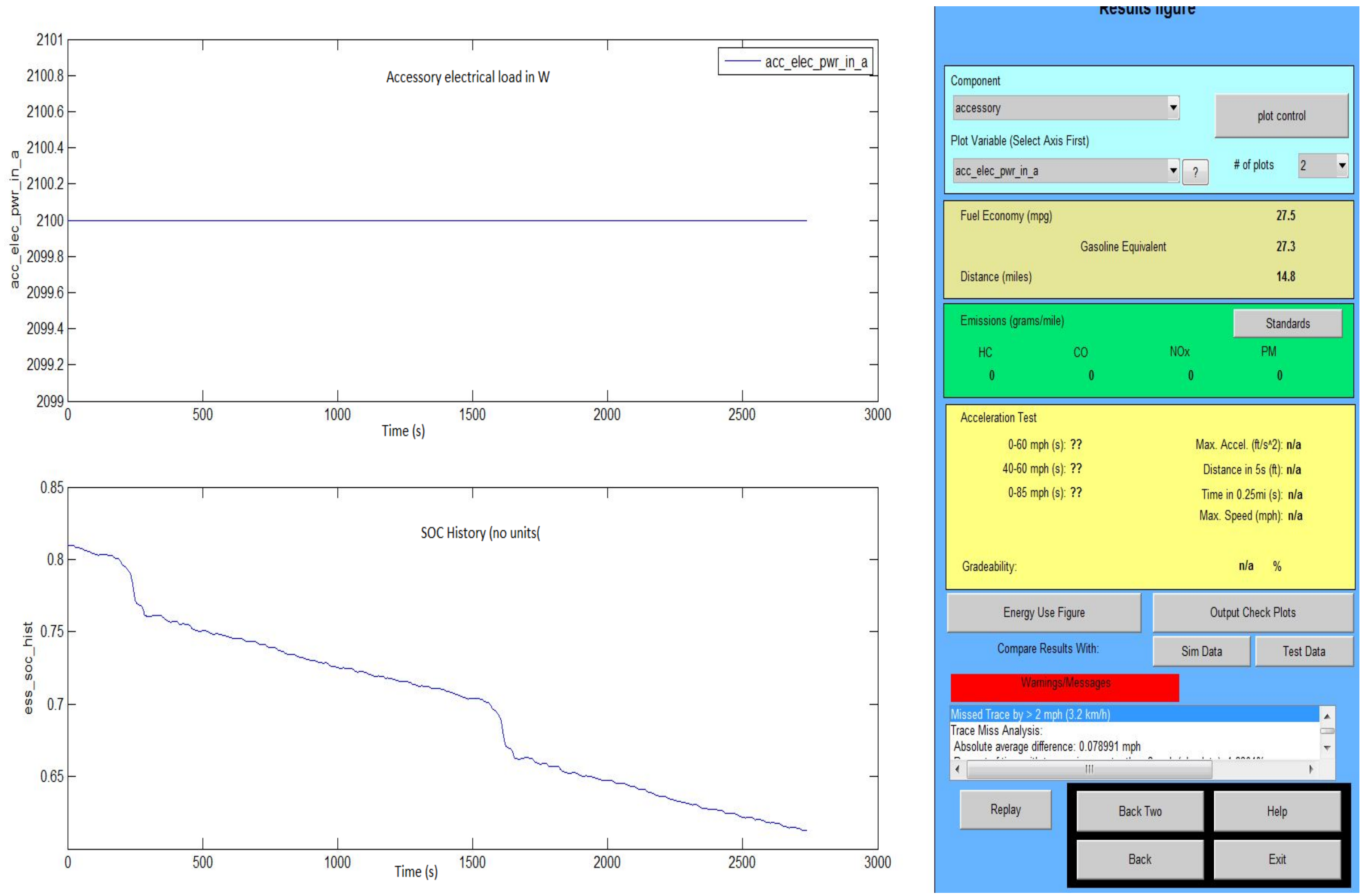

Figure 4. 9. 2100 W Accessory electrical load

(iv) 2800 W accessory load

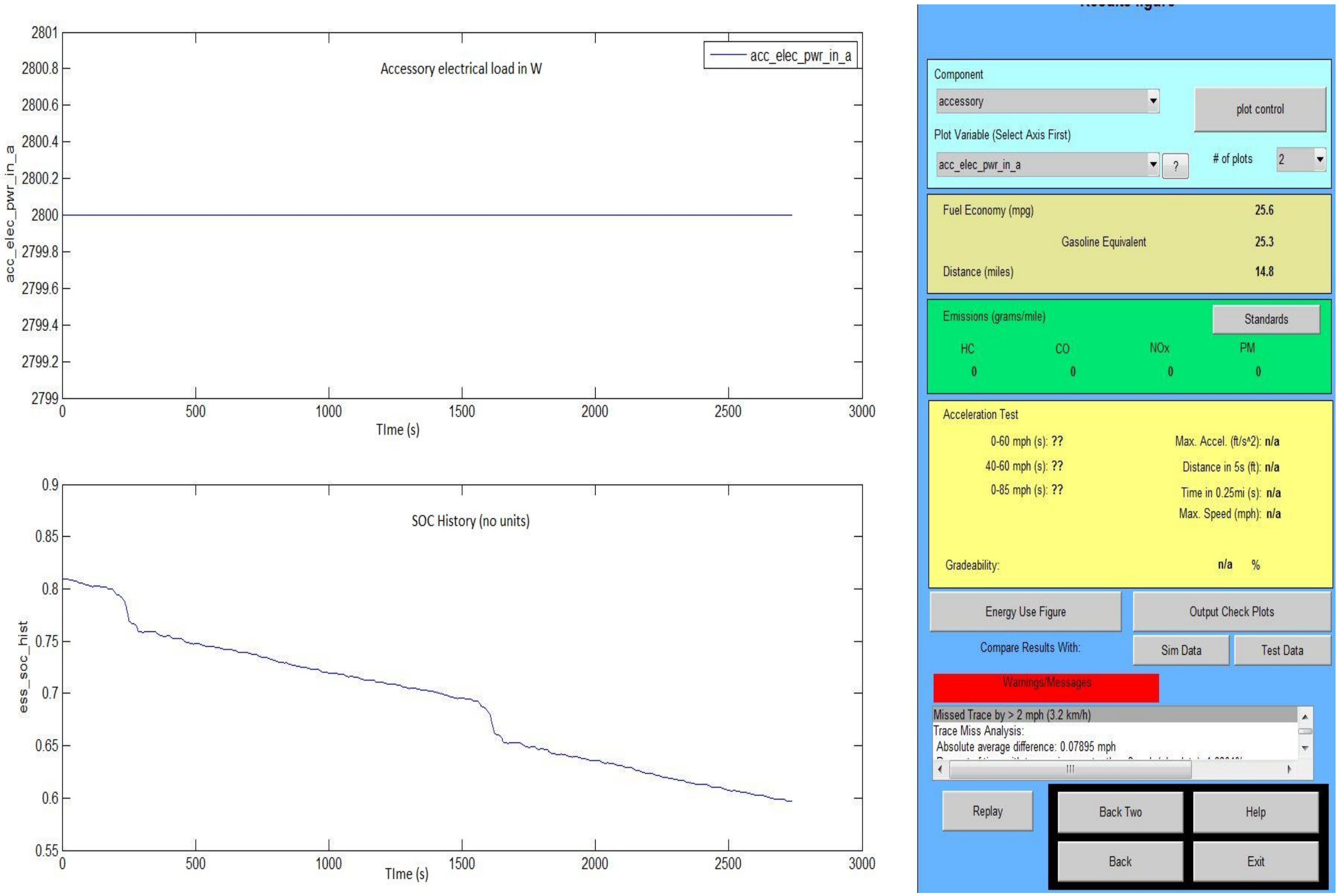

Figure 4. 10. 2800 W Accessory electrical load

(v) 3500 W accessory load

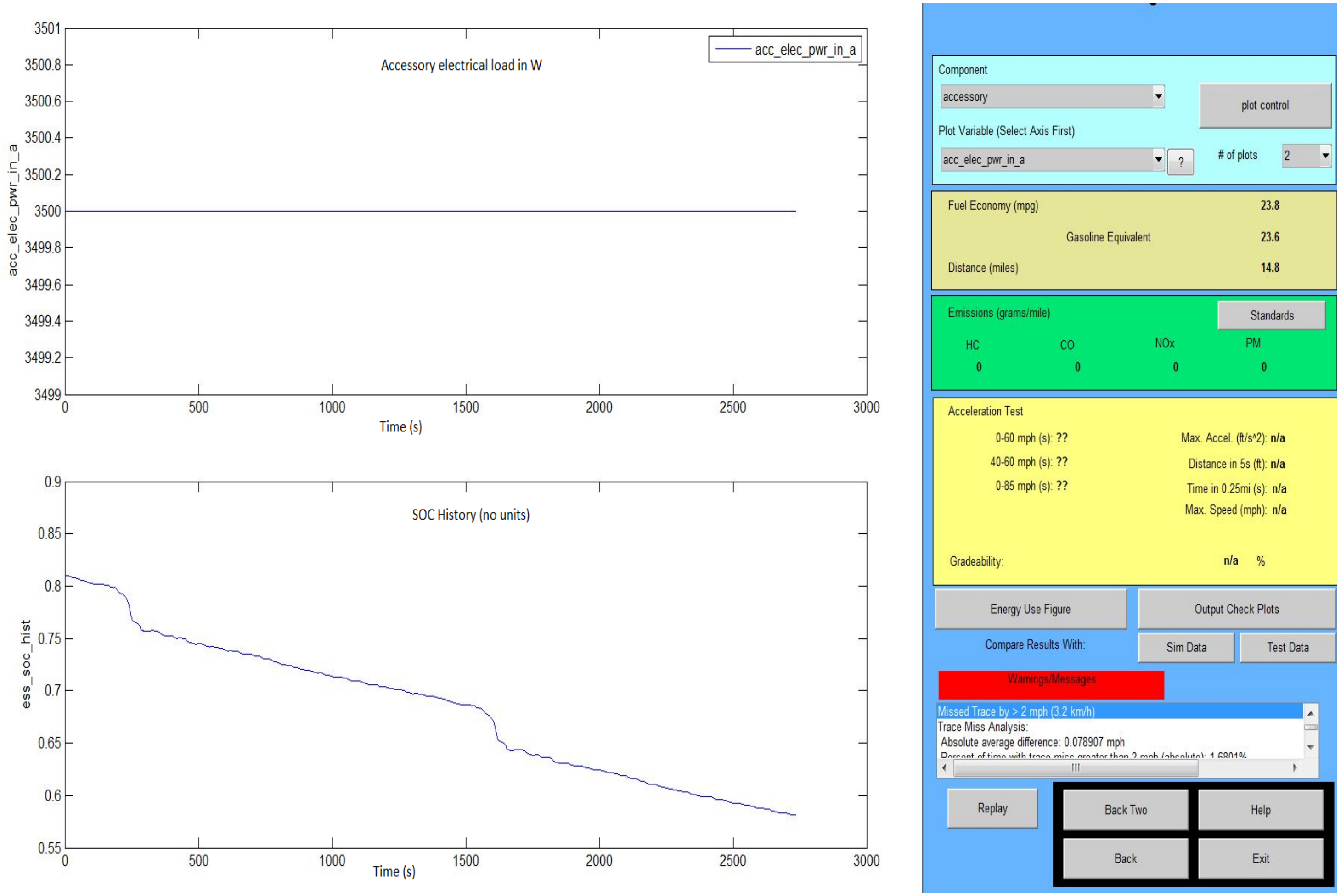

Figure 4. 11. 3500 W Accessory electrical load

4.3. Conclusion

It is evident from the results above or the tabulated comparison below, that the AC system has an adverse impact on the vehicle performance. To produce target cooling of 2125 W or 0.60 TR, which is the vehicle peak heat load, nearly 3500 W of electrical power is required. At 3500 W, the vehicle fuel economy drops 12.1 mpg and the SOC drops another 11%. This is less than the 25 mpg of the stock Malibu 2013 before the conversion.

Table 6. Results comparison

Electrical load, W	Fuel economy, mpg	SOC at 46 minutes
0 (benchmark)	35.9	.66
700	32.4	.645
1400	29.9	.63
2100	27.5	.61
2800	25.6	.59
3500	23.8	.57

However, the entire simulation was run at a constant electrical load. Realistically, the accessory load keeps varying according to the thermal comfort of the passenger.0.6 TR is necessary, only, initially and once a steady, comfortable cab temperature has been reached, the load on the AC also drops linearly. Consequently, the electrical load drops. Hence, the 12.1 mpg drop in the fuel economy or the 11 % less SOC is strictly, a worst case scenario.

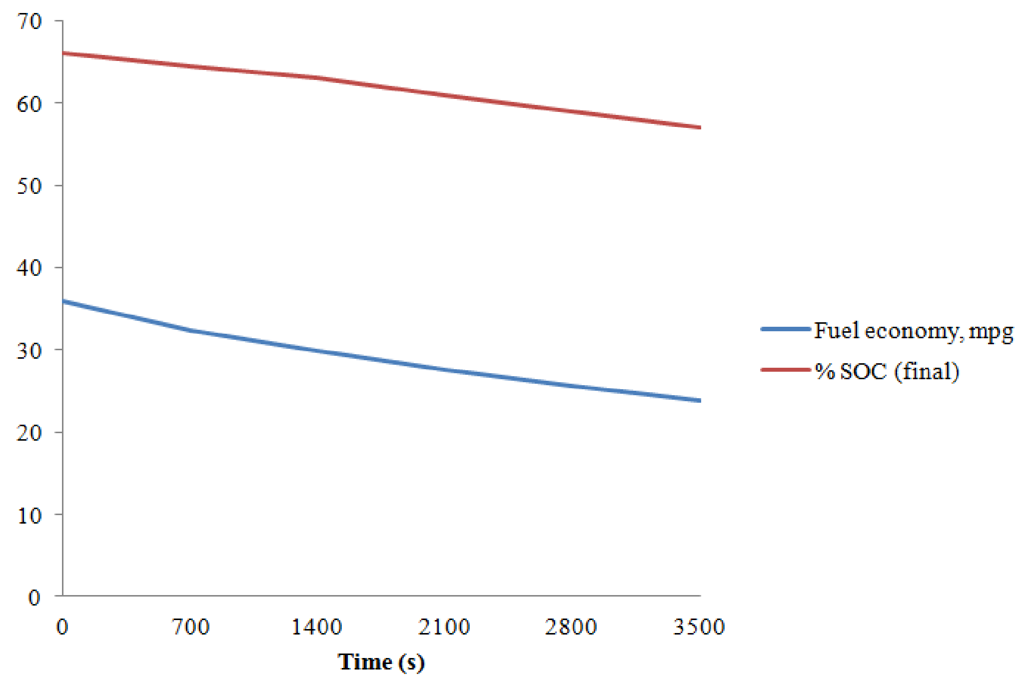

Figure 4. 12. Results

Thus, even with peak air conditioning load, the fuel economy of the *Hybrid-convert* is as high as the fuel economy of the original stock Chevrolet Malibu 2013. This could be further improved by reducing the peak heat load of the vehicle.

5. Reduction of vehicle cooling load

5.1. Introduction

The vehicle cooling load and its consequences have been clearly elucidated in the previous chapters. A reduction in vehicle cooling load will have desirable consequences, not only in air conditioning system performance, but also in the entire vehicle performance and passenger experience. Reduction in fuel economy, emissions, vehicle weight and better thermal comfort are a few of the desirable consequences.

Heat gain through glass, heat gain through infiltration, heat retention due to absorption by vehicle interiors are major contributors to the vehicle cooling load. In this section, methods to reduce the vehicle cooling load (obtained from various sources) will be discussed in detail. Since the EcoCAR2 vehicle was not available, the methods discussed or effects of the changes could not be empirically documented. So, The chapter will take the form of an extended literature survey for cooling load reduction in automobiles. Predictions

NREL has been a pioneer in this field and has produced through path breaking publications through its able researchers. Since the early 1990's, NREL has been leading this field inspiring the automobile manufacturers to devote more attention to AC systems. The various methods to reduce load are:

5.1.1. Reduction of heat through glazing

Glass covers more than 50 % of the vehicle compartment surface area. Depending on the properties of the glass and its tinting, the glass allows the interiors to soak in the rays of the sun light. The vehicle interiors absorb the rays and heat up. Leather (seats) and plastic (dashboards) are common materials found in the vehicle interiors. These materials have specific heat capacity, c_p values on the higher side and tend to generate heat due to the absorption. There is a lag between the absorption and re-radiation by the interiors. This is a major reason for the higher temperature after vehicle soaking under the sun. Also, the glass traps the radiations from passing into the atmosphere due to the “Greenhouse effect”[37]. Hence, any reduction to the heat load through the glass windows will have considerable significance in the overall heat load reduction exercise.

Shimizu et al in [18] explain the importance of using infra-red reflectance glazing for automotive vehicles. Vehicle window glasses should be able to strike a balance between visibility and reflectance. They provide numerical evidence of the effect of infra-red reflectance glass windows in the place of ordinary glass windows (transmissivity of 0.84). The IR glass can cut down the vehicle cooling load of 600-700 W (approximately 30% of their vehicle cooling load).

Considering the vehicle cooling load calculation for EcoCAR2, the transmissivity of the tinted side window glass and rear windows was 0.67. The infrared glass windows mentioned in the above paragraph were said to have 0.52. The windshield was assumed to have a transmissivity of 0.86. Hence changing the glasses to IR reflectance glass could yield anywhere between 15-25% reductions in the vehicle cooling load. Consequently, the fuel mileage could improve 1.5-2 mpg due to the drop in the accessory electrical load.

More recently, improved windshield glasses have been able to reduce the cooling load even further. A study on commercially available and common automobile windshields done in [1] shows that a reduction of 400 W cooling load could be achieved with specific windshield brands. The study involved 3 windshields supplied by PPG. They were *Solex*, *Solar Green* and *Sungate*. Solex is the standard windshield in the US; *Solar Green* is the standard windshield in Europe whereas *Sungate* is an advanced UV and IR reflecting windshield. *Farrington and Rugh* determined from their test that *Sungate* was able to improve fuel economy by at least 0.5 mpg under specific conditions.

Usage of *Sungate* and IR reflectance glasses on EcoCAR2 could effectively improve the fuel economy by at least 2 mpg. Favorable effects on tail pipe emissions, vehicle weight, better AC performance and passenger thermal comfort are advantages which have not been evaluated.

5.1.2. Reduction of heal load from outside air

Air conditioned by the AC system could be 100 % air recirculated (RA) from the cabin, 100 % outside air (OA) or a mixture of both. The decision is usually taken by the passengers who can control the mixture using controls in the cabin. Automatic mixture control is also possible in most of the modern day vehicles.

Depending on the temperature difference between the cab and the ambient conditions, the outside air might assist or retard the AC system. Under normal conditions, OA is usually hotter than RA and the load on the AC system will be higher. Using only RA for air conditioning is a simple option but cabin air would increasingly become slate. Also, OA will improve air quality by replacing VOC and other contaminants with fresh air. In [1], *Farrington and Rugh* prove that RA lowers the vehicle cooling load considerably. But they also mention the effects of VOC and other contaminants. Possible solutions have been listed in [38] which include filtering RA before re-conditioning.

5.1.3. Reduction through interior design improvements

Radiation passing into the cab through the glass cannot be re-radiated back due to the "Greenhouse effect". This results in heat accumulation inside the cabin and thereby, increasing the temperature of vehicle interiors (including cabin air) while decreasing the thermal comfort of the passenger. Few methods have been suggested to reduce this heat accumulation.

In [25], reduction of the accumulation is countered by using heat pipes under the instrument panel. An experiment conducted by them by building heat pipes under a mockup of the instrument panel. They were able to reduce the temperature of the instrument panel by nearly 20 °C and the cabin air by 9-12 °C. Considering the amount of electronics under the instrument panel in cars nowadays, panel cooling could be doubly beneficial.

Liquid cooled seating was another solution considered by NREL research members. These seats improve the thermal comfort of the passenger while decreasing the load by removing heat from the vehicle interiors.

5.1.4. Reduction through opaque surface

Darker colored materials tend to absorb and radiate more heat than light colored counterparts. Heat load through the vehicle body is the second biggest contributor to vehicle heat load. It is also majorly influenced by the color of the vehicle body. *Shimizu et al* prove that a plate which has been painted black has 42 % more absorptance than a white painted plate [18]. Though it is not possible to prescribe only light colored vehicles, the advantages of such vehicles is unmistakable.

A better solution is solar-reflective coating on the opaque surfaces of the vehicle. A test conducted by NREL suggests that they were able to achieve a reduction of around 7°C on the outer skin of the vehicle. This 7°C reduction resulted in a 1°C reduction in the interior cab temperature.

5.1.5. Reduction through insulation

Insulation of the roof seems to be an intuitive solution for heat load reduction. But, if the temperature is higher in the cab than the ambient, insulation will slow down the heat loss through the roof. Tests run by NREL researchers prove that insulation has little to no impact in the vehicle interior temperatures when the vehicle is sun soaked [25]. Hence, insulation is not a successful solution for heat load reduction.

5.2. Conclusion

All the methods mentioned in the above section could reduce the vehicle heat load of EcoCAR2 by 600-800 W. This value is an educated estimate because of the non- availability of the vehicle. This reduction will be a very significant step in the event of retrofitting commercial passenger vehicles as it provides an opportunity to improve the customer's thermal comfort while reducing the cost of AC equipment.

6. **AC system and Components**

The final task to finish retrofitting the vehicle with an AC system is choosing and installing the AC system components. Fundamentally, choosing and installing an AC system is a complex process with many factors. In comparison to a residential AC system, the choices are less for an automobile. For retrofitting an automotive AC system, the choices are even less due to vehicle modeling constraints. During a vehicle development process, changes to vehicle model can be made if the AC system warrants some. When retrofitting the same vehicle with an AC system, such changes are not possible. Even if they are possible, the changes might be expensive.

The steps involved in this final task are:

1. Type of AC system
2. Type of refrigerant
3. Component selection

Step 2 could be skipped depending on the type of the AC system selected.

6.1. Type of AC system

AC systems could be split into various types depending on the method of heat removal. If this research effort was built upon modeling an AC system for an automobile, the next step would be to determine the best AC cycle/system for an automobile.

However, when retrofitting a vehicle with an AC system, major changes to the vehicle design will not be possible. Hence, the choice of AC system that could be fitted into the *Hybrid-convert* narrows down considerably. The most common type of AC system found in automobiles is the vapor compression refrigeration cycle. Other possible alternatives include:

1. CO_2 refrigeration cycle

2. Thermoelectric cooling

The alternatives, mentioned above, do not require major vehicular design changes. While CO_2 refrigeration cycle uses the same components (size wise) as a R134A vapor compression cycle, thermoelectric cooling's smaller size could replace the vapor compression cycle while providing more space.

Considering the scope of the EcoCAR2 competition, the vapor compression cycle was choice in order to aid the team's design and financial targets. Similar to most conventional passenger vehicles in the market, Chevrolet Malibu has a vapor compression cycle run AC system. The team's financial target dictates the choice of components. The cost of the vehicle is kept under the spending threshold by using sponsor donated components. Adopting alternative technologies like CO_2 refrigeration cycle or thermoelectric cooling will result in purchase of components from markets. Hence, vapor compression cycle was finalized for the EcoCAR2 competition.

6.1.1. Vapor compression cycle

This is the most common cycle used in the automobile air conditioning system. The AC system described previously, in this thesis, run under this cycle.

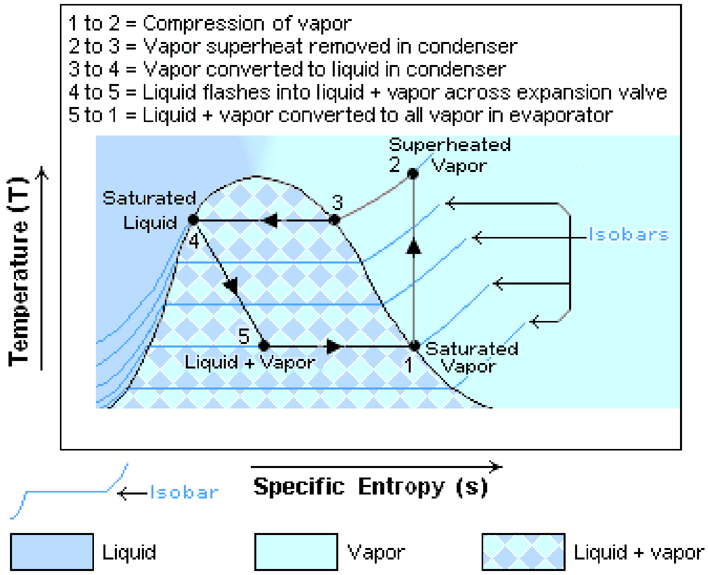

Figure 6. 1. Vapor compression cycle [39]

The state of the refrigerant under different conditions has been provided in the figure above. The TS diagram aids in understanding the theory behind the cooling of the air by the refrigerant. The cycle consists of:

1. Isentropic compression
2. Loss of superheat and then, loss of latent heat to change from vapor to liquid
3. Isentropic expansion
4. Gain latent heat from the cabin air, an isothermal process.

There are 3 main types of vapor compression cycles:

1. Single stage vapor compression cycle:

This is the cycle which has been explained in the earlier sections of this thesis. They have a compressor, evaporator, condenser and an expansion valve.

2. Cascade vapor compression cycle:

Cascade cycle improves the cooling capacity by having another entire AC system coupled with the original AC system. The heat expelled out by the condenser of the system 1 is picked up by the evaporator of AC system 2. To summarize it, the AC system 2 works to cool the refrigerant of AC system 2 thereby improving the possibility of very low air temperatures.

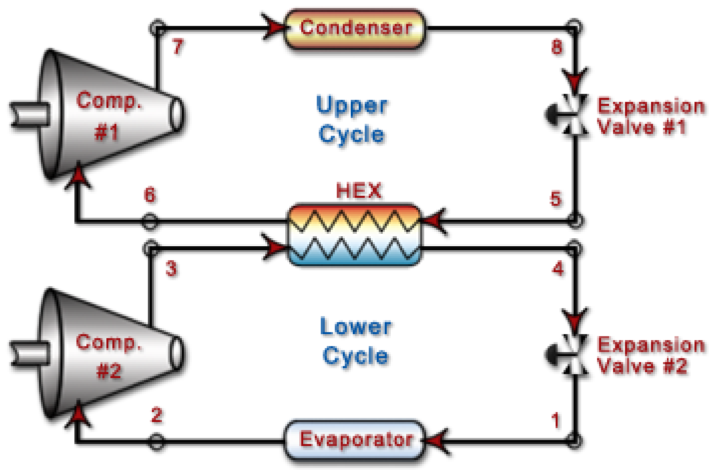

Figure 6. 2. Cascade vapor compression cycle [40]

3. Multi stage vapor compression cycle:

As the name suggests, multi stage involves compression of vapor occurs at more than one stage. Expansion also occurs at equal number of stages as compression. Usually, multi stage compression systems have 2 stages of compression, low-stage and high-stage compression. The vapor is compressed to a lower pressure and then again compressed to a higher pressure by the second stage compressor. The higher pressure vapor is the condensed, expanded twice before sent to the evaporator to be vaporized again. A flash chamber is present in between the 2 expansion valves where saturated vapor is removed from the liquid/vapor mix present after the first expansion.

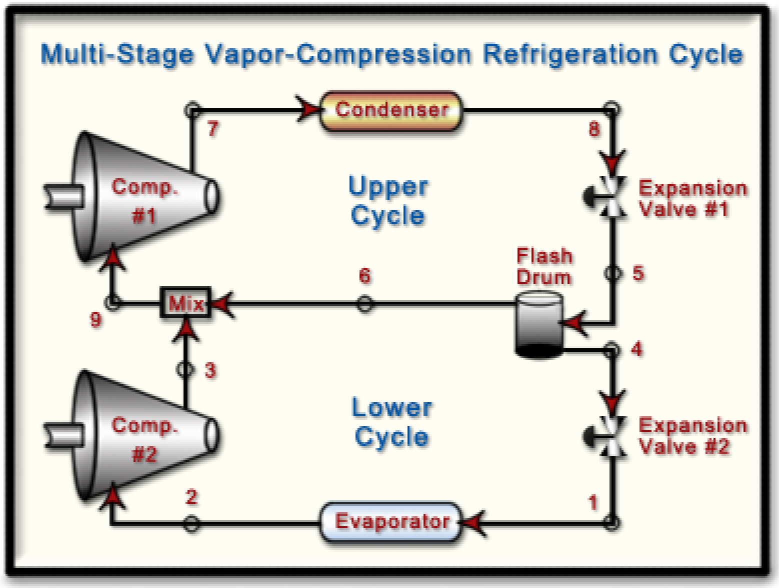

Figure 6. 3. Multi stage vapor compression cycle [41]

Cascade and Multi stage systems provide better cooling. However, they are bulky and expensive systems. One of the main constraints faced in retrofitting AC systems will be space availability. Hence, smaller systems are preferable. From customer's point of view, cost also might be an issue. Multi stage compression systems will be an ideal choice for designers who look for a middle ground between cost and better cooling.

6.2. Type of Refrigerant

The working fluid inside a refrigeration cycle is called a refrigerant. The choice of the refrigerant is a very important step in modeling an AC system because the properties of the refrigerant determine the efficiency of the AC system.

R134 A is the most common refrigerant used in automotive AC systems. R134A was introduced as a substitute to R12/Freon-12. R-12 was banned due under the *Montreal Protocol* as it was found to deplete the ozone layer [42]. There are better alternate refrigerants suggested by experts. However, most of them are dangerous to use (**HFO-1234yf, HFC-152a)** or difficult to operate with (CO_2) [43].

Furthermore, AC components are readily available in the market for R134A. Hence R134A was considered the best choice for EcoCAR2.

6.3. Components

Component selection begins with picking the right sized compressor. Both the heat exchangers are picked as per the temperature difference set by the designer. The expansion valve / orifice tube will be the last equipment to be picked. Installation of the components depends on the available hood space. Component physical sizing is important because of limited hood space available after converting the vehicle into a hybrid.

6.3.1. Compressor

Compressors are the most important component of a compression cycle run AC system. The main function of a compressor is to ensure continuous flow of refrigerant through the AC system. The refrigerant vapor, which carries the heat absorbed across the evaporator, is compressed and converted into a superheated vapor.

Compressors could be split into 2 basic categories [44]:

1. Positive displacement compressors

These compressors increase the pressure by reducing the volume of the chamber. The volume of the chamber is reduced the work input into the compressor. The various compressors which follow such an operating principle are

1. Rotary compressor
2. Reciprocating compressor
3. Scroll compressor
4. Trochoidal compressor

2. Dynamic compressors

These compressors employ the rotation of the members inside the chamber to supply constant angular momentum which increase the vapor refrigerant pressure. Centrifugal compressor falls under this category.

Choosing the right compressor:

The categorization shown below is very basic and there are numerous types of compressors under these basic classifications. ASHRAE's tabulation of various positive displacement compressors will validate the previous statement.

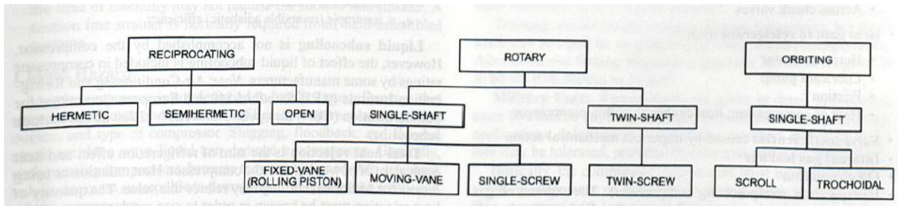

Figure 6. 4. Classification of compressors[44]

The compressor types that are commonly used in the automotive AC system are

1. Axial piston (swash plate) type compressor

2. Vane type compressor

3. Scroll type compressor

ASHRAE Handbook – HVAC Systems and Equipment has an extensive section which explains these compressors in a detailed manner. In normal air conditioning system modeling, the compressors will be evaluated as per various advantages and disadvantages. The compressor would then be selected as per different criteria such as power consumption, COP, discharge pressure and suction pressure.

For a hybrid vehicle, an important criterion to be considered before evaluating advantages of various available compressors is method of power delivery. Conventional automobiles derive power for AC compressor from the engine. The issue with engine driven compressor for hybrid vehicles is that engines of hybrid vehicles run intermittently as per the load requirement. This means the AC system switches whenever the engine switches off. Even in conventional automobiles, the compressor speed varies as the engine speed varies. This results in an inconsistent AC performance. Variable speed compressors that adjust according to engine speed variations are viable alternative to ordinary compressors. Even these compressors are not useful for hybrid vehicles.

Electric compressors are an ideal solution for this problem. Apart from running consistently, these compressors have various advantages. *Ken Matsunaga* and *Kiwama Inui* have empirically proved the superiority of electric compressor in their presentation " *Electric Inverter A/C System for TOYOTA PRIUS Hybrid Vehicle"*[45].

<u>1. Thermal comfort of passengers:</u>

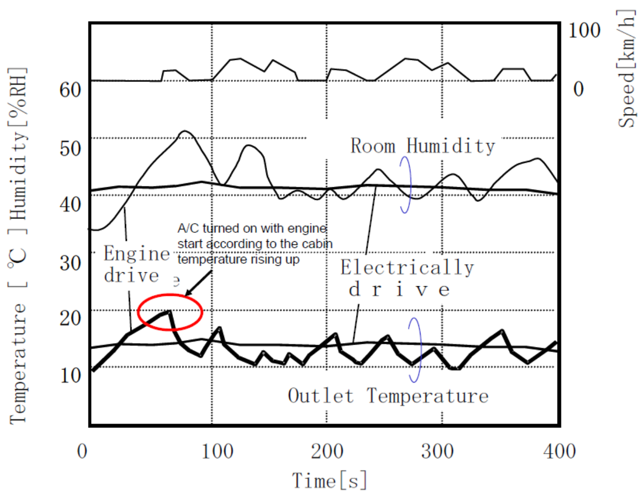

Figure 6. 5. Comparison between belt driven compressors & electrical compressors [45]

Conventional compressors do not provide cooling during stoppage due to engine idling. It is evident from the picture above that there is a peak in the temperature whenever the speed of the engine drops. Electrically driven compressors provide constant consistent cooling irrespective of the speed of the engine. The belt driven compressors which derive power from the engine have an erratic curve due to the dependence on engine power for running the AC system.

2. Fuel economy:

Using the below figure, Mr. Matsunaga and Mr. Inui prove that the new Prius with an electrically driven compressor has a 19 % improvement on fuel economy.

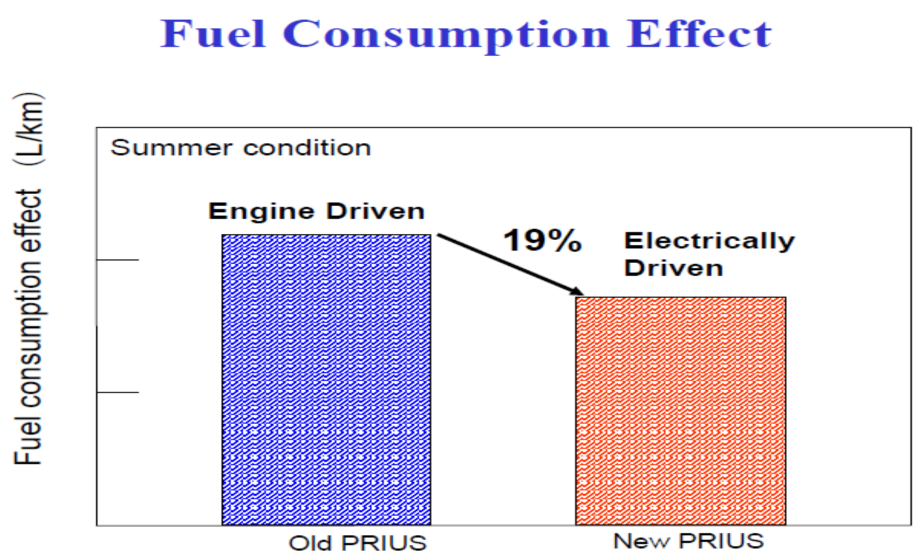

Figure 6. 6. Fuel economy comparison [45]

3. Space and Weight:

An electrically driven compressor provides mass reduction because of the less number of components relative to a belt driven compressor.

Similarly, the space occupied by an electrically driven compressor is much lesser than the conventional compressor. Also, electrically driven compressors are hermetically sealed with motors directly connected to the compressor. Conventional compressors require to be positioned near the engine in some specific position to facilitate power transmission through the belt. Since the motor powering the compressor is locked in along with the compressor compartment (hermetic compressor), there is more flexibility in its positioning under the hood. This is a major advantage for retrofitting AC system components under the hood.

Electrical Compressor Selection:

Electrically driven compressors are relatively new in the market and hence, the selection of compressors in the market is very low. As a result of this, using the usual approach of considering the selection criterion and advantages to select the compressor is not entirely practical.

Also, there are 3 more important factors to be considered while retrofitting a vehicle with a compressor. They are

1. Size: Space available under the hood after conversion into hybrid is much lesser when compared to space before the conversion due to the addition of more components. Also, the layout is completely different after conversion. Hence, the space designated for the compressor originally will not be available. Therefore, the compressor has to be selected as per the space available under the new layout.
2. Weight: Unnecessary weight gain due to a compressor will retard the performance of the vehicle. Additional weight could have an impact on the improved fuel economy.
3. Cost: Since the entire endeavor of retrofitting is geared towards customer satisfaction, cost is a major factor.

Hence, instead of the traditional approach of considering selection criteria and advantages, electrically driven AC compressors for a *Hybrid-convert* (EcoCAR2) was chosen by evaluating them according to the above factors.

Electric compressor for EcoCAR2:

Denso Corporation, *Sanden* and *Delphi* are leading manufacturers of automotive compressors in the Unites States. Among the three, *Denso* and *Sanden* have a selection of electric compressors in the market. *Delphi* is in the process of developing electrical compressors and no information of *Delphi* electrical compressors geared for an electric vehicle is available in the market [46], [47]. Hence, the comparison was between *Denso* and *Sanden* electric compressors. To pick the compressors, specifications needed to be calculated as per the tonnage requirement. Compressors are specified as per the volume flow rate (cc) in the manufacturer catalogs. Hence, the volume flow for 2175 W cooling requirement was calculated. Continuing from calculation on page 36,

$$m_{R134A} = 0.07 \text{ kg/s};$$

The required volume rate can be calculated from the density and mass flow rate. From the table 5 on page 40,

$$\text{Volume} = \text{mass (kg/s)} / \text{Density (kg/m}^3\text{)}$$

$$= 0.07/\ 150.47$$

$$\text{Volume} = 465 \text{ cc/s}.$$

The space available under the hood was calculated from the CAD file developed by the EcoCAR2 team. This file cannot be displayed due to copyright reasons. An approximate representation of the engine bay in a block diagram has been provided below.

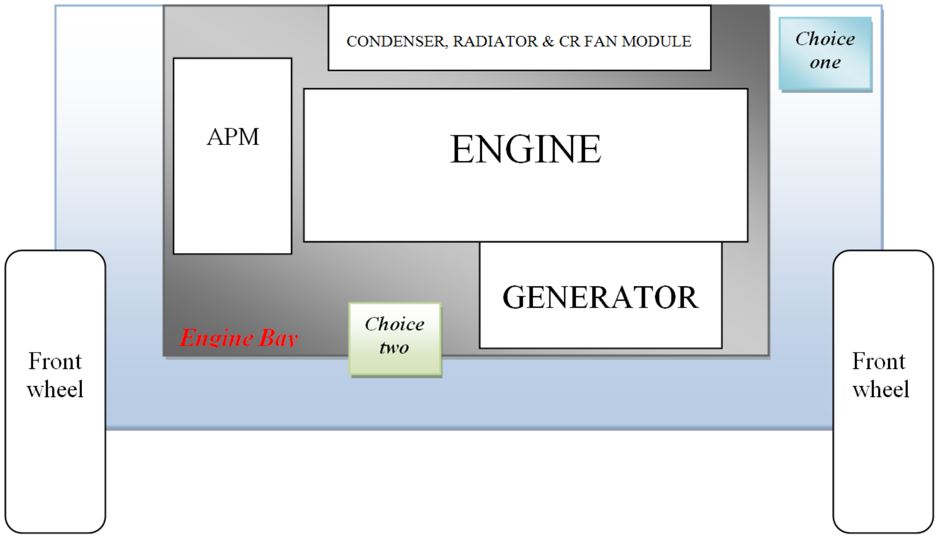

Figure 6. 7. Under the hood

The black shaded area represents the engine bay and the rest of the area under the hood is shaded blue. The areas marked *Choice one* and *Choice two* under the hood are spaces available and selected for positioning the compressor. A minimum area of 0.3 m x 0.3 m is available in either of these spaces.

Choice one was selected for the following reasons:

1. Easier wiring along the periphery of the engine bay.

2. To avoid refrigerant lines travelling over or near the engine. There is a chance that the refrigerant line from the compressor could pick up more heat from the engine while passing nearby. This could result in a drop in the evaporator performance because the refrigerant will not have had enough fall in temperature to pick up heat from the air flowing across the evaporator.

3. Since it is present outside the engine bay, installation is easier.

Once the position was selected and the space available was measured, compressor was selected from the choices available in the market.

Table 7. Compressor selection

Contents	Requirements	*Denso ES18* [45]	*Sanden HBC75115* [48]
Flow rate	465 cc/s	18 cc/rev @ 7500 rpm	15 cc/rev @ 6000rpm
Volume	0.3 x 0.3 x 0.4 m^3	Φ 0.109 x 0.182 m^3	NA
Weight	-	4.8 kg	9 kg

The cost of the compressor is conspicuously missing from the above table. Cost was not available for these models. Similarly, the size of *Sanden* compressor was also not available. However, from the size of the *Denso* compressor, it could be argued that the size of the *Sanden* compressor will not exceed the maximum volume allowed. There is no requirement for the 'Weight'. The lightest possible compressor was considered the best choice.

Considering the capacity of both the compressors, volume flow rate of *ES18* is 2250 cc/s and *HBC75115* is 1500 cc/s. This is more than the requirement specified. Any search to find a smaller capacity was futile. The Denso compressor is clearly lighter and satisfies the space requirement too. Moreover, this model has been used in the Toyota Prius and has been very successful. Hence, the **Denso *ES18*** was the compressor model suggested for EcoCAR2 *hybrid-convert.*

6.3.2. Heat Exchangers

The evaporator and the condenser are the heat exchangers involved in an automotive AC system. The operation of the evaporator and the condenser are similar. Refrigerant flowing through the evaporator picks up heat from the air flowing over the face of the evaporator. The heat picked up across the evaporator is dropped across the condenser with the aid of a draft of air flowing over it. Common automotive AC heat exchanger designs are [3]

(i) Tube and fin with mechanically bonded fins

(ii) Serpentine tube and fin

(iii) Parallel flow

Condensers:

Parallel flow condensers are the most commonly found condensers in automotive applications. Parallel flow condensers are the most efficient of all the 3 types and condenses

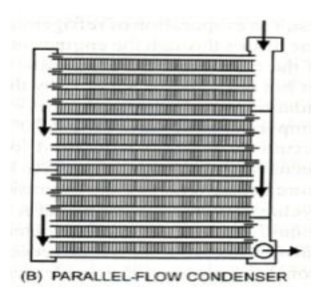

Figure 6. 8. Parallel Flow Condenser

the refrigerant by breaking it up into tiny streams enabling rapid heat transfer. Steven Daly in the book "*Automotive Air Conditioning and Climate Control Systems*" observes the successful design. To quote him "the key to the design is the header tanks/manifolds fitted to the sides of the core allowing the flow to break up into small streams".

Hence, the design choice was a parallel flow condenser. The next step was calculating the required condenser specifications. Using values from the calculations in page 32,

$$m_{R134A} = 0.07 \text{ kg/s}$$

$$h_2 = 453.20 \text{ kJ/kg};\ h_3 = 374.02 \text{ kJ/kg.}$$

$$Q_{out} = 0.07\ (453.20 - 374.02)$$

$$Q_{out} = 5{,}542 \text{ W.}$$

Required area of the condenser, $A = Q_{out} / UF\Delta T_m$

$$U = 1/\ [(1/h_{hot}) + (1/h_{cold})]$$

$$U = 3000 \text{ W/m}^2\text{.K}$$

$$\Delta T_m = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1/\Delta T_2)$$

$$\Delta T_2 = 4°C;\ \Delta T_1 = 20°C;$$

An assumption was made about the temperature of the air under the hood; the temperature of air before and after was assumed to be 80°C and 88°C respectively.

$$\Delta T_m = 16/1.6$$

$$\Delta T_m = 10\ °C$$

Using the above values to determine the required area,

$$A = 0.20 \text{ m}^2$$

And the flow rate required across the condenser, $m_{air} = 5{,}542.60/\ [(1002.32)\ (8)]$

m_{air} = .69 kg/s or 1.44 cfm.

To summarize, with the following specifications, the condenser was chosen.

Area = 0.20 m^2; Q_{out} = 5.5 kW.

Evaporators:

The theory behind the design of automotive condensers and evaporators are very similar. Similar to condensers, parallel flow exchangers are the most common evaporator. Calculations, similar to condenser, were done to determine the evaporator specifications.

Q_{out} = 2175 W.

Required area of the condenser, $A = Q_{out}/ UF\Delta T_m$

$U = 1/ [(1/h_{hot}) + (1/h_{cold})]$

U = 2505.04 W/m^2.K

$\Delta T_m = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1/\Delta T_2)$

ΔT_2 = 8°C; ΔT_1 = 36°C;

An assumption was made about the temperature of the air flowing and from the cabin; the temperature of air before and after was assumed to be 49°C and 20°C respectively.

ΔT_m = 17.5 °C

Using the above values to determine the required area,

A = .054 m^2

And the flow rate required across the evaporator, m_{air}= 2175/ [(1005) (29)]

m_{air}= 0.08 kg/s

The evaporator can be chosen with the following specification.

A = .054 m^2

6.3.3. Expansion Valve

The final important component in a vapor refrigeration cycle is an expansion valve. Expansion valves serve the following purposes:

1. Control the refrigerant inflow into the evaporator. Without the expansion valve, the refrigerant will flow at a very high temperature into the evaporator. Consequently, the refrigerant will not be able to pick up the heat from the air flowing across the evaporator.

2. Separate the high and pressure parts of the AC system.

There are 2 types of expansion valves used in automotive air conditioning systems

i. Thermostatic expansion valve

ii. Fixed orifice valve/tube.

Orifice tubes are very popular with automobiles because of their high reliability and low cost [3]. Orifice tubes guarantee a continuous stream of refrigerant flow into the compressor from the suction line accumulator. The accumulator is a device that will be used along with an orifice tube to avoid floodback from the compressor.

General Motors use orifice tubes in their cars and hence, a fixed orifice tube was suggested for EcoCAR2.

6.4. Component location

For a *hybrid-convert*, compressor selection is the main issue. This is because the compressor is the only component which does not have a specified space in the new layout. The other 3 components have positions which are unlikely to be altered.

The position of a condenser is in the front of the hood near the grill. The condenser needs to be in the front because of the air flow required to cool the condenser down. One issue with condenser and radiator placed one after another is that condenser heats up the air flowing into the radiator and thereby increases the load on the engine cooling system. One alternate solution to avoid this is placing the condenser and radiator adjacent to each other. The advantage of reduction in heat load and the consequent component size reduction helps such a cause.

The evaporator needs to be placed inside the passenger compartment to ensure quick and clear passage of cold air into the same.

6.5. Conclusion

The central idea of this chapter was to explain the selection of components for a *Hybrid-convert* with EcoCAR used as an example. An important point to be noted from this chapter is that selection of components for retrofitting purposes is very different. While modeling an AC system for a new vehicle, the HVAC designer designs components in the traditional method by following handbooks. The traditional handbook approach will lead to specifications that are not useful in a market. For instance, a condenser design would be dominated by terms such as fin size, number of tubes, etc., At the other extreme, a technician who repairs an AC system would pick the components as per models available. Both are two extremes. For any person who wants to understand the theory and model a new AC system for an existing vehicle, the challenge is to understand how to utilize numerical calculations for specifying components based on cooling requirements. The method adopted here and the calculations performed will aid anyone intending to select components for a specific cooling load from the market.

Also, the electric compressor for automotive AC systems is still in infancy and the choices available in the market are limited. With the steady growth of hybrid vehicles and customer awareness about electrical compressors in hybrid vehicles increasing, the market is set to flourish.

7. Future work

Automotive AC systems and their adverse impact have gained attention in the last 2 decades. Research and analysis of improved air conditioning systems which paves way for effective cooling without being a parasite is the need of the hour.

Future research:

An issue which served innumerous roadblocks during the entire research was the dearth of quality papers about automotive AC modeling and information on AC component selection. Large amount of time was spent on combining together information from various sources to understand both these sections and applying it to the above research. Heat load reduction also falls into this category to some extent. However, the 3 papers listed under the literature review section provide a good amount of information.

Better AC modeling for commercial conversions:

Modeling for EcoCAR2 and component selection simplifies the retrofitting venture because of the team targets and various other constraints like finance, resources and non-availability of the vehicle. Attempting this as a commercial venture will provide a range of opportunities for the designer to implement a system which would be better than the system developed here. Thermoelectric systems, which require less space than conventional AC systems, could be a viable option. Similarly, multi stage or cascade vapor cycle run AC systems can be designed for the vehicle. These systems require more space but provide more cooling and very efficient. Dual evaporators, zoned cooling, smart AC systems which are controlled by intelligent algorithms could add value to a hybrid vehicle conversion, since the primary expectation of the customer is to improve his vehicle.

Required initiative from EPA and vehicle manufacturers

With the consumer looking for safer and more comfortable vehicles, it is the automobile manufacturer's duty to devote more resources for AC improvement. Finally, development and modeling of an AC system should not be independent of vehicle performance analysis as there is a marked decrease in performance due to it. The federal regulation making SFTP a standard test is a good starting point. However, SFTP checks emissions and not fuel economy. Considering both will improve passenger safety and comfort while providing better vehicle information to the customer.

8. Conclusion

This research effort was initiated to model an AC system and select AC components for the EcoCAR2 vehicle. Soon after the initial literature review and gaining knowledge about the huge market for hybrid vehicle conversions, the scope of the thesis was altered to include *hybrid-converts* in general. To conclude,

1. A vehicle heat load of 2175 W was calculated under the assumed vehicle and ambient conditions for the EcoCAR2 vehicle.
2. To produce cooling of 2175 W or 0.6 TR, the compressor needs more than 3.5 kW of electrical power supplied to it.
3. A performance analysis of the vehicle was conducted using ADVISOR, a component simulation and performance analysis tool created by NREL. Six tests with the electrical load being added increments of 700 W to the vehicle model were run and the obtained results were analyzed to identify the threshold.
4. With the threshold obtained, methods to reduce the heat load were discussed and the possibility of reducing the peak heat load to a reasonable value was looked at.
5. With the final heat load, calculations were performed to create specifications of required components. Using the values obtained from the calculation, a Denso compressor of 18 cc/rev, a condenser of 0.20 sq.m area, a fixed orifice tube, an accumulator and an evaporator of 0.054 sq.m. were selected to be retrofitted to EcoCAR2.

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APPENDICES

Appendix A

SOLAR IRRADIATION DATA

July 21st TMY data from NSRDB

Time (HH:MM)	Dry-bulb (°C)	R. Humidity (%)
24:00:00	28.9	72
1:00	28.9	70
2:00	28.9	70
3:00	28.9	72
4:00	28.9	70
5:00	28.9	65
6:00	28.3	65
7:00	28.3	63
8:00	30.6	55
9:00	32.8	49
10:00	35	37
11:00	36.7	33
12:00	37.8	30
13:00	39.4	25
14:00	40.6	23
15:00	40.6	23
16:00	41.7	21
17:00	41.7	20
18:00	42.2	19
19:00	41.1	19
20:00	39.4	24
21:00	37.8	30
22:00	36.7	32
23:00	35	36
24:00:00	34.4	37

Interpolated & Corrected data

Time of the day	Interpolated Data (°C)	Corrected Temp (°C)	Time of the day	Interpolated Data (°C)	Corrected Temp (°C)	Time of the day	Interpolated Data (°C)	Corrected Temp (°C)
0.00	32.79	34.09	8.00	30.44	31.74	16.25	40.03	41.33
0.25	32.10	33.40	8.25	30.76	32.06	16.50	40.12	41.42
0.50	31.46	32.76	8.50	31.10	32.40	16.75	40.19	41.49
0.75	30.88	32.18	8.75	31.43	32.73	17.00	40.23	41.53
1.00	30.35	31.65	9.00	31.78	33.08	17.25	40.26	41.56
1.25	29.87	31.17	9.25	32.13	33.43	17.50	40.27	41.57
1.50	29.44	30.74	9.50	32.48	33.78	17.75	40.25	41.55
1.75	29.06	30.36	9.75	32.84	34.14	18.00	40.21	41.51
2.00	28.73	30.03	10.00	33.20	34.50	18.25	40.15	41.45
2.25	28.44	29.74	10.25	33.56	34.86	18.50	40.06	41.36
2.50	28.19	29.49	10.50	33.92	35.22	18.75	39.95	41.25
2.75	27.98	29.28	10.75	34.28	35.58	19.00	39.81	41.11
3.00	27.82	29.12	11.00	34.64	35.94	19.25	39.65	40.95
3.25	27.69	28.99	11.25	34.99	36.29	19.50	39.46	40.76
3.50	27.59	28.89	11.50	35.34	36.64	19.75	39.25	40.55
3.75	27.53	28.83	11.75	35.69	36.99	20.00	39.01	40.31
4.00	27.51	28.81	12.00	36.02	37.32	20.25	38.74	40.04
4.25	27.52	28.82	12.25	36.36	37.66	20.50	38.45	39.75
4.50	27.55	28.85	12.50	36.68	37.98	20.75	38.13	39.43
4.75	27.62	28.92	12.75	36.99	38.29	21.00	37.79	39.09
5.00	27.71	29.01	13.00	37.30	38.60	21.25	37.41	38.71
5.25	27.83	29.13	13.25	37.59	38.89	21.50	37.01	38.31
5.50	27.97	29.27	13.50	37.88	39.18	21.75	36.58	37.88
5.75	28.14	29.44	13.75	38.15	39.45	22.00	36.12	37.42
6.00	28.33	29.63	14.00	38.41	39.71	22.25	35.63	36.93
6.25	28.53	29.83	14.25	38.65	39.95	22.50	35.12	36.42
6.50	28.76	30.06	14.50	38.88	40.18	22.75	34.57	35.87
6.75	29.00	30.30	14.75	39.09	40.39	23.00	34.00	35.30
7.00	29.26	30.56	15.00	39.29	40.59	23.25	33.40	34.70
7.25	29.54	30.84	15.25	39.48	40.78	23.50	32.77	34.07
7.50	29.83	31.13	15.50	39.64	40.94	23.75	32.11	33.41
7.75	30.13	31.43	15.75	39.79	41.09	24.00	31.42	32.72
			16.00	39.92	41.22			

Appendix B

Heat load calculation between 1 – 4 PM

			Side Glass			
			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
4.00	28.00	41.20	0.72	3.72	-13.20	-35.32
4.15	64.96	41.30	0.72	4.30	23.66	73.25
4.30	57.11	41.40	0.72	3.88	15.71	43.91
4.45	55.47	41.50	0.72	3.77	13.97	37.91
5.00	55.06	41.50	0.72	3.74	13.56	36.52
5.15	54.31	41.60	0.72	3.68	12.71	33.68
5.30	52.67	41.60	0.72	3.56	11.07	28.35
5.45	52.17	41.60	0.72	3.51	10.57	26.74
6.00	52.00	41.50	0.72	3.51	10.50	26.54
			Roof Calculation			
PM			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
4.00	28.00	41.20	1.70	5.11	-13.20	-114.63
4.15	64.96	41.30	1.70	5.91	23.66	237.77
4.30	57.11	41.40	1.70	5.34	15.71	142.51
4.45	55.47	41.50	1.70	5.18	13.97	123.04
5.00	55.06	41.50	1.70	5.14	13.56	118.53
5.15	54.31	41.60	1.70	5.06	12.71	109.33
5.30	52.67	41.60	1.70	4.89	11.07	92.03
5.45	52.17	41.60	1.70	4.83	10.57	86.81
6.00	52.00	41.50	1.70	4.82	10.50	86.13
				Doors Calculation		
			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
4.00	28.00	41.20	1.54	3.77	-13.20	-76.72
4.15	64.96	41.30	1.54	4.37	23.66	159.13
4.30	57.11	41.40	1.54	3.94	15.71	95.38
4.45	55.47	41.50	1.54	3.83	13.97	82.34
5.00	55.06	41.50	1.54	3.80	13.56	79.33
5.15	54.31	41.60	1.54	3.74	12.71	73.17
5.30	52.67	41.60	1.54	3.61	11.07	61.59
5.45	52.17	41.60	1.54	3.57	10.57	58.10
6.00	52.00	41.50	1.54	3.56	10.50	57.64

HEAT LOSS

Through Radiatiom

			ε_{steel}	ε_{glass}	σ	Tcab^4-Tamb^4	E
4.00	28.00	41.20	0.97	0.94	0.0000000540	-1537421003.29	-158.57
4.15	64.96	41.30	0.97	0.94	0.0000000540	3287550592.83	339.08
4.30	57.11	41.40	0.97	0.94	0.0000000540	2104448192.42	217.05
4.45	55.47	41.50	0.97	0.94	0.0000000540	1857399358.48	191.57
5.00	55.06	41.50	0.97	0.94	0.0000000540	1799244825.40	185.57
5.15	54.31	41.60	0.97	0.94	0.0000000540	1681408288.29	173.42
5.30	52.67	41.60	0.97	0.94	0.0000000540	1453692773.73	149.93
5.45	52.17	41.60	0.97	0.94	0.0000000540	1383979960.49	142.74
6.00	52.00	41.50	0.97	0.94	0.0000000540	1373622409.21	141.68

By Conduction

Windshield

			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
4.00	28.00	41.20	0.88	3.28	-13.20	-38.16
4.15	64.96	41.30	0.88	3.80	23.66	79.15
4.30	57.11	41.40	0.88	3.43	15.71	47.44
4.45	55.47	41.50	0.88	3.33	13.97	40.96
5.00	55.06	41.50	0.88	3.31	13.56	39.46
5.15	54.31	41.60	0.88	3.25	12.71	36.39
5.30	52.67	41.60	0.88	3.14	11.07	30.63
5.45	52.17	41.60	0.88	3.11	10.57	28.90
6.00	52.00	41.50	0.88	3.10	10.50	28.67

Rear Window

			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
4.00	28.00	41.20	0.86	3.17	-13.20	-35.87
4.15	64.96	41.30	0.86	3.67	23.66	74.40
4.30	57.11	41.40	0.86	3.32	15.71	44.59
4.45	55.47	41.50	0.86	3.22	13.97	38.50
5.00	55.06	41.50	0.86	3.20	13.56	37.09
5.15	54.31	41.60	0.86	3.14	12.71	34.21
5.30	52.67	41.60	0.86	3.04	11.07	28.80
5.45	52.17	41.60	0.86	3.00	10.57	27.16
6.00	52.00	41.50	0.86	3.00	10.50	26.95

Glass Calculation

Windshield

			A, sq.m	U, W/Sq.m. K	CLTDc, °C	Q, W	SC	SCL	Q, W
4.00	28.00	41.20	0.88	3.28	11.30	32.67	0.95	82.00	68.55
4.15	64.96	41.30	0.88	3.80	-26.16	-87.51	0.95	70.50	58.94
4.30	57.11	41.40	0.88	3.43	-18.31	-55.29	0.95	70.50	58.94
4.45	55.47	41.50	0.88	3.33	-16.67	-48.87	0.95	70.50	58.94
5.00	55.06	41.50	0.88	3.31	-16.76	-48.77	0.95	59.00	49.32
5.15	54.31	41.60	0.88	3.25	-16.01	-45.84	0.95	43.00	35.95
5.30	52.67	41.60	0.88	3.14	-14.37	-39.76	0.95	43.00	35.95
5.45	52.17	41.60	0.88	3.11	-13.87	-37.92	0.95	43.00	35.95
6.00	52.00	41.50	0.88	3.10	-13.70	-37.41	0.95	27.00	22.57

Rear Window

			A, sq.m	U, W/Sq.m. K	CLTDc, °C	Q, W	SC	SCL	Q, W
4.00	28.00	41.20	0.86	3.17	11.30	30.70	0.67	679.00	389.42
4.15	64.96	41.30	0.86	3.67	-26.16	-82.26	0.67	652.50	374.22
4.30	57.11	41.40	0.86	3.32	-18.31	-51.97	0.67	652.50	374.22
4.45	55.47	41.50	0.86	3.22	-16.67	-45.94	0.67	652.50	374.22
5.00	55.06	41.50	0.86	3.20	-16.76	-45.84	0.67	626.00	359.02
5.15	54.31	41.60	0.86	3.14	-16.01	-43.09	0.67	490.50	281.31
5.30	52.67	41.60	0.86	3.04	-14.37	-37.38	0.67	490.50	281.31
5.45	52.17	41.60	0.86	3.00	-13.87	-35.64	0.67	490.50	281.31
6.00	52.00	41.50	0.86	3.00	-13.70	-35.16	0.67	355.00	203.60

Side Glass

			A, sq.m	U, W/Sq.m. K	CLTDc, °C	Q, W	SC	SCL,NF	Qnf, W	SCL,SF	Qsf,W	Q
4.00	28.00	41.20	0.72	3.72	11.30	30.23	0.67	98.00	23.64	89.00	21.47	45.10
4.15	64.96	41.30	0.72	4.30	-26.16	-80.99	0.67	108.00	26.05	75.50	18.21	44.26
4.30	57.11	41.40	0.72	3.88	-18.31	-51.17	0.67	108.00	26.05	75.50	18.21	44.26
4.45	55.47	41.50	0.72	3.77	-16.67	-45.23	0.67	108.00	26.05	75.50	18.21	44.26
5.00	55.06	41.50	0.72	3.74	-16.76	-45.14	0.67	118.00	28.46	62.00	14.95	43.42
5.15	54.31	41.60	0.72	3.68	-16.01	-42.43	0.67	112.00	27.01	44.50	10.73	37.75
5.30	52.67	41.60	0.72	3.56	-14.37	-36.80	0.67	112.00	27.01	44.50	10.73	37.75
5.45	52.17	41.60	0.72	3.51	-13.87	-35.09	0.67	112.00	27.01	44.50	10.73	37.75
6.00	52.00	41.50	0.72	3.51	-13.70	-34.62	0.67	106.00	25.57	27.00	6.51	32.08

HEAT GAIN

Time	Tcab, °C	Tamb, °C		Roof Calculation						
PM				A, sq.m	U, W/Sq.m. K	CLTDc, °C	Q, W			
4.00	50.0000	41.20		1.70	4.62	27.30	214.22			
4.15	64.9627	41.30		1.70	5.91	9.84	98.85			
4.30	57.1115	41.40		1.70	5.34	17.69	160.45			
4.45	55.4688	41.50		1.70	5.18	19.33	170.27			
5.00	55.0578	41.50		1.70	5.14	17.24	150.74			
5.15	54.3090	41.60		1.70	5.06	13.99	120.35			
5.30	52.6733	41.60		1.70	4.89	15.63	129.87			
5.45	52.1676	41.60		1.70	4.83	16.13	132.52			
6.00	52.0016	41.50		1.70	4.82	12.30	100.87			
				Doors Calculation						
				A, sq.m	U, W/Sq.m. K	CLTDc (NF), °C	CLTDc (SF), °C	Qnf, W	Qsf, W	Q, W
4.00	28.00	41.20		1.54	3.41	24.30	27.30	63.81	71.69	135.50
4.15	64.96	41.30		1.54	4.37	-17.66	-12.16	-59.39	-40.90	-100.29
4.30	57.11	41.40		1.54	3.94	-9.81	-4.31	-29.78	-13.09	-42.87
4.45	55.47	41.50		1.54	3.83	-8.17	-2.67	-24.08	-7.87	-31.94
5.00	55.06	41.50		1.54	3.80	-7.76	-4.76	-22.70	-13.92	-36.61
5.15	54.31	41.60		1.54	3.74	-7.01	-6.01	-20.18	-17.30	-37.47
5.30	52.67	41.60		1.54	3.61	-5.37	-4.37	-14.94	-12.16	-27.11
5.45	52.17	41.60		1.54	3.57	-4.87	-3.87	-13.38	-10.63	-24.01
6.00	52.00	41.50		1.54	3.56	-4.70	-5.70	-12.90	-15.65	-28.55

Time	Δt, °C
4.15	14.96
4.30	-7.85
4.45	-1.64
5.00	-0.41
5.15	-0.75
5.30	-1.64
5.45	-0.51
6.00	-0.17

Time	Net Gain
4.00	1405.66
4.15	-737.57
4.30	-154.32
4.45	-38.61
5.00	-70.35
5.15	-153.67
5.30	-47.51
5.45	-15.59
6.00	-144.23

Appendix C

Heat load calculation between 11 AM – 1 PM

HEAT GAIN

Time	Tcab, °C	Tamb, °C		Roof Calculation						
				A, sq.m	U, W/Sq.m. K	CLTDc, °C	Q, W			
11.00	**50.0000**	35.90		1.70	5.19	14.30	126.25			
11.15	**49.7291**	36.30		1.70	5.13	18.57	161.97			
11.30	**48.3300**	36.60		1.70	4.96	19.97	168.38			
11.45	**48.0024**	37.00		1.70	4.88	20.30	168.42			
12.00	**48.0529**	37.30		1.70	4.85	24.25	200.04			
12.15	**48.2073**	37.70		1.70	4.83	26.59	218.13			
12.30	**49.1234**	38.00		1.70	4.89	25.68	213.64			
12.45	**49.5772**	38.30		1.70	4.91	25.22	210.58			
13.00	**49.8581**	38.60		1.70	4.91	27.44	229.02			
				Doors Calculation						
				A, sq.m	U, W/Sq.m. K	CLTDc (NF), °C	CLTDc (SF), °C	Qnf, W	Qsf, W	Q, W
11.00	50.00	35.90		1.54	3.84	39.30	-0.70	116.11	-2.07	114.04
11.15	49.73	36.30		1.54	3.79	-7.93	2.57	-23.14	7.50	-15.64
11.30	48.33	36.60		1.54	3.66	-6.53	3.97	-18.42	11.20	-7.22
11.45	48.00	37.00		1.54	3.61	-6.20	4.30	-17.22	11.93	-5.29
12.00	48.05	37.30		1.54	3.59	-4.75	7.25	-13.12	20.01	6.89
12.15	48.21	37.70		1.54	3.56	-3.91	9.09	-10.73	24.96	14.23
12.30	49.12	38.00		1.54	3.62	-4.82	8.18	-13.43	22.77	9.34
12.45	49.58	38.30		1.54	3.63	-5.28	7.72	-14.74	21.58	6.83
13.00	49.86	38.60		1.54	3.63	-4.56	9.44	-12.73	26.37	13.64

					Glass Calculation									
					Windshield		E							
				A, sq.m	**U, W/Sq.m. K**	**CLTDc, °C**	**Q, W**		**SC**	**SCL**	**Q, W**			
11.00	50.00	35.90		**0.88**	**3.34**	**-14.70**	**-43.20**		**0.95**	**261.00**	**218.20**			
11.15	49.73	36.30		**0.88**	**3.30**	**-13.93**	**-40.44**		**0.95**	**197.00**	**164.69**			
11.30	48.33	36.60		**0.88**	**3.19**	**-12.53**	**-35.17**		**0.95**	**197.00**	**164.69**			
11.45	48.00	37.00		**0.88**	**3.14**	**-12.20**	**-33.70**		**0.95**	**197.00**	**164.69**			
12.00	48.05	37.30		**0.88**	**3.12**	**-11.75**	**-32.28**		**0.95**	**133.00**	**111.19**			
12.15	48.21	37.70		**0.88**	**3.10**	**-10.91**	**-29.78**		**0.95**	**127.00**	**106.17**			
12.30	49.12	38.00		**0.88**	**3.15**	**-11.82**	**-32.75**		**0.95**	**127.00**	**106.17**			
12.45	49.58	38.30		**0.88**	**3.16**	**-12.28**	**-34.12**		**0.95**	**127.00**	**106.17**			
13.00	49.86	38.60		**0.88**	**3.16**	**-11.56**	**-32.11**		**0.95**	**121.00**	**101.16**			
					Rear Window		w							
				A, sq.m	**U, W/Sq.m. K**	**CLTDc, °C**	**Q, W**		**SC**	**SCL**	**Q, W**			
11.00	50.00	35.90		**0.86**	**3.23**	**-14.70**	**-40.61**		**0.67**	**121.00**	**69.40**			
11.15	49.73	36.30		**0.86**	**3.19**	**-13.93**	**-38.01**		**0.67**	**127.00**	**72.84**			
11.30	48.33	36.60		**0.86**	**3.08**	**-12.53**	**-33.06**		**0.67**	**127.00**	**72.84**			
11.45	48.00	37.00		**0.86**	**3.03**	**-12.20**	**-31.68**		**0.67**	**127.00**	**72.84**			
12.00	48.05	37.30		**0.86**	**3.02**	**-11.75**	**-30.34**		**0.67**	**133.00**	**76.28**			
12.15	48.21	37.70		**0.86**	**3.00**	**-10.91**	**-27.99**		**0.67**	**197.00**	**112.98**			
12.30	49.12	38.00		**0.86**	**3.04**	**-11.82**	**-30.78**		**0.67**	**197.00**	**112.98**			
12.45	49.58	38.30		**0.86**	**3.05**	**-12.28**	**-32.07**		**0.67**	**197.00**	**112.98**			
13.00	49.86	38.60		**0.86**	**3.05**	**-11.56**	**-30.18**		**0.67**	**261.00**	**149.69**			
					Side Glass		n			**n**		**S**		
				A, sq.m	**U, W/Sq.m. K**	**CLTDc, °C**	**Q, W**		**SC**	**SCL,NF**	**Qnf, W**	**SCL,SF**	**Qsf,W**	**Q**
11.00	50.00	35.90		**0.72**	**3.78**	**-14.70**	**-39.98**		**0.67**	**124.00**	**29.91**	**210.00**	**50.65**	**80.56**
11.15	49.73	36.30		**0.72**	**3.73**	**-13.93**	**-37.43**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
11.30	48.33	36.60		**0.72**	**3.61**	**-12.53**	**-32.55**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
11.45	48.00	37.00		**0.72**	**3.55**	**-12.20**	**-31.19**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
12.00	48.05	37.30		**0.72**	**3.53**	**-11.75**	**-29.87**		**0.67**	**126.00**	**30.39**	**226.00**	**54.51**	**84.90**
12.15	48.21	37.70		**0.72**	**3.51**	**-10.91**	**-27.56**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
12.30	49.12	38.00		**0.72**	**3.56**	**-11.82**	**-30.31**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
12.45	49.58	38.30		**0.72**	**3.57**	**-12.28**	**-31.58**		**0.67**	**125.00**	**30.15**	**218.00**	**52.58**	**82.73**
13.00	49.86	38.60		**0.72**	**3.57**	**-11.56**	**-29.72**		**0.67**	**124.00**	**29.91**	**210.00**	**50.65**	**80.56**

HEAT LOSS

Through Radiatiom

			ε_{steel}	ε_{glass}	σ	Tcab^4-Tamb^4	E
11.00	50.00	35.90	0.97	0.94	0.0000000540	1779714603.98	183.56
11.15	49.73	36.30	0.97	0.94	0.0000000540	1695999282.56	174.93
11.30	48.33	36.60	0.97	0.94	0.0000000540	1473535355.11	151.98
11.45	48.00	37.00	0.97	0.94	0.0000000540	1382561658.28	142.60
12.00	48.05	37.30	0.97	0.94	0.0000000540	1353436153.97	139.59
12.15	48.21	37.70	0.97	0.94	0.0000000540	1325996643.06	136.76
12.30	49.12	38.00	0.97	0.94	0.0000000540	1411915085.14	145.62
12.45	49.58	38.30	0.97	0.94	0.0000000540	1436559693.05	148.17
13.00	49.86	38.60	0.97	0.94	0.0000000540	1438072008.30	148.32

By Convection

Windshield

			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
11.00	50.00	35.90	0.88	3.34	14.10	41.44
11.15	49.73	36.30	0.88	3.30	13.43	38.99
11.30	48.33	36.60	0.88	3.19	11.73	32.92
11.45	48.00	37.00	0.88	3.14	11.00	30.39
12.00	48.05	37.30	0.88	3.12	10.75	29.53
12.15	48.21	37.70	0.88	3.10	10.51	28.69
12.30	49.12	38.00	0.88	3.15	11.12	30.81
12.45	49.58	38.30	0.88	3.16	11.28	31.34
13.00	49.86	38.60	0.88	3.16	11.26	31.28

Rear Window

			A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
11.00	50.00	35.90	0.86	3.23	14.10	38.95
11.15	49.73	36.30	0.86	3.19	13.43	36.65
11.30	48.33	36.60	0.86	3.08	11.73	30.95
11.45	48.00	37.00	0.86	3.03	11.00	28.57
12.00	48.05	37.30	0.86	3.02	10.75	27.76
12.15	48.21	37.70	0.86	3.00	10.51	26.97
12.30	49.12	38.00	0.86	3.04	11.12	28.96
12.45	49.58	38.30	0.86	3.05	11.28	29.46
13.00	49.86	38.60	0.86	3.05	11.26	29.40

				Side Glass			
				A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
11.00	50.00	35.90		0.72	3.78	14.10	38.35
11.15	49.73	36.30		0.72	3.73	13.43	36.08
11.30	48.33	36.60		0.72	3.61	11.73	30.47
11.45	48.00	37.00		0.72	3.55	11.00	28.13
12.00	48.05	37.30		0.72	3.53	10.75	27.33
12.15	48.21	37.70		0.72	3.51	10.51	26.55
12.30	49.12	38.00		0.72	3.56	11.12	28.51
12.45	49.58	38.30		0.72	3.57	11.28	29.01
13.00	49.86	38.60		0.72	3.57	11.26	28.95
				Roof Calculation			
PM				A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
11.00	50.00	35.90		1.70	5.19	14.10	124.48
11.15	49.73	36.30		1.70	5.13	13.43	117.12
11.30	48.33	36.60		1.70	4.96	11.73	98.90
11.45	48.00	37.00		1.70	4.88	11.00	91.29
12.00	48.05	37.30		1.70	4.85	10.75	88.71
12.15	48.21	37.70		1.70	4.83	10.51	86.19
12.30	49.12	38.00		1.70	4.89	11.12	92.55
12.45	49.58	38.30		1.70	4.91	11.28	94.15
13.00	49.86	38.60		1.70	4.91	11.26	93.95
				Doors Calculation			
				A, sq.m	U, W/Sq.m. K	Δt, °C	Q, W
11.00	50.00	35.90		1.54	3.84	14.10	83.31
11.15	49.73	36.30		1.54	3.79	13.43	78.39
11.30	48.33	36.60		1.54	3.66	11.73	66.19
11.45	48.00	37.00		1.54	3.61	11.00	61.10
12.00	48.05	37.30		1.54	3.59	10.75	59.37
12.15	48.21	37.70		1.54	3.56	10.51	57.68
12.30	49.12	38.00		1.54	3.62	11.12	61.94
12.45	49.58	38.30		1.54	3.63	11.28	63.01
13.00	49.86	38.60		1.54	3.63	11.26	62.88

Time	Δt, °C
11.00	-0.27
11.15	-1.40
11.30	-0.33
11.45	0.05
12.00	0.15
12.15	0.92
12.30	0.45
12.45	0.28
13.00	0.93

Time	Net Gain
11.00	-25.45
11.15	-131.44
11.30	-30.77
11.45	4.74
12.00	14.51
12.15	86.06
12.30	42.63
12.45	26.39
13.00	87.28