DESIGN, ANALYSIS, AND PROTOTYPING OF PELTIER COOLING PROJECT

An Interim Report
Presented to
The Advisory Committee and Project Sponsors
Senior Design Course of Fall 2019 & Spring 2020
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MEE 4193/4194: Mechanical Engineering Design

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Abstract

Energy consumption throughout the world has been increasing in recent years and is a major growing concern due its effect on Climate Change. According to NASA, Climate Change has led to stronger hurricanes, more droughts and heat waves, sea level rise, and they predict the arctic will be ice-free by mid-century. One of the most notable energy consumption methods, which negatively affect the climate, is building cooling and heating. Minimizing the need for energy use in buildings for cooling and heating applications is a widely explored industry. While energy efficiency has drastically improved in recent years, the use of harmful refrigerants, such as R-410, has already taken a huge toll on the environment. This presents a need for building cooling and heating without the use of harmful refrigerants. An alternative method of refrigeration is the use of Thermoelectric Modules (TEMs) due to their nature of having no mechanical moving parts, being highly reliable, compact in size, light in weight, requiring no working fluid, and most importantly - they do not require the use of refrigerants. Our project aims to design and create a fully functioning system that uses TEMs as an alternative for building cooling and heating.

- PDR reports are due by 11:59 PM EST on Saturday, November 30th
    - Late reports will **NOT** be accepted
- PDR presentations are due 24 hours in advance
    - Late submissions will **NOT** be accepted
- Peer evaluations are due by 5:00 PM EST on Friday, December 6th
    - Failure to submit, late submissions, or improperly filled out submissions **will result in the decrease of a letter grade**
- Summary of what the report is about
- Highlight important details

Stuff for us to mention:

- Include all advisors and sponsors on front page
- Not every team will have every section ("If applicable"). If you intend on cutting a section out of your report, it is **highly** recommended that you check with the GSAs first!
- Don't start a chapter with a subsection (something should be before 3.1 for example)

- There can (and probably should) be subsections and additions made depending on the project
- Document body: 50 pages maximum

Acknowledgments

We would like to thank Hamidreza Najafi for his contribution to our project and this team.

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Chapter 1: Introduction

1.1 Problem Statement

Energy consumption in the building sector has increased dramatically within the last decade due to worldwide population growth. With that being said, more time is spent inside, therefore elevating the demand for optimal and more efficient cooling and heating systems to provide thermal comfort. This report proposes a thermoelectric based system for cooling and heating application. The system will consist of Thermoelectric Modules (TEMs) embedded into the ceiling envelope of the proposed test building environment, which will provide cooling and heating through radiation and convection. Our project will adhere to the ASHRAE 55 thermal comfort standards. The final product will be utilized for educational purposes in the Heat Transfer laboratory at the Florida Institute of Technology for teaching principles of radiative cooling and heating, as well as the Peltier effect.

1.2 Motivation

Climate Change has been a major growing concern in recent years, with its main contributor being Greenhouse Gas (GHG) emissions. Research and studies are attempting to further identify sources of GHGs, and pinpoint ways to reduce their emissions. According to the US Energy Information Administration (EIA), energy use in the building sector accounts for 20% of global energy consumption, and, in 2018, 40% of energy consumed in the United States came from buildings, which is equivalent to 40 Quadrillion British Thermal Units (BTU). Additionally, cooling and heating alone accounts for over half of the total energy consumption in buildings [1].

[1] "U.S. Energy Information Administration - EIA - Independent Statistics and Analysis." How Much Energy Is Consumed in U.S. Residential and Commercial Buildings? - FAQ - U.S. Energy Information Administration (EIA), https://www.eia.gov/tools/faqs/faq.php?id=86&t=1.

Refrigerants are a commonly used substance for refrigeration. These can vary from Chlorofluorocarbons (CFCs), Hydrochlorofluorocarbons (HCFCs), and Hydrofluorocarbons (HFCs), all of which pose a threat on the environment in numerous ways. The United States Environmental Protection Agency (EPA) has categorized these harmful chemicals in regard to their Ozone Depletion Potential (ODP) and their Global Warming Potential (GWP). ODPs are used to understand the impact different gases have on the ozone through a numerical ranking system where a higher number indicates a more damaging effect on the environment. GWP is used to determine the strength of a gas' greenhouse effect; the higher the GWP, the warmer earth will get with an equal amount of CO_2. Some of these harmful refrigerants have reached as high as 14,000 for a GWP and 16 for an ODP [2]. The EIA also states that, in the past 8 years, emissions due to refrigerants have more than tripled as a result of a dependence on R-410 in air conditioners [1].

Because of the effect refrigerants have on the environment as well as the toxicity these chemicals have on humans, there is a great necessity to find alternatives for cooling and heating systems, as they pose one of the greatest threats to our environment and future welfare. This presents the need for innovative building cooling and heating that doesn't rely on these harmful refrigerants. This will improve both energy efficiency and environmental concerns.

Thermoelectric Modules (TEMs) have recently become a growing topic in research due to their ability to convert DC power into a temperature gradient. The feasibility to utilize TEMs as an alternative to conventional vapor compression HVAC&R (Heating Ventilation Air Conditioning and Refrigeration) systems will completely eliminate the harmful effects of refrigerants as well as reduce overall energy consumption.

[2] "Ozone-Depleting Substances." EPA, Environmental Protection Agency, 31 July 2018, https://www.epa.gov/ozone-layer-protection/ozone-depleting-substances.

1.3 Requirements

- Requirement table
- Explain important requirements in more detail and explain thoroughly how you plan to meet the requirements and how you plan to test that you've met them

1.4 Global, Social, and Contemporary Impact

- What benefit/impact does your project have on society/the world/your target consumer?

Hugo

Our project consists of a fully controlled and energy efficient cooling and heating system. This project uses thermoelectric modules to provide cooling, or heating to the environment. The major difference of this project with conventional air conditioning system is that the conventional AC systems use refrigerant in order to cool or heat the desired environment, and ours don't.

Our system only needs a voltage applied to these modules in order to function. As it is mentioned in the following paper, Iyer Mani, Pooja. *Design, Modeling, and Simulation of a Thermoelectric Cooling System (TEC)*. Western Michigan University.

Almost 10% of the annual consumption of fuel is directed to the use of air conditioning. Most of these systems use refrigerant like R-134 which have adverse effects on the environment being a major greenhouse gas. Different debates are occurring about banning all the depleting compounds, and R-134 is one of them. Our system cools, and heats a desired environments without the use of refrigerant. Thermoelectric cooling is the future due to this. Therefore, our system has a major impact in our society.

Another benefit that our system has compared to conventional air conditioning, is choosing the area that you want to cool within a room. The controllability of temperature with this system is not seen in any of the actual conventional air conditioning system, making it a major benefit to a specific target costumer that needs this controllability.

Controllability is a major benefit, but it goes along with temperature precision. These TE modules allow an excellent temperature precision, which assures very accurate thermal comfort conditions.

2 Background

2.1 Literature Review

- Overview of any and all research you did

Sub-category:

1) **Energy Efficiency**:

For a long time, energy efficiency has been the center for designing most of the refrigerating system. The more energy saved, the less cost is required, and the less energy used, the less wastes are released to the environment. To achieve a better efficiency, we need to increase the cooling/heating power per the total power used. It will directly increase the coefficient of performance (COP) or we need to find a different mean to get more cooling/heating power per actual power input. However, a thermoelectric-based system usually has less COP than a standard HVAC system but better for their design aspect [[3]]. Given that we are trying to implement TE for a cooling/heating system, getting the most usable power per power input is imperative. A different research by Lertsatitthanakorn [[4]] give us the result COP of cooling at

[3] 3 Zhang, H.Y. "A General Approach in Evaluating and Optimizing Thermoelectric Coolers." International Journal of Refrigeration, vol. 33, no. 6, 2010, pp. 1187–1196., doi:10.1016/j.ijrefrig.2010.04.007.

[4] Lertsatitthanakorn, Charoenporn & Srisuwan, Wichan & Atthajariyakul, Surat. (2008). Experimental performance of a thermoelectric ceiling cooling panel. International Journal of Energy Research. 32. 950 - 957. 10.1002/er.1391.

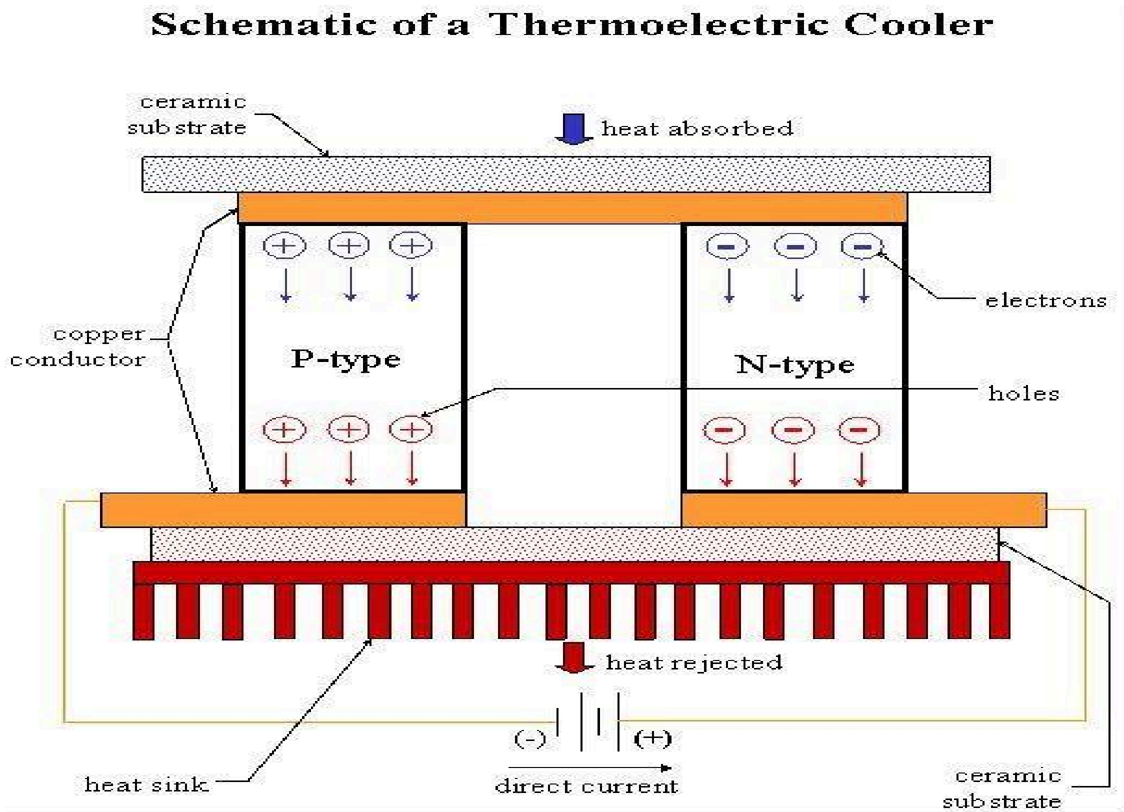
around 0.75 for a TE system that is installed on the ceiling (which is the same objective of our project). That COP value is atrocious compared to the typical COP value of 2 to 4 for standard HVAC system [[5]] (Power Knot, 'COPs, EERs, and SEERs'). Fortunately, that research was made in 2009, which is potentially outdated by now. But to achieve a decent COP for our cooling system, we need to maximized the capabilities of our main parts of the system: the TE modules and the Heat Sink. (Minh)

Eniola TE modules: Solid State cooling is made up of various effects that make use of groundbreaking means to provide cooling to a system. These effects range from Electron tunneling, the Magnetocaloric effect, the Thermoelastic effect and the Peltier effect. The Peltier effect is the only effect that has been commercialized as the others are still under research. The Peltier effect states that when current (Direct Current) goes into a Peltier device, the phenomenon of the absorption of heat by a junction between two different material occurs. It is due to this phenomenon that the one side of the device cools (rejecting heat) and the other heats (absorbing heat).

Peltier devices also called Thermoelectric Modules (TEMs) are devices that effectively transfers heat along an array of pellets in the module in order to make one side hot and the other cold for various applications. The basic build of a TE module consists of P and N type pellets usually made up of Bismuth Telluride material in order to effectively transfer heat, these

[5] "COPs, EERs, and SEERs." Power Knot, 1 Mar. 2011, www.powerknot.com/2011/03/01/cops-eers-and-seers/.

elements are then placed in between two ceramic plates that are both ductile and strong thermal conductors.



There are 5 Basic Principles of Operation of a TE module:

- Thermal Conduction: An action also referred to as Fourier's process which occurs between the pellets in a TE module.
- Joule Heating: The physical process of heat dissipation in the resistive elements.
- See beck Power Generation: The generation of electricity by the process of heating the TE module.
- The Electromotive force of the Module
- Peltier Cooling and Heating

With the different operation modes that occur in a TE module, it was obvious that it would be subjected to losses. These losses were researched in an ASHRAE Journal and they were divided into two major losses.

- Electrical Resistance Loss: This loss occurs between the electrical materials in the TEM and they raise the amount of voltage required to drive current through the module.
- Thermal Conduction Loss: Due to finite thermal conductivity, heat is conducted from the hot side back to the cold side when the module is in operation. This reduces the net flow of heat.

As stated earlier, there are various types of TE modules used for various purposes ranging from small electronic equipment cooling to large enclosure cooling and they are listed below:

i. <u>Standard TEM</u>: Standard modules offer the highest quality in the industry with excellent thermal performance. They are used for regular operations that require the basic understanding of the Peltier effect as well as for testing.

ii. <u>High Performance TEM</u>: High Performance modules have one of two features that set them apart from our standard devices. Some achieve a higher temperature difference through the use of premium quality thermoelectric material. Others have more heat pumping capacity for any given size through the use of shorter thermoelectric elements.

iii. <u>Miniature TEM</u>: Micro TE modules are devices that have semiconductor element footprints of less than 1.0mm square, allowing higher numbers of couples for a given size module. These modules are metallized on both hot and cold surfaces and they are suitable for small scale cooling.

iv. Holed/Circular TEM: Thermoelectric modules with holes allow additional flexibility but have reduced performance. The Center-Hole TEM series is suitable for various cooling and heating applications which generally require medium pumping capacity. Typical application areas include industrial and electrical equipment as well as laboratory and opto-electronics.

v. Multi-stage TEM: Multi-stage thermoelectric modules are modules that are "stacked" to achieve higher temperature differences than can be had with single-stage modules. They are specially constructed with the top and bottom halves sharing a common ceramic in the middle.

With the TE module set to compete with other types of cooling systems we would have to look at the benefits of these devices over popular Vapor Compression Cycle systems available:

Advantages	**Disadvantages**
No Refrigerants (CFC's)	Inefficient at high temperature differences
No moving Parts	Finding materials to reduce inherent losses
Reduced maintenance costs	-
More useable building space	-
Precise temperature control	-
Rapid response time	-
Silent operator	-
Heating and Cooling	-

At first glance, it would seem as though the advantages outweigh the disadvantages but the issue with TEMs being inefficient is a major reason why they are not popular commercially. As the need for a bigger temperature difference is requested, the TEMs require more power and thus become lesser performers than popular Air conditioning systems.

To effectively apply the thermoelectric module for our system, prior knowledge of the components performance is needed. Research Paper published by H.Y Zhang [6] looks into how to effectively determine various properties of a TE module and it was with this knowledge that we were able to determine the Cooling power, Coefficient of Performance (COP), and the ZT for our module. The Cooling power was a measure of how much heat a TEM would need to remove from a system, the Coefficient of Performance (COP) is a value that tells us how efficient our system would be, and the ZT values which relates to the Peltier effect. Based on our target value for the COP we would then have to determine the number of TEMs we would need for our system, as well as the temperature of the hot side and cold side of the module, the calculations involved with these were found on a paper [7] cowritten by Dr. Hamidreza Najafi in which an analysis of a hybrid system consisting of TE modules and PV panels were analyzed and tested.

The Coefficient of Performance (COP) which is the efficiency of our system is the ratio of the cooling power each module provides to the total input electrical power. It should be noted that the lower the COP value we have, the less efficient our system would be and as such we aimed for a higher value. The Figure of Merit (Z) is a term that balances the strength of the thermoelectric effect to the losses in the system. It was also noted a direct relationship existed between the COP and the ZT in which an increase in COP also increases the ZT.

6

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These are what make up the operation and build of the TE module and with proper knowledge of it, we were then able to decide on the most energy efficient module to use in our system.

Heat Sink: Jesse A heat sink is a device used to assist and/or increase the rate of heat transfer form a mechanism, heat engine, heat pump, or any device that produces heat that needs dissipating. Another common way that heatsinks can be represented is that of a reservoir were thermal energy that can hinder the operation of a device can be stored and then dissipated more efficiently. There existents a large number of different heatsinks. The most commonly used heatsinking method is air-cooling with extended surfaces. The method involves either forced convection or natural convection. Where forced convection is the increase in the cooling air's velocity buy aid of a fan or pumping device.

(H. Hana, et al., Heat Sink Design for a Thermoelectric Cooling System, 2008):

Air cooling a method of heat transfer that has been well understood and applied since the beginning of the modern age. In Heat Sink Design for a Thermoelectric Cooling System, the performance of a louver finned, offset strip and plate fin are analyzed using a duct-flow type fan arrangement. The metrics that were used to compare the behaviors of the differing fin arrangements was the thermal resistances, time taken to steady state, and the COP. After the analysis was completed. The heat flux at the heat absorption interface was kept constant using a water-cooled heatsink. The results showed that the plate finned heat sink provided 24.4% higher COP and 27.7% higher heat absorption when compared to the lover finned heat sink.

(C.K. Lou et al, Thermal Characterization of Fan-Heat Sink Systems in Miniature Axial Fan and Micro Blower Airflow 2001):

One very important parameter that effects the overall efficiency and rate of heat transfer from an air cooled finned heatsink. Is the process of forced convection. Forced convection is the method of utilizing a fan as a means of forcing cooling air over the surface of an objected to be cooled to increase the heat transfer rate. In Thermal Characterization of Fan-Heat Sink Systems in Miniature Axial Fan and Micro Blower Airflow 2001 they attempt to investigate which method of forced convection would yield the greater heat transfer or lower thermal resistance. The results of this experiment were found to be inconclusive. It did however give valuable data that could potentially assist in the selection of a fan system.

(PCM 1) liwei Phase change material integrated thermoelectric radiant cooling panels (PCM-TERCP) are proposed. PCM can be frozen during night, so it can maintain the radiant cooling surface temperature when operating. This way, the required temperature for removing the rejected heat is lower, so the temperature difference between two sides of TE models is lower, then the coefficient of performance (COP) get increased, and energy get saved. Validation experiments are produced, and the phase change interval is 195 minutes. In order to increase more operation time of PCM-TERCP, the designs of PCM storage box are proposed. Since the heat conductivity of PCM is low, the physical method, adding fins, are used in the design.

PCM-TERCP with different number of fins are simulated, and the result is that the bottom panel's temperature is lowest with most fins, which means that increasing the overall heat transfer coefficient between the top and bottom panels can increase the operating time. The reason is that a high overall transfer coefficient enables heat to be distributed evenly inside PCM instead of distributed concentratedly on one side of PCM. The minimal overall heat transfer coefficient is found as 615 W/m2·K based on the boundary conditions which are 18℃ for surface temperature and 3 hours for operating time.

Lim, Hansol, et al. "A Numerical Model and Validation of Phase Change Material Integrated Thermoelectric Radiant Cooling Panel." *E3S Web of Conferences*, vol. 111, 2019, p. 01001., doi:10.1051/e3sconf/201911101001.

(PCM 2) Phase change material integrated thermoelectric radiant cooling panels(PCM-TERCP) are proposed. The goal of using phase change materials(PCM) is increasing the COP of the thermoelectric(TEM), because PCM can store a large amount of latent heat with small temperature changes. A mock-up model is constructed to evaluate PCM-TERCP's performance in the solidification and melting processes under design condition. In the experimental setup, n-Alkane is chosen as PCM, which has melting point as 16°C, and that temperature is the general surface temperature of the radiant cooling panel to prevent condensation problems. The thickness of the insulation is 5 mm for 0.16m2 heat transfer area to meet the calculated required heat transfer coefficient, which is 9.5 W/m2K in natural convection conditions. 28 thermocouples are used to measure surface temperature, and there are 7 thermocouples for each surface. In the result, there are two plots show the mean surface temperature variations related with time in both solidification and melting process, and the temperature variations on the bottom side of PCM-TERCP related with time. According to the plots, for solidification process, 4 hours, and 528 Wh input power are needed. The melting process is operated passively for 4 hours, and during most of the time, the temperature of the bottom panel is below 19°C, which is a good surface temperature for radiant cooling. What's more, the temperature differences between maximum and minimum temperature during the melting process is always below 1°C, which satisfies the ASHRAE standard.

Kang, Yong-Kwon, et al. "Experimental Evaluation of Phase Change Material in Radiant Cooling Panels Integrated with Thermoelectric Modules." E3S Web of Conferences, vol. 111, 2019, p. 01002., doi:10.1051/e3sconf/201911101002.

(**arrangement and spacing**) The arrangement and spacing of thermoelectric modules can affect the performance of the thermoelectric radiant cooling panel, so numerical simulation models are developed to find a suitable configuration. Triangular and rectangular configurations are tested since other configurations are combined by triangular and rectangular. The results show that the average temperature differences are 3.1 ± 0.23°K and 3.4 ± 0.41°K between the hot and cold spots in the plate for the triangular and rectangular configurations, which prove that triangular configuration provides better performance. What's more, simulations are developed to investigate the effects of the spacings between the TEMs. The spacings vary from 0.28m to 0.38m with 0.02m interval. By observing a series of plots, the result shows that 0.28m for the spacing between the TEMs provides the lowest average temperature difference as 2.7 ± 0.20°K between the hot and cold spots in the plate, which satisfies the recommendation for radiant cooling panel from ASHRAE. Therefore, triangular configuration with 0.28m spacing is chosen.

Lim, Hansol, et al. "Thermoelectric Radiant Cooling Panel Design: Numerical Simulation and Experimental Validation." *Applied Thermal Engineering*, Pergamon, 20 Aug. 2018, https://www.sciencedirect.com/science/article/pii/S135943111833477X.

2) Thermal Comfort

(Yifei Li) Thermal comfort is the expression of satisfaction with the thermal environment and is assessed by subjective evaluation. Subjective evaluation is expressed in feelings and emotions. Thermal comfort is critical to the building as it not only lays the foundation for architectural design but also influences the field of sustainable design. The current thermal comfort model suggests a very narrow temperature range, and people are almost passively accepting heating or cooling systems, which consume a lot of energy to control the temperature and may not reach the desired ambient temperature. However, our new research and innovative refrigeration and heating design systems are challenging the current model. Thermal comfort depends on many variables, such as ambient temperature, radiant temperature, relative humidity, air motion, metabolic rate and clothing insulation. The use of this design of the thermal comfort model allows us to significantly improve the energy efficiency of the building design, and to directly relate the indoor temperature to the occupants, the activities of the occupants and the outdoor climate.

(Becamon, Mary Rose. "Thermal Comfort: Designing for People." Academia.edu, https://www.academia.edu/35953440/Thermal_Comfort_Designing_for_People.)

Operative temperature is simplified measure of human thermal comfort derived from air temperature, mean radiant temperature and air speed. It's used in heat transfer and thermal comfort analysis in transportation and buildings.

Two.1.3.2 Test Building Environment

The test building environment includes the size, materials, and shape of where the system will be tested. In order to replicate an actual room, a decision was made to have a rectangular shape for our environment.

When looking at the materials to use, several options were evaluated. Different drawbacks, and advantages were faced with the materials in mind. One of the materials evaluated was plywood. This material is lightweight and it is used in small scale room applications. This material can be seen in doll houses, as well as in residential applications. One of the advantages of this material was the lightweight compared to the other materials considered. However, the design of the system must be as energy efficient as possible. Plywood having a low R-value wasn't the best option.

Another option reviewed was sheetrock. Sheetrock is a lightweight drywall. It weights 30% less of regular drywall. The R-value for this material was better compared to plywood. However, it was still very low for the goals and requirements set for the design.

Finally, in order to create an energy efficient environment, the Structural Insulated Panels (SIPs) were studied. Structural Insulated Panels are a high performance building system for residential, and commercial applications. These panels provide an extremely strong, energy efficient, and cost effective system.

3) <u>Control System</u>

2.2 Patent Search

https://patents.google.com/patent/WO2013130424A1/en?oq=thermoelectric+module+air+conditi
onin g

https://patents.google.com/patent/US3252504?oq=thermoelectric+module+air+conditioning 12 https://patents.google.com/patent/US6393842?oq=thermoelectric+module+air+conditioning https://patents.google.com/patent/US9310112?oq=thermoelectric+radiant+cooling All these patents relate to thermoelectric cooling and heating apparatus' and there are several more dealing with cooling and heating, however they're for vehicle applications. There have been inclusions of the use of working fluids as well as incorporation of PCMs in these patents. Any and all patent works for a thermoelectric radiant cooling and or heating for building applications is yet to be established.

2.3 Eniola Current State of the Art

- What products/functions/processes are currently on the market and how/why is yours different?

When it comes to cooling, there are numerous products that are commercially available which are similar to our system in terms of its function and process. These products range from a typical Vapor compression Cycle Air Conditioner, A thermoelectric Refrigerator, and a Peltier Air Conditioning enclosure. While most of these products either perform the same task as our system or incorporate some of the same materials, in terms of our overall system design, no other product has been made.

We have looked into various types of these systems in order to better understand their performance, advantages, and disadvantages so as to make our system better and they are listed below:

- Vapor Compression Cycle Air Conditioner:

Image of Typical VCCAC Unit

This product ranges from various types of typical air-conditions used in houses, offices etc. It is the major product our system is designed to compete with as both our system as well as typical AC units are designed to perform the same function. In order to for our system to compete with this we looked into the advantages and disadvantages of this system and compared them to our Peltier based system.

Peltier System	Vapor Compression Cycle AC
No moving Parts	Moving Parts
No Refrigerants	Refrigerants Present
Inefficient	Efficient
Small Weight and Volume	Large and Heavy
No thermodynamic losses	Pressure losses in coils and connecting tubes

As shown above, we see more advantages from the Peltier System but a major factor for cooling units is efficiency. The issue of our system being inefficient is one we intend to solve and through the use of better TE modules, better heatsinks, and Phase changing materials (PCMs) we can make our system efficient.

- <u>Thermometric Refrigerator</u>:

Image of TR-059 Thermoelectric refrigerating unit by TECA.

This is a thermoelectric fridge that makes use of multiple Peltier units to cool. It offers consistent performance and a long service life with its use originally designed for military transport. There are various reasons why our system is different, one being the actual use. While both systems function to cool, our system doesn't need to cool at a high temperature difference thus making it more efficient than the fridge.

- <u>Peltier Air conditioning Enclosures</u>:

Image of Rittal 341 BtuHr TE Enclosure AC unit.

This AC unit similar to our system performs the same function but at a smaller scale, this Air-cooling unit is used for cooling small enclosures and command panels such as a cupboard or wardrobe. Its small and lightweight design makes it suitable for use on support arm systems and for cooling targeted hotspots. The device in designed this way in order to make use of the TE technology while also making it efficient but at the cost of cooling power. Our system intends to make use of the TEM at a higher cooling power and also making it efficient.

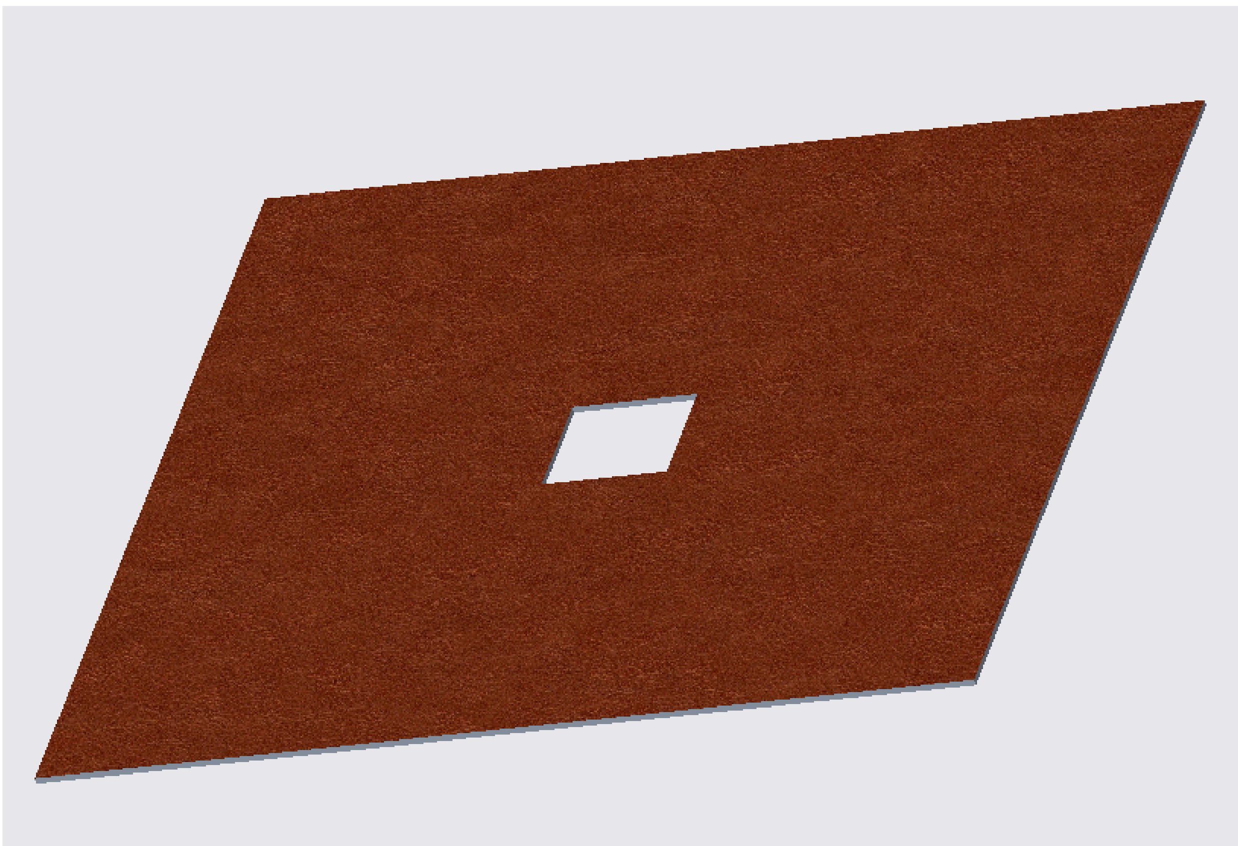
There are other products available that serve to cool an environment or make use of Peltier devices but as discussed, our system offers numerous advantages over these products ranging from size, to cost, to weight. Our system which incorporated the use of TE modules in ceiling tiles to provide radiative cooling is a product that has never been created thus making it the first of its kind.

3 Recommended Solution

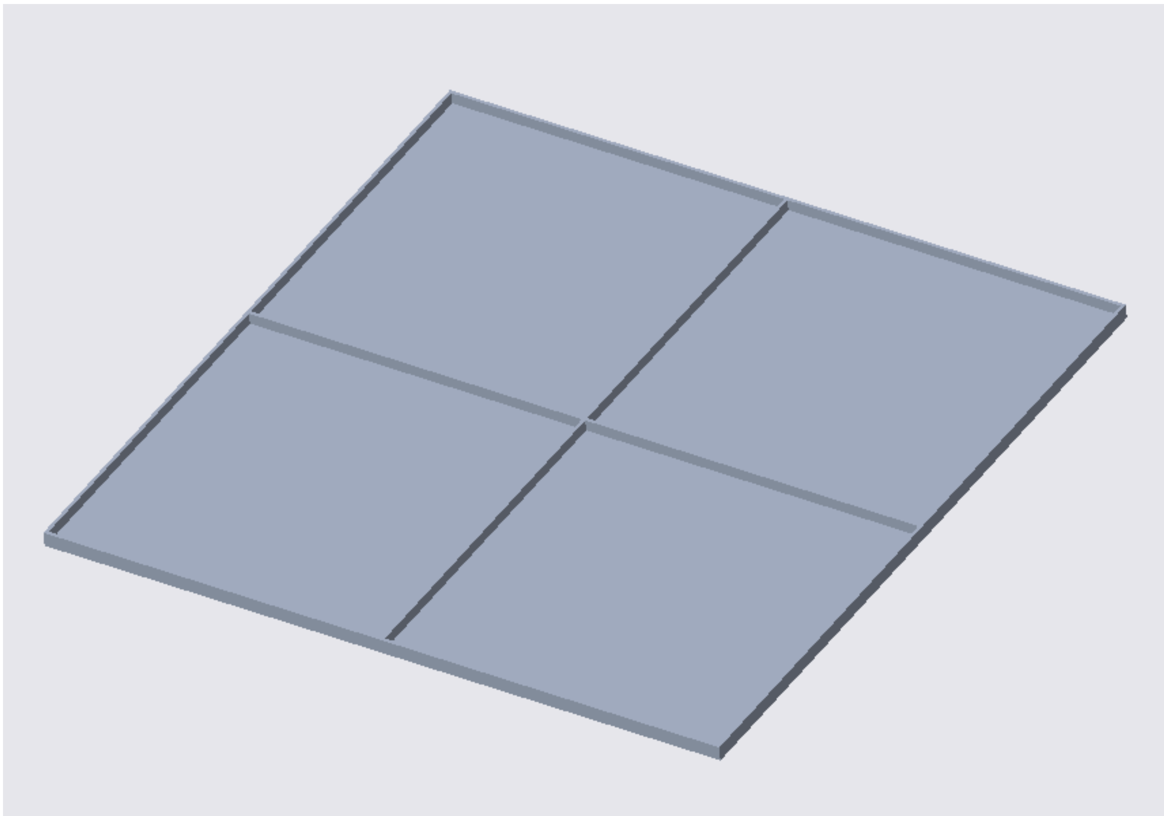
3.1 Proposed Design

- Overall outline of design
- CAD Models

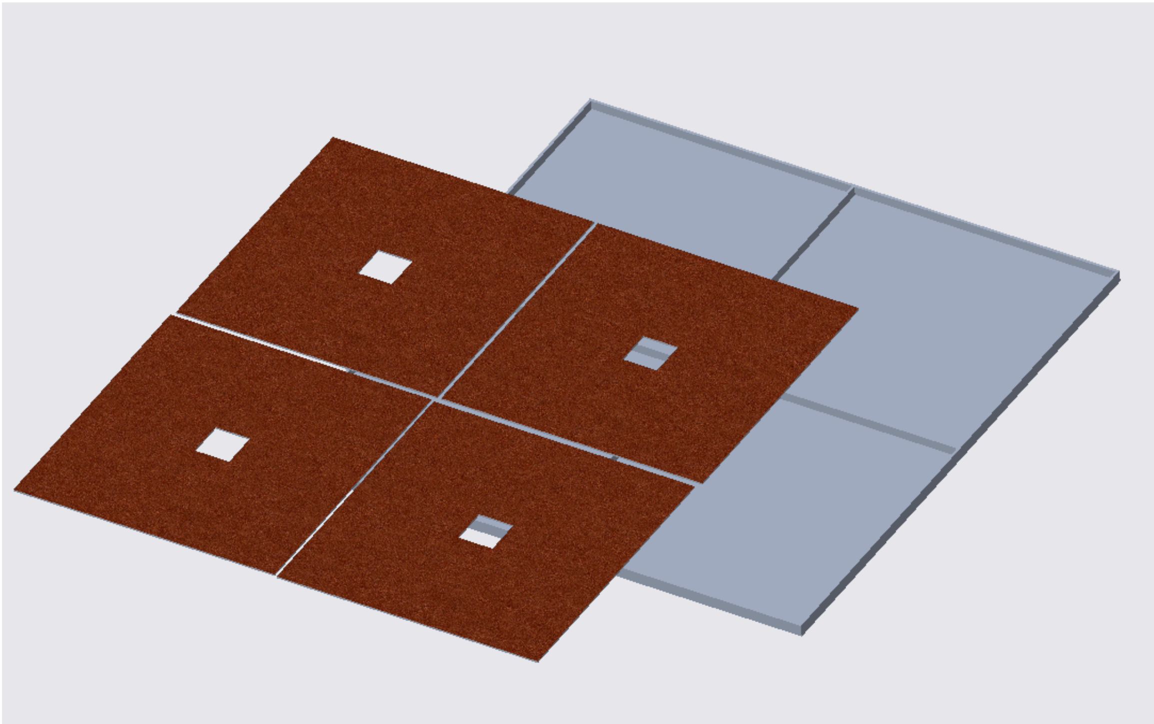
（Yifei Li）

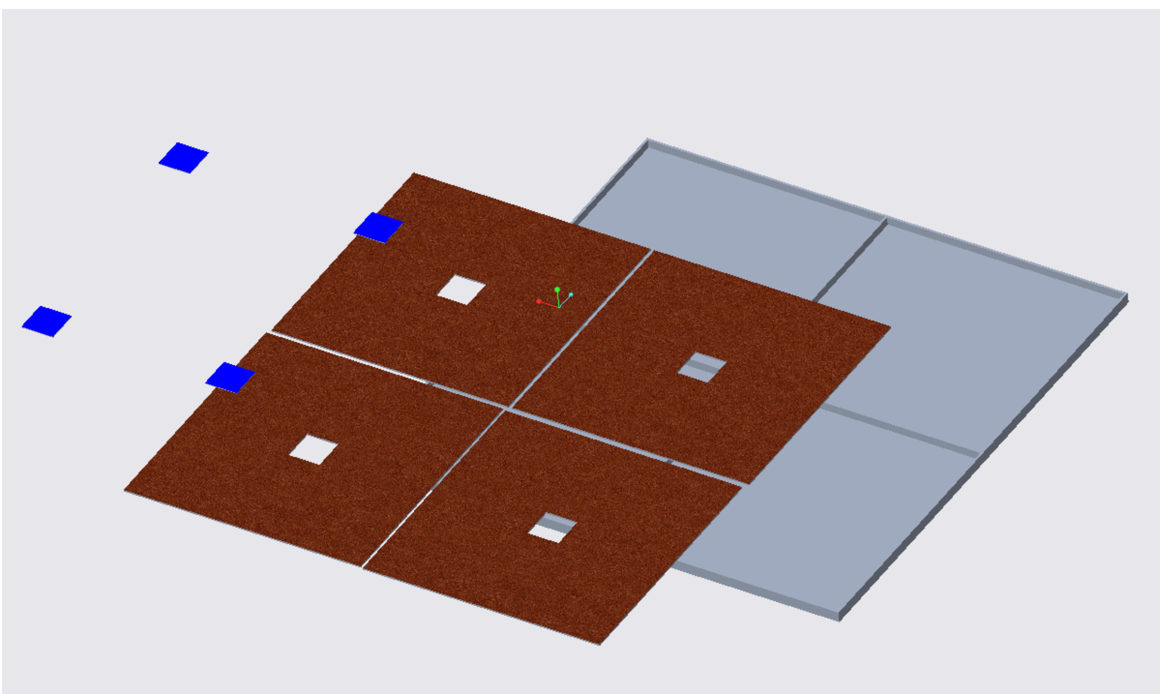


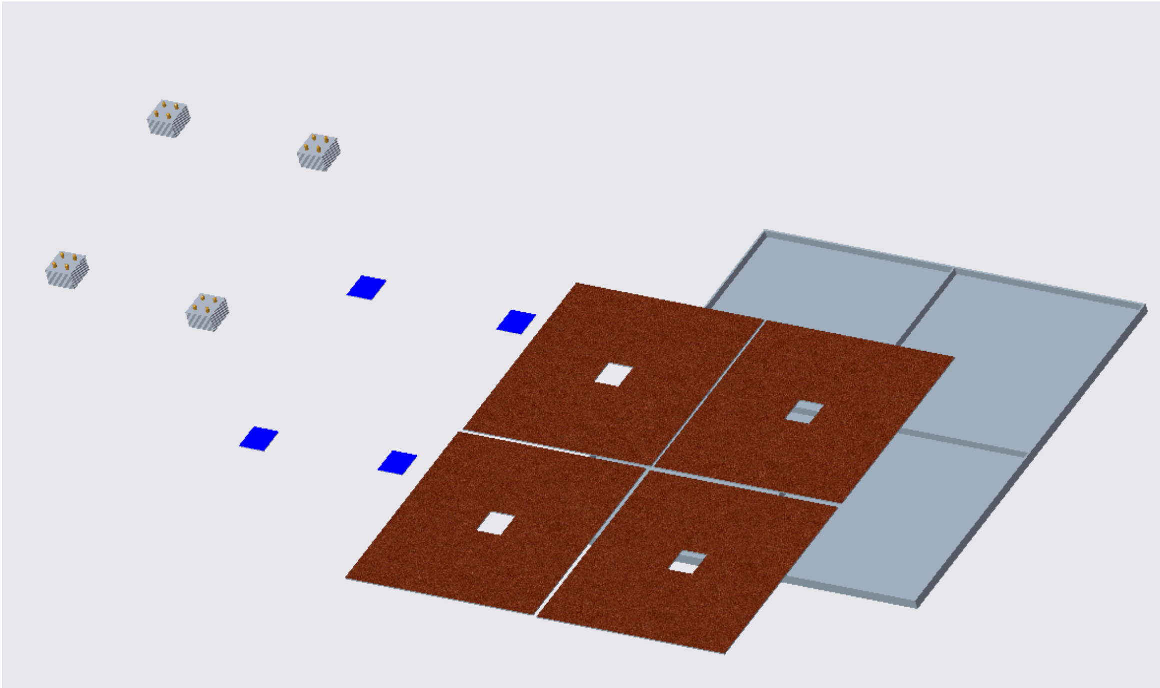
This is a PCM material model, the PCM material surrounds the TE module, and the phase change process of the PCM can release and absorb heat. We use PCM to reduce the working time of the TE module. When the cooling mode is turned on, the TE module reduces the aluminum ceiling temperature and lowers the temperature of the PCM, which may change the PCM from liquid to solid. When the TE module is turned off, the PCM can continue to absorb heat by turning solids into liquid to keep the aluminum ceiling at a lower temperature, thereby extending the temperature holding time and reducing the TE module usage time.



This is an aluminum ceiling, which has a strong heat transfer capacity, so it rapidly drops in temperature when the TE module cooling mode is turned on, so that it can quickly radiation cooling, but it does not store heat for long periods of time, so we need to work with PCM materials.







3.2 Solution Principles

- Breakdown of important functions and design choices to satisfy those functions

Eniola **TE Module: HP-199-1.4-0.8**

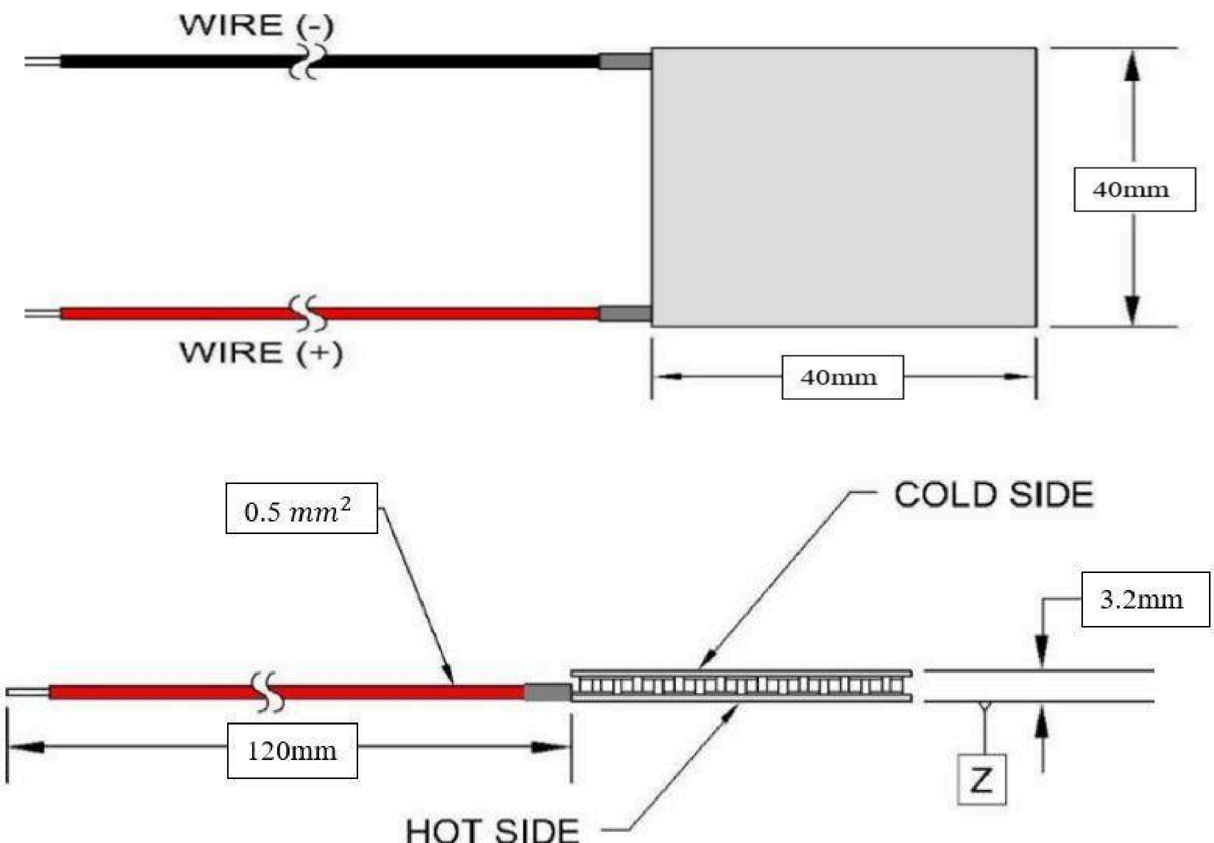
To determine the ideal TE module for our system we had to take into account various factors. We needed a module to be able to remove the required amount of heat from our system for cooling and also provide heat for heating, this was to be determined after we calculated the load of our system. We also needed the TEM to be among the most efficient when operating at our maximum temperature difference of 20 to provide thermal comfort. We also took into consideration the cost of the system as well as the size, and it was with all these factors we decided on our ideal TEM.

It was decided to go with a High powered TE module because it offered more heat pumping capacity meaning it could remove more heat from the system thereby leading to less TE modules needed. It also had a higher maximum load (Q_{max}) which would improve our systems efficiency especially since a high temperature difference was not required for the system. We also compared the High performance (HP) to other types of TE modules below:

Module Comparison	**Advantage**
HP vs Standard	Better Performance
Hp vs High Temperature	Better COP due to less power consumption
HP vs Miniature	Less TE modules needed
HP vs Series-Parallel	Higher cooling load
HP vs Multi-stage	Better cooling and less power consumption

We decided to go with 4 High powered TEMs for Cooling and 2 for Heating, we expect each of the 4 TEMs to provide max of 58W for cooling (Q_c) and a max of 101W for heating (Q_h) based on our calculated loads (see analysis). The size for the TEM was also taken into account and found to not be an issue for our system and compared to most TEMs, the cost was slightly higher than standard modules but offered better performance.

Selected TE Module	HP-199-1.4-0.8
Manufacturer	TE Technology, INC.
Category	High-Performance
Dimensions	40X40X3.2 mm
Cooling Quantity	4
Heating Quantity	2
V_{max}	24.6V
I_{max}	11.3A
Q_{max}	172W



Dimensions of TE-Technologies HP-199-1.4-0.8

3.3 Analysis

- Calculations, testing

Eniola **Load Calculations:**

An environment gains heat from various sources, occupants, electronics, lighting etc. Warm air also enters into the room through the doors, windows, and cracks that we refer to as infiltration however our greatest source of heat entry is from solar radiation. The sum of these sources of heat is called heat gain. It is these heat gains that help us determine the amount of heat we need to remove from our system. Cooling load is the time at which both sensible and latent heat must be removed from a room/space in order to maintain a constant temperature [1]. Sensible heat are the heat gains generated by the source and partly from radiant energy while latent heat is the instantaneous heat load that occurs when moisture is placed in a space either due to internal sources or from outdoor air. These two heat types are what are mainly needed to determine the cooling load for our system.

Project Specifications:

- 4ft X 4ft X 5ft (W X L X H)
- Walls (Structural Insulated panels (SIPS), [or Plywood + Insulation + Drywall)]
- 18 watts Light bulb (LED vs Incadescent0
- Single Room
- ~~2ft X 2ft Window (Clear glass assumed)~~ (Optional)
- Plexiglass wall (acrylic plastic sheets, Single or double walled)
- 4ft X 4ft Roof (SIPS or Plywood +Insulation + Drywall)
- 2ft X 2ft Aluminum Ceiling Tile

- Florida	Cooling Dry Bulb Temperature (°F)		
	0.4%	1%	2%
Melbourne	92.6	**90.8**	89.7

Cooling Load

With the given dimensions for our room, we can calculate the cooling load for our system while also taking into account the various heat sources in the system. When estimating the cooling load, we had to consider various processes because the peak cooling load occurs during the day and the conditions of the outside surroundings vary due to solar radiation and the heat generated from the inside of the room (heat sources) add to the cooling load thus we cannot ignore them.

For our system, we took the cooling load from what we theoretically assumed to be our sources which are as follows:

Cooling Load from Walls: The Cooling load created from the walls are due to solar radiation, which is called the transmission load, this load was calculated using the heat transfer coefficient (U) for the wall material. For the 2 in thick Structurally Insulated Panels (SIPs) we had a resistance value of $R = 12\frac{ft^2 °F}{Btu/hr}$ which was then used to find our U using the formula ((1)Appendix A,1) which we found to be $U_{sips} = 0.0833\frac{Btu/hr}{ft^2 °F}$ and $U_{Plexiglass} = 0.46\frac{Btu/hr}{ft^2 °F}$.

With the calculated U value and gotten U (ASHRAE Handbook of Fundamentals), the area of the wall, as well as the temperature of the outside and inside of the room (ASHRAE handbook 2009) we could then determine our cooling load from the walls. It should be noted that the higher the materials thermal resistance, the lower the heat transfer which is what we are looking for in our system. From our calculations ((2)Appendix A,2) we found that the load for 3 walls plus the plexiglass gave an overall load of $Q_{four\ walls} = 113.8 Whr$.

Cooling load for Roof + Ceiling Tile: Similar calculations for the cooling load of the walls were used when determining the cooling load for the roof but we took into account the size of the ceiling tile as well as the material since different cooling load would be generated by each due to varying heat transfer coefficients. Most of the roof was made up of SIPs materials except the ceiling tile which mainly had aluminum panels used, the heat transfer of each of these materials were individually calculated and added to determine the overall heat transfer from the roof ((3)Appendix A,3) which we found to be $Q_{Roof} = 17.83Whr$

<u>Cooling Load from Window:</u> Similar to the plexiglass calculation, the solar load through clear glass also has two components that needs to be taken into account:

i) Conduction: This involves finding the heat transfer coefficient (U) value (ASHRAE Handbook 2009), the area of the glass and the Temperature difference.

ii) Solar Transmission: Determining the cooling load due to solar transmission involved knowing the shading coefficient (SC) for glass, knowing the zone type, knowing the solar cooling load factor (SCL), and calculating area of the glass. Shading coefficient is the measuring of the thermal performance of glass and it varies with respect to the type of glass.

All values were calculated ((4)Appendix A,4) and our load through the window was found to be $Q_{Window} = 22.05Whr$**.** There are ways of reducing the load generated from widows, adding covers such as drapes to a single glazed window reduces heat loss by 37% and 30% to double glazed while adding insulated drapes reduced the heat loss by 56% and 48% respectively. This is because the thermal resistance of the drapes is added to the thermal resistance of the window.

<u>Cooling Load from Lights:</u> The main source of heat from lighting is from the element inside that emits the light. Calculating the load associated with it in not easy as the rate at which heat gain can be different from the power supplied instantly to those lights. Majority of the heat energy from light is from radiation and only a small amount of the energy from lights is in the form of

convective heat. The load through an 18W light bulb was calculated ((5)Appendix A,5) and found to be $Q_{light} = 21.24Whr$.

The light wattage is obtained from the ratings of the bulb used for general illumination. The light use factor was given assuming the bulb was operational for 24 hours of if room cooling was off at night times. The special ballast allowance factor is the ratio of total installed wattage to the wattage in use, the value while usually being 1 would vary based on the type of light bulb being used as well as the manufacturer.

Cooling Load due to Infiltration: Cooling load Infiltration is the outside air that leaks into a building structure. These leaks could be due to various instances such as an open door, hole and cracks in the walls, or even and open window. For our calculation, it was assumed that an open door was responsible for infiltration as the value could change based on the cause. Our cooling load was calculated ((6)Appendix A,6) and found to be $q_s = 14.28Whr$.

Cooling Load due to an Individual: The final load to take into account is the load generated by individuals. The load people generate varies based on the action they are performing. The Load of people based on the activity they performed was calculated ((7)Appendix A,7) using the heat gain values table ((7)Appendix A,7), the number of people, and the cooling load factor. If the cooling load factor is not given or properly stated, it is always best to assume the value to be one. The latent cooling load was used over the sensible load because as described the latent load was the instantaneous load which was better suited for our system and it did not take into account time delay for heat to be transferred. The value for the heat load due to an individual was found to be $Q_{person,\ latent} = 41Whr$.

Results:

Source of load	Type	Q (Cooling Load) W/hr.
Wall	Structural Insulated Panels (SIPs)	6.84
Wall	Plexiglass	93.28
Roof	Sips roof +Aluminum ceiling tile	17.83
Room	3 SIPs walls + Plexiglass + Roof	131.63
Window	1 Clear glass	22.05
A person	Standing, light work, walking	41
Light bulb	A 18W light bulb	21.24
Infiltration	Door opened occasionally	14.28

Table 2: Cooling Load values for Structural Insulated panels + Plexiglass

	Total = 230.2

Heating Load

Similar to the Cooling load, we are also expected to build our system to provide heating mainly for winter seasons. The Heat Load is simply the amount of heat that must be supplied per unit time to maintain a room temperature at a given level and unlike cooling load, we are mainly looking at conductive heat transfer and latent heat because the heat losses are considered to be instantaneous. The techniques for estimating the heating load are essentially the same for cooling load with the following exceptions:

- Outside Temperature spaces are generally lower than the maintained space temperatures
- Internal heat gains or solar heat gains are not included
- Thermal storage of building structure is ignored
- Thermal conduction effect for walls and roofs are greater in heating load than for cooling load

Calculating the heat load can be done in two forms, the first is an estimated way of determining an approximate value for our heating load. This is done by using your room dimensions assuming they were similar to an office with average insulation and lighting and also adding 500BTU (146.5W) for every additional occupant as shown ((8)Appendix A,8) we found our value to be $Heat\ Load\ estimate\ =\ 820BTU/hr \approx 240.3\ W$.

The issue with this method is simply because it is rough, it doesn't take into account a lot of factors such as, the change in insulations, wall material etc. This method can be used as a way of checking your results to ensure that your calculations are correct because should our heat load calculation be lower than the estimate then, we can tell that some errors were made.

The second form for calculating the heat load is a much more accurate method that takes into account various factors that create a near worst-case scenario for heat load such as the interior(indoor) and exterior(outdoor) conditions, the size and position of windows (shaded or unshaded), the number of occupants, the heat generated by equipment, and the heat generated by lighting, and infiltration and/or ventilation. By individually calculating these heat gains, we can simply add them together to get an overall worst-case heat load value.

Outdoor Conditions: Due to varying weather conditions year to year we have to design our system based on the worst case possible. In order to do this, we made use of the highest values on record for our calculations.

Florida	Heating Dry Bulb Temperature (°F)	
	99.6%	99%
Melbourne	39.0	**43.4**

Table 4: ASHRAE Handbook 2009 Ch 14

Indoor Conditions: The major purpose of the system for heating is to maintain the inside conditions that would keep the occupant the most comfortable but our calculation is obtain values to size the system, and the system will be rarely used to operate at the design conditions. A maximum design dry bulb temperature of **70°F** and **30%** relative humidity is usually recommended for most occupants.

Calculations for the heat load from our various sources were similar to the calculations used in the cooling but more simplified as shown ((9)Appendix A,9) we were then able to generate our table of results.

Source of load	Type	Q (Heat Load) W/hr.
Wall	Structural Insulated panels (SIPs)	14.87
Wall	Plexiglass	81.74
Roof	Sips roof +aluminum ceiling tile	38.76
Room	3 SIPs walls + Plexiglass + Roof	165.11
Window	1 Clear glass	21.4
Infiltration	Door opened occasionally	14.28
		Total = 200.79

Table 5: Heating Load values for Structural Insulated panels + Plexiglass

With our calculated heating load, we are able to see that the values between the calculated and our estimated heat load were off by almost 40W with our calculated being lower. This was because of the fact that we took into account the material type and properties as well as a proper calculation for our infiltration.

With both our cooling as well as our heating load gotten, we are then able to decide the best thermoelectric module as well as how many modules would be needed to remove heat and also provide heat respectively.

Coefficient of Performance (COP):

{Eniola}

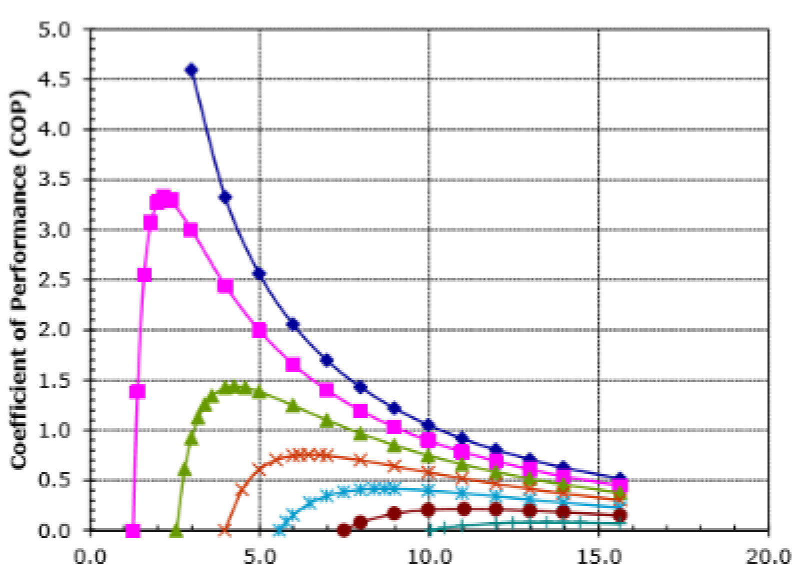
The Coefficient of performance or COP is defined as the efficiency of a system. For our Peltier based system, it is ratio of the Cooling Power from our TE module to the overall input electrical power going into the module or the efficiency of the Peltier elements when cooling.

$$COP = \frac{Q_c}{Q_{te}}$$

Where Qc: Cooling Power

Q_{te}: Power going into the TE module

Typical COP ranges for commercially available Air conditioning units are between 2-4 which is our target COP range for our system. The major issue with Peltier thermoelectric modules is the efficiency, the larger of a temperature difference we have, the less efficient our system would be as depicted on the graph (graph 1) below



Graph: TE-tech manufacturer data of standard TE module

On the left side, we see that the COP is maximum at the lowest temperature difference hence, depending on the dT the corresponding COP would be a varying level. Also following the temperature difference, as it increases, so does the amount of power required but the COP decreases.

The issue of the low efficiency of TE modules is one we intend to solve through the incorporation of Phase Changing Materials (PCMs) as well as maximizing the efficiency of the TEMs by being able to use it at an ideal temperature difference and maximum COP, to achieve this we would need to calculate the COP at various power inputs:

<u>Cooling power (Q_c):</u>

The cooling power as stated earlier is the amount of cold we would need our TE modules to produce. Below is are two different equations on calculating the cooling power based on the basic parameters/properties of the module you are given:

$$* \; Q_c = \alpha I T_c - 0.5 I^2 R - K\left(T_h - T_c\right) \; (better\ form)$$

Or

$$Q_c = 2N(\alpha IT_c - \frac{\rho I^2}{2G} - KG\Delta T)$$

<u>Power of TE module</u> (Q_{te}):

In order to maximize the COP of our TE module we intend to limit the current and voltage at its specified temperature difference. With the equation below, we can determine the amount of power going into the TE module:

$$* \; Q_{te} = \alpha I(\Delta T) + RI^2 \; (better\ form)$$

Or

$$Q_{te} = 2N \, (\alpha I(\Delta T) + \frac{I^2 \rho}{G}$$

<u>Temperature difference</u> (ΔT):

This is found by comparing the temperature of the hot side of the TEM to the cold side of the TEM:

$$* \; \Delta T = T_h - T_c$$

<u>See-beck coefficient</u> (α):

The see-beck effect is the process by which the heating of the junction in a TE module of two different materials produces an electrical potential (current). When estimating the COP, we have to take this into account as small current would be generated within the module:

$$* \; \alpha = \frac{V_{max}}{T_h}$$

<u>Thermal conductivity</u> (K):

As I had stated earlier, the TE module is subject to a loss due to thermal conduction and it has to be taken into account as the heat it generates would reduce the COP of the module:

$$* \ K = \frac{\left(T_h - \Delta T_{max}\right)V_{max} I_{max}}{2T_h \Delta T_{max}}$$

Electrical Resistance (R):

Similar to thermal conduction, electrical resistance is also another form of loss that must be taken into consideration when determining the efficiency:

$$* \ R = \frac{\left(T_h - \Delta T_{max}\right)V_{max}}{T_h I_{max}}$$

Legend

Symbol	Meaning
α	$See - beck\ coefficient\ (\frac{V}{K})$
I	Input electrical current (Amp)
I_{max}	Max Current (Amp)
K	Thermal conductivity ($\frac{W}{K}$)
$T_{h,c}$	$Temperature\ of\ hot\ and\ cold\ side$ (K)
R	Electrical resistance of TE (Ohm)
N	Number of TEC thermocouple
ΔT	Temperature difference
ΔT_{max}	Max Temperature difference
ρ	electrical resistivity
G	Geometry factor

V	Voltage (Volt)
V_{max}	Max voltage (Volt)

Being provided with the current (I), voltage (V), hot side temperature (T_h) and the temperature difference (ΔT) we can determine the COP of the TE module which is shown (Appendix B).

(Minh)

The calculation to get the COP is fairly simple: $COP = \frac{Q_c}{Q_{te}}$ [[8]]

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[8]6 "COPs, EERs, and SEERs." Power Knot, 1 Mar. 2011, www.powerknot.com/2011/03/01/cops-eers-and-seers/.

Eniola

[7] Zhang, H.y. "A General Approach in Evaluating and Optimizing Thermoelectric Coolers." International Journal of Refrigeration, vol. 33, no. 6, 2010, pp. 1187–1196., doi:10.1016/j.ijrefrig.2010.04.007.

Najafi, Hamidreza, and Mohadeseh Seyednezhad. ENERGY AND ECONOMIC ANALYSIS OF A NOVEL HYBRID PHOTOVOLTAIC- THERMOELECTRIC SYSTEM FOR BUILDING COOLING APPLICATIONS. Proceedings of the ASME 2019 International Mechanical Engineering Congress and Exposition , 2019, pp. 1–9.

9 Vi-Hsiang Cheng, Wei-Keng Lin, "Geometric optimization of thermoelectric coolers in a confined volume, 2005
10 M D Kamrul Russel, "A Hybrid Thermoelectric Cooler Thermal Management System For Electronic Packaging", 2011.
11 Evans, Paul. "Cooling Load Calculation - Cold Room." *The Engineering Mindset*, 25 May 2019, theengineeringmindset.com/cooling-load-calculation-cold-room/.
12 *ASHRAE Handbook 2009: Fundamentals*. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 2009.
13 *ASHRAE Handbook 1997: Fundamentals*. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1997.
14 McQuiston, Faye C., and Jeffrey D. Spitler. *Cooling and Heating Load Calculation Manual*. ASHRAE, 1994.

To get the calculation for the COP of the TE module itself however, we'll need a few extra parameters first. Q_c is the cooling power of the TE module, and it can be obtained using:

$$Q_c = \alpha * I * T_c - 0.5 * I^2 * R - K(T_h - T_c)$$

Where: α: See-beck coefficient.

I: Input current

T_c and T_h: Temperature of the cold and hot side respectively.

R: TE internal resistance.

K: Thermal conductivity.

With I, R, T_c, and T_h being able to obtained directly from the data manufacture sheet/table or elsewhere, the rest can be calculated in these equations below:

$$\alpha = \frac{V_{max}}{T_h}$$

$$K = \frac{(T_h - \Delta T)*V*I}{2*T_h*\Delta T}$$

Where ΔT is the difference between T_h and T_c, and V_{max} is the maximum voltage of the TE module (also obtained from the data manufacture sheet/table).

.

15 *Ventilation for Acceptable Indoor Air Quality: ASHRAE Standard*. American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2013.

16 "Heat Load or Heat Gain." *Heat Load Calculations – Heat Gain for Air Conditioner Sizing*, www.tombling.com/cooling/heat-load-calculations.htm.

With those equations, we can estimate the amount of heat the TE can remove from the area. As for the power needed to run the TE module at that level, we need to find the total power required:

$$Q_{te} = 2 * N * \alpha * I * \Delta T + \frac{I^2 * \rho}{G}$$

Where: N is the number of TE module, ρ is the electrical resistivity, and G is the geometry factor.

- ANSYS (tabulate results)
  - Include initial and loading conditions

Xin Hu -Aluminum Ceiling Panel Analysis

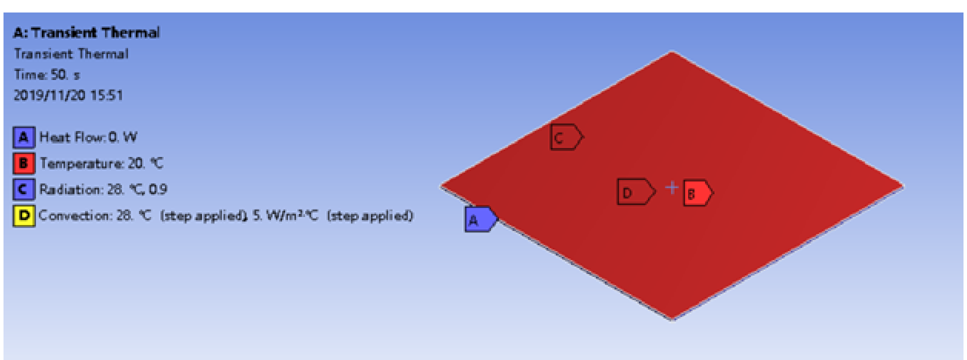
- 

Figure 1 Aluminum Ceiling Panel Analysis Setting

We consider the aluminum ceiling panel as idea condition. The initial temperature is 28 degree(301K). We first assume that TE module can cool down the whole top side of the aluminum ceiling panel, so the top side of the aluminum ceiling panel is constant temperature which is 20 degree (289K). There is convection and radiation on the bottom side of the aluminum ceiling panel. The convective heat transfer coefficient is 5. What's more, the emissivity of the radiation is the 0.9. The surrounding wall is insulation.

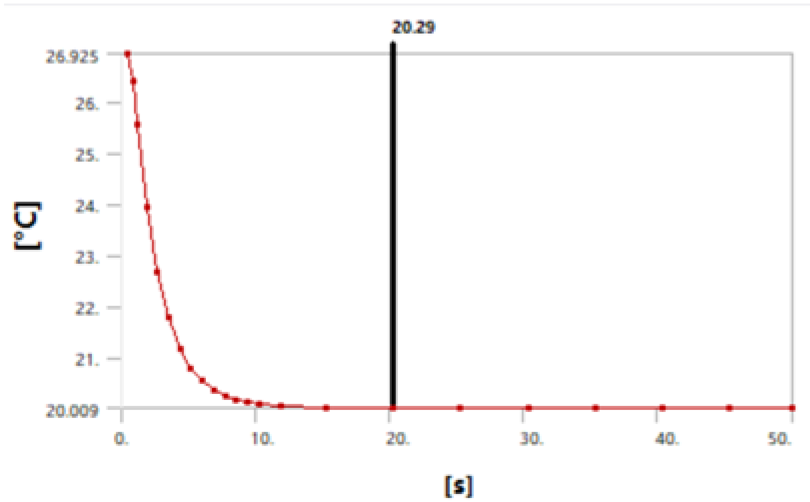


Figure 2 Aluminum Ceiling Panel Analysis Results

We take the probe to know the bottom side of the aluminum ceiling panel temperature. Therefore, for ideal aluminum ceiling panel, it will take 20.29 seconds to achieve the steady state, which is 20 degree (293 K).

We consider that different configuration of TE module distribution will cause some effects on the ceiling cooling system, so the number of the TE module is 12 which can help us to observe the cooling effects. The small square in the following three pictures is the TE module. What's more, TE module can cool down contact region of the top side of the aluminum ceiling panel, so the TE module is constant temperature is 20 degree (289K) and the top side of the aluminum ceiling panel is insulation. What's more, the initial temperature is 28 degree(301K). There is also convection and radiation on the bottom side of the aluminum ceiling panel. The convective heat transfer coefficient is 5 W/(m^2K). What's more, the emissivity of the radiation is the 0.9. The surrounding wall is insulation.

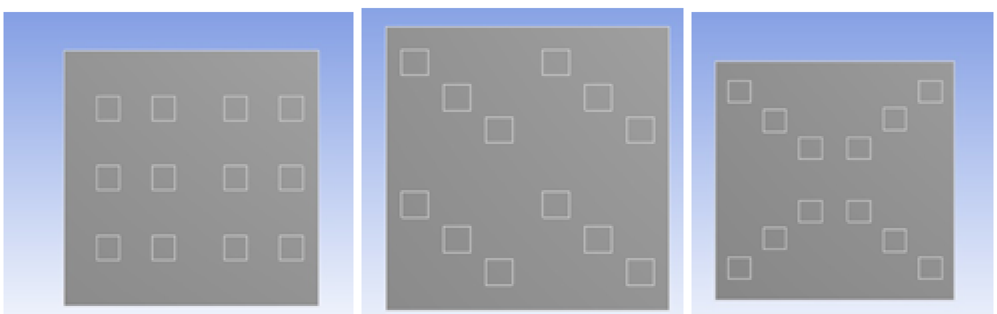


Figure 1-3 Different Configuration of Aluminum Ceiling Panel with 12 TE Module

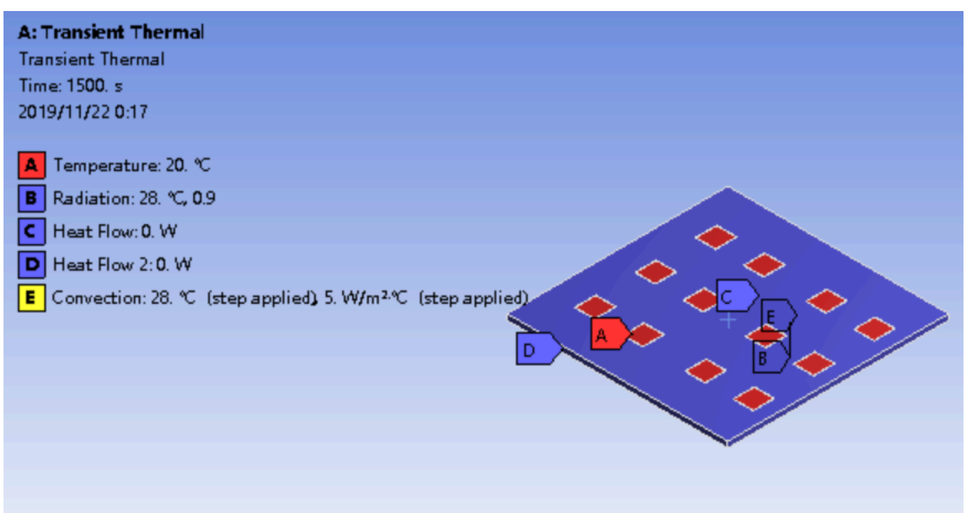


Figure 4 Aluminum Ceiling Panel with 12 TE Module Analysis Setting

	Configuration 1	Configuration 2	Configuration 3
Steady state Achieved Time	813.29	880.89	1000 s

Table 1 Different Configuration of Aluminum Ceiling Panel With 12 TE Module Results

We take the probe to know the bottom side of the aluminum ceiling panel temperature. Therefore, we can know that the configuration 1 is the best choice, which achieve steady state at 813.29 seconds.

Xin Hu Different Number of TE in Aluminum Ceiling Panel Analysis

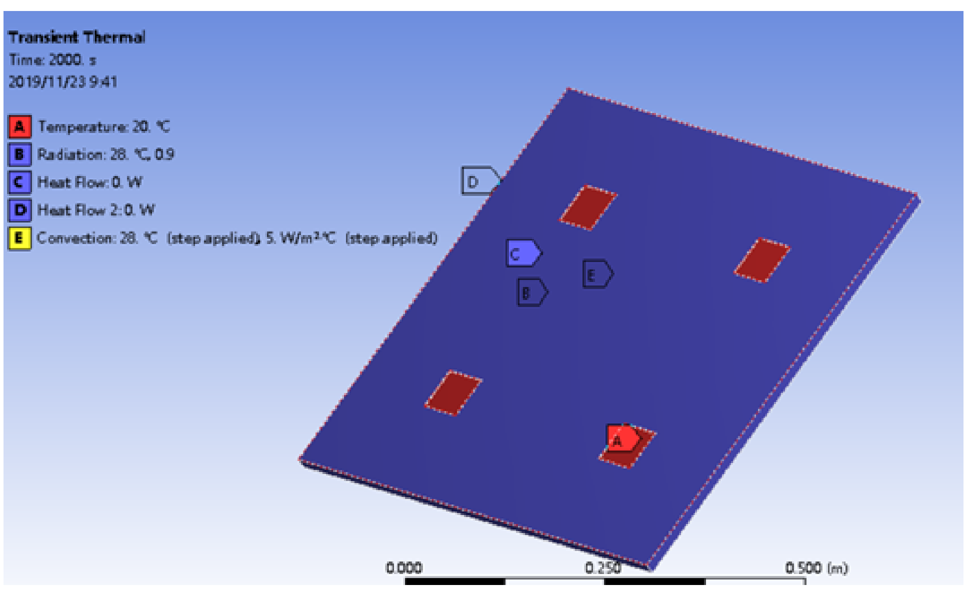


Figure 1 Aluminum Ceiling Panel with 4 TE module Example Analysis Setting

This time we need to determine the number of the TE modules which are used for cooling down the ceiling panels, so we consider that TE module can only cool down contact region of the top side of the aluminum ceiling panel, so the TE module is consider to be is constant temperature is 20 degree (289K) and the top side of the aluminum ceiling panel is insulation. The small square in the picture is the TE module. What's more, the initial temperature is 28 degree(301K). There is convection and radiation on the bottom side of the aluminum ceiling panel. The convective heat transfer coefficient is 5. What's more, the emissivity of the radiation is the 0.9 and the surrounding wall is insulation.

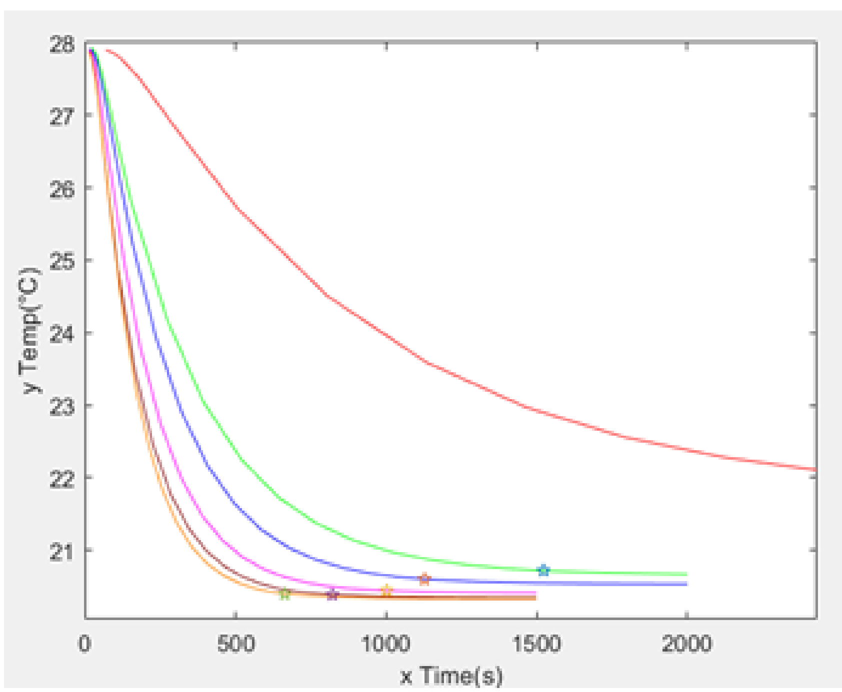


Figure 4 Aluminum Ceiling Panel with Different Numbers of TE Module Analysis Result

Number of TE module	Time to Achieve Steady State
2 TE Modules	5154 seconds
4 TE Modules	1525 seconds
6 TE Modules	1127 seconds
8 TE Modules	1000 seconds
10 TE Modules	821 seconds
12 TE Modules	663 seconds

Table 1 Aluminum Ceiling Panel with different numbers of TE Module Analysis Result

We take the probe to know the bottom side of the aluminum ceiling panel temperature. Therefore, we can know about the steady state time for the bottom side. From this table, we can know that if we use the more TE modules, they can cool down the ceiling panel more quickly. However, we need to consider the cost and performance of TE module, so we choose to use 4 TE module.

Compare with Thickness for PCM Ceiling Panel

Xin Hu

This represents the numerical study and simulation of melting of a PCM for thermal energy storage. This project uses ANSYS as the simulation software, and we will theoretically analyze the ceiling. PCMs are the materials which are used for storing latent heat energy. These materials absorb or release heat during phase change. [6] They can dissipate heat when it changes from a solid to a liquid, or when it changes from a liquid to a solid.

[6] Pielichowska, Kinga, and Krzysztof Pielichowski. "Phase change materials for thermal energy storage." Progress in materials science 65 (2014): 67-123.

Yifei Li **6mm PCM Ceiling Panel Analysis**

After we finish the analysis the steady time of the aluminum ceiling panel, we now start to think the PCM materials. Properties of PCM is in the following table 1. The thickness of the PCM is 6mm. We use the structure Aluminum-PCM. The top side is the aluminum ceiling panel, the size is 1ft*1ft* 6 mm. We put the TE module on it. The bottom side is the PCM panel. Since the configuration of 4 TE module is symmetry, so the cooling effect is the same and we decide to analysis ¼ of the ceiling panels.

Properties of the PCM Ceiling Panel	Value
Size	1ft*1ft* 6 mm
Density	870 kg/m^3
Cp (Specific Heat	2900 j/kg-k
Thermal conductivity	0.2 W/m-K
Viscosity	0.0057933 Kg/m-s

Pure Solvent Melting Heat	190000j/kg
Solidus Temperature	292K
Liquidus Temperature	292.8K

Table 1 Properties of PCM Ceiling

This time TE module contact region with the top side of the aluminum ceiling panel is constant temperature is 20 degree (289K). What's more, the initial temperature is 28 degree(301K). There is convection on the bottom side of the aluminum ceiling panel. The convective heat transfer coefficient is 5 W/(m^2K). What's more, the top side of the aluminum ceiling panel and surrounding wall is insulation.

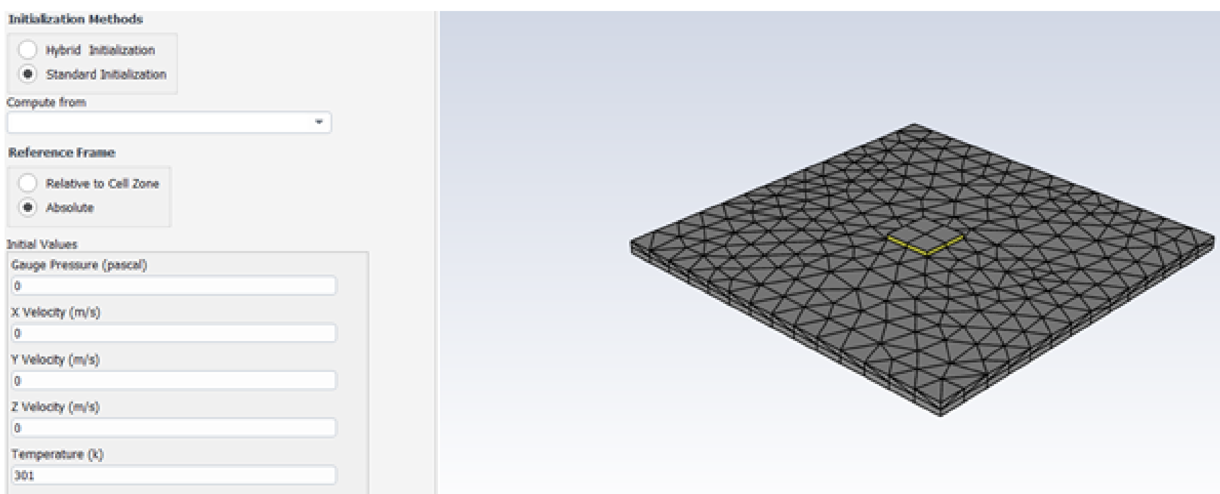


Figure 1 Aluminum-PCM Ceiling Panel with 4 TE module Analysis Setting

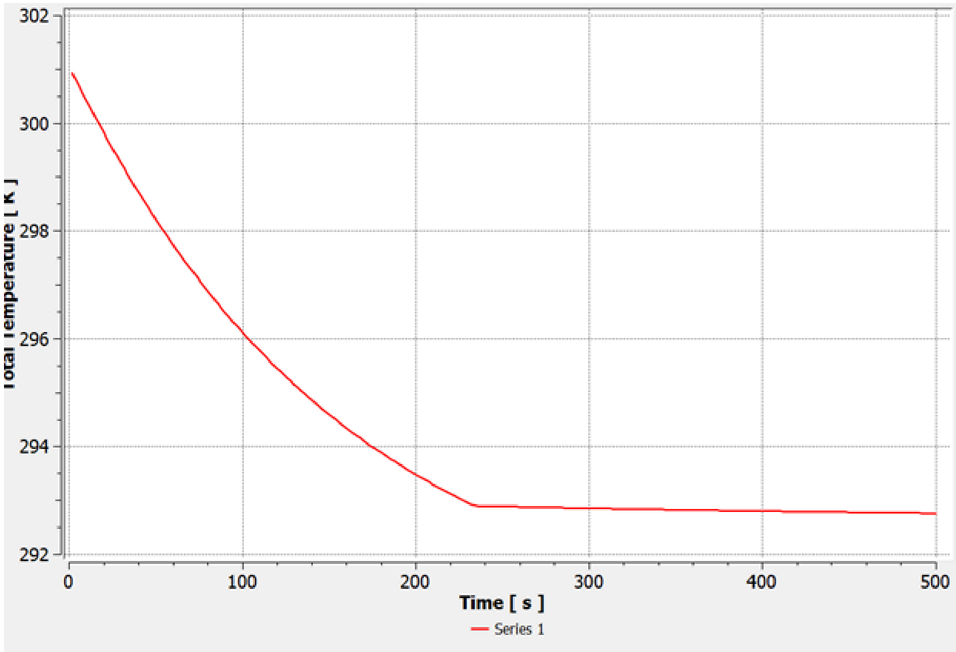


Figure 2 PCM Ceiling Panel with 6 mm Thickness Analysis Result for 500 Seconds

Since the melting temperature is 292k, we can know that the PCM is still not melting after 500 seconds from this graph for structure Aluminum-PCM, of which size is 1ft*1ft* 6 mm for aluminum and PCM ceiling panel.

Xin Hu **1mm PCM Ceiling Panel Analysis**

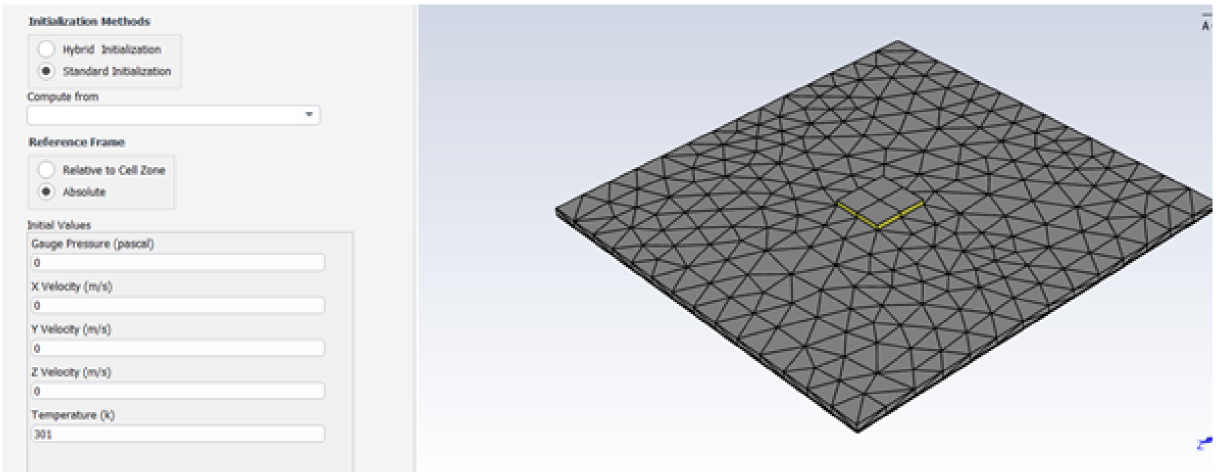


Figure 1 Aluminum-PCM Ceiling Panel with 4 TE module Analysis Setting

Properties of PCM Ceiling Panel	Value
Size	1ft*1ft* 1 mm
Density	870 kg/m^3
Cp (Specific Heat	2900 j/kg-k
Thermal conductivity	0.2 W/m-K
Viscosity	0.0057933 Kg/m-s
Pure Solvent Melting Heat	190000j/kg
Solidus Temperature	292K
Liquidus Temperature	292.8K

Table 1 Properties of PCM Ceiling

After we finish PCM materials, we are going to find out the thickness effects of the PCM ceiling panel. Properties of PCM is in the following table 1. The thickness of the PCM now is 1mm. We use the structure Aluminum-PCM. The top side is the aluminum ceiling panel, the size is 1ft*1ft* 6 mm. We put the TE module on it. The bottom side is the PCM panel. Since the configuration of 4 TE module is symmetry, so the cooling effect is the same and we decide to analysis ¼ of the ceiling panels. This time TE module contact region with the top side of the aluminum ceiling panel is constant temperature is 20 degree (289K). What's more, the initial temperature is 28 degree(301K). There is convection on the bottom side of the aluminum ceiling panel. The convective heat transfer coefficient is 5. What's more, the top side of the aluminum ceiling panel and surrounding wall is insulation.

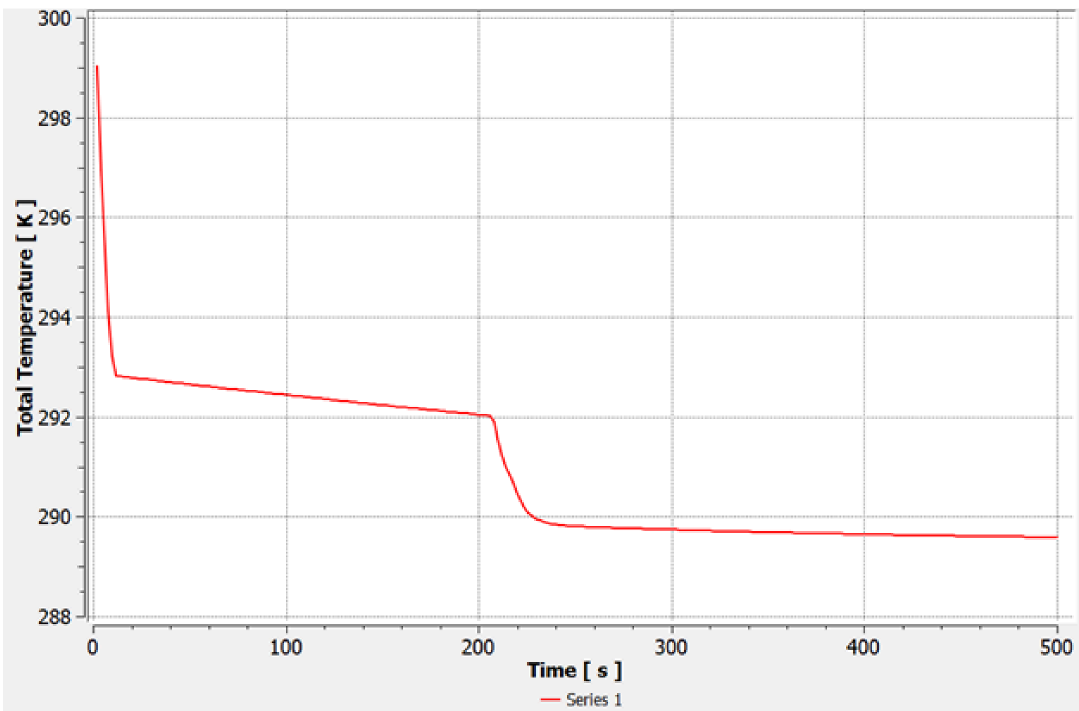


Figure 2 PCM Ceiling Panel with 1mm thickness Analysis Result for 500 Seconds

Compare with the thickness of the 6 mm PCM materials, we can conclude that the performance of the cooling effect of PCM ceiling panel with 1mm thickness is greater than that of PCM ceiling panel with 6mm thickness, so we need to use thin PCM ceiling panel.

Xin Hu- **Reverse Process for PCM with 6mm thickness**

We are going to evaluate the performance for PCM ceiling panel to keep temperature, which is called reverse process. Properties of PCM is in the following table 1. The thickness of the PCM now is 6 mm. We use the structure Aluminum-PCM. The top side is the aluminum ceiling panel, the size is 1ft*1ft* 6 mm.

The bottom side is the PCM panel with constant temperature is 28 degree (301 K). What's more, the initial temperature is 16 degree(289K), so we can see how long can the PCM keeps the temperature. Since the configuration of 4 TE module is symmetry, so the cooling effect is the same and we decide to analysis ¼ of the ceiling panels. What's more, the top side of the aluminum ceiling panel and surrounding wall is insulation.

Properties of PCM Ceiling Panel	Value

Size	1ft*1ft* 6 mm
Density	870 kg/m^3
Cp (Specific Heat	2900 j/kg-k
Thermal conductivity	0.2 W/m-K
Viscosity	0.0057933 Kg/m-s
Pure Solvent Melting Heat	190000j/kg
Solidus Temperature	292K
Liquidus Temperature	292.8K

Table 1 Properties of PCM Ceiling

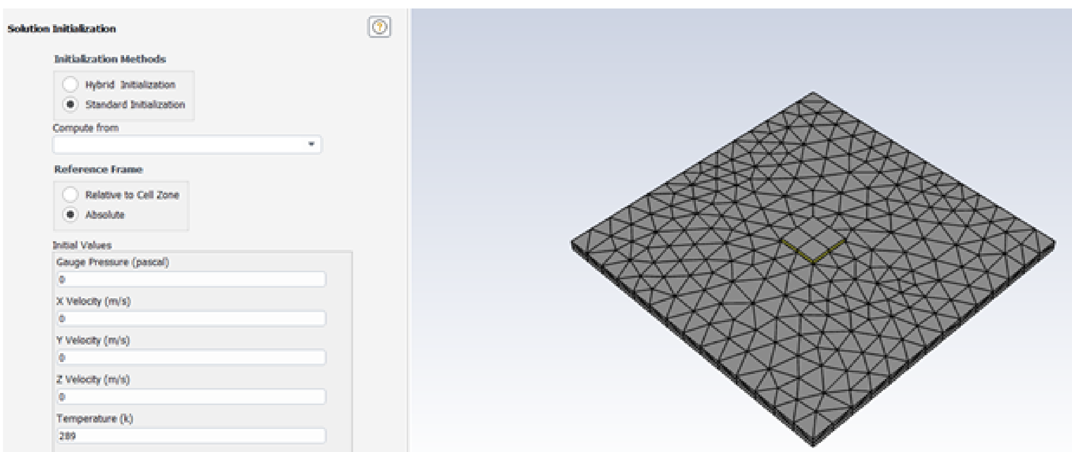


Figure 1 Aluminum-PCM Ceiling Panel with 4 TE module Reverse Analysis Setting

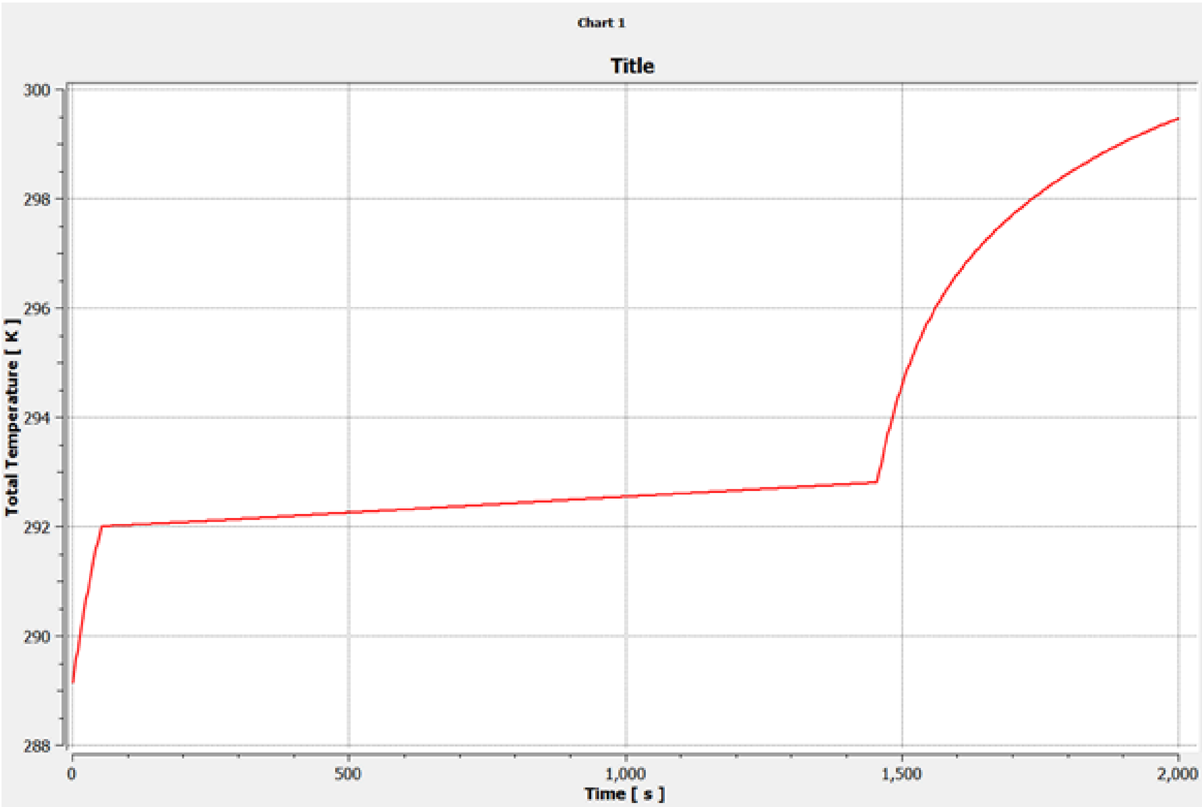


Figure 2 PCM Ceiling Panel with 6 mm thickness Reverse Analysis Result for 2000 Seconds

From this graph, we can know that from 100seconds to 1500 seconds, the whole PCM keeps going the phase change, which means PCM with 6mm thickness can keep temperature for about the 1400 seconds. After 1500 seconds, we can know that the whole PCM is going to the steady state at about 2000 seconds.

Yifei Li- **Reverse Process for PCM with 1 mm thickness**

We are going to evaluate the performance for PCM ceiling panel to keep temperature, which is called reverse process. Properties of PCM is in the following table 1. The thickness of the PCM now is 1 mm.

We use the structure Aluminum-PCM. The top side is the aluminum ceiling panel, the size is 1ft*1ft* 6 mm.

The bottom side is the PCM panel with constant temperature is 28 degree (301 K). What's more, the initial temperature is 16 degree(289K), so we can see how long can the PCM keeps the temperature. Since the configuration of 4 TE module is symmetry, so the cooling effect is the same and we decide to analysis ¼ of the ceiling panels. What's more, the top side of the aluminum ceiling panel and surrounding wall is insulation.

Properties of the PCM Ceiling Panel	Value
Size	1ft*1ft* 1 mm
Density	870 kg/m^3

Cp (Specific Heat	2900 j/kg-k
Thermal conductivity	0.2 W/m-K
Viscosity	0.0057933 Kg/m-s
Pure Solvent Melting Heat	190000j/kg
Solidus Temperature	292K
Liquidus Temperature	292.8K

Table 1 Properties of PCM Ceiling

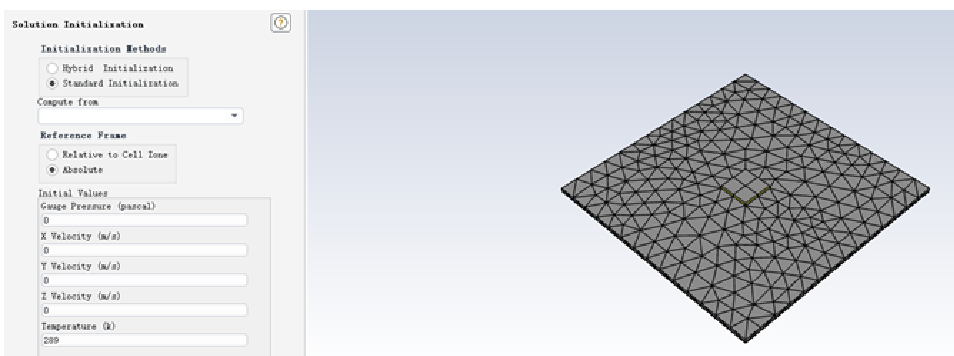


Figure 1 Aluminum-PCM Ceiling Panel with 4 TE module Reverse Analysis Setting

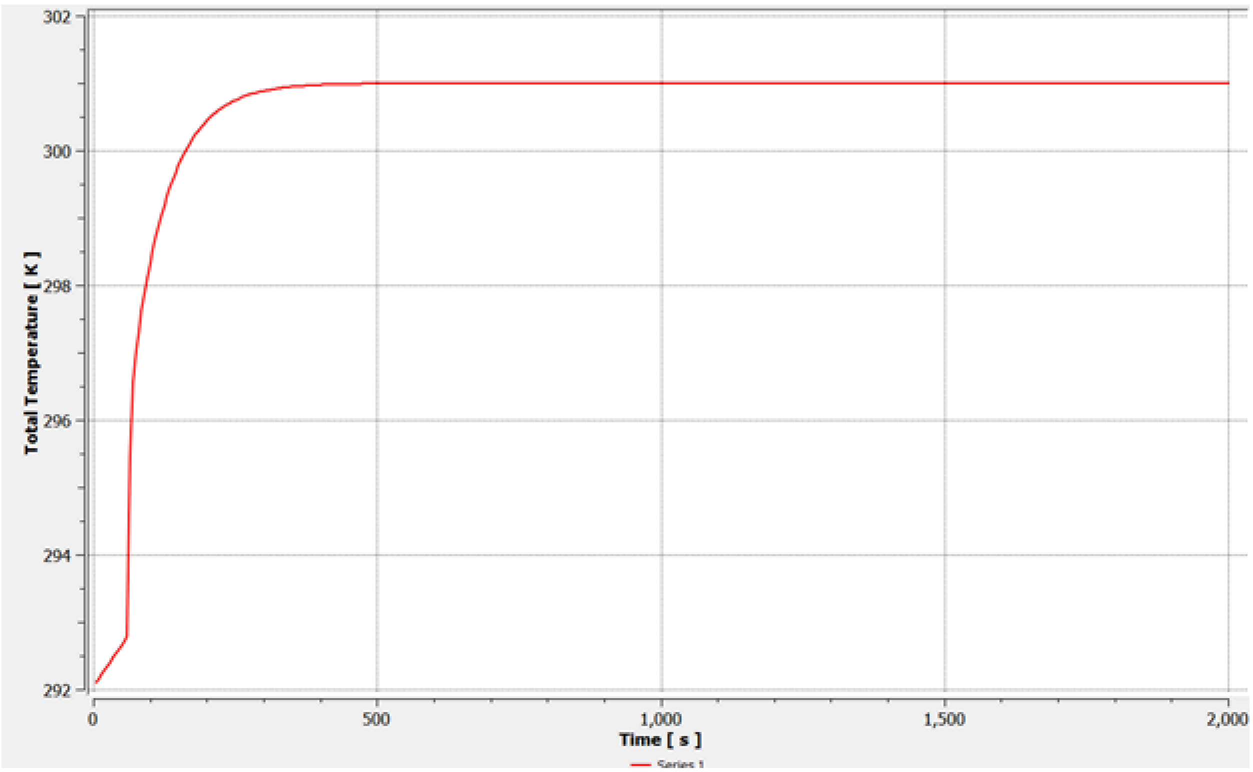


Figure 2 PCM Ceiling Panel with 6 mm thickness Reverse Analysis Result for 2000 Seconds

From this graph, we can know that PCM can only keep the temperature for only 500 seconds for PCM ceiling panel with 1 mm thickness. Comparing with the PCM material with 6mm thickness, we can know that thin PCM cannot keeps temperatures for a long time, so we need to consider the thickness of the PCM to keep the temperature.

Xin Hu- Surrounding PCM Ceiling Panel

We are going to evaluate the performance for surrounding PCM ceiling panel. Properties of PCM is in the following table 1. The thickness of the PCM now is 2 mm. We use the surrounding PCM ceiling panel at the top side. The bottom side is the aluminum ceiling panel, the size is 1ft*1ft* 3 mm.

We put the TE module on PCM. Since the configuration of 4 TE module is symmetry, so the cooling effect is the same and we decide to analysis ¼ of the ceiling panels. This time TE module contact region with the top side of the PCM ceiling panel is constant temperature is 20 degree (289K). What's more, the initial temperature is 28 degree(301K). There is convection on the bottom side. The convective heat transfer coefficient is 5 W/(m^2K). What's more, the top side of the aluminum ceiling panel and surrounding wall is insulation.

Properties of PCM Ceiling Panel	Value
Size	1ft*1ft* 2 mm
Density	870 kg/m^3
Cp (Specific Heat	2900 j/kg-k

Thermal conductivity	0.2 W/m-K
Viscosity	0.0057933 Kg/m-s
Pure Solvent Melting Heat	190000j/kg
Solidus Temperature	292K
Liquidus Temperature	292.8K

roperties of PCM Ceiling

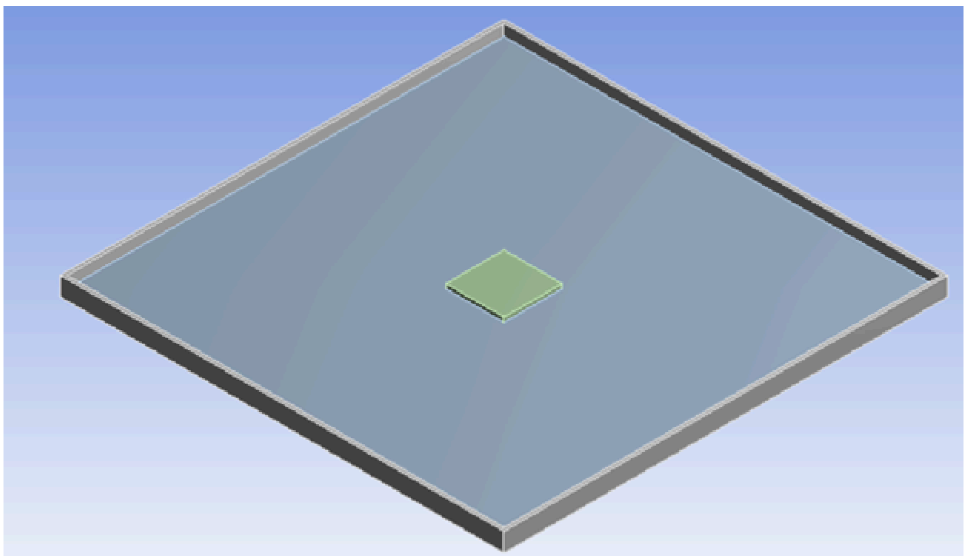


Figure 1 Configuration of Surrounding PCM Ceiling Panel with 4 TE module

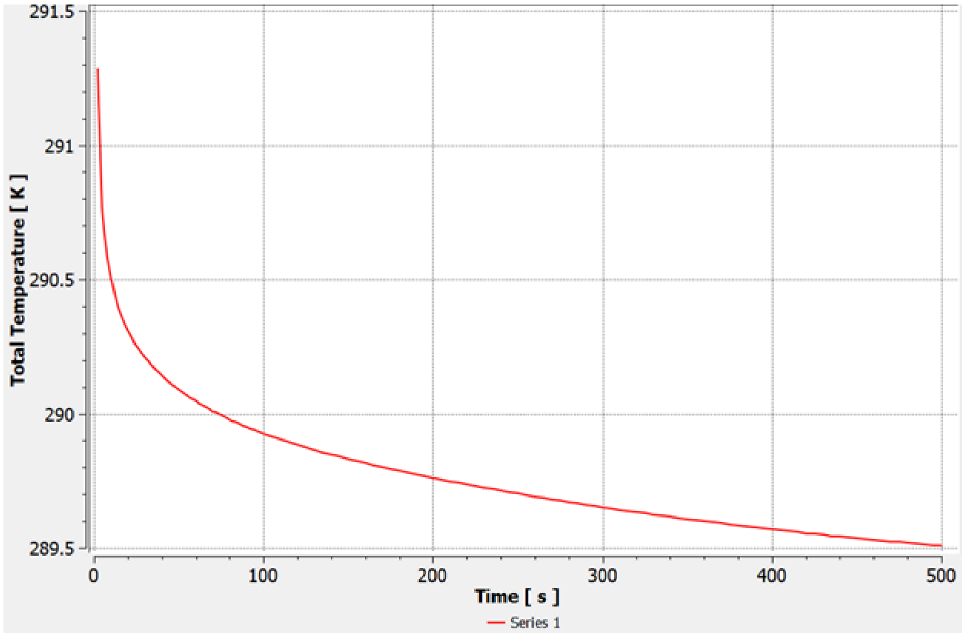


Figure 2 Surrounding PCM Ceiling Panel Analysis Result for 500 Seconds

From this graph, we can know that from Surrounding PCM Ceiling Panel is fast to cool down the bottom side, it takes about the 200 seconds to cool down the Al ceiling panel and it takes 500 seconds to get the steady state.

3.4 FMEA

- Failure Modes and Effect Analysis

3.5 Decision Matrices

- Include criteria for values (-2,-1, 0, 1, 2)

3.6 QFD

- House of Quality

3.7 Bill of Material

- List of materials you plan to use/purchase for the final product (i.e. what materials will a manufacturer/customer need to obtain to create your proposed product)

3.8 Assembly

- CAD models
- How your system will interact/be assembled (theory)

4 Conclusions and Future Work

4.1 Timeline

- Gantt chart
- Important milestones

4.2 Future Work

- Goals for next semester
- How can your project be further developed/improved?
- Other areas/fields to look into

Our goal for next semester is to complete the room model prototype with the ceiling tile and its fully developed control system ready at least one or two months before the senior design showcase date. We will have the time for all the testing and/or fixing any possible errors. We could also add more feature or improving current feature, as well as any details/feature that add on to its aesthetic look and appearances. As of right now the project is designed to cool a room only, but if we have the extra time and with enough funding, we could also work on its heating ability, making it a full-fledge refrigerating system. Since the TE module can work in the other way around, we can also generate power by adding a temperature difference between the two sides, we can also make our system cost less energy by adding that feature, supplying a portion of its required energy by itself. A solar panel to generate power for our system can also be implemented, which makes our system fully “green” is possible for extra research as well.

4.3 Manufacturing and Assembly

- Plan for manufacturing and assembly of system
- Action items

A full manufacturing of the system might not take too much effort, as the biggest part needed to manufacture are just the walls for the room model prototype. The assembly of the system is the tricky one. We have to devise a wiring system in a way that we can put our ceiling tile onto the room, and it wouldn't either tangled with the heat sink above or obstruct the aesthetic of the inside of the room. Putting the walls together can also be troublesome, as we need to find a way so that other people can feel what is happening inside the room, meaning a window or a door is needed, which would then mean that the manufacturing part is going to be tougher. Then we have the ceiling tile itself needed to be assemble. Our research and ANSYS testing of the ceiling tile configuration above give us what we needed to do, but actually doing it can be a different matter entirely. We anticipated that assembling the whole prototype might took us nearly half of the next semester.

4.4 Testing Protocol

- Test procedure and equipment

4.4.1.1.1.1 APPENDICES A

Cooling Load calculations

Cooling Load from Walls:

1) $U_{sips} = \frac{1}{R}$

$$U_{sips} = \frac{1}{12}\ \frac{ft^2°F}{Btu/hr}$$

$$U_{sips} = 0.0833\frac{Btu/hr}{ft^2°F}$$

$$U_{Plexiglass} = 0.8\ \frac{Btu/hr}{ft^2°F}\ (Single\ walled),\ 0.46\ \frac{Btu/hr}{ft^2°F}\ (Double\ walled)$$

Room Area: Area of 4 walls + Area of Roof $A_{Room} = 5.96 + 1.49$ $A_{Room} = 7.45m^2$

2) Sips Material 2in. thick + Plexiglass 0.1875in. thick

$$Q_{sips} = U_{sips} \times A \times (T_0 - T_I)$$

Where $U_{sips} = Heat\ transfer\ coefficient\ of\ Sips\ wall\ used\ (\frac{W}{m^2K})$

A = Area of wall (m^2)

$T_o = Outside\ temperature\ (k)\ for\ Florida\ gotten\ from\ ASHRAE\ Table\ CH\ 14.7)$

$T_i = Temperature\ in\ room\ (K)$

Values used: $U_{sips} = 0.0833 \frac{Btu/hr}{ft^2°F} \approx 0.473 \frac{W}{m^2 K}$ $(R\ value\ of\ 12)$

$$A = 4ft\ X\ 4ft \approx 1.219m\ X\ 1.219m = 1.49m^2$$

$$T_o = 90.8°F \approx 305.7K$$

$$T_i = 73.4°F \approx 296K$$

$$Q_{sips} = 0.473 \times 1.49 \times (305.7 - 296)$$

$$Q_{sips} = 6.84Whr\ (for\ a\ single\ wall)$$

Q for three walls: $Q_{sips} \times 3 = 20.52Whr$ **(3 SIPs walls)**

Or

$$Q_{3\ walls} = U \times A_{3\ walls} \times (T_0 - T_I)$$

Where: $A_{3\ walls} = 1.49\ X\ 3 = 4.47m^2$

$$Q_{3\ walls} = 0.473 \times 4.47 \times (305.7 - 296)$$

$$Q_{3\ walls} = 20.51Whr$$

Determining the heat transfer through the plexiglass involved taking into account two sources of heat generation. The first was the conductive load through the glass and the second was the solar transmission load. These values were then added to give an overall heat transfer value.

i) $Q_{Plexiglass,\ i} = U_{Plexiglass} \times A \times (T_0 - T_I)$

Values used: $U_{Plexiglass} = 0.46 \frac{Btu/hr}{ft^2°F} \approx 2.6 \frac{W}{m^2 K}$

$$Q_{Plexiglass} = 2.6 \times 1.49 \times (305.7 - 296)$$

$$Q_{Plexiglass} = 37.58Whr$$

ii) $Q_{Plexiglass,\, ii} = A \times SC \times SCL$

Where: SHGC = Solar heat gain coefficient (0.77,)

SCL = Solar Cooling Load factor (**42** at 12:00Pm, ASHRAE ONLINE TABLE)

SC = 0.89

$$Q_{Plexiglass,\, ii} = 1.49 \times 0.89 \times 42$$

$$Q_{Plexiglass,\, ii} = 55.70Whr$$

Cooling load for 3 SIPs walls + 1 plexiglass wall: $Q_{four\ walls} = Q_{3sips} + Q_{Plexiglass,i} + Q_{Plexiglass,ii}$

$$Q_{four\ walls} = 20.52 + 37.58 + 55.70$$

$$Q_{four\ walls} = 113.8Whr$$

3) Sips material for Roof

$$Q_{Roof,\, wt} = U_{sips} \times A_{roof,\, wt} \times (T_0 - T_I)$$

Where: $Q_{Roof,\, wt} = Cooling\ load\ of\ Roof\ with\ ceiling\ tile$

$A_{roof,\, wt} = A_{roof} - A_{ceiling\ tile,}$ $(Area\ of\ the\ roof\ without\ the\ ceiling\ tile)$

$A_{roof} = 1.49m^2\ (4ft\ X\ 4ft)$

$A_{ceiling\ tile} = 2ft\ X\ 2ft \approx 0.6096m\ X\ 0.6096m = 0.372m^2$

$$A_{roof,\, wt} = 1.49 - 0.372$$

$$A_{roof,\, wt} = 1.118m^2$$

$$Q_{Roof,wt} = 0.473 \times 1.118 \times (305.7 - 296)$$

$$Q_{Roof,wt} = 5.13Whr$$

-Aluminum Panel for Ceiling tile

$$Q_{Ceiling\ tie} = U_{aluminum,\,p} \times A_{ceiling\ tile} \times (T_0 - T_I)$$

Where: $U_{aluminum,\,p} = 3.52\ \frac{W}{m^2K}$

$$Q_{Ceiling\ tie} = 3.52 \times 0.372 \times (305.7 - 296)$$

$$Q_{Ceiling\ tie} = 12.7Whr$$

Total Q for Roof + Ceiling = 5.13 +12.7 = 17.83Whr

4) <u>Cooling Load from Window:</u>

Conduction: $Q_{Glass,\,con} = U_{glass} \times A_{window} \times (T_0 - T_I)$

Where:

$U_{glass} = 1.93\ \frac{W}{m^2K}\ (Single\ glaze),\ 2.73\ \frac{W}{m^2K}(Double\ glazed),\ 5.34\frac{W}{m^2K}\ (Triple\ glazed)$

$A_{window} = 0.372m^2\ (2ft\ X\ 2ft)$

$$Q_{Glass,\,con} = 2.73 \times 0.372 \times (305.7 - 296)$$

$$Q_{Glass,\,con} = 9.85Whr$$

- Solar Transmission:

$$Q_{Glass,\,sol} = A_{window} \times SC \times SCL$$

Where: SC = $\frac{SHGC}{0.86}$ (Shading Coefficient)

SHGC = Solar heat gain coefficient (0.67, double glazed)

0.86 = Radiant heat transmission property of 3mm clear glass

SCL = Solar Cooling Load factor (**42** at 12:00Pm, ASHRAE ONLINE TABLE)

SC = 0.78

$$Q_{Glass,\, sol} = 0.372 \times 0.78 \times 42$$

$$Q_{Glass,\, sol} = 12.2\, Whr$$

Total Q for window = 9.85 + 12.2 = 22.05Whr

Cooling Load from Lights: $Q_{light} = W \times F_{ut} \times F_{sa} \times CLF$

Where: W = Wattage of light source (18W)

$F_{ut} = Light\ use\ factor\ (1\ for\ 24hr\ run\ time)$

$F_{sa} = Special\ Ballast\ allowance\ factor\ (1.18\ fluorescent\ and\ 1.0\ incandescent)$

$CLF = Cooling\ load\ factor\ by\ hour\ of\ use\ (1\ for\ 24hr\ run\ time)$

$$Q_{light} = 18 \times 1 \times 1.18 \times 1$$

$$Q_{light} = 21.24 Whr$$

5) Cooling Load due to Infiltration: To calculate our infiltration load we made use of the ASHRAE 2009 handbook (Ch16) in order to get the most accurate value for our load and also assumed an open door as our cause of infiltration. Before we could calculate our load due to infiltration, we had to determine the airflow rate (Q_{cfm})

$$Q_{cfm} = C_A \times A \times R_p$$

Where: $Q_{cfm} = airflow\ rate$

$C_A = Arflow\ coefficient\ (for\ door)$

A = Area of door opening

$R_p = Pressure\ factor$

The value for C_A as well as for R_p were gotten from the ASHRAE handbook. The graph shown below is one comparing C_A vs how many times people open\ a door every hour.

Values used: $C_A = 1$

$A = 2ft\ X\ 3\frac{1}{2}ft\ (W\ X\ L) = 7ft^2$

$R_p = 0.3\ (ASHRAE\ 2009\ handbook,\ ch16\ graph)$

$$Q_{cfm} = 1 \times 7 \times 0.3$$

$$Q_{cfm} = 2.1\ cfm$$

With our Infiltration airflow rate calculated, we can now determine the value for our load due to infiltration using the equation.

$$q_s = 1.10(Q_{cfm})(T_i - T_o)$$

Where: $1.10 = assuming\ standard\ air\ indoor\ comfort\ conditons$

$$q_s = 1.10 \times 2.1 \times (294.3 - 273.2)$$

$$q_s = 48.74\frac{Btu}{hr} \approx 14.28Whr$$

6) Cooling Load due to an Individual:

- ASHRAE Tables, Ch 18.4 Journal 2009

	Total Heat			
Activity	Adult Male	Adjusted M/F	Sensible heat ($\frac{Btu}{hr}$)	Latent heat ($\frac{Btu}{hr}$)
Seated, very light work	450	400	240	155

Standing, light work, walking	550	450	250	200
Walking, Standing	550	500	250	250
Seated at rest	**400**	**350**	**210**	**140**
Light bench work	1040	1040	345	695
Moderate dancing	1600	1600	565	1035

Table 1: Sensible and Latent heat gain from person (ASHRAE Handbook)

Sensible heat: Sensible heat gains from people comprises of the heat generated by the person + a percent of radiant energy.

Latent heat: Latent heat is simply instantaneous heat while the sensible heat has to first be absorbed by the surrounding then released (CLF accounts for this), the latent is instant.

(Taking into account seated at rest) 140$\frac{Btu}{hr} \approx 41W$

Calculations: $Q_{person,\ sensible} = N \times SNG \times CLF$

Where: N = Number of People (1)

SNG = Sensible heat gain (210 $\frac{Btu}{hr} \approx 61.5W$)

CLF = Cooling lead factor (1, varies with zone, hour, entering space, and number of hours)

$$Q_{person} = 1 \times 61.5 \times 1$$

$$Q_{person,\ sensible} = 61.5Whr$$

$$Q_{person,\ latent} = N \times LNG$$

Where: $LNG = Latent\ heat\ gain\ (140\frac{Btu}{hr} \approx 41W)$

$$Q_{person,\ latent} = 1 \times 41$$

$$Q_{person,\ latent} = 41Whr$$

<u>Heat Load Calculation:</u>

7) <u>Heat Load Estimate:</u>

$$Heat\ Load\ estimate = Length\ X\ Width\ X\ Height\ X\ 4$$

Where: Length = 4ft (1.219m)

Width = 4ft (1.219m)

Height = 5ft (1.524m)

$$Heat\ Load\ estimate = 4\ X\ 4\ X\ 5\ X\ 4$$

$$Heat\ Load\ estimate = 320 + 500$$

$$Heat\ Load\ estimate = \frac{820BTU}{hr} \approx 240.3\ W$$

8) <u>Calculation of Transmission Heat Losses from walls</u>**:** This calculation takes into account the doors, walls, ceiling and floors and treats them as identical. The equation follows:

$$Q = A \times HF$$

Where A = Area

HF = Heating Load factor

To find the load (Q) we have to first find the heating load factor

$$HF = U \times \Delta T$$

Where: U= heat transfer coefficient of wall material (for us plexiglass and SIPs)

$\Delta T = \left(T_i - T_o\right)$

$T_0 = The\ outside\ heating\ DB\ temperature\ for\ Melbourne$ **(43.4°F** $\approx 273.2K$**)**

$T_i = The\ inside\ design\ temperature$ **(70°F** $\approx 294.3K$**)**

Overall: $Q = U \times A \times \left(T_i - T_o\right)$

Material: Sips 2in. thick (3 walls) + Plexiglass 0.1875in. thick (1 wall)

Structural Insulated Panels (SIPs): $Q_{sips} = U_{sips} \times A \times (T_i - T_o)$

$$Q_{sips} = 0.473 \times 1.49 \times (294.3 - 273.2)$$

$$Q_{sips} = 14.87Whr\ (for\ a\ single\ wall)$$

- Q for three walls: $Q_{sips} \times 3 = 44.61Whr$ **(3 SIPs walls)**

Plexiglass panel: Due to solar heat gain not taken into account when calculating heating load, we ignore the effects on windows and on our plexiglass:

$$Q_{Plexiglass} = U_{Plexiglass} \times A \times (T_0 - T_I)$$

Values used: $U_{Plexiglass} = 0.46\ \frac{Btu/hr}{ft^2°F} \approx 2.6 \frac{W}{m^2K}$

$$Q_{Plexiglass} = 2.6 \times 1.49 \times (294.3 - 273.2)$$

$$Q_{Plexiglass} = 81.74Whr$$

Cooling load for 3 SIPs walls + 1 plexiglass wall: $Q_{four\ walls} = Q_{3sips} + Q_{Plexiglass}$

$$Q_{four\ walls} = 44.61 + 81.74$$

$$Q_{four\ walls} = 126.34Whr$$

9) <u>Calculation of Transmission Heat Losses from Roof:</u> Calculating the heat load from the roof would follow similar steps to the cooling load method except for the changes

in temperature difference. This is because transmission is the only factor taken into consideration when finding the roof load.

Sips material for Roof

$$Q_{Roof,\,wt} = U_{sips} \times A_{roof,\,wt} \times (T_i - T_o)$$

Where: $Q_{Roof,\,wt} = Cooling\ load\ of\ Roof\ with\ ceiling\ tile$

$A_{roof,\,wt} = A_{roof} - A_{ceiling\ tile,}\ (Area\ of\ the\ roof\ without\ the\ ceiling\ tile)$

$A_{roof} = 1.49m^2\ (4ft\ X\ 4ft)$

$A_{ceiling\ tile} = 2ft\ X\ 2ft \approx 0.6096m\ X\ 0.6096m\ = 0.372m^2$

$$A_{roof,\,wt} = 1.49\ -\ 0.372$$

$$A_{roof,\,wt} = 1.118m^2$$

$$Q_{Roof,wt} = 0.473 \times 1.118 \times (294.3\ -\ 273.2)$$

$$Q_{Roof,wt} = 11.16Whr$$

-Aluminum Panel for Ceiling tile

$$Q_{Ceiling\ tie} = U_{aluminum,\,p} \times A_{ceiling\ tile} \times (T_i - T_o)$$

Where: $U_{aluminum,\,p} = 3.52\ \frac{W}{m^2K}$

$$Q_{Ceiling\ tie} = 3.52 \times 0.372 \times (294.3\ -\ 273.2)$$

$$Q_{Ceiling\ tie} = 27.6Whr$$

Total Q for Roof + Ceiling = 11.16 +27.6 = 38.76Whr

10) Calculation of Heating Load from Window: With the removal of solar heat gain that would be caused by solar radiation, we only make use of the conductive heat transfer in our calculation.

$$Q_{Glass,\,con} = U_{glass} \times A_{window} \times (T_i - T_o)$$

Where:

$U_{glass} = 1.93\ \frac{W}{m^2K}\ (Single\ glaze),\ 2.73\ \frac{W}{m^2K} (Double\ glazed),\ 5.34 \frac{W}{m^2K}\ (Triple\ glazed)$

$A_{window} = 0.372m^2\ (2ft\ X\ 2ft)$

$$Q_{Glass,\,con} = 2.73 \times 0.372 \times (294.3 - 273.2)$$

$$Q_{Glass,\,con} = 21.4Whr$$

11) Heating Load from Lights: This is not taken into consideration when calculating the heat load, because the system generates heat which aids the heating process of the room. This is why Lighting and most electrical appliances are not taken into consideration when calculating the heat load.

12) Calculation of Heating Load due to Infiltration: Since air leakages and in this case, cold air leakage is common in building system, heat loss due to this cold air would occur and which is why we must take it into account in our calculation. For this, the dry cold air must be heated to the design temperature indoor and moisture must then be added to increase the humidity.

To calculated our infiltration load we made use of the ASHRAE 2009 handbook (Ch16) in order to get the most accurate value for our load and also assumed an open door as our cause of infiltration. Before we could calculate our load due to infiltration, we had to determine the airflow rate (Q_{cfm})

$$Q_{cfm} = C_A \times A \times R_p$$

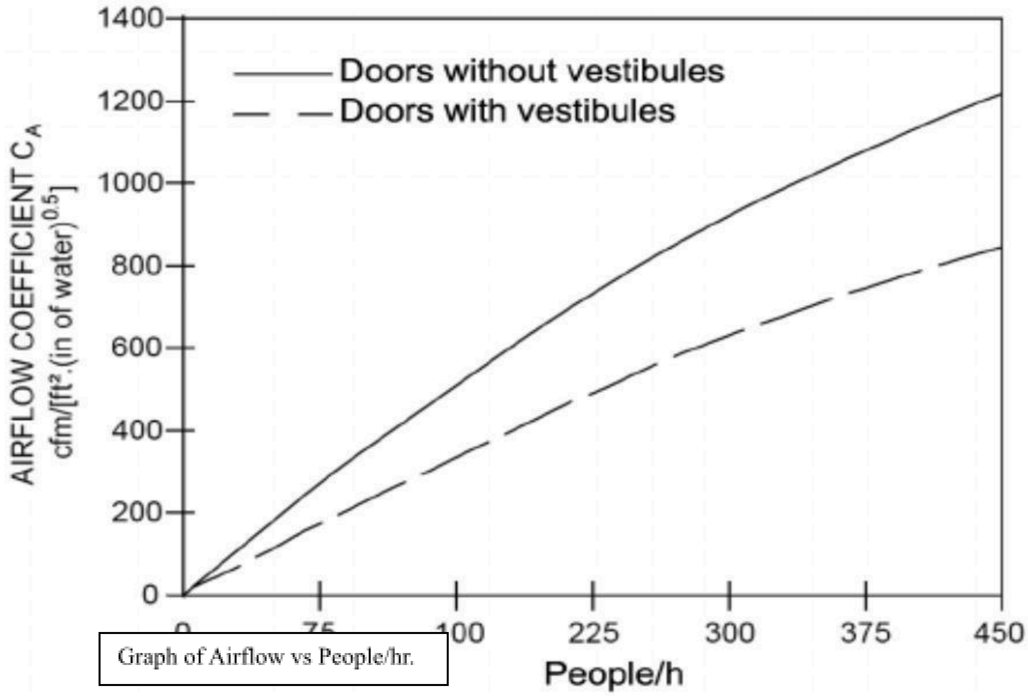
Where: $Q_{cfm} = airflow\ rate$

$C_A = Arflow\ coefficient\ (for\ door)$

A = Area of door opening

$R_p = Pressure\ factor$

The value for C_A as well as for R_p were gotten from the ASHRAE handbook. The graph shown below is one comparing C_A vs how many times people open\ a door every hour.



Graph of Airflow vs People/hr.

Values used: $C_A = 1$

$A = 2ft\ X\ 3\frac{1}{2}ft\ (W\ X\ L) = 7ft^2$

$R_p = 0.3\ (ASHRAE\ 2009\ handbook,\ ch16\ graph)$

$$Q_{cfm} = 1\times7\times0.3$$

$$Q_{cfm} = 2.1\ cfm$$

With our Infiltration airflow rate calculated, we can now determine the value for our load due to infiltration using the equation.

$$q_s = 1.10(Q_{cfm})(T_i - T_o)$$

Where: $1.10 = assuming\ standard\ air\ indoor\ comfort\ conditons$

$$q_s = 1.10\times2.1\times(294.3 - 273.2)$$

$$q_s = 48.74\frac{Btu}{hr}\approx14.28Whr$$

13) <u>Heating Load due to an Individual:</u> Similarly, for lighting, the heating load from an individual is ignored because the occupant is already generating heat based on the action, they are performing so it aids in the heating of the room.

4.4.1.1.1.1.1

Appendix B

Sample COP calculation:

Given Values: $V_{max} = 24.6\ volts$

$I_{max} = 11.3\ amps$

$\Delta T_{max} = 69$

I = 4.4 amps

V = 9.2 volts

ΔT = 19

$T_h = 300K$

$T_c = 281K$

Solution:

- $\alpha = \frac{24.6}{300}$

$$\alpha = 0.082 \frac{V}{K}$$

- $K = \frac{(300-69)\times 24.6\times 11.3}{2\times 300\times 69}$

$$K = 1.55\ \frac{W}{K}$$

- $R = \frac{(300-69)\times 24.6}{300\times 11.3}$

$$R = 1.68\ Ohms$$

- $Q_c = (0.082\times 4.4\times 281) - (0.5\times 4.4^2\times 1.68) - 1.55(300 - 281)$

$$Q_c = 55.65W$$

- $Q_{te} = 0.082\times 4.4\times 19 + (1.68\times 4.4^2)$

$$Q_{te} = 39.36W$$

- $COP = \frac{55.65}{39.36}$

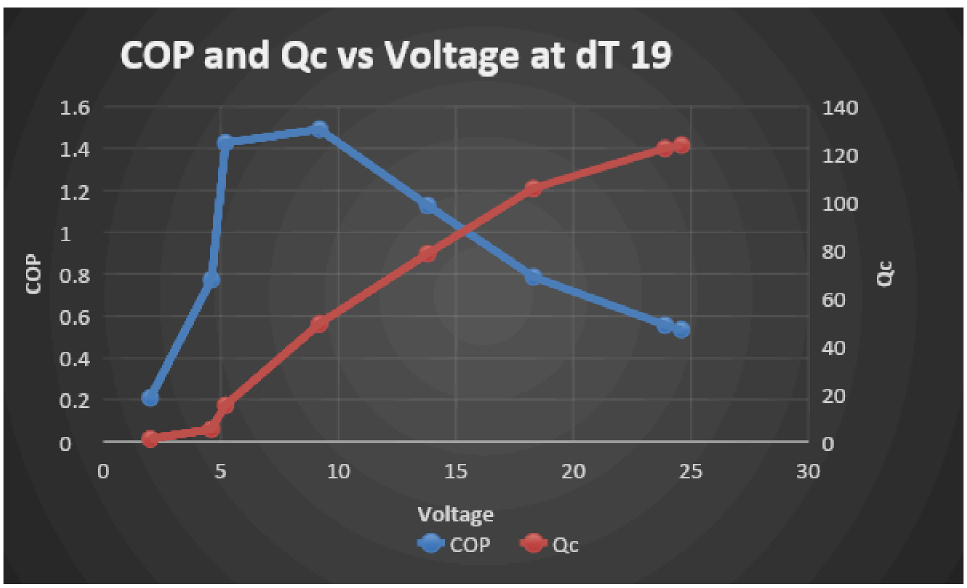
$$COP = 1.41$$

Results at ΔT of 19

Using the equations, we were able to find the COP at varying Current, Voltage, and Temperature Difference. Shown below is a table calculating our COP at varying values for current and voltage.

V_{max}	I_{max}	**Voltage (V)**	**Current (I)**	Th (hotside temperatrue) in K	dT (Temperature difference) in K	Tc (Cold Side temp) in K
24.6	11.3	**2**	**1.4**	300	19	281
24.6	11.3	**4.6**	**1.6**	300	19	281
24.6	11.3	**5.2**	**2.1**	300	19	281
24.6	11.3	**9.2**	**4**	300	19	281
24.6	11.3	**13.8**	**6**	300	19	281
24.6	11.3	**18.3**	**8.5**	300	19	281
24.6	11.3	**23.9**	**11**	300	19	281
24.6	11.3	**24.6**	**11.3**	300	19	281

Max dT	∝	K (Thermal Cond)	R(Thermal Res)	Qc (Cooling pwr)	Qte (Power into TE)	COP
69	0.082	1.551047826	1.676283186	1.146133782	5.466715044	0.209657
69	0.082	1.551047826	1.676283186	5.251648826	6.784084956	0.774113
69	0.082	1.551047826	1.676283186	15.22208688	10.66420885	1.4274
69	0.082	1.551047826	1.676283186	49.28782582	33.05253097	1.491197
69	0.082	1.551047826	1.676283186	78.60899396	69.69419469	1.127913
69	0.082	1.551047826	1.676283186	105.8313612	134.3544602	0.787703
69	0.082	1.551047826	1.676283186	122.5769586	219.9682655	0.557248
69	0.082	1.551047826	1.676283186	123.8823913	231.65	0.534783



4.4.1.1.1.1.2 Title of appendix here

Appendix text.

4.4.1.1.1.1.3 Title of appendix here

Appendix text.

4.4.1.1.1.1.4 Title of appendix here

Appendix text.

4.4.1.1.1.1.5 Title of appendix here

Appendix text.

4.4.1.1.1.1.6 Title of appendix here

Appendix text.

4.4.1.1.1.1.7 Title of appendix here

Appendix text.

4.4.1.1.1.1.8 Title of appendix here

Appendix text.

4.4.1.1.1.1.9 Title of appendix here

Appendix text.

4.4.1.1.1.1.10 Title of appendix here

Appendix text.

4.4.1.1.1.1.11 Title of appendix here

Appendix text.

REFERENCES