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SUSTAINABLE ENERGY TECHNOLOGIES TO IMPROVE FOOD SECURITY AND SUSTAINABILITY OF FOOD CHAINS IN INDONESIA

Year-1 implementation of 2 years research program

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SUMMARY

In Indonesia, postharvest losses of perishable foodstuffs can be up to 50% of primary production and the country faces challenges of food security and sustainability. Postharvest losses can be reduced significantly through the development of an effective cold chain to ensure that not only food security is improved but income of small holder farmers is increased to improve living standards. This research project will make a significant contribution in this field, by evaluating cold chain requirements and developing a road map for its development as well as developing cold storage technologies to improve the sustainability of the food chain.

The project will quantify and evaluate the capacity of cold chain at particular stages of the food supply chain. A comprehensive review of the literature and stakeholder surveys will be carried out to inform the research. The results from the review and survey will be used for evaluation of: (i) the current refrigerated storage capacity in Indonesia; (ii) prediction of the appropriate capacity to satisfy current and projected requirements of food demand and food security in Indonesia, a numerical model will be established for this purposes; (iii) analysis of demand for cold storage facilities to reduce post-harvest losses and food waste. The research will also investigate the utilization of sustainable energy technologies for food-storage applications to reduce fossil-fuel use and environmental impacts. The innovative approach will combine the advantages of the use of natural refrigerants and renewable energy and bio-phase change materials (bio-PCMs) and vacuum wall technology. Proof of concept of the innovative approach will be established. Prior to the establishment of the proof of concept, a numerical model will be developed to simulate its energy performance at different requirement and operating conditions. The environmental impact and economic viability of the innovative approach will also be simulated.

The expected results of this research will include: (i) comprehensive data of refrigerated storage in Indonesia, including prediction of appropriate capacity to fit current and future demand to satisfy the needs of the food supply chain; (ii) a Road Map for cold chain development particularly cold storage requirements; (iii) proof of concept of an innovative cold storage system with sustainable energy technologies employing integrated thermal energy storage using bio-based phase change materials (bio-PCMs) and natural refrigerants such as CO_2 and hydrocarbons to minimize or eliminate grid energy demand and greenhouse gas (GHG) emissions; (iv) at least 2 publications in high quality international journals and 2 international conference publications. Alternatives of the high quality international journals include: *Applied Thermal Engineering*, *International Journal of Refr*igeration, or *Applied Energy;* (v) briefing documents (an international book with ISBN) to inform the research findings to the policymakers in Indonesia and in the UK.

In order to achieve the research targets, a collaboration has been established with the Institute of Energy Futures, Brunel University London (UK). This collaboration will provide opportunities to carry out part of the research at the Institute due to insufficient research resources at Bali State Polytechnic. Inkind support has been provided by the research partner which cover the use of research facilities such as modelling, renewable energy and CO_2 refrigeration test facilities, data logging system (DataScan Logger) as well as access to international journals through Brunel University e-journal system.

Keywords: Food supply chain, cold storage, sustainable energy technologies, environmental impact, energy performance and economic viability.

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CHAPTER I: INTRODUCTION

1.1 Background

Indonesia is an agriculture country with population number of 258.5 million. Poultry, beef and veal production are anticipated to increase 3 to 5 percent annually through 2020, while consumption is expected to rise 4 to 6 percent annually (Miller, 2016). The country has experienced unbalanced food supply and demand which may need to be well-adjusted through import policy. As an example, beef supply of the country in 2016 was estimated 348,020 tons, while the demand was 651,420 tons (BPS, 2016). Unbalanced food supply will boost the price of food, sometimes reaching unreasonable level which very much affects the lives of low income populations. This is one of the challenges for the country in improving food security and sustainability.

Furthermore, FAO (2016) stated that Indonesia is the second highest producers of fishery and aquaculture products in the world. Fishery and aquaculture production of the country in 2014 was 14.33 million tons. In 2016, the production reached 23.03 million tons which 27.6% came from marine fisheries and 72.4% from aquaculture (MMAF-RI, 2016). Additionally, Indonesian territory consists of 2/3 of water, has given enormous benefits for Indonesia, especially fishermen. To improve the economic level of fishermen requires efforts to develop proper facilities. One of the efforts is by improving the quality of products which can be marketed in the regional and international levels. It is certainly need the support of the existence of various fishery facilities, one of which is cold storage (Putri and Munaf, 2013).

The development of cold storage for fishery industry in Indonesia is less than satisfactory. As the second highest producers of fishery and aquaculture products in the world, the country has only 2 out of total 6 of its ocean fishing ports possess cold storage facilities. Moreover, only 4 out of total 14 national fishery ports own cold storage facilities (Putri and Munaf, 2013). Indonesia is also on the list of low index cold storage markets with abundant natural resources (Miller, 2016). The lack of cold storage facilities has greatly restricted the development of its fishery industry. The current infrastructure is also too poor to exploit the resources efficiently. Under this circumstance, technologies that can encourage development of infrastructure including cold storage in the country is in extremely great demand.

The demand for refrigerated facilities such as refrigerated warehouse, cold storage and retail refrigeration in Indonesia is expected to increase with the country’s economic development because they have a vital role to play in reducing post-harvest losses, improving quantity and quality of fishery and aquaculture products and maintain food supply to consumers. The facilities will enable to store over supply of foodstuffs during crop season and use them when there is no crops. The refrigerated facilities are also essential for food quality preservation which are required for the development of national health, proper food costing and improvement of the competitive power of farmers and fishermen.

With respect to environmental aspect, global trend of increasing consumption of food products has an increase impact on greenhouse gas emissions (GHG) or climate change due to energy consumption. It is estimated that for Western Europe the food industry is responsible for between 20% and 30% of GHG emissions (Tassou and Suamir, 2010). A major source of emissions is energy use by manufacturing processes, food distribution and retail. In the UK, food distribution and retail are responsible for approximately 7% of total GHG emissions which accounts for 4 $MtCO_{2\text{-}e}$ (million tons CO_2 equivalent) as reported by DEFRA (2005). In Indonesia, the government is committed to reduce GHG emissions by 26% (on the own) and by 41% (with international support) from emission baseline level in 2020. GHG emissions of the commercial sector was estimated for about 0.7% (accounted for 3.8 $MtCO_{2\text{-}e}$) of the country's emissions of 540 $MtCO_{2\text{-}e}$ (Sugiyono *et al*., 2014). Refrigeration technology is responsible for 15% of all electricity consumed worldwide (Coulomb, 2008) and approximately 72% of the global warming impact of refrigeration plant is due to energy consumption (Cowan et al., 2010). Reducing the energy consumption of refrigeration plant has therefore become one of the key priorities in the reduction of GHG emissions of the food chain.

There is considerable concern about the large quantities of food wasted globally and food security. Indonesia has ambitions to increase food security through increased local production and reduction of waste through improvements in infrastructure alongside reductions in greenhouse gas emissions to meet the COP21 Paris agreement.

There is little information in the current state of cold chain infrastructure in Indonesia and future needs to meet local food production and food security aspirations. This research project is planned to meet this gap through in-depth studies and the development of a roadmap for the development of cold chain infrastructure in Indonesia. There are also research and development activities on the development of solar powered refrigeration systems for food cooling and preservation applications but activities in Indonesia are limited. The current project will contribute to the international effort through the numerical study of innovative integrated renewable energy technologies that: a) minimize thermal energy requirements of cold storage facilities through the employment of advanced insulation materials, such as vacuum insulation; b) development and use of vegetable based phase change material (bio-PCM) storage to enable operation during periods of low solar insolation and maximum demand on the grid; c) use of solar powered refrigeration systems that employ natural refrigerants; can be easily tailored to specific food product needs and applications.

1.2 Problem Identification and Research Urgency

As described in Section 1.1 previously, several problems have identified with regards to the food chain as follows:

- Indonesia and other developing countries in the world are currently facing problems of their food chain which include food supply and low quality food products. The countries frequently experience unstable food supply and fluctuated food price, over supply of food during harvest and lack of supply when the crops are failed. Most of the over supply food are wasted due to insuffcient refrigerated storage.
- There is a need to evaluate current refrigerated storage capacity in Indonesia in order to predict appropriate capacity to fit the food demand as well as to estimate future demand of such facility in improving a smooth food supply especially perishable food such as onion, garlic, fishery and aquaculture products.
- The use of renewable energy in the food chain will reduce the impact to the environment. There is, however, still a very minor utilization of renewable energy in Indonesia compared to its renewable energy potential.
- Required technologies which integrates the advantages of the use of natural refrigerants; solar energy and phase change material (PCM).
- There is a need of comprehensive investigations to examine the influence of energy efficient and environmentally friendly technologies to food supply chain of the country especially the remote areas on spread islands.

1.3 Collaboration with Overseas University

Research collaboration was firstly conducted with School of Engineering and Design of Brunel University which has been established since 2013. This was still at early stage of collaboration which involve principal researcher and Head of the School of Engineering and Design, Brunel

University. In 2017, the collaboration has been formalised under a Memorandum of Understanding (MoU) which involves both institutions: Bali State Polytechnic (BSP) Indonesia and Institute of Energy Futures, Brunel University London (UK). The collaboration is now improving to a further level where researches take place in both institutions. The research area covers refrigeration, air conditioning, building energy, controls and renewable energy technology specifically for the application in the food chain. The MoU and Research Agreement can be seen in Appendix 4.

For this project the research partner, the Institute of Energy Futures, Brunel University London (IEF), will allow the researcher to use IEF research facilities such as CFD (Computational Fluid Dynamics) and EES modelling facility to establish numerical model. The research partner will also allow the researcher to use the renewable energy and CO_2 refrigeration test facilities to practically learn and explore the technologies applied.

Other in-kind contributions from the research partner will include: an access to international journals through Brunel e-journal system (2 years). The research partner will also assist the researcher from Indonesia to arrange a visit to the IEF Brunel University London to carry out the research. For this visit the Indonesia institution will fully fund the accommodation and transportation for a maximum of 6 weeks every year.

Works that will be accomplished in this collaboration include:

(i) Prepare, test and execute appropriate questionnaires to collect data and information through surveys and carry out literature review to evaluate and determine current cold chain capacity in Indonesia (BSP);

(ii) Evaluate data and design approaches and develop models to estimate future cold storage requirements (BSP);

(iii) Develop a Road Map for the development of cold chain infrastructure and cold storage systems to 2040 (BSP and IEF).

(iv) Design and develop a numerical approach to proof of concept stage an innovative renewable energy cold storage system that incorporates bio-based PCM energy storage, natural refrigerants and vacuum insulation to reduce energy demand and emissions (BSP); Vacuum insulation walls with integrated PCM (IEF). CO_2 refrigeration investigations (IEF). Hydrocarbon refrigeration (BSP). Integrated system and controls (IEF&BSP).

CHAPTER II: LITERATURE REVIEW

2.1 Refrigeration Technology in the Food Chain

Refrigeration is important in food chain both maintaining the safety and quality of many foods and enabling food to be supplied from productions to consumers. Refrigeration has also a vital role to play in reducing post-harvest losses. Less than 10% of such perishable foodstuffs are in fact currently refrigerated and it is estimated that post-harvest losses currently account for 30% of total production (Coulomb, 2008). IIR (2009) reported that total global food production was 5500 million tons, at least 33% required refrigeration but only 7% were preserved through refrigeration; this results in huge losses.

Indonesia as one of lower-middle income economies is in the burgeoning stages of retail and cold chain development. In 2016, the country has cold storage capacity of 12.3 million m^3 with Cold Chain Competiveness Scorecard 4 out of 7 (Miller, 2016 and IARW, 2016). Indonesia needs extensive investment for efficient cold chain systems.

2.2 Technologies towards Security and Sustainability in the Food Chain

Refrigeration in the food chain is necessary for maintaining the quality and prolonging the shelf-life of fresh, frozen and perishable products after harvest, after fishing, during transportation, store, display and consumers. Refrigeration also helps stabilize market prices and maintains a smooth food supply to the consumers. Refrigeration, however, consumes considerable amount of energy in the food chain. Within refrigerated storage facilities, for example, 60-70% of the electrical energy used is for refrigeration (Evans and Gigiel, 2010). Tassou, *et al*. (2009) reviewed the potential of alternative refrigeration technologies to reduce energy consumption in food refrigeration. Their review focused on seven systems which include: trigeneration, air cycle, sorption systems, thermoelectric, Stirling cycle, thermoacoustic and magnetic refrigeration.

Optimization of cold storage for improving energy performance also reported by Kozak *et al*. (2017) that experimentally and theoretically study of cold storage packages containing PCM. The arrangement could sustain the product cold longer through taking advantage of the PCM latent heat. Oró, *et al*. (2014) also reported that the implementation of PCM in cold storage could reduce CO_2 emissions by a range between 5% and 22%. While utilization of solar radiation to provide cooling can be as an alternative technology to reduce energy use in food chain. Application of solar energy in cold storage utilizes both solar thermal and photovoltaic system was reported by Basu and Ganguly (2016). Wang and Dennis (2015) investigated energy performance of battery storage and PCM storage in a PV cooling system. It was reported that best performing PCM storage could provide credible savings compared to a system without PCM.

To date, there is limited information of research and development activities on the development of solar powered refrigeration systems for food cooling and preservation applications in Indonesia. The current project will contribute to the international effort through the numerical study of innovative integrated renewable energy technologies that minimize thermal energy requirements of cold storage facilities through the employment of advanced insulation materials, such as vacuum insulation; develop and use bio-PCM storage to enable operation during periods of low solar insolation; and use solar powered refrigeration systems that employ natural refrigerants. The technologies can be easily tailored to specific food product needs and applications.

2.3 Previous Researches and Achievements

The previous researches were focused on the investigation of approaches and technologies for energy sustainability in the food sector particularly supermarkets. Some researches also investigated sustainable energy system for application in commercial buildings especially hotel buildings. The researches in the food sector include: integration of trigeneration and CO_2 based refrigeration systems for energy conservation in the food industry; integrated thermal energy storage (PCM) in food refrigeration equipment for energy and CO_2 emissions reduction; Application of hydrocarbon refrigerant R-1270 on a vertical multideck display cabinet; Investigation on CO_2 gas cooler for commercial refrigeration applications; and temperature performance investigation on a serve-over display cabinet with CO_2 as refrigerant. Some of the research works have been published through international journals and conferences. The publications of the research works can be seen in Appendix 5.

2.4 Description of the Proposed Sustainable Energy Technology

The technology proposed is a cold storage system for food supply chain. The proposed system will combine the utilization of sustainable and recent refrigeration technologies with natural refrigerants (hydrocarbon and CO_2) incorporating bio-PCM as thermal energy storage. PV system will generate electricity from solar energy and will be used as the main drive of the refrigeration system. There will be no battery storages used in this system. The refrigeration system comprises a main refrigeration system with hydrocarbon as its primary refrigerant as shown in Figure 2.1. The main refrigeration system cools a secondary refrigerant (CO_2) which is circulated to the cold room using a CO_2 pump. The liquid CO_2 absorbs heat in a flooded evaporator, then returns back to the main refrigeration system. This arrangement will utilize advantages of energy efficient refrigeration system using hydrocarbon refrigerant (Suamir *et al*., 2012a) and highly efficient CO_2 secondary fluid system (Suamir *et al*., 2012b). PCM arrangement in the cold room will be designed to enable the storage room remains cold when the electricity is not available especially during night time or off cycle. During off cycle, CO_2 refrigerant in the secondary loop will be kept cold by utilizing cooling from the cold room provided by PCM through the flooded evaporator. The secondary loop system will be properly insulated to minimize heat gain. The technology will be highly flexible for Indonesia application as an archipelago country. It can be an excellent solution for remote areas on distributed islands with shortage electricity supply. However for the island which are well electrified, the technology can be run as hybrid cold storage systems.

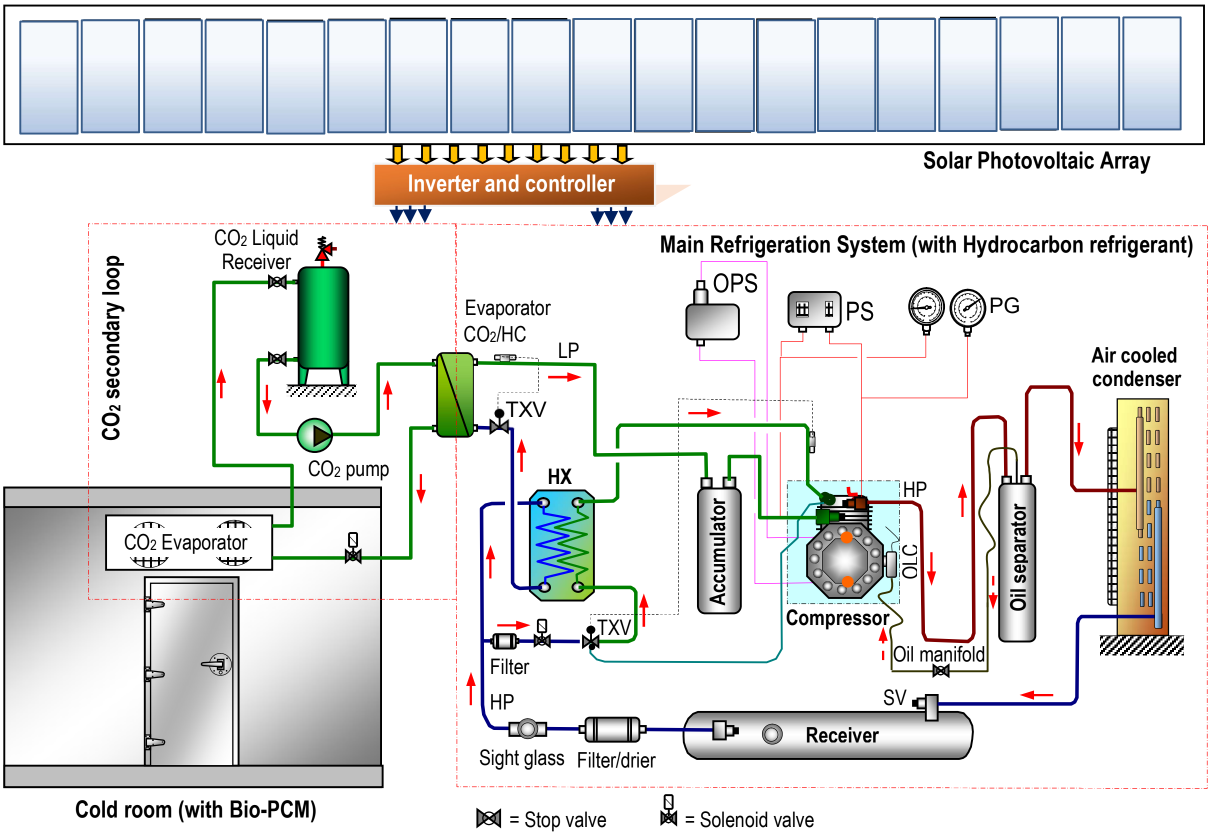


Figure 2.1: Proposed technology: a solar energy driven cold storage incorporated environmentally friendly refrigerants and bio-PCM

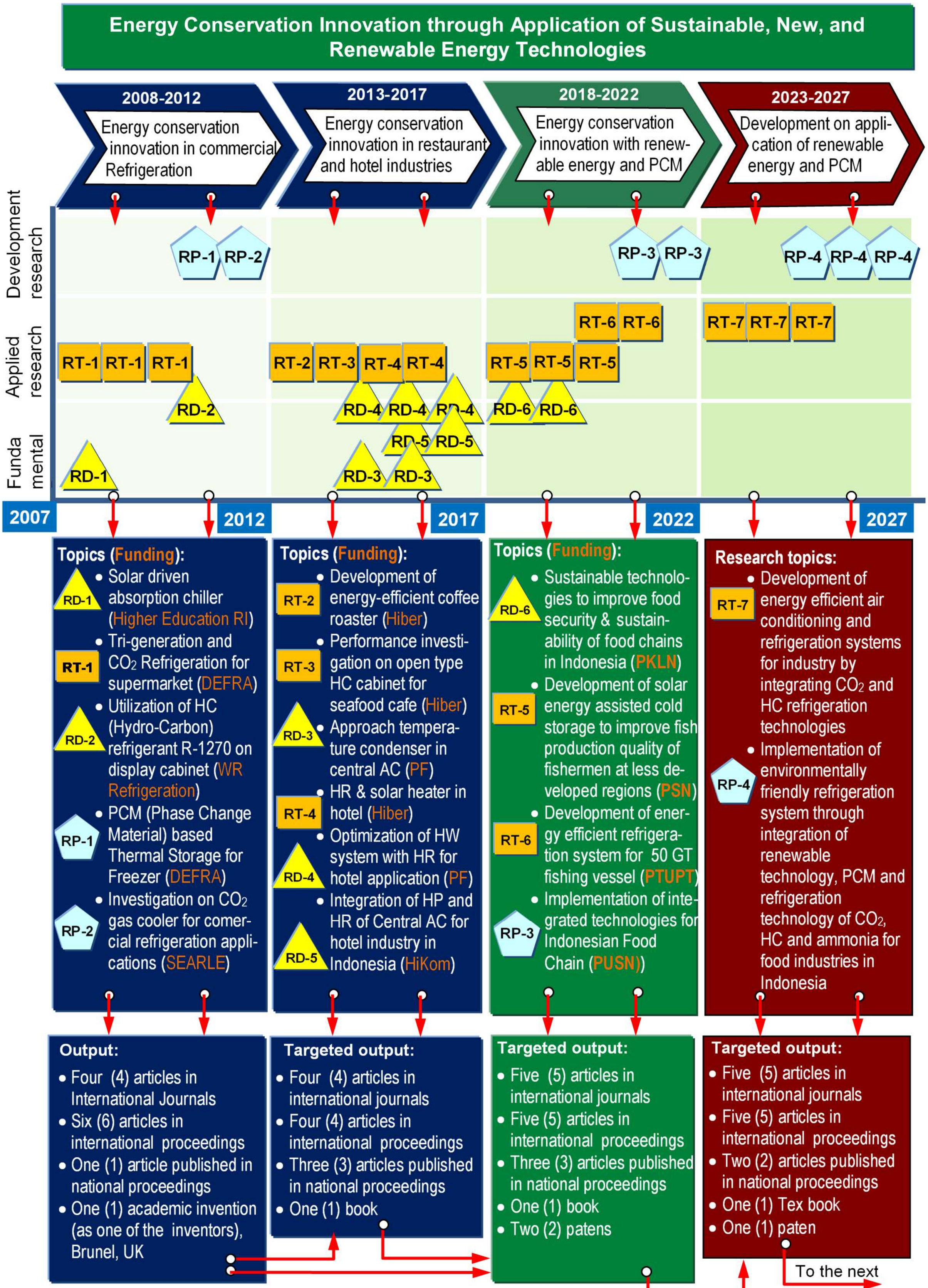
2.5 Contribution of the Proposed Research

The proposed research will make contributions to the overall effort to reduce the energy consumption and environmental impact of the Indonesian food chain through utilization of sustainable energy and energy efficient refrigeration technologies. The research will also deliver specific contributions as follows:

- Provide information of the need of sufficient refrigerated storage capacity for post-harvest and after fishing of food and fish productions to the policy maker in Indonesia with regards to the problem solutions of an unstable perishable food supply to the community due to post-harvest losses or fish wasted.
- The proof of concept of the technology proposed, where renewable energy can be integrated within refrigeration system should improve confidence in the reliability of practical application of the concept in the food chain.
- The numerical model developed and knowledge gained enable the design, sizing and optimisation of refrigerated storage for storing perishable food to ensure maximum performance and utilization efficiency.

2.6 Research Roadmap

Research roadmap of the principal researcher illustrates the research projects that have been completed, the current researches and the future research projects. In general the road map is divided into four sub-topics (focuses) within 5 years time period for each focus. The research roadmap has also been adjusted according to the new research guide book XI published in 2017 by DRPM Dikti. The researches are categorized into three levels in accordance with technological readiness, i.e.; fundamental, applied and development research. Detailed research road map of the principal researcher can be seen in Figure 2.2.



Note:
DEFRA, WR Refrigeration, SEARLE = funding sources from UK, PCM = Phase Change Materials
Hiber, PF, Hikom, PKLN, PSN, PUSN, PDUPT, PTUPT = research schemes from Ministry of Research, Technology and Higher Education
AC = Air Conditioning, HP = Heat Pump, HR = Heat Recovery, HW = Hot Water, HC = Hydro-Carbon

Figure 2.2: Research roadmap of the principal researcher

CHAPTER III: RESEARCH AIMS

3.1 Research Aims

The principal aim of the project is to set the foundations for long term collaboration between the Bali State Polytechnic and Institute of Energy Futures in the UK to make contributions to the improvement of the integrity of the cold food chain in Indonesia.

Specific objectives include:

(i) Evaluation of the current state of the cold chain in Indonesia and the development of a road map for its development to satisfy current and future needs of the country.

(ii) Evaluation of cold storage needs to satisfy specific product requirements and future food distribution patterns paying particular attention to local value chains.

(iii) Develop and demonstrate to proof of concept stage an innovative cold storage system with sustainable energy technology that minimizes environmental impacts and easily adaptable to specific conditions and product requirements.

(iv) Raise awareness of policymakers on the potential impacts of the project and technologies to ensure speedy adoption and implementation.

3.2 Justification of the Research Conducted with Research Partner

The development of numerical model of the technology which integrates the advantages of the use of natural substances as refrigerants; solar energy and phase change material (PCM) will be carried out at the IEF Brunel University London due to unavailability of the softwares at BSP. Numerical study of sustainable technology in food storage system incorporate PCM and natural refrigerants (CO_2 and Hydrocarbon) will also be carried out in the UK. This is because the technology of the PCM storage, PV system and CO_2 test facilities are not available in BSP.

Detailed of the research that will be carried out at the IEF Brunel University London are:

- ***Year-1***: Evaluation of the current state of the cold chain in Indonesia and the development of a road map for its development to satisfy current and future needs of the country (under supervision of research partner).
- ***Year-2***: Development of numerical model of the investigated technology approach applying the technology of solar PV assisted food storage system incorporate PCM and natural refrigerants (CO_2 and Hydrocarbon). The model will require CFD and EES softwares which include as inkind supports from the research partner. The model validation and preparation for briefing document as well as publication to the

international community through international conference. The activities will be assisted by research partner who has the proficiency in cold chain of the food supply chain in the UK.

CHAPTER IV: RESEARCH METHOD

4.1 Research Method and Program Outline

The proposed research involves literature reviews, data gathering and analyses, identification of requirements and needs and technology road mapping and investigation and development of innovative cold storage technologies powered by solar energy. Surveys will be used to obtain data on the needs of the cold chain and development requirements – this will involve all stakeholders: manufacturers, traders, retailers, consumers and relevant government departments. Methodologies and models will be developed to map food production and consumption, seasonal variations to determine current capacity and future needs of cold storage capacity. Modelling, simulation and design studies will be performed to develop appropriate renewable powered cold storage technologies for the Indonesian markets.

The methods used to perform the investigation comprise stages which include: (1) collect data and information through survey and literature review about the current capacity of cold storages and food chain in Indonesia; (2) evaluating the data and designing approaches to estimate proper capacity of the cold storages in the future to develop a security and sustainable food chain; (3) identification of requirements, needs and technology road mapping; (4) investigation on innovative cold storage technologies powered by solar energy which includes numerical modeling and designing a new technology approach of cold storage through utilizing renewable energy, efficient and environmentally friendly refrigeration system, and bio-PCM for cold thermal storage; (5) validation of the numerical model; (6) analysis the influence of the sutainable technology to the stability and sustainability of the food chain; (7) preparation of a briefing documents (an international book with ISBN) to inform policymakers in Indonesia and the UK; (8) publications and reporting. The research will be conducted at Bali State Polytechnic and the Institute of Energy Futures Brunel University London within 2 years. The research program outline is described in Figure 3.1. The program outline in every year can be detailed as follows:

The 1st year research program: The first year programs comprises two main activities: (i) quantification and evaluation of refrigerated storages for post-harvest products and after fishing storages in the food chain and (ii) Development matemathical approach for designing approaches to estimate proper capacity of the cold storages in the future to develop a security and sustainable food chain including identification of requirements, needs and technology road mapping. The first activity will be carried out through surveys and a comprehensive literature review. The stakeholders involve manufacturers, retailers, consumers as well as governmental offices related to the food chain. Based on the data obtained, numerical approaches will be developed for evaluatiing the data and designing approaches to estimate proper capacity of the cold storages in the future to develop a security and sustainable food chain including identification of requirements, needs and technology road mapping. The road map will plot the currently refrigerated storage capacity and the future demand to minimise post-harvest losses and food wasted in the food production and to improve food quality supplied to the community. The numerical approaches **were developed in the research facilities of the Institute of Energy Futures Brunel University London (IEF)**. For publication and reporting, one international and one national journal papers as well as one international conference paper including annual implementation report of the project have also been prepared.

***Output of the 1st year program**:* Comprehensive data of refrigerated storage in Indonesia; Numerical model which will be used to predict appropriate capacity to fit current and future demand to satisfy the needs of the food supply chain in Indonesia; Road Map for cold chain development particularly cold storage requirements; one international journal paper, one international conference paper, one national agredited journal paper, one draft of briefing document and 1st year reports.

The 2nd year research program: Two main activities are set in the second year program. They are (i) Development of numerical model for designing and simulating the proof concept of the proposed technology for cold storage applied in the cold chain of food supply chain in Indonesia and validation of the model; (ii) analysis of the research findings and their influences to the stability, sustainability of the food chain. **The model of the sustainable technologies will be developed in IEF Brunel University London**. **The model will also be validated using data obtained from the experimental test at research facility available in the IEF**. The validated model can be used to estimate the performance of similar refrigerated storage at different operation conditions and capacities. For publications, one paper will be prepared for international journal. Others include one paper for international conference, one paper for national journal, one international book (briefing document) and one final implementation report.

Output of the 2nd year program: one international book with ISBN containing briefing documents and proof concept of an innovative cold storage system with sustainable energy technologies employing integrated thermal energy storage using bio-based phase change materials (bio-PCMs) and natural refrigerants such as CO_2 and hydrocarbons to minimize or eliminate grid energy demand and greenhouse gas (GHG) emissions. Other outputs include: one international journal paper, one international conference paper, one non-acredited national journal paper and final report of the project.

4.2 Testing and Data Processing

For model validation some experimental tests such as test on the CO_2 refrigeration as secondary fluid system, solar energy supply system, PCM and vacuum wall storage systems **will be conducted in the IEF UK due to such test facilities are not available at Bali State Polytechnic**. Testing for performance evaluation of a cold storage system will be conducted in the IEF UK. The test will refer to the ASHRAE Standard 72 (2005). The purpose of this standard is to prescribe a uniform method of testing commercial refrigerators and freezers for evaluations on energy consumption, product temperature performance, and other performance factors.

The complete performance testing will require proper data logging system which comprises data acquisition modules and a recording and display system. The data acquisition modules utilize the Datascan 7000 series which include a Datascan processor 7320 and expansion modules 7020. ***The* Datascan processor 7320 and 8 expansion modules 7020 including Corriolis flowmeter will be in-kindly supported by IEF Brunel University London.**

Collected data were processed in a spread sheet program. The properties of the refrigerants were derived from the EES software, while the properties of bio-PCM will be determined by using Differential Scanning Calorimetry (DSC) which will be conducted through other scheme of the research team.

4.3 Publications

Scientific papers will be prepared and published at least one paper every year. Possible international journals and conferences for the papers: *Alternative Journals*: International Journal of Refrigeration, Applied Thermal Engineering, and Applied Energy. *Possible conferences*: 2nd International Conference on Sustainable Energy and Resource Use in Food Chains 2018 (Wednesday 17 to Friday 19 October 2018 Coral Bay Hotel & Resort, Paphos, Cyprus), 25th International Congress of Refrigeration (2019, Montreal, Canada), 20th and 21st

International conference on applied energy in April 2018 and in April 2019 respectively, Brisbane Australia.

Alternatives for international coferences held in Indonesia: International Conference on Quality in Research (QiR) held by Indonesia University; International Conference on Mechanical Engineering (ICOME) – organised by ITS.

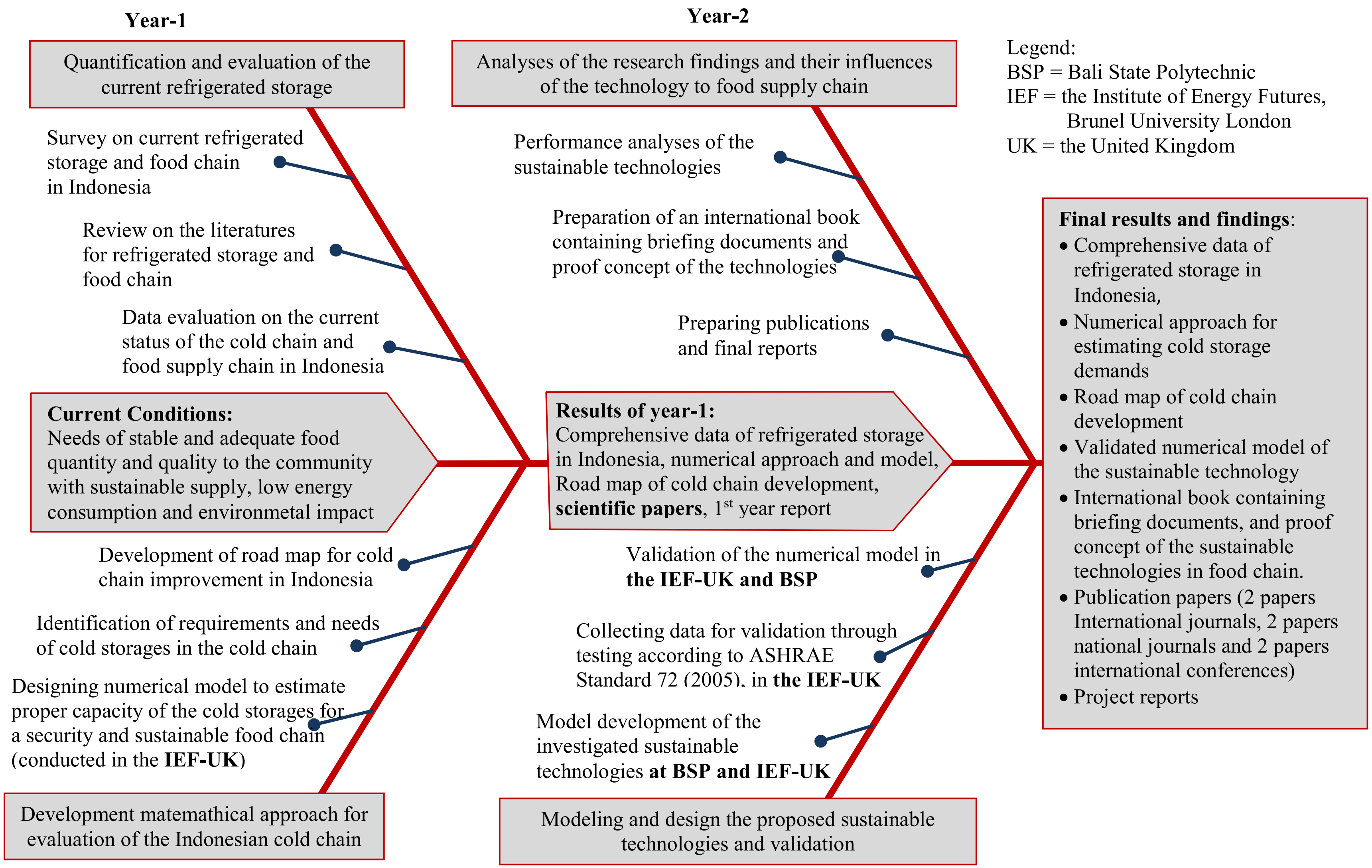


Figure 3.1: Flow chart of the proposed research in a fish-bone diagram

CHAPTER V: RESULTS

5.1 Development of Sustainable Evaporator

The Mathematical Model

Two main models were established to investigate the performance of CO_2 evaporator coils under different geometry, circuit arrangement and different operating conditions. The first model was for the investigation of the performance of MT CO_2 flooded evaporator coil. The second model was for the simulation of the performance of CO_2 DX evaporator coils for both chilled and frozen temperature levels. A third model was also developed to analyze the coil performance using refrigerant R-404A for comparative analyses. The models can also be used to design the geometry and tube arrangement of evaporator coils for a given refrigeration capacity. The numerical models apply standard plate fin specification to determine fin and tube pattern, height and width of the coil. The models were developed using the software EES.

To simulate the flooded and DX evaporator coils, some main assumptions were made as follows: steady state flow conditions; one dimensional flow for refrigerant inside tubes and air across the coil; negligible thermal losses to the environment; uniform temperature and air flow; constant air side convective heat transfer coefficient over the entire coil; intermediate pressure (P_{int}) to be considered as condensing pressure for CO_2 DX evaporator coil, negligible refrigerant pressure drops of less than 2 K saturated temperature equivalent for DX coils and less than 1 K for flooded coils; the same number of tubes in each circuit with the same fraction of total mass flow rate; quasi steady frosting process; maximum pressure drop at air side after frost to be lower than 0.175 kPa.

To simulate the geometry and circuitry of the evaporator coils, certain assumptions were made as follows: constant air side convective heat transfer coefficient over the entire coil; condensing pressure of 12.7 bar_a (corresponds to 29 ºC condensing temperature), evaporating temperature of -7 ºC; negligible refrigerant pressure drop of less than 2 K saturated temperature drop equivalent; the same number of tubes in each circuit with the same fraction of total mass flow rate; quasi steady frosting process; maximum pressure drop at air side after frost to be lower than 0.175 kPa.

Figure 5.1 shows the basic geometry of the finned tube evaporator considered in the models. The tubes are arranged in coordinates along width, depth and height axes (i, j, k) as can be seen in the figure. The number of rows and tube pattern can be used to determine the size of the coil and the tube interconnections within the coil circuits. If the coil has more than one circuit, the number of tubes in each circuit should be evenly balanced.

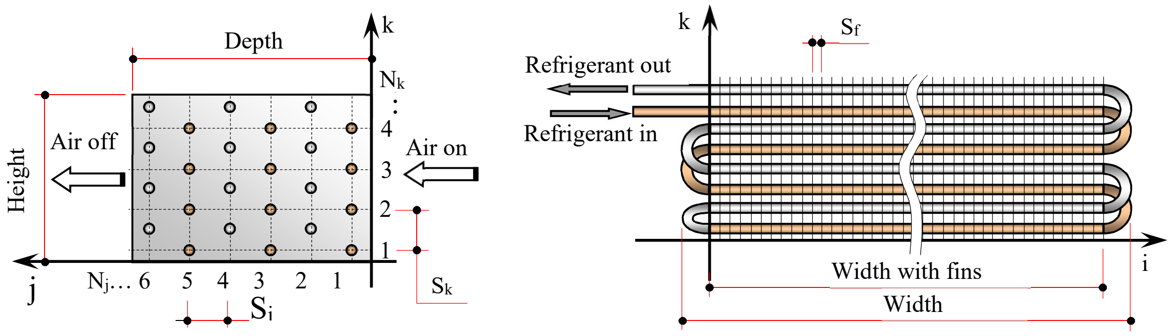
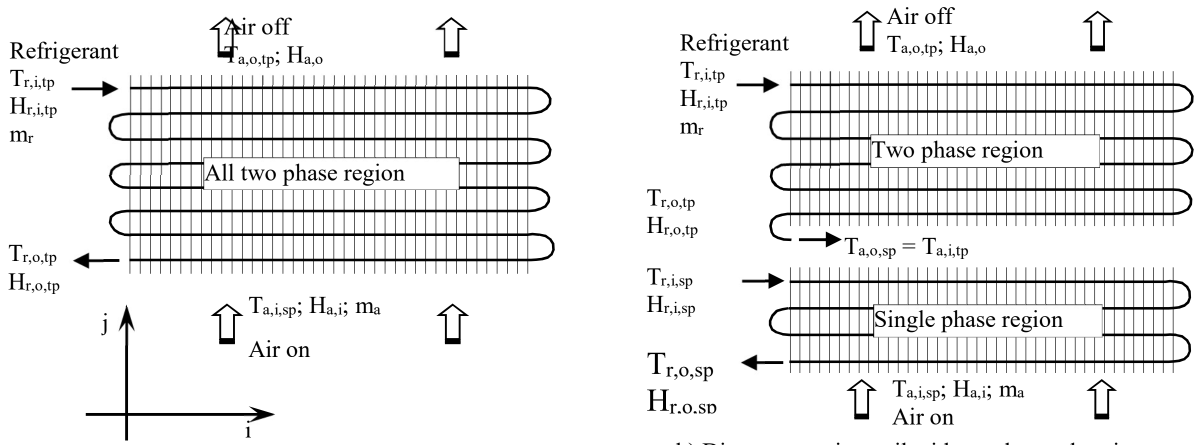


Figure 5.1 Geometry of a finned tube evaporator model

The mathematical models used the lumped element technique by which the evaporator coil can be divided into the superheated and two phase regions. A DX coil has two lumped regions (single and two phase regions), while a flooded evaporator coil only has a one region, the two phase region as shown in Figure 5.2. Each region is considered as a single control volume. The fraction of the coil area in each control volume in a DX coil is calculated in proportion to the amount of heat transfer in each control volume.



a) Flooded coil with single lumped region

b) Direct expansion coil with two lumped regions

Figure 5.2 Schematic of flooded and direct expansion evaporator coils with single and two control volumes

The mass and energy balance principles are applied to each control volume, which is summarized in equations (1) to (3). For the flooded evaporator coil, a single phase region does not exist, thus the heat transfer rate component for single phase region ($Q_{r,sp}$) is omitted. ΔT_{lm} is the logarithmic mean temperature difference of each control volume. The air side surface area of the coil (A_a) and other geometric parameters of the coil such as free flow area and free flow area with frost were calculated.

$$Q_e = Q_{r,tp} + Q_{r,sp} = Q_a \tag{1}$$

$$Q_e = m_r\,(h_{r,o,tp} - h_{r,i,tp}) + m_r\,(h_{r,o,sp} - h_{r,i,sp}) = m_a\,(h_{a,i} - h_{a,o}) \tag{2}$$

$$Q_e = U_{a,tp} A_{a,tp} \Delta T_{lm,tp} + U_{a,sp} A_{a,sp} \Delta T_{lm,sp} \tag{3}$$

The overall heat transfer coefficient (U_a) of the coil with frost can be calculated from equation (4). For frost free coil the component of frost resistance is not included. For the DX coil the external and internal heat transfer areas as well as the internal heat transfer coefficient depend on the mode of heat transfer, single or two phases.

$$\frac{1}{U_a} = \frac{1}{\eta_f h_a} + \frac{\delta_{frost}}{\eta_f \lambda_{frost}} + \frac{A_a \ln\left(\dfrac{d_o}{d_i}\right)}{2\pi\lambda_t L_r} + \frac{A_a}{A_r h_r} \tag{4}$$

Correlations of heat transfer coefficient and pressure drop at refrigerant side

The local heat transfer coefficients and pressure drop correlations were selected for each flow regime as it changes with the flow and evaporation of refrigerant in the evaporator. Thus the correlations reliably capture the variation of two phase heat transfer coefficient and frictional pressure drops at different mass velocities and vapor qualities.

Equations (5) and (6) show the general equations for the two phase heat transfer coefficient and pressure drop on the refrigerant side of the evaporator.

$$h_{tp} = \frac{\theta_{dry} h_v + (2\pi - \theta_{dry})\left[(Sh_{nb})^3 + h_{cb}^3\right]^{(1/3)}}{2\pi} \tag{5}$$

$$\Delta P_{total} = \Delta P_{static} + \Delta P_{momentum} + \Delta P_{frictional} \tag{6}$$

Correlations of heat transfer coefficient and pressure drop on the air side

The air side heat transfer coefficient can be calculated from:

$$h_a = h_{c,a} + h_{lat,a} \tag{7}$$

$$h_{c,a} = \frac{j\,Cp_a\,G_a}{\Pr^{2/3}} \tag{8}$$

Calculation of frost accumulation

The rate of frost accumulation was determined from equation (9) and the amount of frost accumulated on the surface of the evaporator and frost thickness after Δt time, from equation (10).

$$m_{frost} = m_a(\omega_i - \omega_o) \tag{9}$$

$$\Delta m_{frost} = m_{frost}\,\Delta t \quad \text{and} \qquad \delta_{frost} = \frac{\Delta m_{frost}}{\rho_{frost}\,A_a} \tag{10}$$

Test results from the experimental CO_2 test facility were used to validate the models. The model of conventional evaporator coil with R-404A was validated against data provided by the manufacturer. Comparison between predictions and experiments under design conditions was found to be satisfactory for the refrigeration capacity as shown in Table I. The pressure drop estimations were, however, lower than the experiment results mainly because the pressure drops across the distributor and lead tubes were not included in the model. For synthetic refrigerants these pressure drops can be as high as 89% of total pressure drops in the evaporator coil.

The validated models were used to design 8 evaporator coils with different geometry and circuitry. The evaporator coils were simulated at evaporating temperature of -8ºC for MT coils and -30ºC for LT coils. Tubes and fins were made from copper and aluminum respectively. Equilateral fin and tube pattern in a staggered arrangement was applied. The temperature of chilled food display cabinet was in the range of -1 to 1ºC and frozen food display cabinet was in the range of -19 to -21ºC.

Table 5.1 Model And Experiment Results

Parameters		[a] MT CO_2 Models Flooded	DX	[b] DX LT CO_2 model	[c] DX R-404A model
Q_e (kW) full load, ΔT_a = 10 K	Model/	5.19	5.09	3.00	-
	Expe-riment	5.10	4.93	2.89	-
Q_e (kW) steady state load, ΔT_a = 9 K for MT and ΔT_a = 8 K for LT	Model	4.55	4.46	2.35	3.65
	Expe-riment	4.42	4.30	2.12	3.60*
ΔP_r (kPa) steady state	Model	11.31	9.22	6.01	40.92
	Expe-riment	21.15	16.87	14.87	148.28*

Evaporator coil investigated:
[a] Tube arrangement: staggered; d_o = 12.70 (mm); N_k = 4; N_j = 6; number of circuits = 4; fins pitch 4 fins/inch
[b] Tube arrangement: staggered; d_o = 12.70 (mm); N_k = 4; N_j = 8; number of circuits = 3; fins pitch 3 fins/inch
[c] Tube arrangement: inline; d_o = 15.87 (mm); N_k = 2; N_j = 16; number of circuits = 2; fins pitch 3 fins/inch
* Data from manufacturer

Table 5.1 shows the geometry of the evaporator coils together with their performance parameters. It can be seen the physical sizes of the CO_2 evaporator coils are varied and are generally much smaller compared to the coils with R-404A.

Table 5.2 Geometry of the designed coils with their performance parameters

Parameters	MT evaporator coils					LT evaporator coils		
	DX CO_2		Flooded CO_2		DX R-404A	DX CO_2		DX R-404A
	EC-1	EC-2	EC-3	EC-4	EC-5	EC-6	EC-7	EC-8
Tube outside diameter (mm)	9.52	12.70	9.52	12.70	15.87	9.52	12.70	15.87
Number of rows high	2	2	2	2	2	2	2	2
Number of rows deep	21	17	13	10	20	12	10	12
Number of circuits	2	1	2	1	2	2	1	2
Total tube length (m)	91.1	73.8	56.4	43.4	86.6	48.7	40.6	48.7
Height (mm)	63.5	63.5	63.5	63.5	76.2	63.5	63.5	76.2
Depth (mm)	577.4	467.4	357.5	275.0	659.9	330.0	275.0	395.9
Width with fins (mm)	2170	2170	2170	2170	2170	2030	2030	2030
Refrigerant volume (L)	4.14	6.78	2.56	3.99	13.47	2.22	3.74	7.57
G_r (kg s^{-1} m^{-2})	171.0	168.7	199.8	198.1	109.0	100.4	99.0	72.2
CR	-	-	1.2	1.2	-	-	-	-
Q_e (kW)	3.75	3.75	3.75	3.76	3.76	2.25	2.25	2.25
$Q_{e,frost}$ (kW)*	3.26	3.12	3.24	2.96	3.44	2.16	2.14	2.15
Fin efficiency	0.85	0.88	0.85	0.88	0.87	0.89	0.92	0.90
h_r (kW m^{-2} $^oC^{-1}$)	2.899	3.107	3.206	3.473	0.482	2.521	2.802	0.337
h_a (kW m^{-2} $^oC^{-1}$)	0.062	0.073	0.064	0.075	0.063	0.042	0.050	0.046
ΔP_r (kPa)	65.13	30.89	42.34	19.27	50.47	33.15	16.24	31.79
$\Delta P_{a,frost}$ (kPa)	0.016	0.024	0.030	0.043	0.013	0.018	0.020	0.010

For the given refrigeration duty, the flooded MT coil with tube diameter 9.52 mm (EC-3) has the smallest size with refrigerant volume about 62% of the MT DX coil using the same tube diameter (EC-1) and about 19% of that in the R-404A evaporator coil (EC-5). The CO_2 coils also need less refrigerant charge as shown in Figure 5.3, assuming 25% and 35% of the evaporator volume was filled with liquid for the DX and flooded evaporator coils respectively.

The CO_2 evaporator coils with smaller tube diameter require more tube rows and longer tubes to meet the designed refrigeration duty. Using single circuit arrangement results in high pressure drop particularly for the DX type coils. As can be seen in Table 5.2 the pressure drops of the CO_2 coils (EC-1, EC-3 and EC-6) are still higher than the coils with larger tube diameter (EC-2, EC-4 and EC-7) even in two circuit arrangement. The pressure drop will be higher if the pressure drop across the distributor and lead tubes is taken into account. Moreover, the physical size of the coils, except for in the case of the flooded coil EC-3, is larger which may increase their production cost.

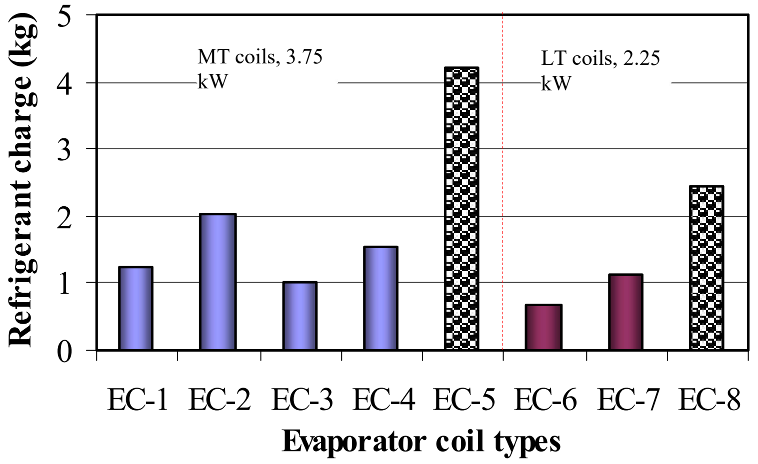


Figure 5.3 Refrigerant charge comparisons

Figure 5.4 shows the performance variation of CO_2 MT DX coils with evaporating temperature. Increasing the evaporating temperature can slightly improve the refrigeration capacity and reduce the pressure drop. Similar effect was also found on the LT DX and flooded evaporator coils.

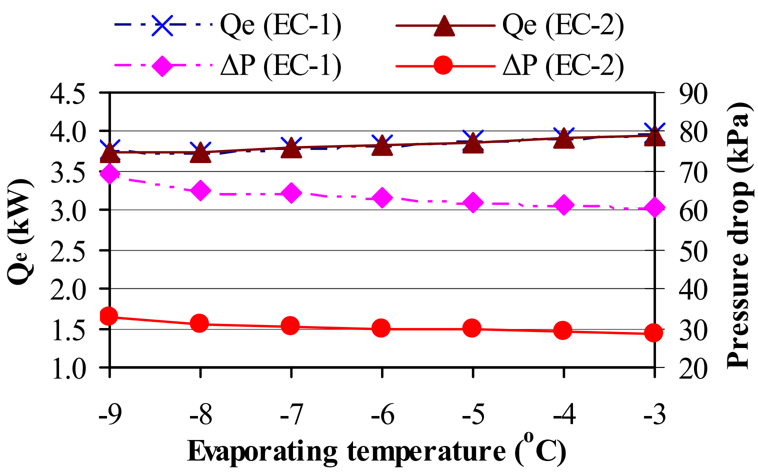


Figure 5.4 The influence of evaporating temperature

In Figure 5.5 and 5.6, the performance of the flooded CO_2 evaporator coils at different circulation ratios (CR) is shown. As the CR increases, the refrigeration duty slightly improves due to the enhancement of the evaporation heat transfer coefficient. However, the increase of the CR considerably increases the pressure drop and refrigerant mass velocity which increases the power consumption of the CO_2 pump and causes a reduction in the coefficient of performance of the refrigeration system. The CR, therefore, should be chosen to be as low as possible in the range of the designed refrigeration capacity. The experimental tests showed the optimum CR to be in the range 1.1 and 1.3.

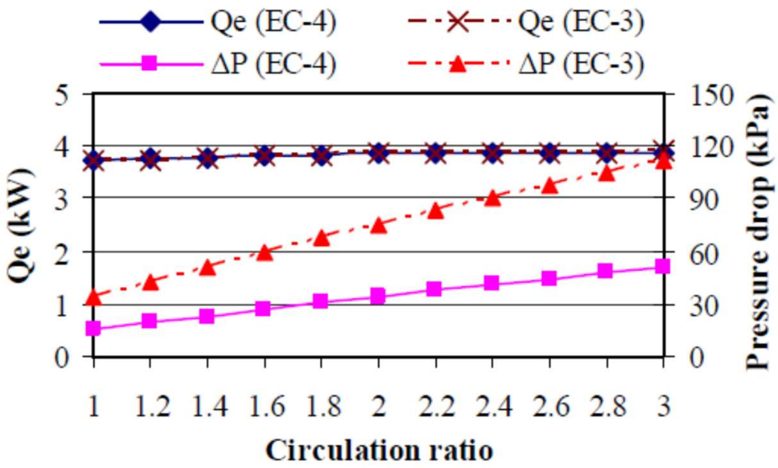


Figure 5.5. Refrigeration duty and pressure drops with circulation ratio (CR)

Numerical models have been developed and validated for design and performance simulation of finned tube flooded and direct expansion coils. Different geometry and circuitry arrangements were simulated using CO_2 and R-404A as refrigerants. The investigation found that for a given refrigeration capacity, CO_2 evaporator coils had considerably smaller size and lower refrigerant charge compared to the coils using R-404A refrigerant.

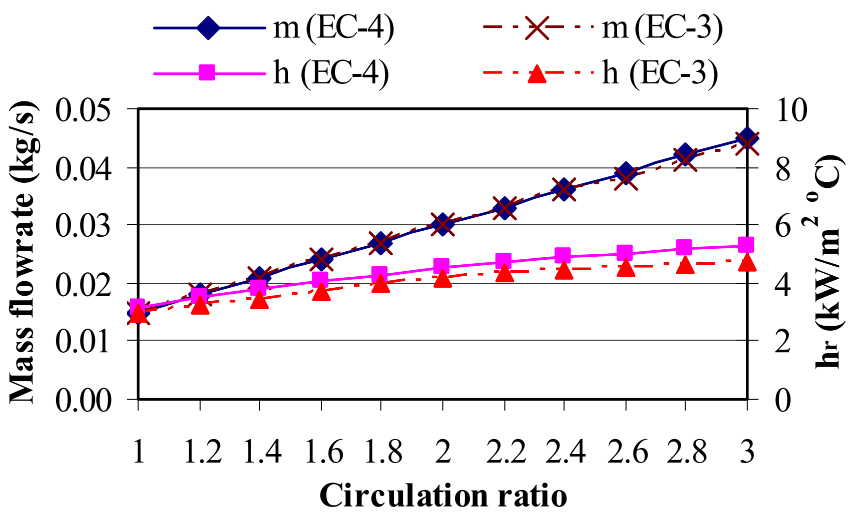


Figure 5.6 Refrigerant mass flow rate and heat transfer coefficient with circulation ratio

The investigation among the CO_2 evaporator coils showed that for the refrigeration capacity examined, using the larger tube diameter, the CO_2 DX evaporator coils were found to be more compact due to smaller number of rows along the depth of the oil. The pressure drop of the coils was also found to be lower. However the coil had more refrigerant charge compared to the coil with the smaller tube diameter. For the CO_2 flooded evaporator application, the use of smaller tube diameter was found to be more favorable in terms of coil size and refrigerant charge.

Table 5.3 Geometry of the designed coil with its performance parameters

Parameters	R1270 coil	R404A coil
Tube outside diameter (mm)	12	12
Number of rows high	2	2
Number of rows deep	10	14
Number of circuits	2	2
Total tube length (m)	70	98
Height (mm)	70	70
Depth (mm)	350	490
Width with fins (mm)	3500	3500
Refrigerant volume (L)	7.1	7.8
Refrigerant charge (kg)+	0.640	2.438
G_r (kg s^{-1} m^{-2})	67.66	206.1
Q_e (kW)	4.2	4.2
$Q_{e,frost}$ (kW)*	3.4	3.6
Fin efficiency**	0.83	0.84
h_r (kW m^{-2} ºC^{-1})	0.738	1.318
h_a (kW m^{-2} ºC^{-1})	0.052	0.048
ΔP_r (kPa)	8.53	97.38
$\Delta P_{a,frost}$ (kPa)*	0.016	0.010

* After frost accumulation of 240 minutes
**Fin thickness: 0.2 (mm); Fin pitch: 8 (mm) for R1270 and 10 (mm) for R404A
+ Assumption 25% of the coil volume filled with liquid refrigerant.

Table 5.3 details the geometry of the R1270 evaporator and original R404A coil together with performance parameters obtained from the EES model. Because the R1270 refrigeration system is comprised of two circuits, the coil geometry given in the table is for only one circuit. Similar assumptions were made for the R404A coil. A 7 ºC superheat was also assumed for both coils. The results show that for the same load the R1270 coil would require 70 m of copper pipe and 0.64 kg of refrigerant charge compared to 98 m pipe for the R404A coil and 2.44 kg of refrigerant charge.

5.2 Design and Simulation of Sustainable Technology Food Storage

The technology used is a cold storage system for food supply chain. The system will combine the utilization of sustainable and recent refrigeration technologies with natural refrigerants (hydrocarbon and CO_2) incorporating bio-PCM as thermal energy storage. PV system generates electricity from solar energy. It is also used as the main drive of the refrigeration system. There is no battery storages used in this system. The refrigeration system comprises a main refrigeration system with hydrocarbon as its primary refrigerant as shown in Figure 5.7.

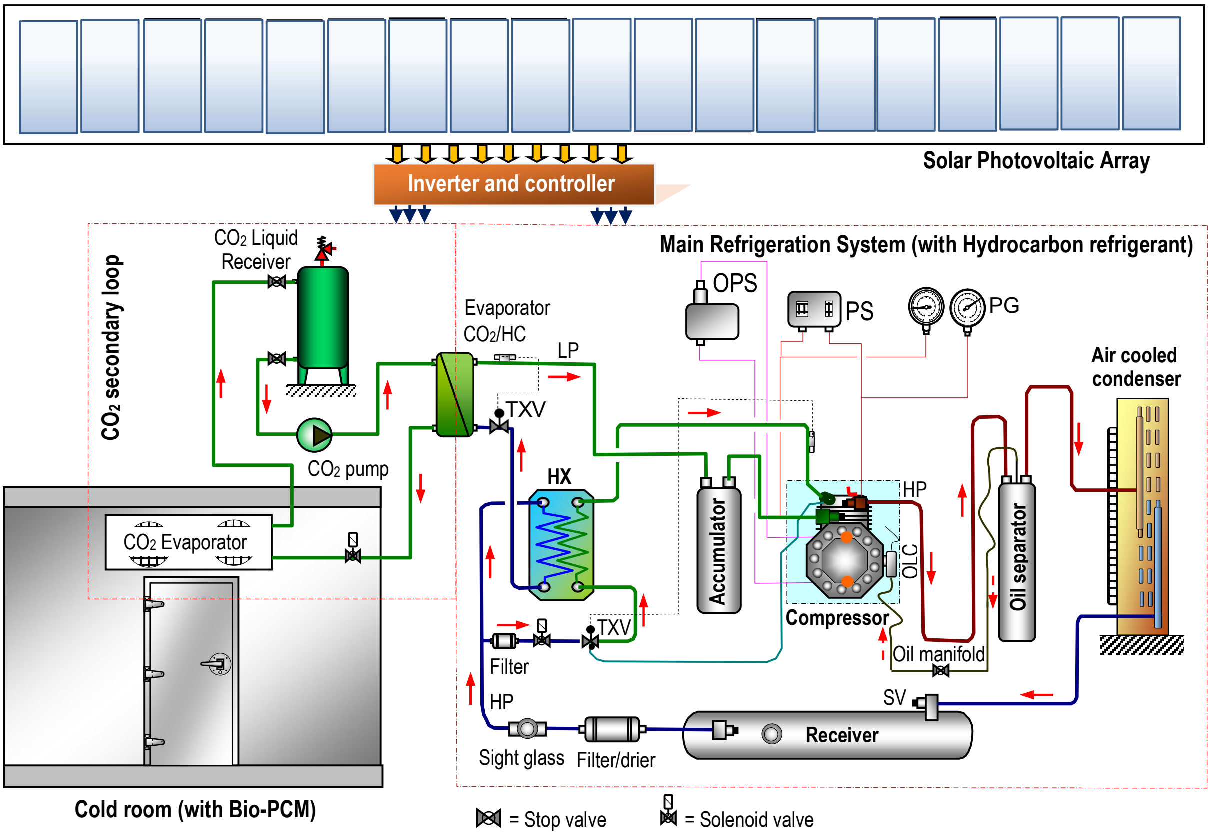


Figure 5.7 Technology implemented: a solar energy driven cold storage incorporated environmentally friendly refrigerants and bio-PCM

The main refrigeration system cools a secondary refrigerant (CO_2) which is circulated to the cold room using a CO_2 pump. The liquid CO_2 absorbs heat in a flooded evaporator, then returns back to the main refrigeration system. This arrangement will utilize advantages of energy efficient refrigeration system using hydrocarbon refrigerant and highly efficient CO_2 secondary fluid system. PCM arrangement in the cold room will be designed to enable the storage room remains cold when the electricity is not available especially during night time or off cycle.

During off cycle, CO_2 refrigerant in the secondary loop will be kept cold by utilizing cooling from the cold room provided by PCM through the flooded evaporator. The secondary loop system will be properly insulated to minimize heat gain. The technology will be highly flexible for Indonesia application as an archipelago country. It can be an excellent solution for remote areas on distributed islands with shortage electricity supply. However for the island which are well electrified, the technology can be run as hybrid cold storage systems.

CHAPTER VI: NEXT RESEARCH PLAN

Research activities of the first year will be completed until the end of the year 2018. Based on the data obtained, numerical approaches will be used to evaluate data and to finish the design of estimating proper capacity of the cold storages in the future in order to develop a security and sustainable food chain including identification of requirements, needs and technology road mapping. The road map will plot the currently refrigerated storage capacity and the future demand to minimise post-harvest losses and food wasted in the food production and to improve food quality supplied to the community. The numerical approaches **will be developed in the research facilities of the Institute of Energy Futures Brunel University London (IEF)**. For publication and reporting, one international and one national journal papers as well as one international conference paper including annual implementation report of the project will also be prepared.

The 2nd year research program: Two main activities are set in the second year program. They are (i) Development of numerical model for designing and simulating the proof concept of the proposed technology for cold storage applied in the cold chain of food supply chain in Indonesia and validation of the model; (ii) analysis of the research findings and their influences to the stability, sustainability of the food chain. **The model of the sustainable technologies will be developed in IEF Brunel University London**. **The model will also be validated using data obtained from the experimental test at research facility available in the IEF**. The validated model can be used to estimate the performance of similar refrigerated storage at different operation conditions and capacities. For publications, one paper will be prepared for international journal. Others include one paper for international conference, one paper for national journal, one international book (briefing document) and one final implementation report.

Output of the 2nd year program: one international book with ISBN containing briefing documents and proof concept of an innovative cold storage system with sustainable energy technologies employing integrated thermal energy storage using bio-based phase change materials (bio-PCMs) and natural refrigerants such as CO_2 and hydrocarbons to minimize or eliminate grid energy demand and greenhouse gas (GHG) emissions. Other outputs include: one international journal paper, one international conference paper, one non-acredited national journal paper and final report of the project.

CHAPTER VII: CONCLUSIONS

7.1 Conclussions

August 2018, the research activities planned for year 1 can be completed around 100% of the target research activities Year 1. The results of the research activities in year 1 are outlined in Table 7.1

Table 7.1: Research achievement Year-2018

No	Target output Year 1	Achievement
1	Comprehensive data of refrigerated storage in Indonesia	Data collection has been done some data is still in progress (target: comphrehensive data can be obtained from various sectors).
2	Numerical model which will be used to predict appropriate capacity to fit current and future demand to satisfy the needs of the food supply chain in Indonesia	Model has been developed and finalized, validatedin the UK.
3	Road Map for cold chain development particularly cold storage requirements.	Roadmap of Indonesia coldchain is being prepared will be continued in the next year program
3	International journal paper, one international conference paper, one draft of briefing document	Three papers have been prepared: 1. *The 3rd International Join Conference on Science and Technology (IJCST) October 2018* (submitted) 2. *The 1st International Conference on Mechanical Engineering Research and Application, October 2018 (submitted)* 3. The 2nd International conference on sustainable Energy and Resource Use in Food Chains October 2018.
		Article for international journal and briefing document is being prepared and continued to the second year program
5	Progress and annual reports year 1	Progress and annual reports year 1 have been completed

7.2 Suggestion

It is necessary to affirm the financial administration system in the implementation of research so that it can alleviate research tasks that are supportive but more focused on achieving research output.

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APPENDICES

1. Availability of research facilities
2. Structure of the research management
3. Manuscript for International Conference on Mechanical Engineering Research Application (ICOMERA) Unibraw Malang
4. Manuscript for ICSEF (international conference on Sustainable Energy and Resource in Food Chain)
5. Manuscript for IJCST Nusa Dua Bali

Appendix-1: AVAILABILITY OF RESEARCH FACILITIES

The research activities will be done in Mechanical Engineering Department of Bali State Polytechnic and the Institute of Energy Futures, Brunel University London. Most of the lab works will be conducted in the IEF-UK. The IEF has the facilities required for this research including: modeling laboratory, test facilities (Solar Energy Generation System, CO_2 Test system, and Cold storage test system). Their test facilities are completed with calibrated instrumentation system. However, Bali State Polytechnic has test facility just for conventional refrigeration systems. The laboratories and workshop with their facilities that are available at Bali State Polytech can be described as follows:

1. Laboratory of Applied Refrigeration (Commercial Refrigeration Section)
   Activities involved: investigation of conventional cold storage system and the new approach of food storage system.

Conventional cold room system

Multirack refrigeration system

Condensing unit cascade system

Ice mechine - refrigeration system

2. Laboratory of Instrumentation and Control
   Activities involved: Installation of control and instrumentation systems of the cold storage with integrated PCM incorporating renewable energy source.

Automation control system

Instrumentation panel construction room

3. Mechanical workshop
   Activities involved: Fabrication of PCM packages and racking equipment for loading system, solar PV support and other mechanical components production


Mechanical production machines


Mechanical production machines

4. Availability of the space for installation of the proposed refrigerated food stotage system


Available room in the Refrigeration Laboratory

5. Our labs at Bali State Polytechnic has also been completed with proper instrumentation which procured through previous research grants.


Three phase power analyser
4-wire

Ultrasonic Flowmeter

Data logging system
(DataScan Logger)

Appendix2: STRUCTURE OF RESEARCH MANAGEMENT

BSP project Team: Dr Suamir has over 10 years' research experience in energy conservation and sustainability, He supervised many projects on refrigeration and energy conservation. Dr Santosa has expertise in modelling of refrigeration systems and has supervised a number of projects in the field.

Principal Researcher	:	I Nyoman Suamir, ST, MSc, PhD
NIDN	:	0025036514
Field of Science	:	Refrigeration and Sustainable Energy
Alocated time	:	12 (hours/week)

Job descriptions:

- Manage the research activities
- Quantify and evaluate the energy consumption at specific stages of cold chain in Indonesia
- Review the current state of refrigerated storage, estimate the appropriate capacity and estimate the future demand of such facility.
- Investigate the use of low GWP and natural refrigerants for cold chain applications.
- Modelling and design a new approach of sustainable energy system incorporating renewable energy for food storage applications (EES model)
- Validate the established energy models
- Publication through international journals and conferences
- Reporting

Member of researcher (1)	:	I Dewa Made Cipta Santosa, ST, MSc, PhD
NIDN	:	0021127202
Field of Science	:	Refrigeration and Air Condiitioning
Alocated time	:	8 (hours/week)

Job descriptions:

- Survey the current capacity of refrigerated storages in Indonesia for food chain.
- Review annual domestic production capacity of seasonal food and value for storage
- Review the energy system apply in Indonesian food chain.
- Investigate environmental impact and economic viability of the proposed energy system
- Data processing and analysis
- Reporting and publications

Member of researcher (2)	:	Dr. I.G.A.B. Wirajati, ST, MEng
NIDN	:	0015047107
Field of Science	:	Refrigeration
Alocated time	:	8 (hours/week)

Job descriptions:

- Quantify and evaluate the energy consumption at specific stages of cold chain in Indonesia
- Review the current state of refrigerated storage, estimate the appropriate capacity and estimate the future demand of such facility.

- Build a numerical model for analysing of the prediction of the future demand of refrigerated storages in Indonesia and for evaluating their energy consumption (EES model).
- Data processing, analysis and reporting
- Publication through international journals and conferences

The UK Team comprises of: Prof. Tassou – considerable research management experience and expertise in food chains, food refrigeration systems and system integration and optimization. Dr Singh- Substantial expertise on solar energy and solar PV systems as well as vacuum insulation. Dr Ge- Expertise in modelling of energy and refrigeration systems.

International partner	:	Prof. Savvas A. Tassou
Field of Science	:	Food refrigeration, energy system, building energy, Heat transfer and heat exchangers, Wastewater filtration

Job descriptions:
- Supervise research work
- Manage the Newton Fund research Scheme
- Arrange researcher visit to UK
- Assist the modelling, design and construction of the proposed system
- Establish further collaboration to companies in the UK to support the project
- Review CO_2 test facility for food chain application.
- Publication through international journals and conferences

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Appendix-3: INKIND SUPPORT FROM RESEARCH PARTNER

In kind surprising support has been provided by IEF Brunel University for research capacity building at Bali State Polytechnic

Two sets of DataScan Logger worth more than 240 million IDR

Thermocouples pressure transducers worth more than 30 million IDR

Appendix-4: MANUSCRIPT FOR INTERNATIONAL CONFERENCE ON MECHANICAL ENGINEERING RESEARCH APPLICATION (ICOMERA) UNIBRAW MALANG

(Presented in the next page)

Experimental Study on the Influences of Air Flow in an Integral Hydrocarbon Display Cabinet to its Temperature and Energy Performances

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Abstract. This paper presents results on temperature energy performance tests of an integral refrigerated hydrocarbon (HC) display cabinet for retail food applications. Heat from the condensing unit of the cabinet is rejected to both ambient air and water/glycol mixture flowing in a closed water circuit. Air flow in the cabinet loaded with M-packages as test products was studied in order to analyze effects of air flow rate in the cabinet to its temperature and energy performances. The product and air temperatures as well as energy consumption of the cabinet were measured. The tests were conducted in a test chamber at climate class 3. It was found that the HC display cabinet with integral condensing unit was found to provide excellent energy performance with an Energy Efficiency Index below the requirement to qualify for enhanced capital allowances. The refrigeration system of the cabinet could also achieve a COP of 3.15. The study also found that higher air flow rate in the cabinet could make the product temperatures a little bit better, but the energy consumption increased approximately 7% when air flow rate was increased from 1200 $m^3\ h^{-1}$ to 1800 $m^3\ h^{-1}$. Air flow distribution in the cabinet was necessary to be optimized in order to comply with M0 classification cabinet.

Introduction

Vertical refrigerated display cabinets are commonly used in retailed food stores to ensure safety of the food products. These cabinets keep and display food for the customers at different levels of temperature within the retail stores. Various types of vertical refrigerated display cabinets can be found in retail stores including the stand-alone or centralized systems. Stand-alone units are self-contained refrigeration systems. For the centralized applications, the display cabinet evaporators in sales area are feed by the centralized refrigeration system which located in the machinery room. With regards to their opening, there are two types of vertical refrigerated display cabinets: open-type and door type. The open-type refrigerated display cabinets are widely used in retail stores to attract the costumers and increase the sales. The absence of any physical obstacle like a glass door between the customer and product display area is preferred for commercial reasons. The main advantage of the open type refrigerated display cabinets over the door type ones is to allow consumers free access to food [1].

Retail refrigeration systems using HFC refrigerants are responsible for substantial greenhouse gas emissions from leakage of refrigerant to the ambient and indirect emissions from the electrical power used by the compressors, fans and other ancillary equipment [2]. One way to significantly reduce or completely eliminate direct emission is through the use of natural refrigerants, such as hydrocarbons,

CO_2 and ammonia [3]. Considerable research has been carried out on the development and application of retail refrigeration systems employing natural refrigerants. Most systems are either trans-critical booster CO_2 systems, cascade all CO_2 systems, or subcritical CO_2 systems cascaded with a hydrocarbon system on the high pressure side for heat rejection [4-6]. An interesting approach also developed and applied by some retail chains involves the use of integral or stand-alone hydrocarbon display cabinets with heat rejection to the air in the retail [7] and combination of heat rejection to the air and water in a closed loop system [8]. The heat in the water circuit can be either upgraded through a boiler or heat pump and used for domestic hot water and/or space heating, or rejected to the ambient through a dry cooler. This approach can provide energy integration between the refrigeration and space conditioning systems in the store and offers the potential for energy savings if the system is appropriately designed and controlled.

Another way to reduce greenhouse gas emissions from retail refrigeration systems is by improving their energy efficiency. Sun *et al*. [9] reported the use of guiding strips at front face of the shelves of the open-type vertical refrigerated display cabinets could improve temperature performance and energy efficiency. Cooling capacity required to maintain the food product chilled decreased by 34%. Investigations on energy efficiency of retail cabinets have also been reported in [10,11,12]. They reported the amount of warm air entering open-type display cabinet was due the turbulence intensity, shape of the mean velocity profile at the discharge air grill and the Reynolds number. Furthermore, experimental results on energy performance of refrigerated display cabinets have been presented by [13,14]. They compared different patterns on the rear duct panel to improve the air distribution as well as to reduce heat extraction rate of the display cabinet. The authors reported optimum air distribution that could reduce heat extraction rate was achieved when 67% of the total air circulation to be delivered from the rear panel and the remaining 33% from the air curtain of the display cabinet.

This paper presents results of experimental investigations carried out for the development of a 3.81 meter low front open multi-deck chilled food display cabinet with 37% back flow ratio. The results indicated that regulating air flow rate could not make an open-type display cabinet comply with the M0 classification of product temperature range from -1°C to +4°C as described in BS EN ISO 23953-2 [15]. Further optimization on air flow distribution of the display cabinet would be required as discussed in [14].

Cabinet Description, Test Conditions and Methods

Cabinet Description

The cabinet tested was a vertical multi-deck display cabinet with compression type and built-in condensing units. The cabinet is a 3.81 m long low front cabinet originally designed for R-404A refrigerant. The height of the cabinet is 2.22 m and the depth 1.14 m. The cabinet has a Total Display Area (TDA) of 6.9 m^2. The evaporator coil and circulation fans are located in the rear flow tunnel. In the conversion to an integral Hydrocarbon (HC) system, the condensing units are located at the top of the cabinet. The cabinet uses hydrocarbon (HC) refrigerant, R1270. The refrigerant is flammable and carries an A3 safety classification [16,17] which limits the concentration of the refrigerant in an occupied space, in the event of leakage, to below its lower flammability limit. To minimize the refrigerant charge in each refrigeration circuit, the cabinet was designed with two completely independent condensing units and evaporator coils. Each circuit was charged with 0.72 kg R1270. This charge is within the safety limit specified by [16].

The cabinet consists of 5 shelves and 1 base (bottom) deck for loading the test packages. Ticket-bars are attached in the front edge of the 5 shelves. The selves are also completed with acrylic-risers which seat on the gap between the front edge and the ticket-bars. Heat rejection from the condensing unit takes place through two heat exchangers, an air cooled heat exchanger rejecting heat to the ambient air for direct retail food store space heating, and water cooled heat exchangers rejecting heat to water/glycol for centralized thermal energy management. A horizontal scroll compressor is used to keep the height of the condensing unit to the minimum. To facilitate testing and development in the laboratory, a water/glycol heat rejection system was developed which enables control of flow rate and temperature to the condensing units. A schematic diagram of the heat rejection system together with the refrigeration system of the cabinet is shown in Figure 1.

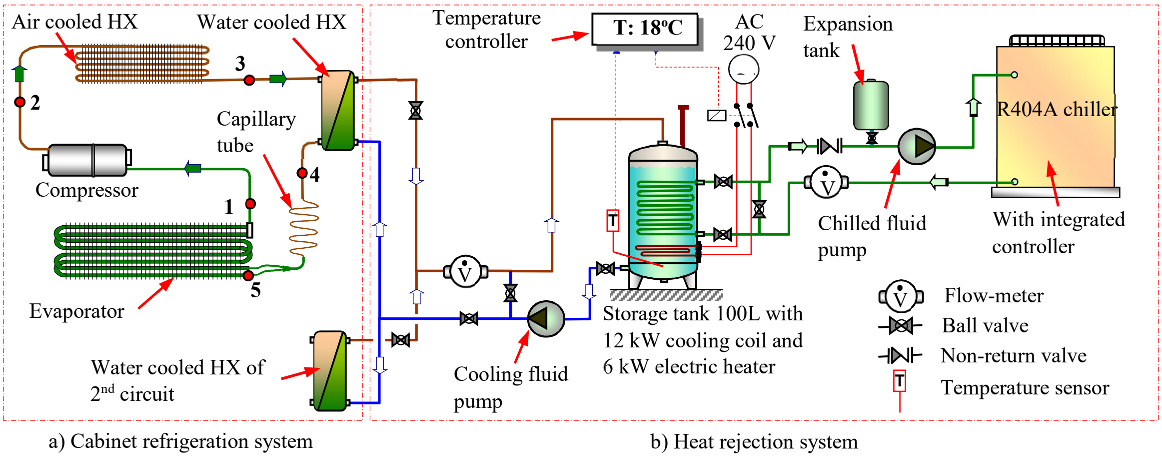


Figure 1. Schematic diagram of heat rejection system

Test Conditions

The tests were carried out in a test room conforming to [15]. The test room walls and ceiling are thermally insulated and are equipped with inner metal skin. The useful dimensions of the test room can accommodate overall dimensions of the tested cabinet. The lighting in the test room was provided by fluorescent lights with lighting level in the range between 500 and 650 lux which complies with the standard of 600±100 lux at a height of 1 meter above floor level. Ambient conditions in the test room were tightly controlled by a proportional-differential controller which modulates humidifier, heating system and the opening of three ways valve of the water chiller system. The ambient conditions were monitored by temperature and RH sensors linked to the measurement system. Mean horizontal air velocity (cross flow) was in the range between 0.1 and 0.2 m s^{-1}. The room conditions were set to Climate Class 3 with dry bulb temperature of 25°C and 60% RH as shown in Figure 2.

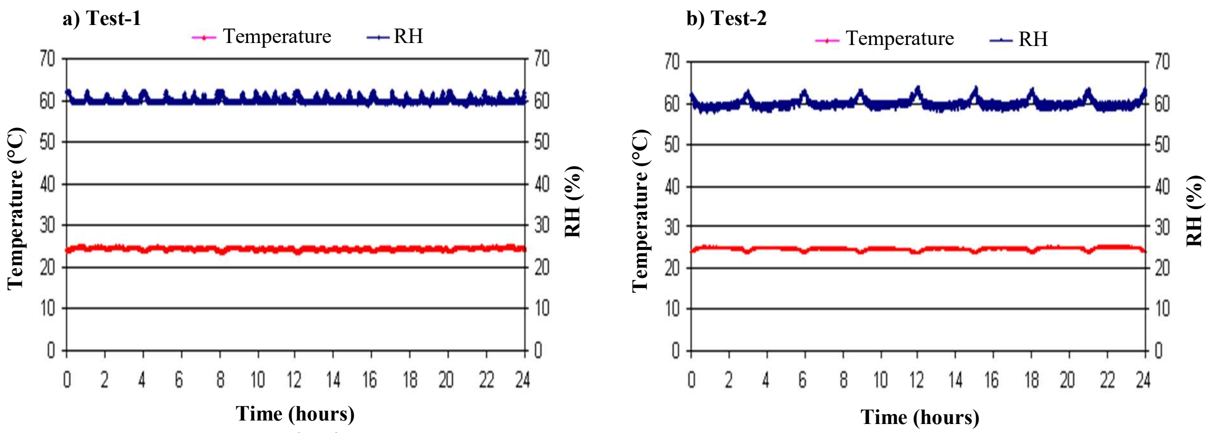


Figure 2. Room conditions (Temperature and RH) during the test

Methods

The tests were performed according to [15] which comprised two tests: (i) Test-1 Cabinet with lower air flow rate (1200 $m^3 h^{-1}$); (ii) Test-2 Cabinet with higher air flow rate (1800 $m^3 h^{-1}$). Air flow in the cabinet is shown in Figure 3. The tested display cabinet had a original back panel from manufacturer with perforated ratio 1.25% and could provide back flow ratio of 37%. The remaining 63% of air flow was for the air curtain.

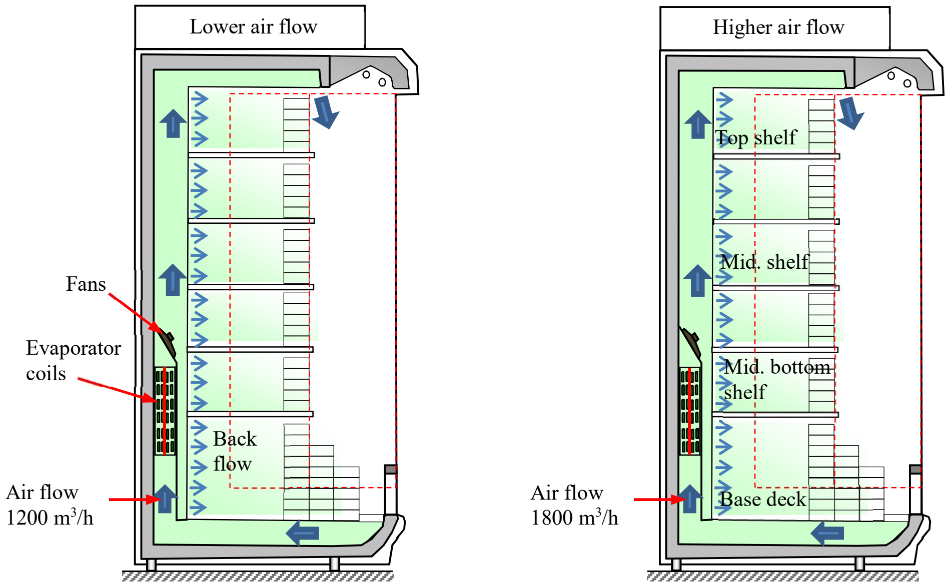


Figure 3. Air flow in the cabinet applied for the tests

In order to comply with the standard, two measurement systems were applied. The first one measured parameters of the refrigeration systems (pressure, temperature and mass flow rate of the cooling fluid), parameters of air, M-packages in the cabinet and parameters of the test room (temperature and RH). The measurement system consisted of some sensors which include temperature sensors with accuracy better than ±0.5°C, pressure transducers (accuracy ±1%), RH sensor (accuracy ±3 unit), mass flow meter (accuracy ±1%) and air velocity meter with accuracy 10%; data logging system (Labtech software and Datascan modules) and recording system (computer set and monitor). The second measurement system monitored and recorded power consumption of the cabinet which comprised a programmable power meter (HM8115-2 from Hameg Instrument; connected in series with the main supply). Temperature sensors (T type thermocouples) in the M-packages were placed on 4 shelves: top shelf, middle shelf, mid-bottom shelf and bottom deck as identified in Figure 3. The positions of the sensors were placed in accordance with [15].

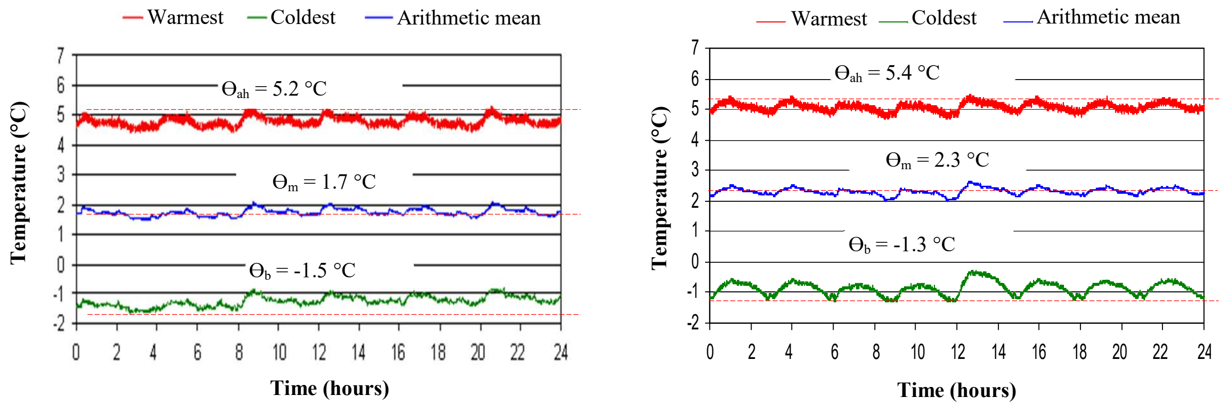
All measurements were recorded every 10s. This interval provided possibilities to check all temperature measurements at every 60s, mass flow rate and pressure measurements at every 20s as specified in [15]. Recorded data from the measurement systems were processed and analyzed. Performance parameters such as mean, the overall mean of the M-packages temperatures and energy consumption were calculated. M-packages represented foods that are being stored in the cabinet. EES (Engineering Equation Solver) software was applied to calculate refrigerant mass flow rate from the energy balance of the water-cooled condenser. The software was also used to determine the state of refrigerant in the compression cycle and to check whether the energy balance equation of the water-cooled condenser was valid for calculating the refrigerant mass flow rate. Further calculations and graphs manipulation were processed by using spread sheet program.

Results and Discussion

Temperature Performance

Temperature variation of the warmest and the coldest M-packages for Test-1 is presented in Figure 4a. The highest temperature of the warmest M-package was +5.2°C and the lowest temperature of the coldest M-package was -1.5°C. The overall mean temperature of the all M-packages was found to be 1.7°C. During the 24-hour test 2 M-packages on the middle shelf and 3 M-packages on the base deck laid above +4°C. All of the M-packages on the top shelf were below 3°C, but 2 of them were below -1°C.

a) Test-1 **b) Test-2**



Θ_{ah} = highest temperature of warmest M-package

Θ_m = Overall mean temperature of M-packages

Θ_b = Lowest temperature of coldest M-packages

Figure 4. Variation of the warmest, the coldest and the arithmetic mean temperature of M-packages for Test-1 and Test-2

Time based curve of the temperature of the warmest and the coldest M-packages for Test-2 is shown in Figure 4b. The highest temperature of the warmest M-package was +5.4°C and the lowest temperature of the coldest M-package was -1.3°C. The overall mean temperature of the all M-packages was found to be 2.3°C. Similar with the results of the Test-1, for 24-hour test some M-packages on the middle shelf and on the base deck laid above +4°C. In addition, some M-packages on the top shelf were below -1°C.

Based on the temperature of the M-packages, results of the Test-1 and Test-2 indicated that the cabinet with lower and higher air flow rate could not comply with the standard for M0 classification cabinet. Further optimization on the air flow rate distribution inside the cabinet would be required.

Energy Performance

Total Energy Consumption (TEC) of the cabinet was calculated according to [15]. For the cabinet equipped with integral condensing unit, TEC equals to Direct Energy Calculation (DEC) which can be calculated from:

$$TEC = DEC = \sum_{n=1}^{n=N_{max}} W_n \, x \, \Delta t \ \text{(kWh)} \tag{1}$$

W_n = instant power consumption of the cabinet (kW) over 24 hours ($W_n = 0$ during stopping and defrost time), Δt = period of measurement (h).

Table 1. Operational time of the cabinet during the tests

Operational time	Test-1	Test-2
Compressor ON/OFF frequency in 24 (h)	27	1
Running time t_{run} (h)	21.4	21.8
Defrost time t_{deft} (h)	1.0	2.0
Stopping time t_{stop} (h)	1.6	0.17
Percentage of the running time t_{rr} (%)	93.0	99.2

Operational parameters of the both tests include compressor On and OFF frequency, running, defrost, stopping and percentage of running time is presented in Table 1. It can be seen that Test-1 with lower air flow rate, number of compressor switching ON and OFF due to thermostat setting is much higher than it is in Test-2. This indicated that the refrigeration system of the cabinet stopped in every 50 minutes which make any frost accumulated in the evaporator melt. Therefore for the cabinet with lower air flow rate, frost would not be a problem.

Power consumption including total energy consumption (TEC) of the cabinet can be seen in Table 2. TEC of the cabinet was found to be 57.61 kWh.day^{-1} for Test-1 and 62.24 kWh.day^{-1} for Test-2. This results showed that the cabinet with lower air flow rate is more energy efficient compared to the cabinet with higher air flow rate.

Table 2. Power consumption and TEC of the cabinet

Power and Energy consumption	Test-1	Test-2
Maximum power (kW)	3.00	2.99
Minimum power (kW)	1.52	1.50
Average power (kW)	2.69	2.85
TEC (kWh.day^{-1})	57.61	62.24

The variation of instant power consumptions of the cabinet are shown in Figure 5a and 5b respectively for Test-1 and Test-2. From the figure can be seen compressor cycles in both tests. For Test-2 most of the compressor cycle due to defrost which was set in every 3 hours. This showed that higher air flow rate caused higher infiltration load and the system run continuously which could make problems with frost accumulation in the evaporator.

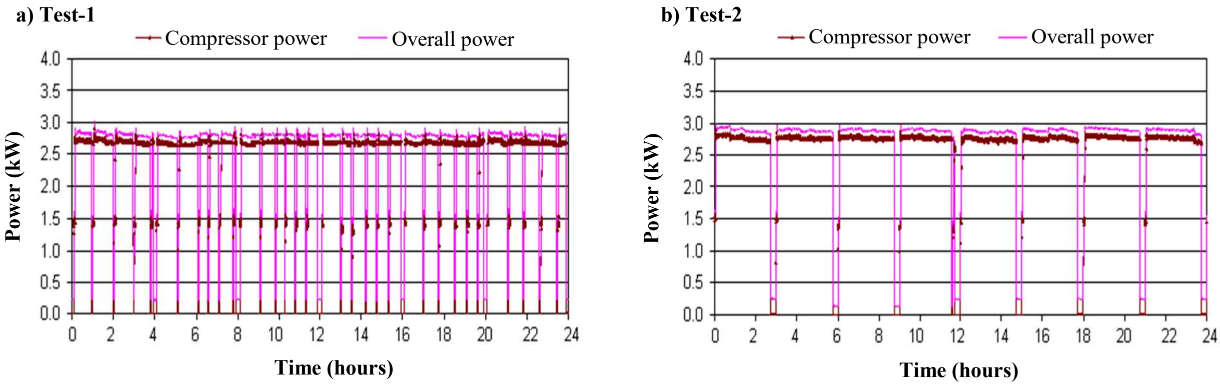


Figure 5. Power consumption of the cabinet for Test-1 and Test-2

An ECA (Enhanced Capital Allowance) performance criterion of a refrigerated display cabinet is expressed as Energy Efficiency Index (EEI) which can be calculated from:

$$EEI = \frac{TEC}{TDA} \text{ (kWh day}^{-1}\text{ m}^{-2}\text{)} \qquad (2)$$

The energy efficiency index of the cabinet and arithmetic mean coefficient of performance (COP) of the refrigeration systems of both tests are presented in Table 3.

Table 3. EEI and COP of the cabinet

	Test-1	Test-2
Energy efficiency index (EEI) kWh day^{-1} m^{-2}	8.28	8.95
Coefficient of performance (COP)	3.17	3.03

The EEI threshold for M0 classification cabinet with integral condensing unit is **12.50** (kWh.day^{-1}.m^{-2}) [18]

The Energy Efficiency Index (EEI) of the cabinet at for lower air flow rate is found to be 8.28 kWh day^{-1}m^{-2} and 8.95 kWh day^{-1}m^{-2} the higher air flow rate. This EEI value is far below the enhanced capital allowances (ECA) threshold of 12.50 kWh day^{-1}m^{-2} [18].

Conclusions

The HC cabinet with integral condensing unit was found to provide excellent energy performance with an Energy Efficiency Index below the requirement to qualify for enhanced capital allowances. The refrigeration system of the cabinet could also achieve a COP of 3.15 at climate class 3 conditions. Higher air flow rate in the cabinet could make the product temperatures a little bit better, but the energy consumption increased approximately 7% (from 58 kWh day^{-1} to 62 kWh day^{-1}). Higher air flow rate caused higher infiltration load and the system run continuously which could make problems with frost. For the lower air flow rate frost was not a problem due to the system switching ON and OFF every 50 minutes. Total fans capacity from 1400 to 1500 m^3 h^{-1} would be ideal for this cabinet. However, air flow distribution in the cabinet is necessary to be optimized in order to comply with M0 classification cabinet.

Acknowledgements
The authors would like to acknowledge the technical and financial contributions of WR Refrigeration and Arneg SPA to the project.

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Appendix-5: MANUSCRIPT FOR ICSEF (INTERNATIONAL CONFERENCE ON SUSTAINABLE ENERGY AND RESOURCE IN FOOD CHAIN)

(Presented in the next page)

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2nd International Conference on Sustainable Energy and Resource Use in Food Chains, ICSEF 2018

Development of Corn-Oil Ester and Water Mixture Phase Change Materials for Food Refrigeration Applications

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Abstract

This research aims to investigate development of corn-oil ester and water mixtures as novel solid-liquid phase change material candidates for chilled and frozen food refrigeration applications. Thermal properties of both water and its mixture with corn-oil ester were tested by DSC and T-history methods. The results showed that corn oil could mix well in water solutions. Phase transition temperatures of the mixtures were lower than those of individual water. Corn-oil ester in the mixtures was acted as a nucleate agent and it was able to lower freezing point and to trigger ice nucleation in water which could diminish super-cooling. Addition of corn oil ester by 5% to 35% in water solutions could decrease freezing temperature from 0°C down to respectively -3.5°C to -28°C. The PCM candidates were also found to have excellent thermal properties that could fulfill requirements of thermal energy storage systems for food refrigeration applications.

Keywords: corn-oil ester; phase change materials; thermal energy storage; food refrigeration system

Introduction

Phase change material (PCM) is one of thermal energy storage (TES) technology which can improve the performance and reliability of energy systems. The technology could also potentially provide energy savings, which in turn could reduce environmental impact related to energy use [1]. Phase change materials (PCMs) store heat by using latent heat, commonly from solid to liquid, as they can exhibit latent heat of phase change and have attracted interest as possible heat thermal storage [2]. Phase change storage with PCMs is one of the most efficient ways to store thermal energy [3]. One advantage of the PCMs is it has high energy storage density with small temperature variation during the process of phase change [4]. They have been used to improve the TES capacity of different systems [5]. PCMs with large latent heat of fusion are also increasingly being used for thermal management of air conditioning in buildings in order to achieve a better balance between cooling supply and demand [6]. In the last 15 years PCMs have also been gradually used for food storage and transportation system [7].

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Nomenclature	
COP	Coefficient of Performance
DSC	Differential Scanning Calorimeter
GCMS	Gas Chromatography Mass Spectrometry
PCMs	Phase Change Materials
TES	Thermal Energy Storage

Many researchers have also carried out investigations on the feasibility of application of PCMs in improving the performance of refrigerated cabinets, chest freezer and domestic refrigerators. Applications of integrated thermal energy storage with PCMs have the potential to increase the energy efficiency of the refrigeration systems. The improvement can be achieved by reducing compressor cycling frequency and cycling losses. Moreover, the use of PCMs can also maintain product temperature within a safe temperature range in the event of electrical power failure [8,9]. Experimental investigations on the performance of household refrigerators using PCM were carried out by Azzouz *et al.* [10,11]. The PCM was placed in a container at the back of the evaporator plate between. The results showed that the response of the refrigerator to the addition of PCM and its efficiency were strongly dependent on the thermal load. The integration of PCM allowed 5 to 9 hours of continuous operation without electrical supply. This could increase the coefficient of performance (COP) of the system by 10% - 30% depending on the cooling load. While studies on the use of PCMs in freezers were reported by [9,12,13]. The PCM was introduced close to the evaporator wall. The results indicated that, during electrical power failure the use of PCM could maintain product temperature for longer compared to the freezer without PCM. Lu *et al.* [14] and Jouhara *et al.* [15] investigated the use of PCM and heat pipes to provide product temperature uniformity on the shelves of vertical multi-deck food display cabinets. The PCMs were placed within the structure of the shelves of the display cabinet. The results showed that the use of heat pipes could homogenize the temperature profile of the products and improve the heat transfer between the cabinet, the shelves and the products.

PCMs are generally grouped into organic and inorganic compounds [16,17]. Organic PCMs are very important class materials because of their unique thermal properties such as congruent melting process and narrow melting-freezing temperature ranges [18,19]. Paraffin, the most commonly used organic PCM, have been widely used for energy storage due to its wide range of phase change temperatures, negligible super-cooling, no corrosive behavior and chemical stability [6,20]. However, paraffin relatively has higher cost, high volume change, lower latent heat and lower thermal conductivity. Another serious issue of the paraffin is its high flammability [20]. The low thermal conductivity of paraffin requires heat transfer enhancement methods such as the incorporation of materials with high thermal conductivity [21-23], increasing heat transfer surface area [24-28] or application of compact heat exchanger [6,29]. While salt water solutions are very common inorganic PCMs. The solutions have advantages of higher thermal conductivity, fusion heat and density, and lower flammability. However, salt water solutions possess serious issues of corrosion and super-cooling.

The best-known PCM is water. It has very good thermal properties such as reliability, low cost, high specific heat, high density, high latent capacity of 335 kJ/kg and safe [30]. Unfortunately, water cannot be used on its own as a PCM in food refrigeration of temperature range below 0 °C [31]. Water also has a big degree of super-cooling during solidification process [32]. In some applications, degree of super-cooling can have major effect on a system performance [33]. In order to make water applicable as PCMs at temperatures below 0 °C, nucleation agent could be added to trigger heterogeneous nucleation. This could also eliminate the super-cooling of water [34]. A food grade antifreeze or nucleation agent in the water would be required [35]. This will maintain the high percentage of water in the solution and the high latent heat of the PCM making it a good candidate for applications just below 0 °C.

The main objective of this paper is to develop phase change material candidates for medium and low temperature food refrigeration applications. The PCM candidates were made by mixing corn-oil ester which worked as nucleation agent. Corn-oil ester and water solutions to be investigated are applicable for medium and low temperature food refrigeration of evaporating temperature of the system between -35 °C and -8 °C. The solutions contain only small portion of corn oil ester. Larger part of the solutions is water which makes them become strong PCM candidates for food refrigeration applications. Moreover, corn-oil ester also contains various types of fatty acids which have many superior properties as organic PCM materials [36-38]. Fatty acids are also derivatives of materials that are readily found in nature such as vegetable oils and labeled as bio-based materials [39]. However, fatty acid ester is more expensive compared with corn-oil ester. Another advantage is that corn-oil ester offers a continuous supply [40,41], no corrosive behavior, non-flammable, and non-toxic, therefore it is suitable for food refrigeration.

Materials and Characterization

Materials

Materials used in this study were tap water and corn oil ester as nucleating agent resulted from esterification of corn oil. Corn oil ester was chosen because it contains a lot of unsaturated fatty acids so it has low freezing and melting points. The corn oil ester is composed mainly by methyl esters of 38.54%. The oil ester also contains benzene (17.45%), 1,3-cyclohexadiene (8.29%), beta-sesquiphellandrene (23.83%) and others of about 11.89%. Esters are polar molecules that have a very important role on the solubility of corn oil in water. These chemical compositions of corn-oil ester were obtained from Gas Chromatography Mass Spectrometry (GCMS) [42]. The test method comprised analysis of corn oil ester which was performed on a GC-MS Shimadzu type QP 2010 with a split/split less injector. The GCMS test results are presented in Table 1.

Table 1. Chemical composition of investigated corn-oil ester

Component name	Formula	Area (%)
3-Isopropoxy-1,1,1,7,7,7-hexamethyl-3,5,5-tris (trimethylsiloxy)	$C_{18}H_{52}O_7Si_7$	0.61
Benzene, 1-(1,5-dimethyl-4-hexenyl)	$C_{15}H_{22}$	17.45
1,3-Cyclohexadiene, 5-(1,5-dimethyl-4-hexenyl)	$C_{15}H_{24}$	8.29
Copaene	$C_{15}H_{24}$	0.28
8-Nonenoic acid, 5,7-Dimethylene-, methyl ester	$C_{12}H_{18}O_2$	0.50
Cyclohexene, 1-methyl-4-(5-methyl-1-methylene-4-hexenyl)	$C_{15}H_{24}$	8.45
Dodecanoic acid, methyl ester	$C_{13}H_{26}O_2$	10.92
Beta-sesquiphellandrene	$C_{15}H_{24}$	23.83
Hexadecanoic acid, methyl ester	$C_{17}H_{34}O_2$	13.28
3-Butoxy-1,1,1,7,7,7-hexamethyl-3,5,5-tris (trimethylsiloxy)	$C_{19}H_{54}O_7Si_7$	0.68
Dodecanoic acid, (2,2-dimethyl-1,3-dioxolan-4-yl) methyl ester	$C_{18}H_{34}O_4$	
Hexadecanoic acid, (2,2-dimethyl-1,3-dioxolan-4-yl) methyl ester	$C_{22}H_{42}O_4$	2.95
2-Heptadecanone, 1- (2,2-dimethyl-1,3-dioxolan-4-yl) methoxy	$C_{23}H_{44}O_4$	
Anodendroside G, monoacetate	$C_{32}H_{42}O_{11}$	0.48
9-Octadecenoic acid (Z), methyl ester	$C_{19}H_{36}O_2$	
7-Hexadecenoic acid, methyl ester	$C_{17}H_{32}O_2$	6.21
9-Octadecenoic acid, methyl ester	$C_{19}H_{36}O_2$	
Cyclopropanebutanoic acid	$C_{25}H_{42}O_2$	1.38
Oxiraneoctanoic acid, 3-octyl, methyl ester, trans	$C_{19}H_{36}O_3$	
Heptasiloxane, hexadecamethyl ester	$C_{16}H_{48}O_6Si_7$	1.63
Octadecanoic acid, methyl ester	$C_{19}H_{38}O_2$	1.99
Heptasiloxane, hexadecamethyl ester	$C_{16}H_{48}O_6Si_7$	1.06

Methyl ester, which mainly contained in corn oil ester, is a small ester with single carbon chain. Small esters are soluble in water. This is a key role the solubility of corn oil ester in water. The solubility of corn oil ester can also be explained that certain acid molecules of ester in water solution having -OH cluster are ionized by releasing hydrogen atom to make ion H+. Even though esters cannot hydrogen bond with themselves but esters can hydrogen bond with water molecules. Individual positive hydrogen atom in a water molecule can be attracted to one of the single pairs on one of the oxygen atoms in an ester for a hydrogen bond to be formed. Moreover, there are also dispersion forces and dipole-dipole attractions between the ester and the water molecules which release energy. This helps to supply energy required to separate water molecule and ester molecule from others before they can mix together [43]. This explains why corn oil ester dissolves in water. The corn-oil esters were chosen as nucleating agents for the purpose of obtaining a food grade PCM which was considered as one important factor for food refrigeration applications. PCMs made from the mixture of tap water and corn oil esters are also economically competitive compare with paraffin based PCM. At present, the market price of fatty acid esters is relatively high. This is because of the cost of producing the fatty acid esters is higher than that of corn oil esters due to production line of fatty acid ester includes purification process. While the corn-oil esters can be used without further purification.

Characterization of PCM

Thermal properties of the PCM candidates (of corn-oil ester in water mixtures) were measured by differential scanning calorimeter (Perkin Elmer Jade DSC). The properties included melting and freezing temperatures and latent heat of melting and freezing. The analyses were performed at temperatures between 25 °C and -100 °C for cooling and from -100 °C to 25 °C for heating at 2 °C per minute of cooling and heating rate. The analyses were also performed under a constant stream of nitrogen gas at flow rate of 20 mL per minute. The temperature accuracy was ± 0.01 °C and heat flow repeatability was 0.2 μW. A 30 mg sample of PCM candidate was sealed in an aluminum pan. The melting and crystallization points were taken as onset temperatures.

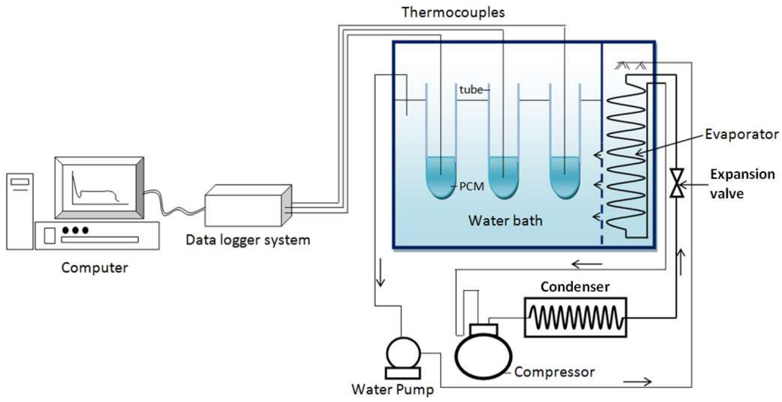

Fig. 1. Schematic diagram of T-history method

The latent heat of PCM candidates was determined by numerical integration of the area of peak thermal transition. Even though phase change temperatures of the solutions can be measured by the DSC system, the specimen used in DSC is very small (of about 10-30 mg) which is not applicable for practical use especially for samples that contain water with high degree of super-cooling [44,45]. Whereas degree of super-cooling is an important parameter for PCMs. In this research, phase change temperatures and degree of super-cooling of the PCM candidates were tested by using T-history method which is considered more suitable for this application. Schematic diagram of the T-history method is shown in Fig. 1. The PCM candidates tested include the mixtures of 5%, 7.5%, 10%, 12.5%, 15%, 20%, 25%, 30% and 35% corn oil ester in water.

Results and Discussion

Super-cooling analysis

Super-cooling occurs when the temperature of a liquid is lowered below its freezing point without becoming a solid [46]. Fig. 2a shows that tap and mineral water was super-cooled to reach -7.5 °C and -8.5 °C respectively before the ice formation process started. The ice crystallization process involves combination of nucleation and growth of ice crystals within a crystalline structure. Ice crystal formation occurs after nucleation, at which the water molecules join the already formed nuclei. For comparison in Fig. 2b is illustrated super-cooling of propylene glycol solutions. Super-cooling occurs at lower concentration of propylene glycol solution. At higher concentration, the super-cooling disappears.

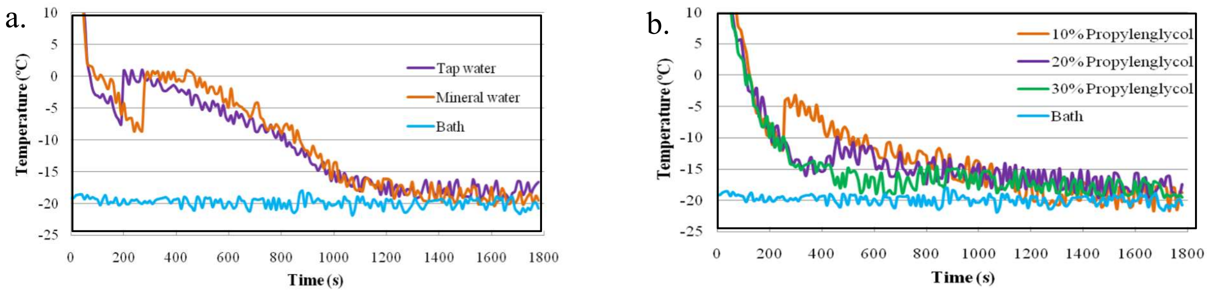

Fig. 2. Super-cooling: tested at bath temperature of -20 °C; a. pure water, b. propylene glycol in water solutions

Fig. 3a and 3b shows that PCM candidates with different concentration of corn-oil ester are able to initiate formation of ice nuclei quickly at somewhat higher temperature than its approaching freezing-point of the solution. For the history-T method, the test can be done up to 25% corn oil ester in water solution due to limitation of the minimum bath temperature. It can be seen the additions of 5%, 7.5%, 10%, 15%, 20% and 25% of corn-oil ester in the PCM candidates can decrease tap water freezing point to -3.5 °C, -6 °C, -7.5 °C, -10 °C, -15 °C and -19.5 °C respectively. They can also reduce super-cooling of the pure water. The addition of corn-oil ester as solute particle into the tap water as solvent produce some ions that contribute to intermolecular force between solvent and solute particles.

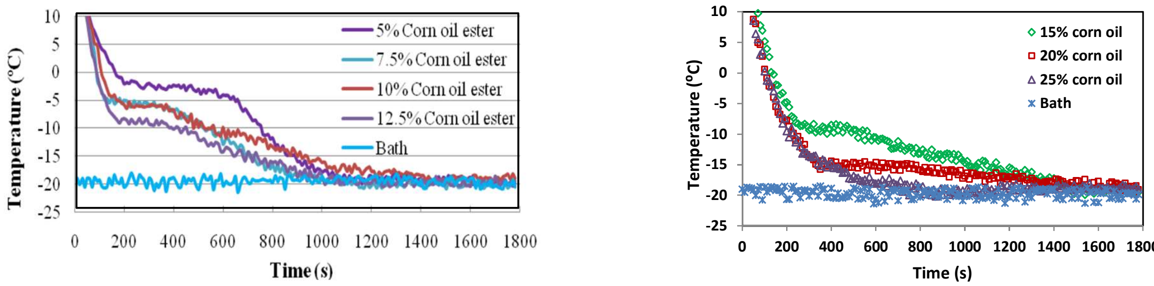

Fig. 3. Cooling process of corn-oil ester in tap water solutions at bath temperature of -20 °C

During cooling process, the pulling force between solvent and solute particles release more heat hence the freezing point of the solution is lowered. Therefore, corn-oil ester solution is able to reduce or even eliminate super-cooling due to: (i) faster nucleation and (ii) lower freezing point.

Thermal properties of the PCMs

In order to compare thermal properties and phenomena in melting and freezing processes of the PCM candidates, the results of DSC for melting and freezing processes of tap water is also presented in Table 2. The melting and freezing temperatures of tap water resulted from DSC were 0 °C and -19.5 °C respectively, and the latent heat of melting and freezing were 297.4 J/g and 102.4 J/g respectively. It is noteworthy that, whatever the sample size, ice melts at 0 °C. On the contrary, freezing occurs at different temperatures, depending on the water sample size [45]. From nucleation theory, it has been shown that the smaller the volume, the lower the freezing temperature. For bulk water, freezing occurs at -14 °C for a volume of 1 cm3 and at around -24 °C for a volume of 1 mm^3, while for micro-sized droplets (1 μm^3) freezing is found around -39 °C [44]. The energy released during the freezing process is evidenced on the DSC result as an exothermic peak with imperfect bell shape when compared with melting process.

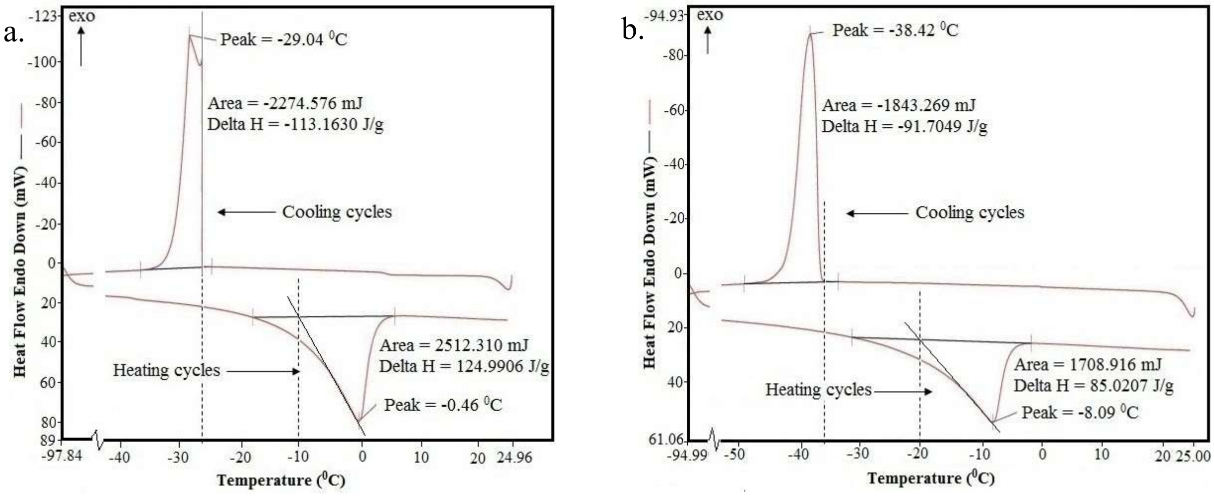

Fig. 4. DSC curves of heating and cooling processes: a. 15% corn-oil ester, b. 25% corn oil ester

Thermal properties of PCM candidates which contain tap water and various compositions of corn-oil ester can be seen in Table 2. While Fig. 4 shows only two DSC curves in order to show super-cooling. Other curves of the corn oil ester solutions are not presented. Fig. 4 shows thermal properties of PCM candidate with 15% and 25% corn-oil ester. From the figure it can be seen that the addition of 15% corn-oil ester into tap water still demonstrates the occurrence of super-cooling. Increasing the concentration of the corn-oil ester to 20% (it is not shown in the figure) causes degree of super-cooling of the PCM solution decreases. Degree of super-cooling is totally disappeared as the concentration of corn-oil ester reaches 25% (Fig. 4b). This is indicated by a perfect bell shape shown in Fig. 4b.

Table 2. Thermal properties of the PCM candidates, DSC Test Results

Samples (Vol. %)	DSC			
	Heating process		Cooling process	
	Melting temp. (Tm, °C)	Latent heat of melting (ΔH_m, J/g)	Freezing temp. (T_f, °C)	Latent heat of freezing (ΔH_f, J/g)
Tap water	0	297.4	-19.5	-102.4
5/95 (E/W)	-3.5	227.8	-22	-103.3
7.5/92.5 (E/W)	-6.0	222.7	-20	-120.3
10/90 (E/W)	-7.5	171.7	-25	-135.2
15/85 (E/W)	-10.5	125.0	-27	-113.2
20/80 (E/W)	-15	107.3	-33	-109.8
25/75 (E/W)	-19.5	85.0	-36	-91.7
30/70 (E/W)	-23	78.6	-38	-84.7
35/65 (E/W)	-27	68.7	-43	-67.2

E/W = Corn-oil ester in water

The freezing temperatures vary from one sample to another, because nucleation is a stochastic phenomenon. Results of DSC test method are also summarized in Table 2. The table clearly shows that melting temperatures of the PCM candidates are lower than those of the tap. The melting and freezing temperatures of corn-oil ester in water solutions of concentration between 5% and 35% (by volume) range from -10 °C to -27 °C. While melting latent heat varies from 68.7 J/g to 227 J/g respectively. The results indicated that by increasing concentration of corn-oil ester in water solution can reduce melting temperature and minimize or even negate the super-cooling. These properties make the solutions potential to be PCMs with large latent heat and suitable phase change temperatures for medium and low temperature food refrigeration applications. For comparison, Fig. 5 shows melting temperature of PCM candidate corn oil ester (COE) in water solutions, propylene glycol solutions and NaCL solutions at different concentrations. The PCM candidates (Corn oil ester solutions) require lower concentration to achieve the same melting temperature below 0 ºC. This indicates less material is needed for the corn oil ester PCM.

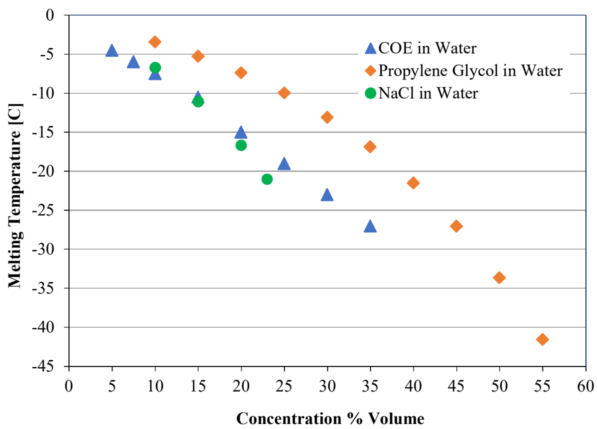

Fig. 5. Melting temperatures of Corn oil ester solutions in comparison to NaCl and Propylene glycol solutions

Conclusions

Corn oil ester in tap water mixtures have been investigated for development of phase change materials (PCMs) as thermal energy storages that can be applied for food temperature refrigeration systems. DSC and T-history thermal analyses were applied in the investigation and it has been found that the water-based mixtures contain 5% up to 35% corn-oil ester have freezing temperatures -3.5 ºC to -27 ºC respectively. The investigation also found that the PCM candidates at test conditions have minimum or even without degree of super-cooling. Additionally, corn-oil ester and water solutions offer a continuous supply and cheaper compared with fatty acid esters that frequently used for below 0 °C applications, no corrosive behavior as well as non-toxic. These make the mixture of corn oil ester and water become applicable as low cost novel PCMs for food refrigeration applications.

Acknowledgements

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Appendix-6: MANUSCRIPT FOR IJCST (INTERNATIONAL JOIN CONFERENCE ON SCIENCES AND TECHNOLOGIES)

(Presented in the next page)

Performance Analyses on Evaporator Coils of Sustainable Cold Storages for Food Chain in Indonesia

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Abstract— **The development of cold storage in Indonesia is less than satisfactory. As the second highest producers of fishery and aquaculture products in the world, the country has only 2 out of total 6 of its ocean fishing ports possess cold storage facilities. Indonesia is also on the list of low index cold storage markets with abundant natural resources. The lack of cold storage facilities has greatly restricted the development of its fishery and food industries. The current infrastructure is also too poor to exploit the resources efficiently. Under this circumstance, technologies that can encourage development of sustainable cold storages applied for food chain and related components such evaporator coils in the country is in great demand. To date there has not been much research into cold storage evaporator coil design specifically for natural refrigerants such as Hydrocarbon (HC) and CO_2. This paper presents results of theoretical investigation on the main design parameters of natural refrigerant evaporator coils which can be applied in sustainable cold storage systems. The parameters included tube diameter, number of refrigerant circuits, evaporator temperature and circulation ratio. A lumped element technique was applied to develop evaporator models which were used to design and to simulate different coil geometry and circuit arrangements. The results from published papers were used for model validation. The evaporators considered were direct expansion and flooded evaporator coils for low temperature cold storage applications. The paper also presents comparison analyses of natural refrigerant (HC and CO_2) evaporator coils and direct expansion coils using the synthetic refrigerant R-404A.**

Keywords— Evaporator coil, natural refrigerants, sustainable cold storage, performance analyses

I. Introduction

Indonesia is an agriculture country with population number of 258.5 million. Poultry, beef and veal production are anticipated to increase 3 to 5 percent annually through 2020, while consumption is expected to rise 4 to 6 percent annually [1]. The country has experienced unbalanced food supply and demand which may need to be well-adjusted through import policy. As an example, beef supply of the country in 2016 was estimated 348,020 tons, while the demand was 651,420 tons [2]. Unbalanced food supply will boost the price of food, sometimes reaching unreasonable level which very much affects the lives of low income populations. This is one of the challenges for the country in improving food security and sustainability.

Furthermore, FAO [3] stated that Indonesia is the second highest producers of fishery and aquaculture products in the world. Fishery and aquaculture production of the country in 2014 was 14.33 million tons. In 2016, the production reached 23.03 million tons which 27.6% came from marine fisheries and 72.4% from aquaculture [4]. Additionally, Indonesian territory consists of 2/3 of water, has given enormous benefits for Indonesia, especially fishermen. To improve the economic level of fishermen requires efforts to develop proper facilities. One of the efforts is by improving the quality of products which can be marketed in the regional and international levels. It is certainly need the support of the existence of various fishery facilities, one of which is cold storage [5].

The development of cold storage for fishery industry in Indonesia is less than satisfactory. As the second highest producers of fishery and aquaculture products in the world, the country has only 2 out of total 6 of its ocean fishing ports possess cold storage facilities. Moreover, only 4 out of total 14 national fishery ports own cold storage facilities [5]. Indonesia is also on the list of low index cold storage markets with abundant natural resources [1]. The lack of cold storage facilities has greatly restricted the development of its fishery industry. The current infrastructure is also too poor to exploit the resources efficiently. Under this circumstance, technology that can encourage development of infrastructure including cold storage in the country is in extremely great demand.

The demand for refrigerated facilities such as refrigerated warehouse, cold storage and retail refrigeration in Indonesia is expected to increase with the country's economic development because they have a vital role to play in reducing post-harvest losses, improving quantity and quality of fishery and aquaculture products and maintain food supply to consumers. The facilities enable to store over supply of foodstuffs during crop season and use them when there are no crops. The refrigerated facilities are also essential for food quality preservation. However, the increase of refrigerated facilities can provide impact to the environment due to refrigerant leakage and energy use. It has been well-known that refrigeration systems consume intensive energy.

The use of natural refrigerants such as CO_2 and Hydrocarbon offers the opportunity to reduce not only the direct impacts of systems employing HFC refrigerants that possess high global warming potential but also the indirect impacts by improving energy efficiency. Another advantage of CO_2 over HFC refrigerants is its better heat transfer properties that can lead to an increase in the evaporating temperature. A consequence of this is a potential increase in the refrigeration capacity of the coil and a reduction in the rate of frost formation on the coil surface.

Supermarkets have two refrigeration temperature levels, medium temperature (MT) and low temperature (LT) refrigeration. Evaporating temperature of MT refrigeration system is -8ºC and -30ºC for LT refrigeration system. The refrigeration systems employed can be direct expansion or of the secondary loop type. In conventional supermarkets, the direct expansion refrigeration system is the most commonly used to provide refrigeration to display cabinets located in the store. For supermarkets which have used natural refrigerants such as CO_2, the applications of secondary refrigeration loops are of particular interest. As CO_2 has low viscosity, the use of CO_2 refrigerant as volatile secondary fluid can significantly improve the performance of the refrigeration system due to its small pumping power. Analyses on secondary loop refrigeration systems using CO_2 as secondary fluid has been reported by [6,7].

Finned tube heat exchangers are commonly used as forced air evaporators of display cabinets in supermarkets. Performance of the evaporator coil directly affects the temperature performance of a display cabinet and the overall performance of the supermarket refrigeration system. The influences of geometry and configuration of finned tube coils using synthetic refrigerants have been intensively investigated by many researchers. Romero-Mendez *et al.* [8] investigated the effects of fin spacing to the hydrodynamics and heat convection of a plate fin and tube heat exchanger. Liang *et al.* [9] and Jiang *et al.* [10] showed the impacts of circuiting on performance and parameter distributions within the tubes and across the coil. Getu and Bansal [11] developed a model of R-404A evaporator coil to analyze the performance of the coils in LT supermarket display cabinets. Chandrasekharan and Bullard [12] developed a design tool for a fin and tube display cabinet evaporator to predict local and overall effects of frost accumulation.

To date there has not been much research into evaporator coil design specifically for CO_2 refrigerant. Aidoun and Ouzzane [13] established a numerical model to study the effects of circuitry of CO_2 finned tube evaporators and found that it was possible to use longer circuits, thus reducing the number of circuits for a given refrigeration capacity. Authors in [14,15] investigated the impact of the geometry, tube circuitry and tube diameter on the performance of CO_2 evaporators and showed that by reducing the number of circuits could increase the velocity of refrigerant and reduce the total length of pipe. This study investigated the performance of CO_2 evaporator coils under different geometry, circuitry arrangement and different operating conditions for chilled food and frozen food display cabinets in supermarket applications. Comparison analyses with evaporator coils using R-404A refrigerant were also performed.

Supermarket refrigeration systems using HFC refrigerants are responsible for substantial greenhouse gas emissions from leakage of refrigerant to the ambient and indirect emissions from the electrical power used by the compressors, fans and other ancillary equipment [16]. Direct emissions from refrigerant leakage can sometimes be as high as indirect emissions, and for this reason, legislation at has been aimed at effecting reductions in direct emissions. Another way to significantly reduce or completely eliminate direct emissions is through the use of natural refrigerants, such as hydrocarbons, CO_2 and ammonia. In recent years, considerable research has been carried out on the development and application of supermarket refrigeration systems employing natural refrigerants. Most systems deployed in the field are either trans-critical 'booster' CO_2 systems, cascade all CO_2 systems, or subcritical CO_2 systems cascaded with a hydrocarbon system on the high pressure side for heat rejection [17,18,19].

An interesting approach developed and applied by some supermarket chains involves the use of 'integral' hydrocarbon cabinets with heat rejection to the air in the supermarket [20] and combination of heat rejection to the air as well as water in a closed loop system [21]. The heat in the water circuit can be either upgraded through a boiler or heat pump and used for domestic hot water and/or space heating, or rejected to the ambient through a dry cooler. This approach can provide energy integration between the refrigeration and space conditioning systems in the store and offers the potential for energy savings if the system is appropriately designed and controlled.

II. MATEMATHICAL MODEL APPROACH

A. The Mathematical Model

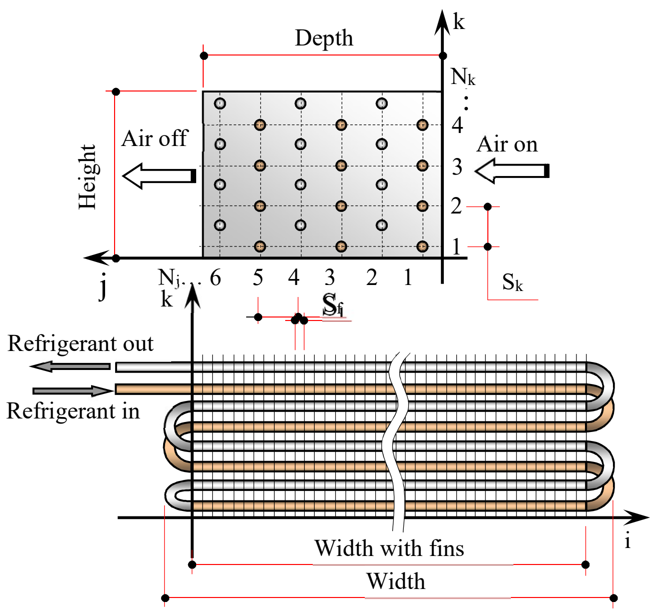
Three main mathematical models were established to investigate the performance of CO_2 and HC evaporator coils under different geometry, circuit arrangement and different operating conditions. The first model was for the investigation of the performance of MT CO_2 flooded evaporator coil. The second model was for the simulation of the performance of CO_2 DX evaporator coils for both chilled and frozen temperature levels. The third model was for HC (R-1270) DX evaporator coil. The additional model was also developed to analyze the coil performance using refrigerant R-404A for comparative analyses. The models can also be used to design the geometry and tube arrangement of evaporator coils for a given refrigeration capacity. The numerical models apply standard plate fin specification from [22] to determine fin and tube pattern, height and width of the coil. The models were developed using the software EES.

To simulate the flooded and DX evaporator coils, some main assumptions were made as follows: steady state flow conditions; one dimensional flow for refrigerant inside tubes and air across the coil; negligible thermal losses to the environment; uniform temperature and air flow; constant air side convective heat transfer coefficient over the entire coil; intermediate pressure (P_{int}) to be considered as condensing pressure for CO_2 DX evaporator coil, negligible refrigerant pressure drops of less than 2 K saturated temperature equivalent for DX coils [22] and less than 1 K for flooded coils; the same number of tubes in each circuit with the same fraction of total mass flow rate; quasi steady frosting process; maximum pressure drop at air side after frost to be lower than 0.175 kPa.

To simulate the geometry and circuitry of the evaporator coils, certain assumptions were made as follows: constant air side convective heat transfer coefficient over the entire coil;

condensing pressure of 12.7 bar_a (corresponds to 29 ºC condensing temperature), evaporating temperature of -7 ºC; negligible refrigerant pressure drop of less than 2 K saturated temperature drop equivalent; the same number of tubes in each circuit with the same fraction of total mass flow rate; quasi steady frosting process; maximum pressure drop at air side after frost to be lower than 0.175 kPa.

The mathematical modeling approach and design strategy for the coil followed the process described by [15]. The evaporation heat transfer coefficient for refrigerant R1270 was determined from the correlation by [23]. The two phase pressure drop was calculated from [24,25]. The heat transfer coefficient and pressure drop correlations were associated with the flow pattern map developed by [26].



Geometry of a finned tube evaporator model

Fig. 1 shows the basic geometry of the finned tube evaporator considered in the models. The tubes are arranged in coordinates along width, depth and height axes (i, j, k) as can be seen in the figure. The number of rows and tube pattern can be used to determine the size of the coil and the tube interconnections within the coil circuits. If the coil has more than one circuit, the number of tubes in each circuit should be evenly balanced.

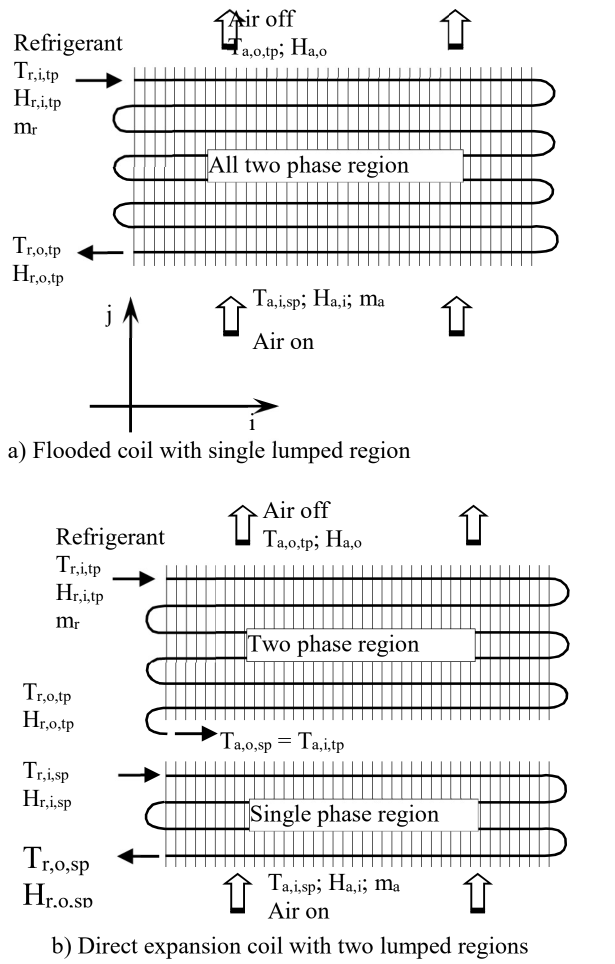
The mathematical models used the lumped element technique by which the evaporator coil can be divided into the superheated and two phase regions. A DX coil has two lumped regions (single and two phase regions), while a flooded evaporator coil only has a one region, the two phase region as shown in Fig. 2. Each region is considered as a single control volume. The fraction of the coil area in each control volume in a DX coil is calculated in proportion to the amount of heat transfer in each control volume.

The mass and energy balance principles are applied to each control volume, which is summarized in equations (1) to (3). For the flooded evaporator coil, a single phase region does not exist, thus the heat transfer rate component for single phase region ($Q_{r,sp}$) is omitted. ΔT_{lm} is the logarithmic mean temperature difference of each control volume. The air side surface area of the coil (A_a) and other geometric parameters of the coil such as free flow area and free flow area with frost were calculated as in [27].

$$Q_e = Q_{r,tp} + Q_{r,sp} = Q_a \tag{1}$$

$$Q_e = m_r\,(h_{r,o,tp} - h_{r,i,tp}) + m_r\,(h_{r,o,sp} - h_{r,i,sp}) = m_a\,(h_{a,i} - h_{a,o}) \tag{2}$$

$$Q_e = U_{a,tp} A_{a,tp} \Delta T_{lm,tp} + U_{a,sp} A_{a,sp} \Delta T_{lm,sp} \tag{3}$$



Schematic of flooded and direct expansion evaporator coils with single and two control volumes

The overall heat transfer coefficient (U_a) of the coil with frost can be calculated from equation (4). For frost free coil the component of frost resistance is not included. For the DX coil the external and internal heat transfer areas as well as the internal heat transfer coefficient depend on the mode of heat transfer, single or two phases.

$$\frac{1}{U_a} = \frac{1}{\eta_f h_a} + \frac{\delta_{frost}}{\eta_f \lambda_{frost}} + \frac{A_a \ln\left(\frac{d_o}{d_i}\right)}{2\pi\lambda_t L_r} + \frac{A_a}{A_r h_r} \tag{4}$$

B. Correlations of heat transfer coefficient and pressure drop at refrigerant side

The local heat transfer coefficient and pressure drop for the two phase flow of CO_2 were calculated using the correlations reported by [28,29,30]. The correlations for the two-phase frictional pressure drop for CO_2 were based on the correlations proposed by [24,25.

The local heat transfer coefficients and pressure drop correlations were selected for each flow regime as it changes with the flow and evaporation of refrigerant in the evaporator. Thus the correlations reliably capture the variation of two phase heat transfer coefficient and frictional pressure drops at different mass velocities and vapor qualities.

Equations (5) and (6) show the general equations for the two phase heat transfer coefficient and pressure drop on the refrigerant side of the evaporator. The detailed correlations for each flow regime and for the single phase region can be found in [29,30].

$$h_{tp} = \frac{\theta_{dry} h_v + (2\pi - \theta_{dry})\left[(Sh_{nb})^3 + h_{cb}^3\right]^{(1/3)}}{2\pi} \tag{5}$$

$$\Delta P_{total} = \Delta P_{static} + \Delta P_{momentum} + \Delta P_{frictional} \tag{6}$$

The evaporation heat transfer coefficient for refrigerant R-404A was determined from the correlation by [23]. The two phase pressure drop was calculated from [24,25]. The heat transfer coefficient and pressure drop correlations were associated with the flow pattern map developed by [26].

C. Correlations of heat transfer coefficient and pressure drop on the air side

Air side convective heat transfer coefficient has been calculated using the Colburn j-factor proposed by [31], while the total heat transfer coefficient for wet coil and pressure drop were calculated based on the equations by [11]. The air side heat transfer coefficient can be calculated from:

$$h_a = h_{c,a} + h_{lat,a} \tag{7}$$

$$h_{c,a} = \frac{j\, Cp_a\, G_a}{\Pr^{2/3}} \tag{8}$$

D. Calculation of frost accumulation

Frost accumulation on the evaporator surface has been estimated using the method proposed by [11]. The rate of frost accumulation was determined from equation (9) and the amount of frost accumulated on the surface of the evaporator and frost thickness after Δt time, from equation (10).

$$m_{frost} = m_a(\omega_i - \omega_o) \tag{9}$$

$$\Delta m_{frost} = m_{frost}\,\Delta t \quad \text{and} \quad \delta_{frost} = \frac{\Delta m_{frost}}{\rho_{frost} A_a} \tag{10}$$

Detailed calculations of density, thermal conductivity and diffusivity of the frost can be found in [11].

III. Results and Discussion

Test results from the experimental CO_2 test facility were used to validate the models. The model of conventional evaporator coil with R-404A was validated against data provided by the manufacturer. Comparison between predictions and experiments under design conditions was found to be satisfactory for the refrigeration capacity as shown in Table I. The pressure drop estimations were, however, lower than the experiment results mainly because the pressure drops across the distributor and lead tubes were not included in the model. For synthetic refrigerants these pressure drops can be as high as 89% of total pressure drops in the evaporator coil [32].

Model and experiment results

Parameters		[a] MT CO_2 Models		[b] DX LT CO_2 model	[c] DX R-404A model
		Flooded	DX		
Q_e (kW) full load, ΔT_a = 10 K	Model/	5.19	5.09	3.00	-
	Experiment	5.10	4.93	2.89	-
Q_e (kW) steady state load, ΔT_a = 9 K for MT and ΔT_a = 8 K for LT	Model	4.55	4.46	2.35	3.65
	Experiment	4.42	4.30	2.12	3.60*
ΔP_r (kPa) steady state	Model	11.31	9.22	6.01	40.92
	Experiment	21.15	16.87	14.87	148.28*

Evaporator coil investigated:
[a] Tube arrangement: staggered; d_o = 12.70 (mm); N_k = 4; N_j = 6; number of circuits = 4; fins pitch 4 fins/inch
[b] Tube arrangement: staggered; d_o = 12.70 (mm); N_k = 4; N_j = 8; number of circuits = 3; fins pitch 3 fins/inch
[c] Tube arrangement: inline; d_o = 15.87 (mm); N_k = 2; N_j = 16; number of circuits = 2; fins pitch 3 fins/inch
* Data from manufacturer

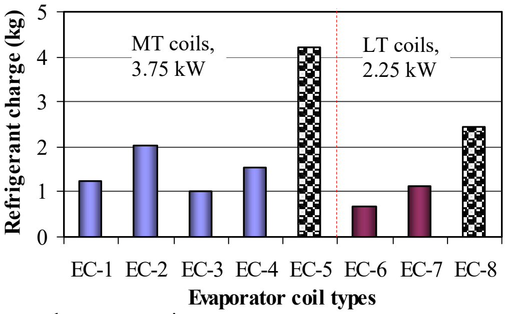
The validated models were used to design 8 evaporator coils with different geometry and circuitry. The evaporator coils were simulated at evaporating temperature of -8ºC for MT coils and -30ºC for LT coils. Tubes and fins were made from copper and aluminum respectively. Equilateral fin and tube pattern in a staggered arrangement was applied. The temperature of chilled food display cabinet was in the range of -1 to 1ºC and frozen food display cabinet was in the range of -19 to -21ºC.

Table II shows the geometry of the evaporator coils together with their performance parameters. It can be seen the physical sizes of the CO_2 evaporator coils are varied and are generally much smaller compared to the coils with R-404A.

GEOMETRY OF THE DESIGNED COILS WITH THEIR PERFORMANCE PARAMETERS

Parameters	MT evaporator coils					LT evaporator coils		
	DX CO_2		Flooded CO_2		DX R-404A	DX CO_2		DX R-404A
	EC-1	EC-2	EC-3	EC-4	EC-5	EC-6	EC-7	EC-8
Tube outside diameter (mm)	9.52	12.70	9.52	12.70	15.87	9.52	12.70	15.87
Number of rows high	2	2	2	2	2	2	2	2
Number of rows deep	21	17	13	10	20	12	10	12
Number of circuits	2	1	2	1	2	2	1	2
Total tube length (m)	91.1	73.8	56.4	43.4	86.6	48.7	40.6	48.7
Height (mm)	63.5	63.5	63.5	63.5	76.2	63.5	63.5	76.2
Depth (mm)	577.4	467.4	357.5	275.0	659.9	330.0	275.0	395.9
Width with fins (mm)	2170	2170	2170	2170	2170	2030	2030	2030
Refrigerant volume (L)	4.14	6.78	2.56	3.99	13.47	2.22	3.74	7.57
G_r (kg s^{-1} m^{-2})	171.0	168.7	199.8	198.1	109.0	100.4	99.0	72.2
CR	-	-	1.2	1.2	-	-	-	-
Q_e (kW)	3.75	3.75	3.75	3.76	3.76	2.25	2.25	2.25
$Q_{e,frost}$ (kW)*	3.26	3.12	3.24	2.96	3.44	2.16	2.14	2.15
Fin efficiency	0.85	0.88	0.85	0.88	0.87	0.89	0.92	0.90
h_r (kW m^{-2} °C^{-1})	2.899	3.107	3.206	3.473	0.482	2.521	2.802	0.337
h_a (kW m^{-2} °C^{-1})	0.062	0.073	0.064	0.075	0.063	0.042	0.050	0.046
ΔP_r (kPa)	65.13	30.89	42.34	19.27	50.47	33.15	16.24	31.79
$\Delta P_{a,frost}$ (kPa)	0.016	0.024	0.030	0.043	0.013	0.018	0.020	0.010

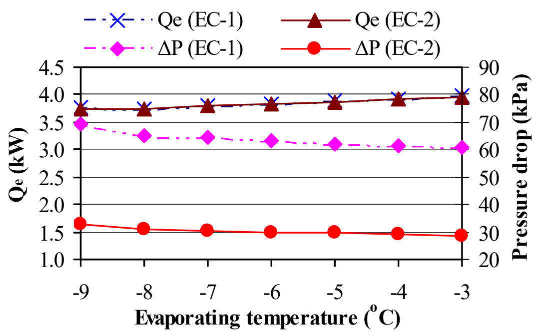
For the given refrigeration duty, the flooded MT coil with tube diameter 9.52 mm (EC-3) has the smallest size with refrigerant volume about 62% of the MT DX coil using the same tube diameter (EC-1) and about 19% of that in the R-404A evaporator coil (EC-5). The CO_2 coils also need less refrigerant charge as shown in Fig. 3, assuming 25% and 35% of the evaporator volume was filled with liquid for the DX and flooded evaporator coils respectively.



Refrigerant charge comparisons

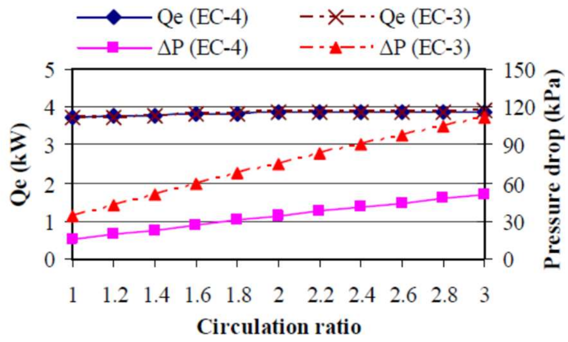
The CO_2 evaporator coils with smaller tube diameter require more tube rows and longer tubes to meet the designed refrigeration duty. Using single circuit arrangement results in high pressure drop particularly for the DX type coils. As can be seen in Table II the pressure drops of the CO_2 coils (EC-1, EC-3 and EC-6) are still higher than the coils with larger tube diameter (EC-2, EC-4 and EC-7) even in two circuit arrangement. The pressure drop will be higher if the pressure drop across the distributor and lead tubes is taken into account. Moreover, the physical size of the coils, except for in the case of the flooded coil EC-3, is larger which may increase their production cost.

Fig. 4 shows the performance variation of CO_2 MT DX coils with evaporating temperature. Increasing the evaporating temperature can slightly improve the refrigeration capacity and reduce the pressure drop. Similar effect was also found on the LT DX and flooded evaporator coils. In Fig. 5 and 6, the performance of the flooded CO_2 evaporator coils at different circulation ratios (CR) is shown. As the CR increases, the refrigeration duty slightly improves due to the enhancement of the evaporation heat transfer coefficient. However, the increase of the CR considerably increases the pressure drop and refrigerant mass velocity which increases the power consumption of the CO_2 pump and causes a reduction in the coefficient of performance of the refrigeration system. The CR, therefore, should be chosen to be as low as possible in the range of the designed refrigeration capacity. The experimental tests showed the optimum CR to be in the range 1.1 and 1.3.

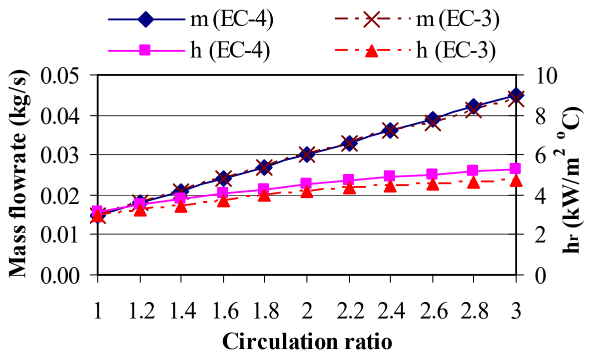


The influence of evaporating temperature

Numerical models have been developed and validated for design and performance simulation of finned tube flooded and direct expansion coils. Different geometry and circuitry arrangements were simulated using CO_2 and R-404A as refrigerants. The investigation found that for a given refrigeration capacity, CO_2 evaporator coils had considerably smaller size and lower refrigerant charge compared to the coils using R-404A refrigerant. The investigation among the CO_2 evaporator coils showed that for the refrigeration capacity examined, using the larger tube diameter, the CO_2 DX evaporator coils were found to be more compact due to smaller number of rows along the depth of the oil. The pressure drop of the coils was also found to be lower. However, the coil had more refrigerant charge compared to the coil with the smaller tube diameter. For the CO_2 flooded evaporator application, the use of smaller tube diameter was found to be more favorable in terms of coil size and refrigerant charge.



Refrigeration duty and pressure drops with circulation ratio (CR)



Refrigerant mass flow rate and heat transfer coefficient with circulation ratio

GEOMETRY OF THE DESIGNED COIL WITH ITS PERFORMANCE PARAMETERS

Parameters	R1270 coil	R404A coil
Tube outside diameter (mm)	12	12
Number of rows high	2	2
Number of rows deep	10	14
Number of circuits	2	2
Total tube length (m)	70	98
Height (mm)	70	70
Depth (mm)	350	490
Width with fins (mm)	3500	3500
Refrigerant volume (L)	7.1	7.8
Refrigerant charge (kg)+	0.640	2.438
G_r (kg s^{-1} m^{-2})	67.66	206.1
Q_e (kW)	4.2	4.2
$Q_{e,frost}$ (kW)*	3.4	3.6
Fin efficiency**	0.83	0.84
h_r (kW m^{-2} ºC^{-1})	0.738	1.318
h_a (kW m^{-2} ºC^{-1})	0.052	0.048
ΔP_r (kPa)	8.53	97.38
$\Delta P_{a,frost}$ (kPa)*	0.016	0.010

* After frost accumulation of 240 minutes
**Fin thickness: 0.2 (mm); Fin pitch: 8 (mm) for R1270 and 10 (mm) for R404A
\+ Assumption 25% of the coil volume filled with liquid refrigerant.

Table III details the geometry of the R1270 evaporator and original R404A coil together with performance parameters obtained from the EES model. Because the R1270 refrigeration system is comprised of two circuits, the coil geometry given in the table is for only one circuit. Similar assumptions were made for the R404A coil. A 7 ºC superheat was also assumed for both coils. The results show that for the same load the R1270 coil would require 70 m of copper pipe and 0.64 kg of refrigerant charge compared to 98 m pipe for the R404A coil and 2.44 kg of refrigerant charge.

IV. CONCLUSIONS

Numerical models have been developed and validated for design and performance simulation of finned tube flooded and direct expansion coils which can be applied in sustainable cold storage systems. Different geometry and circuitry arrangements were simulated using natural refrigerant CO_2 and Hydrocarbon (HC). For comparison simulation also performed for R-404A evaporator. The investigation found that for a given refrigeration capacity, CO_2 evaporator coils had considerably smaller size and lower refrigerant charge compared to the coils using R-404A refrigerant. The investigation among the CO_2 evaporator coils showed that for the refrigeration capacity examined, using the larger tube diameter, the CO_2 DX evaporator coils were found to be more compact due to smaller number of rows along the depth of the coil. The pressure drop of the coils was also found to be lower. However, the coil had more refrigerant charge compared to the coil with the smaller tube diameter. For HC evaporator coil, the results showed that, at the same refrigeration load, the HC coil would require 40% shorter copper pipe and 30% refrigerant charge compared to R404A evaporator coil.

ACKNOWLEDGMENT

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NOMENCLATURE

A	Area (m^2)		*Greek symbols*
Cp	Specific heat (kJ kg^{-1} ºC^{-1})	δ	Thickness
CR	Circulation ratio	η	Efficiency
d	Diameter (m)	θ	Dry angle
EC	Evaporator coil	λ	Conductivity (kW m^{-1} ºC^{-1})
DX	Direct expansion	ρ	Density (kg m^{-3})
G	Mass velocity (kg s^{-1} m^{-2})	ω	Humidity ratio (kg kg$_{da}^{-1}$)
H	Enthalpy (kJ kg^{-1})	*Subscript*	
j	Colburn j-factor	a	Air; air side
h	Heat transfer coefficient (kW m^{-2} ºC^{-1})	c	convective
LT	Low temperature	cb	Convective boiling
MT	Medium temperature	e	Evaporator
m	Mass (kg); mass flow rate (kg s^{-1})	f	Fin
N	Number of rows	i	In; width axis
P	Pressure (kPa)	j	Depth axis
Pr	Prandtl number	k	Height axis
Q	Refrigeration load (kW)	lat	Latent
Re	Reynold number	nb	Nucleate boiling
RH	Relative humidity	o	Out
S	Suppression factor (m)	r	Refrigerant
T	Temperature (ºC)	sp	Single phase
t	Time (s)	tp	Two phase
U	Overall heat transfer coefficient (kW m^{-2} ºC^{-1})	v	Vapor

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