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Review on performance and working of cold room components

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***Abstract—* A cold room is a refrigerated enclosure intended for the storage of chilled and/or frozen foodstuff or other perishable items. Cold room is the storage facility where large quantities of products could be stored for longer duration at lower temperature. Cold chain is now recognized as a sunrise sector in India. The current scenario reveals that there is a tremendous scope for the development of cold chain facilities and Cold Rooms form the heart of the cold chain. The cold room like every other refrigerating systems of the same magnitude employs the vapor compression refrigeration cycle. Vapor compression refrigeration is widely used for air conditioning of buildings and automobiles. It is also used in domestic and commercial refrigerators, large-scale warehouses for chilled or frozen storage of foods and meats, refrigerated trucks and railroad cars, and a host of other commercial and industrial services**.

***Key words*- cold room, vapor compression refrigeration**

I. INTRODUCTION

A cold room is refrigeration room whose walls, ceiling and floor is assembled in any size from various sections of modular insulated panels. These modules all fit together and lock into place providing a thermally insulated and structurally sound room. Cold room is the storage facility where large quantities of products could be stored for longer duration at lower temperature.

The vapour-compression refrigeration cycle is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps. Figures represent the T-S & P-V diagram for simple vapour compression refrigeration system. It consists of four processes:

1-2 Isentropic compression in a compressor
2-3 Constant-pressure heat rejection in a condenser
3-4 Throttling in an expansion device
4-1 Constant-pressure heat absorption in an evaporator

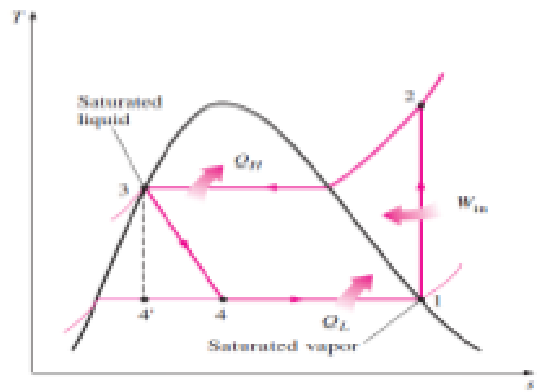

Fig. 1 T-S diagram for vapour compression refrigeration system

In an ideal vapour-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapour and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during this isentropic compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as superheated vapour at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings.

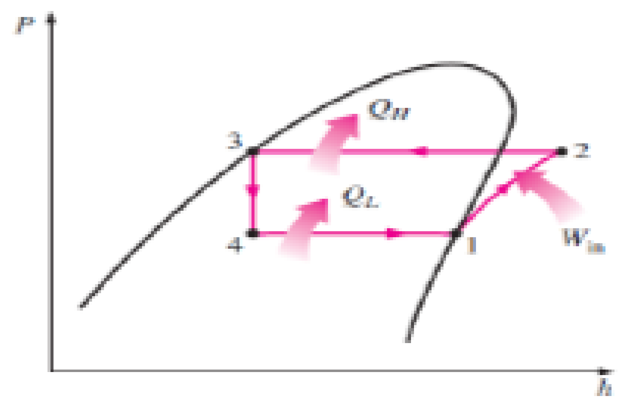

Fig. 2 P-H diagram for vapour compression refrigeration system

The saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drop below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture, and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapour and re-enters the compressor, completing the cycle.

II. LITERATURE SURVEY

Ugwu et al. [1] 2012 designed cold storage room for Umudike to provide a better storage facility. It is an adaptive design aimed at designing the cold room to suit the prevailing factors of Umudike community with reference to some design calculations. The cold room has an estimated total refrigeration capacity of 0.82TR (about 4Hp), and a maximum COP of 6.09. Its operating ambient temperature is 36 $^{\circ}$C with a rated evaporator capacity of 1.85HP and a rated condenser capacity of 2.15HP, respectively.

The author had calculated the total heat load which consists of the amount of heat to be removed from a cabinet during a certain period. It is dependent on two main factors: heat leakage or heat transfer load, and heat usage or service load, respectively. Thus, the following types of heat loads were considered in the design of this cold room Heat Leakage Load, Heat Usage Load, Air Change Heat Load, product heat load, miscellaneous heat load and occupancy load.

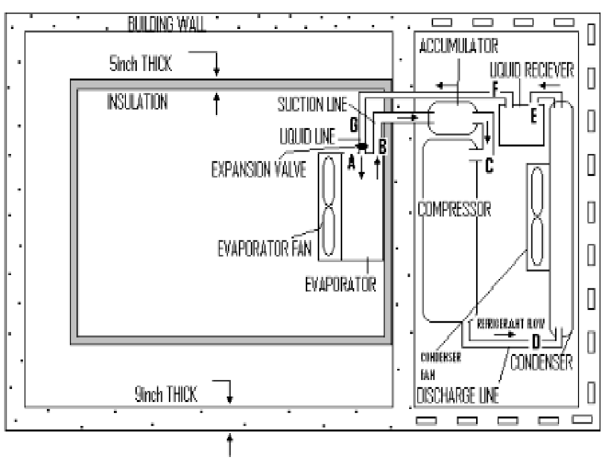

Fig.3 Schematic diagram of the cold room, indicating the component parts and the direction of refrigerant flow in them[1]

The author had designed the compressor by calculating mass flow rate, compressor capacity and COP to find the power required by the compressor.

The author had also calculated the capacity, surface area and tube diameter required by the condenser and evaporator. The author shows a comparative analysis of their design with traditional 4 HP capacity cold room.

A M Foster et al. [2] 2003 had discussed the measurements of infiltration through different size entrances of a cold store at two different cold store temperatures were taken and compared against established analytical models and computational fluid dynamics (CFD) models. The analytical and CFD models generally tended to over predict the infiltration. The analytical model developed by Gosney et al. provides the closest comparison with the various experiments. The CFD models were more accurate than the fundamental analytical models but less accurate than those based on a semi-empirical approach. For the experimental configurations examined, CFD offered no real advantage over these empirical analytical models.

A predictive model of the experimental test room was created using CFX 5.4 (CFDS, AEA Technology), a commercially available CFD code. A predictive model of the test room and its entrance was created.

To simplify the model, a number of assumptions were made;

(i) There was no heat flow through the walls of the test room.

(ii) The test room had no thermal mass.

(iii) Humidity had no effect on the flow rate through the door (it will, however, have an effect on heat transfer through the door). How the door was opened did not affect the air-flow through it.

(v) The simplification of outside room conditions had no effect.

(vi)The room was leak proof i.e. air could only move through the entrance.

The author had created a tetrahedral mesh and then refined until a converged solution was obtained. Turbulence was modeled using the k–ε (k–epsilon) model, this is the industry standard two-equation turbulence model. The final mesh size varied depending on the size of entrance but varied between 95 000 and 250 000 tetrahedral elements and 18 000 and 46 000 nodes.

Predictions were obtained for the cases that were measured in the proceeding sections; three different sizes of entrance and two different initial room temperatures. Assumption v was checked by extending the domain of the model from 3 to 6 m outside of the walls of the cold room. Due to the extra memory and computing time required this was only carried out at one condition (the 2.3 m wide entrance). This is referred to as the large boundary model. In reality the cold room was not contained within exterior walls 3 or 6 m beyond the cold room walls, the geometry was much more complex and difficult to model. Extending the domain showed what effect this simplistic assumption had on the predictions.

Assumption vi was checked by measuring the background leakage rate from the refrigerated test room with the door closed.

M.L. Hoang et al. [3] 2000 investigated airflow inside a cold stored using computational fluid dynamics. The airflow model is based on the steady state incompressible, Reynolds-averaged Navier Stokes equations. The turbulence is taken into account using a k-ε model. The standard as well as the Renormalisation-Group (RNG) version of the k-ε model is investigated. The forced-circulation air cooler unit is modelled with an appropriate body force and resistance, corresponding to the characteristics of the fan and the tube-bank evaporator. The finite volume method of discretisation is used. The validation of the model has been performed by a comparison of the calculated time-averaged velocity magnitudes with the mean velocities measured by means of a hot film type omni-directional velocity sensor. A relative error on the calculated air velocities of 26% was observed. The RNG k-ε model does not help to improve the prediction of the recirculation. Both a finer grid and enhanced turbulence models are needed to improve the predictions.

For the empty room, the model solution reached equilibrium for an air flow of 4800 m^3/h, which is 5% lower than the value derived from the design pressure drop of the heat exchanger unit (78 Pa) and the head capacity curve of the propeller fan, rotating at 1380 rev/min. When the room is loaded with four palloxes, the calculated air flow rate was 5.5% lower than the derived value. This indicates that the resistance caused by the palloxes is very small compared to the resistance of the cooler, for this configuration.

A.S. Dalkilic et al. [4] 2013 focussed on cold-room systems in Turkey regions. They were also compared with each other in terms of the alteration of alternative refrigerants, insulation thickness, energy and exergy analyses of the cold rooms' refrigeration system. The coefficients of performance (COP), refrigerant charge rates, volumetric refrigeration capacity and capacities of each component of the refrigeration system for the refrigerants CFC-12, HCFC22 and their alternatives, such as HFC-134A, HFC-404A, HFC-407C, HFC-410A and HFC-507, were determined by considering the effects of the main parameters of the performance analysis, such as refrigerant type, degree of sub-cooling, superheating and compressor efficiency. The first law efficiencies, exergy efficiencies and irreversibility rates were also obtained and discussed in this theoretical study.

The ideal refrigeration cycle was considered for the working substances that changed phases during the cycle. Actual refrigeration cycle systems have some deviations from the ideal one due to actual pressure losses due to fluid flow and heat transfer exchange in the surroundings. Cycle performance determination is performed to ease theoretical calculations by means of several assumptions: steady state operating conditions, negligible alterations of kinetic and potential, an adiabatic (well-insulated) heat exchanger, saturated liquid at the outlet of the condenser, saturated vapour at the inlet of the compressor, neglect of pressure drops and heat loss in the environment from evaporators and condensers isenthalpic flow across the expansion valves and the assumed isentropic efficiency for the compressor.

An ideal vapour compression refrigeration system was used for the performance analysis of alternative new refrigerants as substitutes for CFC-12, HFC134A and CFC-22. Considering the comparison of COP and refrigerant charge rates of the tested refrigerants, refrigerants of R407C andR410A were found to be the most suitable alternatives among refrigerants tested for R12 and R22, respectively. The COP of the system decreased with increasing condensing temperature for a constant evaporating temperature in the analysis. All systems were improved by analyzing the effect of the superheating/subcooling case. Further, the effect of compressor efficiency on COP was also discussed in various case studies.

There is not any refrigerant that meets all of the necessary requirements due to the changes in the thermodynamic physical properties of the refrigerant. The thermo-physical properties, such as performance and efficiency, limitations and restrictions related to safety, environmental impact and associated legislation, are the most significant factors in choosing a new refrigerant. Low viscosities of liquid and vapour phases, high liquid specific heat, high thermal conductivities of liquid and vapour phases, high latent heat, and limited temperature glide are the desired thermo-physical properties of refrigerant/refrigerant mixtures.

A.S. Dalkilic et al. [5] 2010 shows theoretical performance study on a traditional vapour-compression refrigeration system with refrigerant mixtures based on HFC134a, HFC152a, HFC32, HC290, HC1270, HC600, and HC600a was done for various ratios and their results are compared with CFC12, CFC22, and HFC134a as possible alternative replacements. In spite of the HC refrigerants' highly flammable characteristics, they are used in many applications, with attention being paid to the safety of the leakage from the system, as other refrigerants in recent years are not related with any effect on the depletion of the ozone layer and increase in global warming. Theoretical results showed that all of the alternative refrigerants investigated in the analysis have a slightly lower performance coefficient (COP) than CFC12, CFC22, and HFC134a for the condensation temperature of 50°C and evaporating temperatures ranging between -30 °C and 10 °C. Refrigerant blends of HC290/HC600a (40/60 by wt %) instead of CFC12 and HC290/HC1270 (20/80 by wt %) instead of CFC22 are found to be replacement refrigerants among other alternatives in this paper as a result of the analysis. The effects of the main parameters of performance analysis such as refrigerant type, degree of subcooling, and superheating on the refrigerating effect, coefficient of performance and volumetric refrigeration capacity are also investigated for various evaporating temperatures.

E. Elgendy [6] 2013 describe use of an ejector as an expansion device as one of the alternative ways. Aims to evaluate the performance improvement of a vapour compression refrigeration cycle under a wide range of operating conditions. A numerical model is developed and a parametric study of important parameters such as condensation (30 to 50°C) and evaporation temperatures (-20 to 5°C), nozzle and diffuser efficiencies (0.75-0.95), subcooling and superheating degrees (0-15°C) are investigated. The model verification gives a good agreement with the literature data. The simulation results revealed that condensation temperature has the highest effect (129%) on the performance improvement ratio while superheating has the lowest one (6.2%). Among ejector efficiencies, the diffuser efficiency has a significant effect on the COP of ejector expansion refrigeration cycle. The COP improvement percentage decreases from 10.9% to 4.6% as subcooling degrees increases by 15K. Fig.2.2 shows a schematic diagram of ejector expansion refrigeration cycle (EERC) and the corresponding p-h diagram. The primary flow from the condenser (state 1) and the secondary flow from the evaporator (state 2) are expanding through primary and secondary nozzles, respectively (1-1b and 2-2b) to mixing chamber pressure, mixing at constant pressure (3m). The mixed flow is discharged through the diffuser (3m-3) of the ejector and then separated in forms of vapor (state 4) and liquid (state 6) so that this ratio matched with the inlet ratio of primary and secondary flows. Then the liquid circulates through the expansion valve (6-7) and then evaporates in the evaporator (7-2), whereas the vapour circulated through the compressor (4-5) and then condensate in the condenser (5-1).

In this way, the compressor inlet pressure in this system is relatively higher than that in a basic cycle and hence less work is used to operate the compressor in the EERC. For constant pressure mixing ejector, the primary nozzle exit located within the suction nozzle in front of the constant-area section and the static pressure is assumed to be constant through the mixing process.

The ejector expansion vapor compression refrigeration cycle has been modeled based on the mass, momentum, and energy conservations. To simplify the theoretical model and set up the equations per unit mass flow rate at the ejector exit, the following assumptions have been made.

- Neglect the pressure drop in the condenser, evaporator, separator, and the connection tubes.
- No heat transfer with the environment for the system except in the condenser.
- Both the motive stream and the suction stream reach the same pressure at the inlet of the constant pressure mixing section of the ejector.
- Kinetic energy of the refrigerant at the ejector inlet and outlet are negligible.

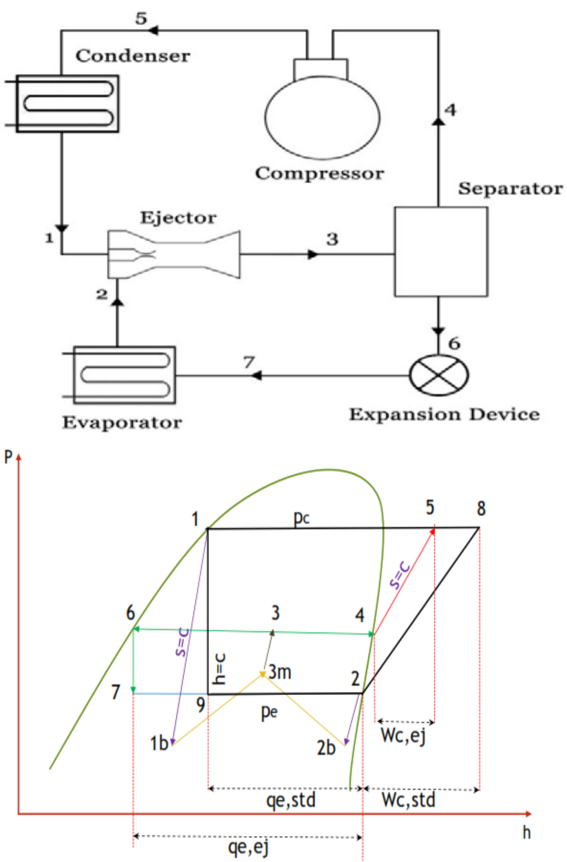

Fig.4 Schematic diagram of ejector expansion refrigeration cycle and p-h diagram[6]

Suhas D. Kshirsagar et al. [7] 2013 proposed combined vapour compression-ejector refrigeration system which uses the waste heat of condenser of simple vapour compression system and this heat is utilized to drive the binary ejector refrigeration system. The cooling effect produced by this binary system can be considered as input to the cooling effect of basic vapour compression system. Thermal design of this combined vapour compression ejector refrigeration system (VCR-VER) is based on energy and mass conservation in each component.

The system performance is first analysed for the on design conditions. The results show that the COP is improved by 3.086% for the proposed system. The system is then analyzed for variation of four important variables. Matlab Simulink software is used to model the combined VCR-VER system. The system analysis shows that this refrigeration system can effectively improve the COP by the ejector cycle with the refrigerant which has high compressor discharge temperature.

The refrigeration system is a combination of a basic compression refrigeration cycle and the ejector cycle. The governing equations are based on energy and mass balances for each component in the two cycles.

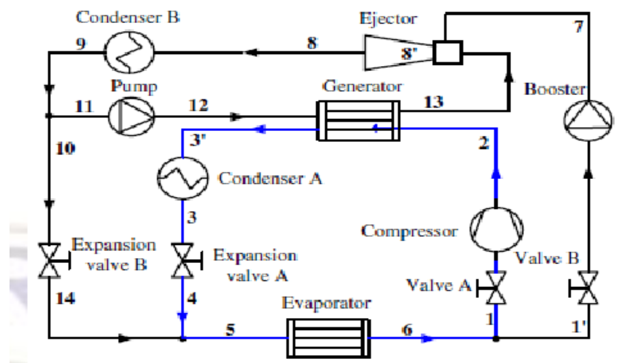

Fig.5 combined vapour compression-ejector refrigeration system[7]

A combined VCR-VER refrigeration system was developed by combining a basic vapour compression refrigeration cycle with an ejector cooling cycle in parallel. The governing equations of each component were derived based on the energy and mass conservation laws. The system performance was analysed as a function of five important variables. The results show that the combined VCR-VER system is superior to the basic refrigeration system over a wide range of operating conditions. The main results can be summarized as follows.

1. The combined VCR-VER refrigeration system with the parallel ejector cycle significantly improves the COP when the compressor discharge temperature is larger than 100 °C. Simulations give an average COP increase for the combined VCRVER system with R600a is 3.086%.

2. As with the basic vapor compression refrigeration system, the COP of the combined VCR-VER system increases with the evaporating temperature and decreases with the condensing temperature.

3. The ejector is the key component of the combined VCR-VER refrigeration system. The ejector geometries and operating conditions greatly influence the ejector performance and the whole refrigeration system. A reduced primary flow inlet pressure or increased area ratio and secondary flow inlet pressure increases the COP of both the ejector cycle and the combined VCR-VER cycle. However, the two variables of primary flow inlet pressure and area ratio are contradictory since a high critical pressure ratio is required for a great area ratio to make the ejector operate in the critical mode. In practice, the pressure ratio and the ejector area ratio need to be

carefully designed to optimize the combined VCR-VER system for the best COP.

Amit Prakash [8] 2013 describe have to improve coefficient of performance of VCR system. To improve the coefficient of performance, it is to require that compressor work should decrease and refrigerating effect should increase.

Modifications in condenser are meant to increase degree of sub-cooling of refrigerant which increased refrigerating effect or more cooling water is required in condenser. The purpose of a compressor in vapour compression system is to elevate the pressure of the refrigerant, but refrigerant leaves the compressor with comparatively high velocity which may cause splashing of liquid refrigerant in the condenser tube, liquid hump and damage to condenser by erosion. It is needed to convert this kinetic energy to pressure energy by using diffuser. By using diffuser power consumption is less for same refrigerating effect so performance is improved.

In the present cycle, the vapour refrigerant leaves the compressor with high velocity. This high velocity refrigerant directly impinges on the tube of condenser which may damage to it by vibration and erosion. It results undesirable splashing of refrigerant in the condenser tube. It also results a phenomenon called as “liquid hump”. Liquid hump refer to a rise in the level of the condensed refrigerant liquid in the central portion of the condenser as compared to the level at the ends of the condenser. It reduces the heat transfer surface area so reduce condenser efficiency. Thus reducing the velocity of refrigerant a diffuser is attached after compressor.

Diffuser is a device which converts the velocity into pressure energy. It smoothly decelerates the incoming refrigerant achieving minimum stagnation pressure losses and maximizes static pressure recovery. Due to pressure recovery, at same refrigerating effect compressor to do less work. Hence, power consumption of the compressor will be reduced which results improvement in system efficiency. Superheated vapour is passed through condenser where condensation takes place. More amount of water is flow in condenser so refrigerant vapour is cooled below the condensing temperature at constant pressure thus sub-cooling is achieved. By sub cooling enthalpy of vaporization of refrigerant is increases thus more amount of heat is absorbed by refrigerant in evaporator for evaporation takes place, so refrigerating effect is increases thus performance of cycle is increased. After sub-cooling liquid refrigerant is passed through expansion valve where expansion takes place and passes through evaporator where absorb the latent heat from storage space and evaporate. Thus cooling is achieved in storage space.

Amit M Patel, Prof. R I Patel [9] 2012 optimized different parameter of cold storage for energy conservation. Energy conservation is required in the cold storage system so The Design of Experiment is used to Optimization of different parameters of cold storage on the bases of performance experiments. In This Experiment, three levels of Thickness, Area of wall and Compressor are kept as the control parameters, The Insulating wall material was taken as polyurethane foam (PUF) and different energy were taken as a result in the experiment.

The analysis is being done with the help of Minitab-15 software. The analysis of variance ANOVA is also performed as identified the statistical significance of parameters. The result of the experiments are the optimum value of insulating thickness, energy consumption rate with the help of ANOVA, After the using Taguchi method determine the feasibility of improving cooling capacity of cold storage, establish the mathematical models relating the cold storage performance parameters & control parameters by regression analysis and obtained set of optimal cold storage parameters for better performance.

S. van Mourik et al [10] 2009 presented a temperature model of a bulk storage room and derived open loop control law. The sensitivity of the system performance to parameter variations was evaluated numerically for the nominal system. The performance analysis showed the following trade offs. The temperature difference and energy costs are decreased by a more powerful fan. A high capacity fan uses more energy when switched on, but it is still more economical since it is switched on for a much shorter time. Such a fan will be more expensive to purchase, but on the other hand it is a onetime investment with a continuous benefit.

A lower wall conductivity considerably decreases the energy costs. It does not seem to influence the temperature uniformity much. A good wall insulation is, like a more powerful fan, a onetime investment that continually saves costs. A higher cooling element temperature is beneficial for both energy costs and uniformity. The energy costs seem to have a local minimum at 273 K. This is caused by a trade off between fan energy and the energy used to cool the element. A warmer cooling element requires less energy, but consequently the fan has to be on for a longer time. The bulk height has no large influence on either cost. The bulk volume is kept constant, so the amount of heat that is produced remains the same. A higher bulk gives more flow resistance and therefore the energy costs rise a little. The tradeoff is that for a fixed bulk volume, a lower bulk means a larger floor area, which is usually more expensive than a higher roof. One should keep in mind that the conclusions that are drawn here are only valid inside the experimental parameter range.

Michael Bergera et al. [11] 2012 installed small linear Fresnel collector for a pilot system of a solar driven cold room at the office building of the company Kramer in Umkirch near Freiburg / Germany. The cold room, as well as the balance of plant have been installed during June 2012, and commissioned could take place on July 5th, 2012. The cold room with roughly 100 m³ of cooled volume has a dimension of roughly 8m x 4m x 3m (LxWxH), its walls consist of 120mm thick polyurethane sandwich elements, and the steel structure is completely outside. The cold room is being cooled by one 12 kW water ammonia chiller, driven by a Fresnel collector from Industrial Solar, consisting of four modules with a total aperture area of 88 m² at an operation temperature of 180 °C. The produced cold can be delivered to the cold room or the ice storage alternatively by use of a water glycol mix as heat carrier.

The system, which is small for such cold rooms, will serve mainly for demonstration of the sustainable technology and help in leading to succeeding projects in the sunny target region. By developing an integrated as well as modular solution for solar thermal driven cold storages, the system shall serve as an example and help to decrease the effort of installation as well as engineering of such systems.

Jing Xie et al. [12] 2005 proposed a two-dimensional mathematical model for the inside a mini type constructional cold store [4.5 m(l)x 3.3m(w)x2.5m(h)] and a related computer program with SIMPLE algorithm and crisscross girding technique were developed. The simulation results reflected the characteristics of airflow and temperature distribution. After that, several design parameters (corner baffle, the stack mode of foodstuffs, etc.) which would affect the flow field in the cold store were analyzed. The results of calculation indicated that all these design parameters especially the stack mode of foodstuffs influenced the flow and temperature fields inside the cold store greatly. It was demonstrated that CFD was a powerful tool for designing and optimizing the flow field in cold store.

III. CONCLUSIONS

- A cold storage room for Umudike community and her environs has been designed. The cold room has an estimated total refrigeration capacity of 0.82TR (about 4Hp), and a maximum COP of 6.09. Its operating ambient temperature is 36°C with a rated evaporator capacity of 1.85Hp and a rated condenser capacity of 2.15Hp, respectively.[1]
- The ability to model the lag time and the drop of region would have put the CFD predictions at an advantage for very short and very long door opening times respectively. CFD allows factors such as the effect on the infiltration to temperatures inside and outside of the room to be investigated. It also allows the effect of air curtains on the entrances to be investigated.[2]
- By using CFD with an average difference of 26% between calculated and measured velocities, it becomes only possible to have a qualitative insight into the airflow pattern for different product stacks, different fan rotation speeds and different room designs, without the need for additional experimental determination of boundary conditions for the velocity.[3]
- Based on the results of the energy analyses, all the alternative refrigerants HFC-134A, HFC-404A, HFC-407C, HFC-410A and HFC-507 have a slightly lower COP and require lower refrigerant charge rates than CFC-12 and HCFC-22 for condensation temperatures ranging from 36-47 °C.
    - Based on the results of the exergy analyses, the compressor has the greatest irreversibility, followed by the expansion valve, condenser and evaporator as the components of the refrigeration system.[4]
- Considering the comparison of co-efficients performance (COP) and pressure ratios of the tested refrigerants and also the main environmental impacts of ozone layer depletion and global warming, refrigerant blends of HC290/HC600a (40/60 by wt.%) and HC290/HC1270 (20/80 by wt.%) are found to be the most suitable alternatives among refrigerants tested for R12 and R22 respectively.
- The refrigeration efficiency, the performance coefficient (COP) of the system, increases with increasing evaporating temperature for a constant condensing temperature in the analysis. [5]
- Condensation temperature has the highest effect of the performance improvement ratio. As condensation temperature increases from 30 to 50°C, the performance improvement ratio is doubled.
- The rate of increase in COP of VCRC (14%) is much higher than those of COP of EERC (9%) as subcooling degrees increases with 15K. Superheating at the evaporator exit has the lowest effect of the COPs of both VCRC and EERC cycles and improvement percentages.[6]
- The combined VCR-VER refrigeration system with the parallel ejector cycle significantly improves the COP when the compressor discharge temperature is larger than 100 °C. Simulations give an average COP increase for the combined VCR-VER system with R600a is 3.086%. As with the basic vapour compression refrigeration system, the COP of the combined VCR-VER system increases with the evaporating temperature and decreases with the condensing temperature.[7]
- COP of Vapour Compression Cycle is increased by lowering the power consumption /work input or increasing the refrigerating effect. By using sub-cooling and using diffuser at condenser inlet refrigerating effect increases and power consumption or work input decreases. Thus performance of cycle is improved. High velocity refrigerant has various serious affect on vapour compression refrigeration system such as liquid hump, undesirable splashing of the liquid refrigerant in the condenser and damage to the condenser tubes by vibration, pitting and erosion. [8]
- Author concluded that orthogonal array and Grey relation design both gave same result as best optimum value of Thickness 0.100 Mt, area of wall 0.96 Mt^2 and compressor 0.167 HP and also study of Regression analysis to develop Mathematical model for calculating direct optimum value of all parameters.[9]
- The main results were that the energy costs as well as the bulk temperature uniformity are substantially improved by a larger fan capacity. Also, a lower wall conductivity considerably decreases the energy costs. The interesting trade off is that both good wall insulation and more fan capacity are one time investments that give a continuous benefit. Another interesting tradeoff is the local minimum in energy costs for the cooling element temperature, caused by decreasing costs for a less cold cooling element versus the increasing costs for the extra ventilation that is required when the air is cooled down less. The bulk shape only had a small influence, with the trade off that a lower bulk costs a little less energy due to reduced airflow resistance. [10]

- The technology demonstrated leads the way towards an environmentally friendly and carbon neutral operation of cold rooms, which is an important step especially for agriculture and food industry in regions of sunny, hot climate. [11]
- The airflow structure of cold store was simulated by using CFD and results were agreed with the experiments. The approximations in the model formulation made the model simpler, but limited the quantitative accuracy of the model. However, the model developed in this paper was rather practicable, and can be used to predict the flow pattern in a cold store.[12]

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