ENERGY - EXERGY BASED PERFORMANCE ANALYSIS OF A SOLID ADSORPTION SOLAR REFRIGERATOR

BY

NDEKE CHIBUIKE (B.ENG.)

REG. NO. 20094698578

A THESIS SUBMITTED TO THE SCHOOL OF POST GRADUATE STUDIES

FEDERAL UNIVERSITY OF TECHNOLOGY, OWERRI - NIGERIA

IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF THE DEGREE: MASTER OF ENGINEERING, M.ENG. IN MECHANICAL ENGINEERING (ENERGY AND POWER)

MAY, 2013

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CERTIFICATION

This is to certify that **Ndeke, Chibuike, 20094698578;** of the Department of Mechanical Engineering, Federal University of Technology, Owerri, carried out this work as an original research work and that this work satisfies in part, the requirement for the award of the degree of Masters of Engineering (M.ENG.) in Mechanical Engineering (Energy & Power), Federal University of Technology, Owerri – Imo State, Nigeria.

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DEDICATION

This work is dedicated to GOD ALMIGHTY, YAHWEH. LORD of all. The FATHER of Spirits. He owns my life and gave me all I have. It is by His special grace that I embarked on this study and finished it. He is my all.

Special thanks to Elder & Deaconess Innocent Ugwonali Ndeke and my entire family members, this work is partly dedicated to you and to numerous others, who gave their prayers, encouragements, support and more.

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My fiancée Miss Uko Chioma Queen and her family were supportive all through the period of this thesis. The Ononogbo's family is deeply appreciated for their hospitality and care.

All glory to GOD. THE I AM, THAT I AM.

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SYMBOLS, SUBSCRIPTS AND ABBREVIATIONS

Symbols

c_p	Specific Heat Capacity (KJ/kgK)
D	Constant in D-A Equation
E_x	Exergy (Heat Exergy) (KJ)
H	Enthalpy (KJ/kg)
h_{sg}	Isosteric Heat of Sorption (KJ/kg)
L	Latent Heat of Vaporization (KJ/kg)
M	Mass (kg)
P	Pressure (bar)
Q	Heat Generated (W/m^3)
S	Entropy (KJ/kgK)
T	Temperature (K)
U	Internal Energy (KJ/kg)
W_o	Maximum Capacity of Sorbent (L/kg)
W_{max}	Maximum Work Potential
X	Concentration (kg/kg)
Δ	Change in
ε	Mass Exergy
τ	Exergetic Temperature Factor
ψ	Psi (Exergy Efficiency)
Π	Pi (Exergy Loss or Destroyed)
Σ	Sigma (Summation)

Subscripts

A	Adsorbent, Adsorption, Adsorbed
C	Condenser, Condensed
D	Desorption, Desorbed
E	Evaporator

F	Final
G	Generated/Generated
I	Initial
L	Cooled chamber
0	Environment
R	Refrigerant/Refrigerator

Abbreviations

CAAC	Compact Adsorption Air Conditioner
CCHP	Combined Cooling Heat Power
COP	Coefficient of Performance
COP_{ct}	Carnot Coefficient of Performance
COP_N	Net Solar Coefficient of Performance
FLT	First Law of Thermodynamics
P-T-X	Pressure Temperature Concentration
SCP	Specific Charge Power
SI	International System of Units
SJTU	Shanghai Jiao Tong University
SLT	Second Law of Thermodynamics
UN	United Nations

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ABSTRACT

The Energy - Exergy based performance analysis of the Solid Adsorption Solar Refrigerator is presented. The system comprises the collector/generator/adsorber, the condenser and the evaporator. It utilizes activated carbon/methanol as its adsorbent/adsorbate working combination. The exergy performance analysis of this refrigerator involves detailed assessment of all the following sub-processes that make up the refrigeration cycle, which are isosteric heat-up phase, desorption phase, isosteric cool-down phase and re-adsorption phase. The exergy performance of the Condenser and the Evaporator in a quasi-static regime of operation were also reported. Exergy destruction equations were generated for each sub-process and utilized in estimating the amount of exergy destroyed with reference to varying ambient environment temperatures recorded at 30 mins interval and at constant (average) temperature of 301 K for all test days. The ranges of exergy destruction during each sub-process, for varying reference temperature analysis for test days I, II and III were determined as 0.000 – 3.362 kJ, 0.000 – 2.720 kJ and 0.000 – 5.516 kJ for the isosteric heat-up phase; 302.059 – 2874.119 kJ, 144.778 – 3249.376 kJ and 574.044 – 2999.723 kJ for the desorption phase; 0.247 – 8.357 kJ, 0.350 – 8.737 kJ and 0.007 – 5.933 kJ for the isosteric cool-down phase; 0.000 – 32.183 kJ, 0.000 – 281.807 kJ and 0.000 – 84.715 kJ for the re-adsorption phase; 1.746 – 21.485 kJ, 0.790 – 16.706 kJ and 2.809 – 17.275 kJ in the condenser and in the evaporator as 6.868 – 41.350 kJ, 1.344 63.580 kJ and 1.515 – 54.507 kJ. The ranges of exergy destruction during each sub-process, based on constant (average) reference temperature analysis for test days I, II and III were determined as 0.000 – 3.384 kJ, 0.000 – 2.720 kJ and 0.000 – 5.533 kJ for the isosteric heat-up phase; 292.510 – 2736.713 kJ, 141.392 – 3190.508 kJ and 565.081 – 2912.175 kJ for the desorption phase; 0.248 – 8.248 kJ, 0.352 – 8.651 kJ and 0.008 – 5.880 kJ for the isosteric cool-down phase; 55.327 – 128.443 kJ, 86.251 – 139.388 kJ and 0.000 – 124.244 kJ for the re-adsorption phase; 0.524 – 3.353 kJ, 0.347 – 8.975 kJ and 1.641 – 5.918 kJ in the Condenser and in the Evaporator as 16.234 – 64.322 kJ, 8.738 – 97.613 kJ and 10.514 – 77.084 kJ. The exergy efficiency values for the varying reference temperature analysis were obtained for test days I, II and III as 0.01331, 0.04380 and 0.01226 respectively and for constant (average) reference temperature as 0.02275, 0.05933 and 0.02162 for test days I, II and III respectively. The exergy analysis indicates losses of the highest magnitude during desorption and re-adsorption phases.

Choices of collector/generator/adsorber with lower heat emissions during desorption and re-adsorption, with more efficient cooling in the condenser, will improve the overall efficiency of the system.

KEY WORDS: ADSORPTION, DESORPTION, EXERGY, EXERGY EFFICIENCY, EXERGETIC TEMPERATURE FACTOR, EXERGY DESTRUCTION.

CHAPTER ONE

INTRODUCTION

1.1 BACKGROUND OF THE STUDY

Adsorption is the adhesion of atoms, ions, or molecules from a gas, liquid, or dissolved solid to a surface. This process creates a film of the adsorbate on the surface of the adsorbent and it differs from absorption, in which a fluid (the adsorbate) permeates or is dissolved by a liquid or solid (the absorbent), McMurry (2003). It is important to note that adsorption is a surface-based phenomenon while absorption involves the whole volume of the material. The term sorption encompasses both processes, while desorption is the reverse of adsorption. It is also a surface phenomenon.

There are two types of adsorption processes; these are physical adsorption and chemical adsorption or chemisorptions. The categorization of which is based on the binding energy of the adsorbate to the substrate, Nic et al. (2006).

In physical adsorption, Van der Waal forces are responsible for the adsorption process and species so adsorbed are characterized by low heats of adsorption, relative ease of desorption and a significant temperature dependence upon the equilibrium capacities. On the other hand, chemical adsorption results from purely chemical interaction of adsorbent and adsorbate. It is characterized by very high heat of adsorption, Anyanwu and Ogueke (2001).

Similar to surface tension, adsorption is a consequence of surface energy. In a bulk material, all the bonding requirements (be they ionic, covalent, or metallic) of the constituent atoms of the material are filled by other atoms in the material. However, atoms on the surface of the adsorbent are not wholly surrounded by other adsorbent atoms and therefore can attract adsorbate. The exact nature of the bonding depends on the details of the species involved, but the adsorption process is generally classified as physisorption (characteristic of weak Van der Waals forces) or chemisorptions (characteristic of covalent bonding). It may also occur due to electrostatic attraction, Ferrari et al. (2010).

Adsorption is present in many natural physical, biological, and chemical systems, and is widely used in industrial applications such as activated charcoal, capturing and using waste heat to provide cold water for air conditioning and other process requirements (adsorption chillers), synthetic resins, increase storage capacity of carbide-derived carbons for tunable nonporous carbon, and water purification. Adsorption, ion exchange, and chromatography are sorption processes in which certain adsorbate are selectively transferred from the fluid phase to the surface of insoluble, rigid particles suspended in a vessel or packed in a column. Lesser known, are the pharmaceutical industry applications as a means to prolong neurological exposure to specific drugs or parts thereof, Ferrari et al. (2010).

An adsorption refrigeration system use solid adsorbent beds to adsorb and desorbs a refrigerant to obtain cooling effect. These solid adsorbent beds adsorb and desorb the refrigerant vapor in response to changes in the temperature of the adsorbent. The basic adsorption refrigeration system, commonly referred to as the adsorption heat pump loop, or an adsorption refrigeration circuit, usually consists of four main components: a solid adsorbent bed, a condenser, an expansion valve and an evaporator.

The solid adsorbent bed desorbs refrigerant when heated and adsorb refrigerant vapor when cooled. In this manner, the bed can be used as a thermal compressor to drive the refrigerant around the system to heat or cool a heat transfer fluid or to provide space heating or cooling. The refrigerant is desorbed from the bed as it is heated to drive the refrigerant out of the bed and the refrigerant vapor is conveyed to a condenser. In the condenser, the refrigerant vapor is cooled and condensed to liquid. The refrigerant condensate then expands to a lower pressure through an expansion valve, and the low pressure condensate passes to an evaporator where the low pressure condensate is heat exchanged with the process stream or space to be conditioned to vaporize the condensate. When further heating no longer produces desorbed refrigerant from the adsorbent bed, the bed is isolated and allowed to return to the adsorption conditions. When the adsorption conditions are established in the bed, the refrigerant vapor from the evaporator is reintroduced to the bed to complete the cycle. To obtain continuous and stable cooling effect in the operation of adsorption refrigeration system, generally two or more adsorbent beds are used in the system. In a typical two-bed cycle, one bed is heated during desorption period and the other bed is cooled during the adsorption period.

The time for the completion of a full cycle of adsorption and desorption is known as the “cycle time”. The heating and cooling steps are reversed when the beds reach the desired upper and lower temperature limits of the adsorption cycle. The upper and lower temperatures will vary depending upon the selection of the refrigerant fluid and the adsorbent. The performance indicator in cooling is called the “coefficient of performance” (COP) and is generally the ratio of the cooling effect divided by the heat input. Another performance indicator is named as “specific cooling power” (SCP) and is defined as the ratio between the cooling production and the cycle time per unit of adsorbent weight. The specific cooling power reflects the size of the system.

For a nominal cooling load, higher SCP values indicate the compactness of the system, Li (2007).

Diverse concepts of Adsorption Refrigerators were designed though with no valves, all well spelt out in the History and Development of Adsorption Refrigerators.

In physics, energy is an indirectly observed quantity. It is often understood as the ability a physical system has to do work on other physical systems. Since work is defined as force acting through a distance (a length of space), energy is always equivalent to the ability to exert pulls or pushes against the basic forces of nature, along a path of a certain length, Harper (2007).

The total energy contained in an object, is identified with its mass and energy (like mass) cannot be created or destroyed. When matter (ordinary material particles) is changed into energy (such as energy of motion, or into radiation), the mass of the system does not change through the transformation process. However, there may be mechanistic limits as to how much of the matter in an object may be changed into other types of energy and thus into work, on other systems. Energy, like mass, is a scalar physical quantity. In the International System of Units (SI), energy is measured in joules, but in many fields other units, such as kilowatt-hours and kilocalories, are customary. All of these units translate to units of work, which is always defined in terms of forces and the distances that the forces act through.

A system can transfer energy to another system by simply transferring matter to it (since matter is equivalent to energy, in accordance with its mass). However, when energy is transferred by means other than matter-transfer, the transfer produces changes in the second system, as a result of work done on it. This work manifests itself as the effect of force(s) applied through distances within the target system. For example, a system can emit energy to another by transferring (radiating) electromagnetic energy, but this creates forces upon the particles that absorb the radiation. Similarly, a system may transfer energy to another by physically impacting it, but in that case the energy of motion in an object, called kinetic energy, results in forces acting over distances (new energy) to appear in another object that is struck. Transfer of thermal energy by heat occurs by both of these mechanisms: heat can be transferred by electromagnetic radiation, or by physical contact in which direct particle-particle impacts transfer kinetic energy, Lofts et al. (2004).

Energy may be stored in systems without being present as matter, or as kinetic or electromagnetic energy. Stored energy is created whenever a particle has been moved through a field it interacts with (requiring a force to do so), but the energy to accomplish this is stored as a new position of the particles in the field - a configuration that must be "held" or fixed by a different type of force (otherwise, the new configuration would resolve itself by the field pushing or pulling the particle back toward its previous position). This type of energy "stored" by force-fields and particles that have been forced into a new physical configuration in the field by doing work on them by another system is referred to as potential energy. A simple example of potential energy is the work needed to lift an object in a gravity field, up to a support. Each of the basic forces of nature is associated with a different type of potential energy, and all types of potential energy (like all other types of energy) appear as system mass, whenever present. For example, a compressed spring will be slightly more massive than before it was compressed. Likewise, whenever energy is transferred between systems by any mechanism, an associated mass is transferred with it.

Any form of energy may be transformed into another form. For example, all types of potential energy are converted into kinetic energy when the objects are given freedom to move to different position (as for example, when an object falls off a support). When energy is in a form other than thermal energy, it may be transformed with good or even perfect efficiency, to any other type of energy, including electricity or production of new particles of matter. With thermal energy, however, there are often limits to the efficiency of the conversion to other forms of energy, as described by the second law of thermodynamics.

In all such energy transformation processes, the total energy remains the same, and a transfer of energy from one system to another, results in a loss to compensate for any gain. This principle, the conservation of energy, was first postulated in the early 19th century, and applies to any isolated system. According to Noether's theorem, the conservation of energy is a consequence of the fact that the laws of physics do not change over time, Lofts et al. (2004).

Internal energy is the sum of all microscopic forms of energy of a system. It is the energy needed to create the system. It is related to the potential energy, e.g., molecular structure, crystal structure, and other geometric aspects, as well as the motion of the particles, in form of kinetic energy. Thermodynamics is chiefly concerned with changes in internal energy and not its absolute value, which is impossible to determine with thermodynamics alone, Klotz (2008).

In thermodynamics, the exergy of a system is the maximum useful work possible during a process that brings the system into equilibrium with a heat reservoir, Perrot (1998). When the surroundings are the reservoir, exergy is the potential of a system to cause a change as it achieves equilibrium with its environment. Exergy is the energy that is available to be used. After the system and surroundings reach equilibrium, the exergy is zero. Determining exergy was also the first goal of thermodynamics.

Energy is never destroyed during a process; it changes from one form to another (see First Law of Thermodynamics). In contrast, exergy accounts for the irreversibility of a process due to increase in entropy (see Second Law of Thermodynamics). Exergy is always destroyed when a process involves a temperature change. This destruction is proportional to the entropy increase of the system together with its surroundings. The destroyed exergy has been called anergy, Perrot (1998). For an isothermal process, exergy and energy are interchangeable terms, and there is no anergy.

Exergy analysis is performed in the field of industrial ecology to use energy more efficiently. The term was coined by Rant (1956), but the concept was developed by Gibbs (1873). Ecologists and design engineers often choose a reference state for the reservoir that may be different from the actual surroundings of the system.

Exergy is a combination property of a system and its environment because unlike energy it depends on the state of both the system and environment, Cengel and Boles (2008). The exergy of a system in equilibrium with the environment is zero. Exergy is neither a thermodynamic property of matter nor a thermodynamic potential of a system. Exergy and energy both have units of joules. The Internal Energy of a system is always measured from a fixed reference state and is therefore always a state function. Some authors define the exergy of the system to be changed when the environment changes, in which case it is not a state function. Other writers prefer a slightly alternate definition of the available energy or exergy of a system where the environment is firmly defined, as an unchangeable absolute reference state, and in this alternate definition exergy becomes a property of the state of the system alone.

The term exergy is also used, by analogy with its physical definition, in information theory related to reversible computing. Exergy is also synonymous with: availability, available energy, exergic energy, essergy (considered archaic), utilizable energy, available useful work, maximum (or minimum) work, maximum (or minimum) work content, reversible work, and ideal work, Cengel and Boles (1998).

Exergy has the characteristic that it is conserved only when all processes occurring in a system and the environment are reversible. Exergy is destroyed whenever an irreversible process occurs. When an exergy analysis is performed on a plant such as a power station, a chemical processing plant or a refrigeration facility, the thermodynamic imperfections can be quantified as exergy destructions, which represent losses in energy quality or usefulness (e.g., wasted shaft work or wasted potential for the production of shaft work). Like energy, exergy can be transferred or transported across the boundary of a system. For each type of energy transfer or transport there is a corresponding exergy transfer or transport.

Exergy analysis takes into account the different thermodynamic values of different energy forms and quantities, e.g., work and heat. The exergy transfer associated with shaft work is equal to the shaft work. The exergy transfer associated with heat transfer, however, depends on the temperature at which it occurs in relation to the temperature of the environment, Ibrahim and Marc (2007).

1.2 STATEMENT OF THE PROBLEM.

It is a known fact that all heat generating systems lose some heat generated to member components and to the environment in some degree, no matter how lagged they may appear to be designed. In some machines, the heat loss amounts to a greater proportion compared to those utilized in doing useful work, of which the refrigeration machine under review is no exception. In all such heat systems, the energy analysis gives just a theoretical efficiency far from the ideal, as just the quantity and not the quality of the energy produced are utilized in the analysis.

In this work, the energy - exergy based performance analysis of the solid adsorption machine is to be carried out, with more emphasis on exergy analysis as its analysis - which is based on expended heat generated by the system; provides an efficiency value close to the actual efficiency value for the system analyzed, because it utilizes the quantity and quality of the energy produced by the machine, which accounts for irreversibilities - that is lost energy generated by the system.

1.3 OBJECTIVE OF THE STUDY.

The major objective of this work is to provide a detailed energy – exergy based performance analysis of a Solid Adsorption Solar Refrigerator, which utilizes activated carbon/methanol, as its adsorbent/adsorbate working pair. The specific objectives of the study are outlined below

(i) Develop Exergy equations of the established sub-processes of the refrigeration namely isosteric heating, desorption, isosteric cooling and re-adsorption,

(ii) Determine the Exergy losses in the Condenser and Evaporator in a quasi-static regime of operation,

(iii) Following objectives i and ii above, performance evaluation of the components and units of the refrigeration machine are to be carried out, namely the Collector/Generator/Adsorber, the Condenser and the Evaporator.

(iv) Determine the Exergy optimal performance parameters of the refrigerator and proffer upgrade options.

1.4 JUSTIFICATION.

Exergy unlike Energy analysis, yields efficiency values which provide a true measure of how nearly the actual performance of the system analyzed is reached. Exergy analysis also identifies more clearly than energy analysis the causes and locations of thermodynamic losses. Consequently exergy analysis can assist in improving and optimizing system designs. Exergy analysis, thus becomes an effective tool for

(i.) Combining and applying the conservation of energy principles together with the second law of thermodynamics for the design and analysis of energy systems.

(ii.) Improving the efficiency and other resource use (by identifying efficiencies that always measure the approach to reality as well as the locations, types and true magnitude of wastes and losses).

(iii.) Revealing whether or not and by how much it is possible to design more efficient systems by reducing their inefficiencies in existing systems.

(iv.) And identifying whether a system contributes to achieving sustainable development or is unsustainable; Exergy analysis is thus justified.

1.5 SCOPE OF THE STUDY.

This thesis is based on a Solid Adsorption Solar Refrigerator using activated carbon/methanol adsorbent/adsorbate pair as its working combination, designed and tested in Nsukka, Nigeria; Anyanwu (1996). Its scope is limited to development of exergy equations of its refrigeration cycle, comprising isosteric heating, desorption, isosteric cooling and re-adsorption. Also the exergy equations of the Condenser and Evaporator in a quasi-static regime of operation are determined. All of these would permit the development of the exergy performance parameters of the refrigerator.

CHAPTER TWO

LITERATURE REVIEW

2.1 BRIEF HISTORY OF ADSORPTION REFRIGERATION.

The development of sorption refrigeration systems powered by solar energy emerged in the late 1970's following the pioneering work by Tchernev who studied a basic sorption cycle with the working pair zeolite-water, Tchernev (1989).

Since then a number of studies have been carried out both numerically and experimentally, but the costs of these systems still made them non-competitive for commercialization. More research has since been focused though on cost reduction and more on increasing the efficiency of the machine; around which promising results is being obtained.

Based on the result of his previous study/research, Pons and Guilleminot (1986) concluded that the solid adsorption system process could be the basis of more efficient solar powered refrigerators, and they developed a prototype with the pair activated carbon/methanol. This machine produced almost 6kg of ice per m^2 of solar panel when the insulation was about 20MJ/day; with a solar COP of 0.12. This rate of ice production remains one of the highest obtained by a solar powered icemaker, Pons et al. (1999).

Critoph et al. (1992, 2002.), designed a solar powered refrigerator, which was recommended by the U.N. for vaccine storage in poor areas. Recently, Monma and Mizota (2005) disclosed an adsorption refrigerator as shown in Fig. 2.1. Components of this refrigerator are: a refrigeration chamber, an evaporator, an adsorbent bed. The refrigeration chamber is comprised of a thermal conductive member surrounding a periphery. The evaporator is arranged in an outer side surrounding the periphery of the refrigeration chamber. The evaporator is composed of a first heat insulating layer surrounding an outer side periphery. By transferring cold generated in the evaporator into the refrigeration chamber via the thermal conductive member composing the refrigeration chamber, the cooling range by the evaporator of the refrigeration chamber can be significantly enlarged, and the initial cooling in reducing the temperature of refrigeration chamber can be carried out in a short time. A similar adsorption type cooler has been developed by Hidaka et al. (2005), for cooling objects such as drinks in containers.

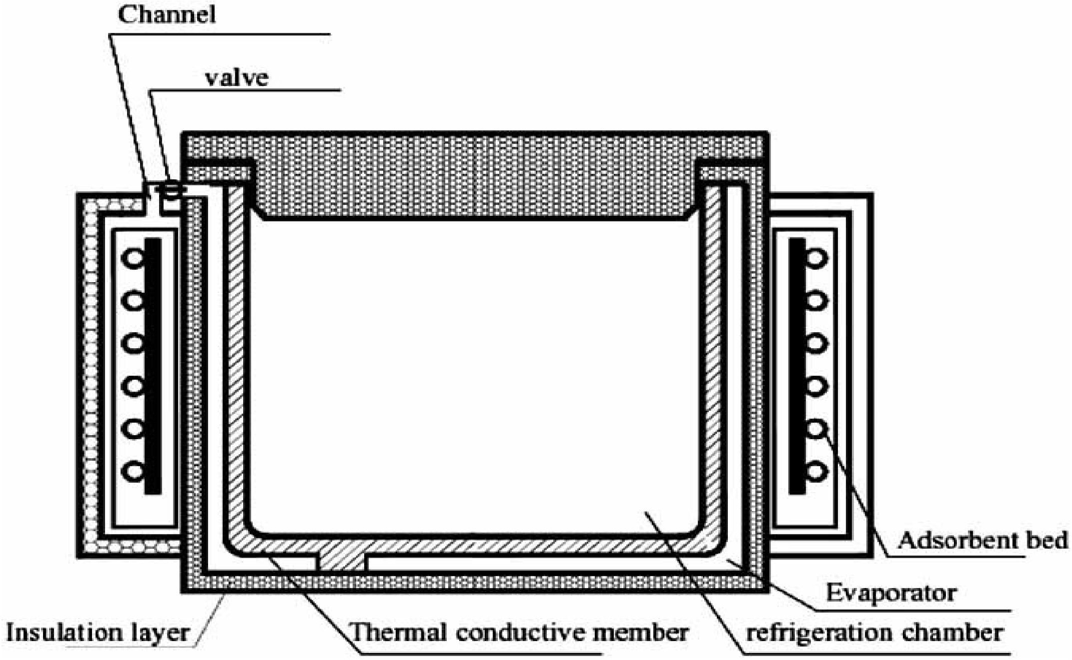


Fig. 2.1: Adsorption refrigerator invented by Monma and Mizota (2005).

Patzner (2001), disclosed a cooling method which uses at least one adsorber with zeolite as adsorbent, water as refrigerant, with an evaporator. As illustrated in Fig. 2. 2; both containers are connected via at least one line containing a blocking valve. The adsorber and the evaporator each have at least one orifice and are selectively connectable to an over-pressure generator and a condenser. The selective separation and connection of individual components or component groups for carrying out regeneration and adsorption cycles take place in each in a pressure tight or vacuum-tight manner. An electrical heater can be used in this refrigerator. The capacity of this system can be extended by employing a number of identical absorbers.

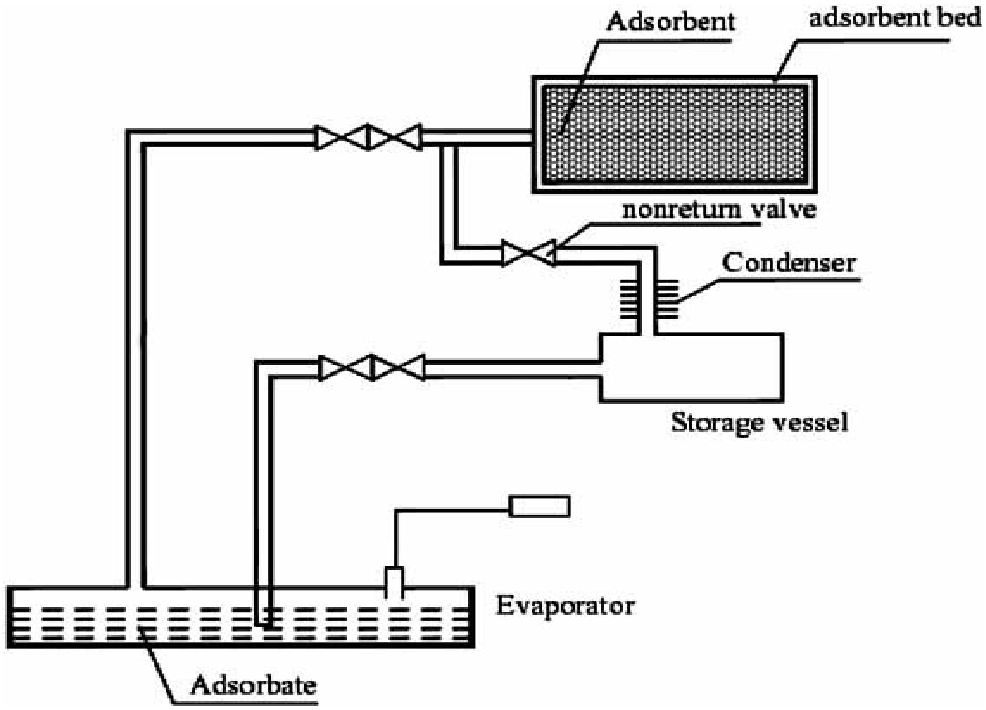


Fig. 2.2: Adsorption refrigerator invented by Patzner (2001).

Endo and Komori (2005), designed an adsorption refrigerating machine which comprises an evaporator, an adsorber, and a condenser. Water is used as the refrigerant, the adsorbent satisfying that a relative pressure is 0.14-0.2, and an adsorption swing amount is changed to be more than 0.05 kg/adsorbent (1kg) is filled. The adsorption swinging operation and the desorbing operation are performed in an area of the adsorption of 0.04 kg/adsorbent (1kg) or more. As the adsorbing/desorbing operation is performed in an area of capillary condensation, the heat of adsorption is inhibited, the heating quantity necessary for regeneration is reduced, and the freezing capacity and the energy efficiency can be improved.

The above mentioned systems all utilized the novel one bed system adsorption refrigerator to provide the needed adsorption effect; the one bed system though could only provide cooling intermittently, also it has a lower system performance in terms of COP and SCP. To solve this problem, two or more beds systems were patented and tested.

Sato et al. (1998), proposed a multi-stage multi-bed strategy, which involves cooling of the adsorber with refrigerant emanating from one or more evaporators. It may be desirable and more practical to have the evaporator devoted to cooling the chilled water, with the evaporated refrigerant being superheated at the adsorber. In their invention: Fig. 2.3; an adsorber of n-th order and an adsorber of (n+1)-th order of a number of adsorber located in an adsorption stage are thermally connected in series and adsorbent in the n-th adsorption device of a former stage is cooled by adsorption of refrigerant with the adsorbent of the (n+1)-th adsorption device of a later stage. That is, the adsorbent in a left side first stage for adsorbing vapor of refrigerant such as water flowing out of an evaporator is highly cooled while vapor of refrigerant such as water flowing in a cooling pipe being adsorbed into the adsorbent of the left second stage adsorbing device. A high adsorbing capability is generated by the lowering of temperature, resulting in that the evaporator may realize a high freezing capability.

To cool air which has been diffused indoors to a satisfactory level while raising coefficient of performance of an adsorption type refrigerator, Sato et al. (2001), invented an adsorption type refrigerator which has four adsorbers in the system with two adsorbers different from the other two in adsorption capacity. Total adsorption capacity of refrigerant of second and fourth adsorbers is rendered to be not less than 0.2 times and not more than 0.8 times of the total adsorption capacity of refrigerant of first and third absorbers. Thus, coefficient of performance of an adsorption type refrigerator is raised while maintaining a sufficient cooling capacity for utilization. In addition, the sizes of the second adsorber and the fourth adsorber can be made compact. Another work from Sato et al. (2003), mentioned an adsorption-type refrigerating apparatus to make use an adsorbent having a temperature-dependent characteristic in which an amount adsorbed in an adsorption step is larger than an amount adsorbed in a desorption step, even when a vapor pressure rate in the adsorption step is equal to or lower than a vapor pressure rate in the desorption step. When the temperature of the outside air for cooling the adsorbent increases, a sufficient cooling capacity may be obtained. In addition, a difference between the amount adsorbed in the adsorption step and the amount desorbed in the desorption step can be made larger.

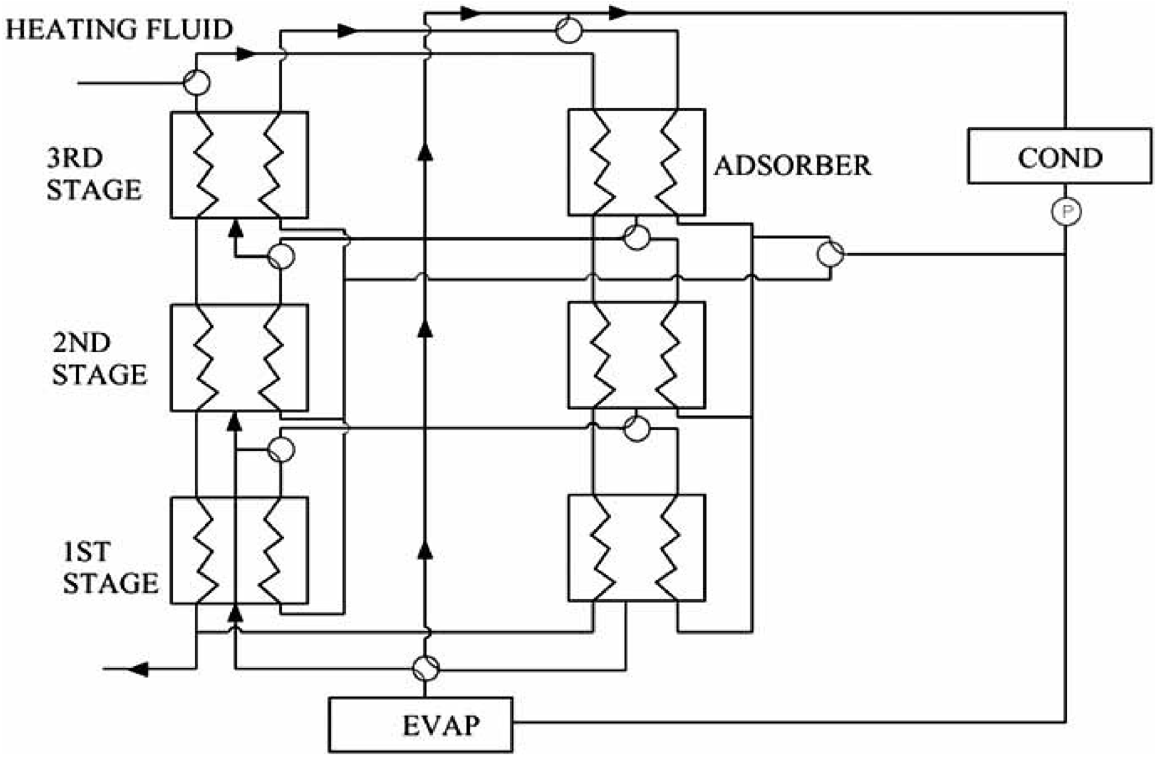


Fig. 2.3: Multi-stage, multi-adsorber adsorption refrigerator, Sato et al (1998).

In major attempts to increase the heat and mass recovery of the solid adsorption system, three main technologies have been extensively developed: regenerative process with temperature front (also named thermal wave process), Shelton et. al. (1996), regenerative process with heat and mass recovery, Aittomaecki and Haerkoenen (1992) and a third technology involving rotating adsorbers; Critoph et al (2001).

Shelton (1996), designed a heat driven heat pump system wherein a temperature gradient is established lengthwise in the adsorbent bed in order to establish a thermal wave in the bed. The so-called thermal wave system is shown in Fig. 2.4, as a heat transfer fluid is circulated through the system by a reversible pumping means, the beds are cycled between an upper and a lower operating temperature, creating the thermal wave within the bed of solid adsorbent. The heat transfer fluid always flows serially from a heater through a bed heat exchanger heating that bed while cooling the heat transfer fluid. Then the heat transfer fluid is passed through the cooling heat exchanger to further cool the heat transfer fluid, and the further cooled heat transfer fluid is passed through the other bed heat exchanger to cool that bed while heating the heat transfer fluid. Finally, the heated heat transfer fluid is returned to the heater to raise the heat transfer fluid to the original temperature. The adsorbent beds are constructed of one or more tubes through which the heat transfer fluid is passed and around which the solid adsorbent is held by a housing shell.

Aittomaecki and Haerkoenen (1992), noted that there is a draw back to the Shelton cycle which creates a thermal wave in the bed traveling in the direction of the flow of the heat transfer fluid. The cycle time must be short enough to maintain the operation temperatures of the out-flowing fluids at the desired level. However, short cycles decrease the net adsorption and lead to a decrease in the COP of the basic process. Thus, there are finite limits to the thermal wave processes which must have a cycle time long enough to maintain adsorbent regeneration efficiency, but short enough to maintain the overall COP.

Chua et al. (2002), introduced a regenerative adsorption process and a multi-reactor regenerative adsorption chiller assembly. The system includes a condenser, an evaporator and a number of reactors, each being able to operate in adsorption and desorption modes and having a coolant inlet to directly or indirectly receive coolant when operating in adsorption mode before, after or simultaneous with the condenser. A waste heat inlet for directly or indirectly receiving thermal energy from a waste heat source when operating in desorption mode. Each adsorber alternately operates in adsorption and desorption modes for substantially identical time intervals. Each adsorber has an equal chance of being the first adsorber to receive the coolant when operating in adsorption mode and the waste heat from the waste heat source when operating in desorption mode.

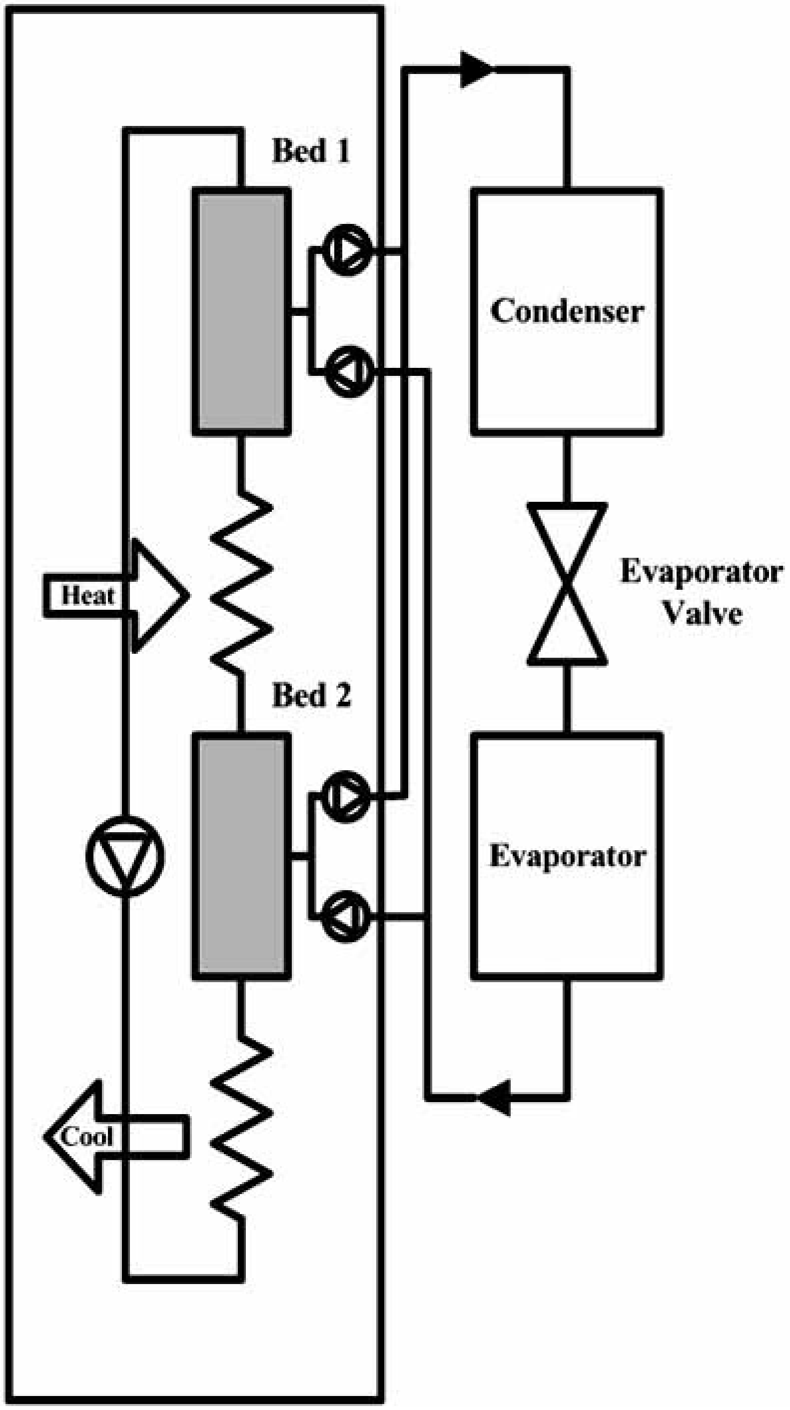


Fig. 2.4: Thermal wave refrigeration system, Shelton (1986).

Critoph and Thorpe (1996), developed an adsorption cooling system comprises two adsorbent beds each with an associated thermal management system. The thermal management systems are identical and consist of a circulating supply of a control fluid which passes through the adsorbent bed, a pump, a heat exchanger and an inert bed. Heat removed from the adsorbent beds by the control fluid is supplied to the inert beds and is store to be subsequently regenerated to heat the adsorbent beds in a later half of the operating cycle of the thermal compressor. The thermal compressor is energy efficient by virtue of the heat recycling (recovery) which is performed. In a recent work by Critoph (2002), another thermal driven adsorption system was described to provide energy-efficient heating or cooling by exploiting heat regeneration in an adsorption system.

In the paper issued by Xia et. al. (2005), mass recovery was applied in the two bed system. The improvements were in the order of 7 - 22% for the SCP and 20 - 30% for the COP, depending on the application and operation conditions. Rotary structure has been used for heat regeneration in adsorption system and providing continuous cooling effect.

Critoph (2001), Critoph and Tamainot-Tello (2003), described a rotary thermal regenerative adsorption system, which has a number of adsorbent modules circumferentially about a rotational axis partly within a toroidal conduit Fig. 2.5. A heat transfer fluid flows from an inlet of the conduit to the outlet in counter-flow with respect to the rotational movement of the adsorbent modules. Separate fluid channels encase the evaporation/condensation zones of the vessels to enable transfer of heat between the vessels and the fluid flowing in channels. In this system, heat is regenerated in a particularly simple and convenient manner. As a result the compressive device is capable of achieving higher efficiencies than existing adsorption devices.

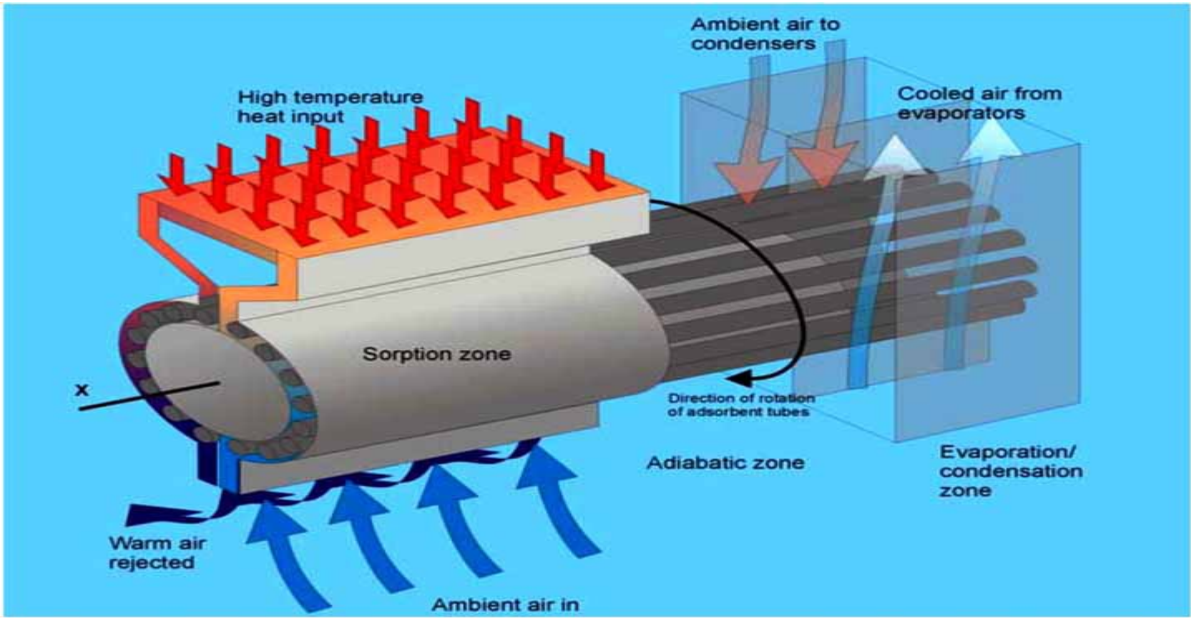


Fig. 2.5: Rotary thermal regenerative sorption device, Critoph (2001).

The high initial costs of the machines and the low heat transfer rates of the adsorbers are among the limitations to commercialize adsorption systems. Based on the principle of heat pipe, a novel silica gel - water adsorption chiller was designed in Shanghai Jiao Tong University (SJTU) in China to solve these problems; Xia et al. (2005). The system comprised three vacuum chambers in the adsorption chiller, two adsorption/desorption (or evaporation/condensation) vacuum chambers and one heat pipe working vacuum chamber as the evaporator. One adsorber, one condenser and one evaporator are housed in the same chamber to constitute an adsorption/desorption unit. The evaporators of two adsorption/desorption units are combined together by a heat pipe heat exchanger Fig. 2.6, to obtain continuous cooling effect. Since heat pipe can achieve a high heat flux, and no moving parts are used to drive the heat transfer fluid, the whole system can be made inexpensive and more reliable. In this chiller, a vacuum valve is installed between the adsorption/desorption vacuum chambers to increase its performance especially when the chiller is driven by a low temperature heat source. The reliability of the chiller has been enhanced significantly due to using fewer valves.

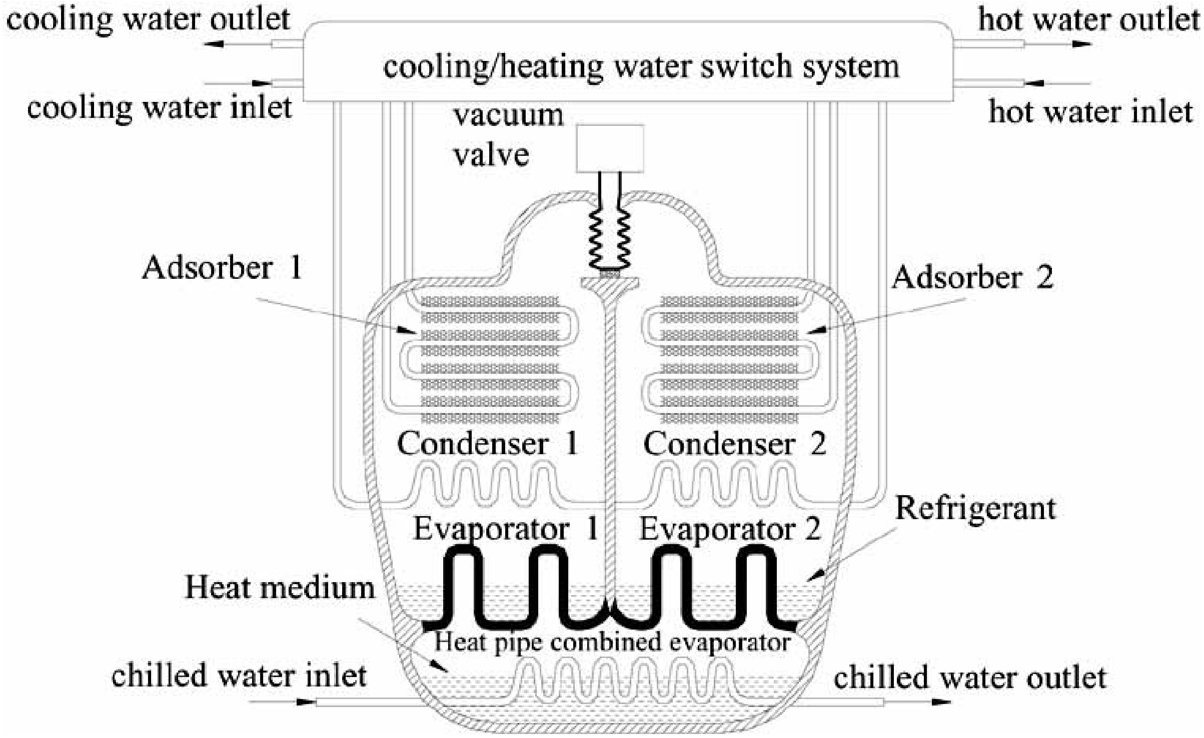


Fig. 2.6: Schematic diagram of the heat pipe type silica gel-water adsorption chiller invented by SJTU, Xia et al (2005).

Another recent patent assigned to the SJTU, Wang et al. (2005,) employed a similar evaporator heat pipe device in a compact adsorption air conditioner (CAAC). This chiller was designed for combined utilization with a fuel cell for home use, and which could be considered as a mini Combined Cooling Heating and Power (CCHP) system. There are two vacuum chambers in the chiller, and each one encloses one adsorber (using silica gel as adsorbent), one condenser and one evaporator. As these equipments are inside the same chamber, there is no need of vacuum valves among them.

Besides the application of gravity, heat pipes in adsorption chillers powered by low temperature heat sources, recent research in the SJTU also used heat pipes in the development of adsorption ice makers for fishing boats, Wang et al. (2005). The utilization of this technology increased the heat transfer inside the adsorber and allowed direct use of exhaust gases as heat source and seawater as heat sink without producing corrosion in the adsorber.

There are various valves and other components which need to be controlled during the operation of the adsorption system. To obtain a better performance and stable operation, novel control strategies have been devised based on newly developed technology. In a two-bed adsorption refrigeration apparatus, the supply of a heating fluid to a heat-exchange channel of an adsorber is alternated with the supply of a cooling fluid thereto so as to make an adsorbent release and adsorb the refrigerant vapor alternately. There might be some heating fluid remaining in the heat-exchange channel from flowing into the cooling fluid-supply channel in switching period.

For air cooled system, Okumura et al. (2004), at Denso described a method to prevent device breakage and stabilize device service life due to temperature fluctuation. The temperature control system is provided to suppress a temperature fluctuation width of a heating medium within a certain range during switching period. The rotational frequency of a fan for cooling an outdoor heat exchanger is controlled in response to following temperatures: 1) inlet and outlet temperatures of outdoor air inputted and outputted in the outdoor heat exchanger; 2) temperatures (measured by temperature sensors) of the heating medium inputted and outputted in the outdoor heat exchanger; 3) at least one inputted and outputted heating medium temperature of a number of indoor heat exchangers. This suppresses the temperature fluctuation of heating medium within a certain range and limits the number of fluctuations of unpleasant noise caused by fluctuation of a fan rotational velocity in the adsorption refrigeration machine. Also from Denso Co., Shinoda et al. (1999), disclosed a method to control the cooling capacity by varying the rotate velocity of blower for cooling the adsorber. This method can enhance the durability of an adsorption refrigeration machine.

Rosser and Dunne (1998), developed a process for controlling a sorption cooling or heating process. The heating and cooling of the adsorption zone wherein a heat exchange zone is in intimate indirect thermal contact with the adsorption zone is provided during a cyclic process. It was discovered that by conserving hot working fluid at or above a reference temperature in a hot working medium reservoir, fluctuations in the heating requirements at the hot source can be minimized and significant cost savings could be achieved. This invention can provide significant benefits in the refining and petrochemical process industry where product streams from catalytic reactions at elevated temperatures must be cooled to ambient conditions for storage or transport.

Since the pressure in adsorbent bed is low at the end of adsorption process and high at the end of desorption process. Kan et al. (2005), disclosed a method to utilize this characteristic. This leads to improvement system performance and enhancement in cooling power. Inaoka (2005), at Toyota Motor Co. designed an adsorption type heat pump capable of quickly starting operation even under an environment heated at high temperatures.

Efforts have been made to improve refrigerating capacity and relax a temperature change of cooling liquid circulating by including energy storage in the adsorption refrigeration system. Researchers at Denso, Endo and Nishikawa (2003), designed an adsorption refrigeration machine which uses a tank for cooling liquid storage. Fig. 2.7 shows the machine with two adsorbers sealed with an adsorbent to adsorb a steam refrigerant and a liquid refrigerant, providing refrigerating capacity. A snubber is used when switching from the desorption process to the adsorption process is affected. The temperature variation of cooling liquid is relaxed by mixing together cooling liquid circulating the absorbers and cooling liquid stored in the snubber.

Maier-Laxhuber et al. (2004), at Zeolith Tech., proposed an adsorption cooling apparatus with buffer reservoir. This adsorption cooling device is so formed that a liquefaction device is connected to a buffer material storage device, and the buffer material storage device buffers a part of liquefied heat of vapor of an operating means and guides the stored heat again to the circumference during adsorption. The liquefaction device is connected to the buffer material storage device to enable quickly desorption and high desorption capacity derived there from.

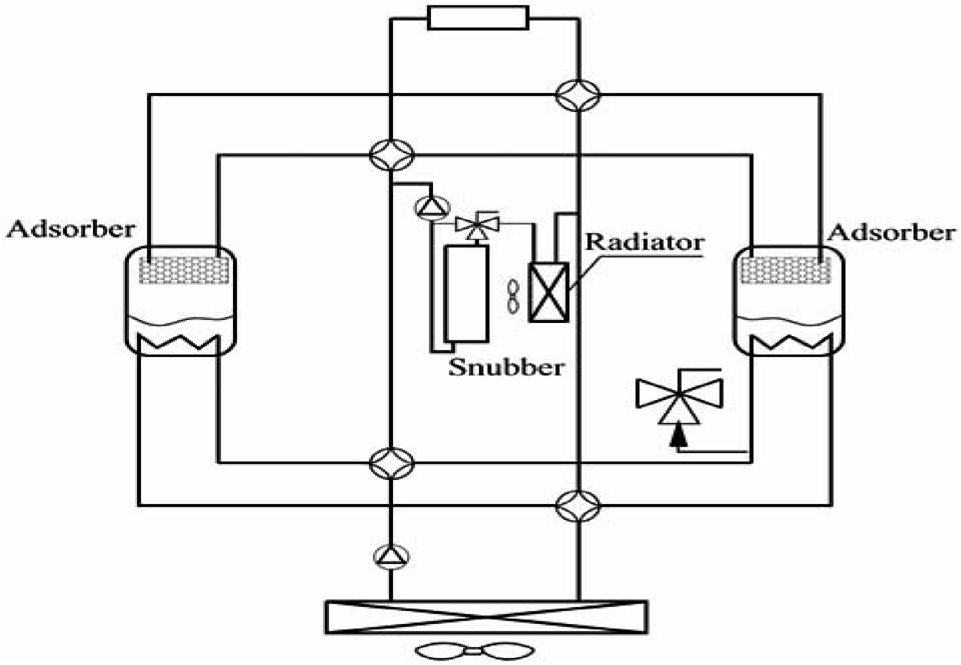


Fig. 2.7: Adsorption refrigerating machine with energy storage invented by Denso Co. Nikishawa (2003).

In some designs, adsorption is combined with other type of refrigeration system to obtain a higher cooling efficiency. Kakiuchi et al. (2009), developed an adsorption heat pump Fig. 2.8; comprising two adsorbent beds, an evaporator, a condenser, and a refrigerant steam pressure machine for raising the pressure of the refrigerant steam and transporting it. A refrigerant steam pressure machine is designed in between the adsorber and condenser. The refrigerant steam pressure machine is used for reducing the pressure of the adsorbent bed in their regeneration period, raising the pressure of the refrigerant steam and transporting it. The advantage of this heat pump is that its structure can be easily made compact and it can be used, even when the temperature of a cooling source for cooling an adsorbent in an adsorption period is higher and the temperature of a heat source for heating the adsorbent in a regeneration period is higher.

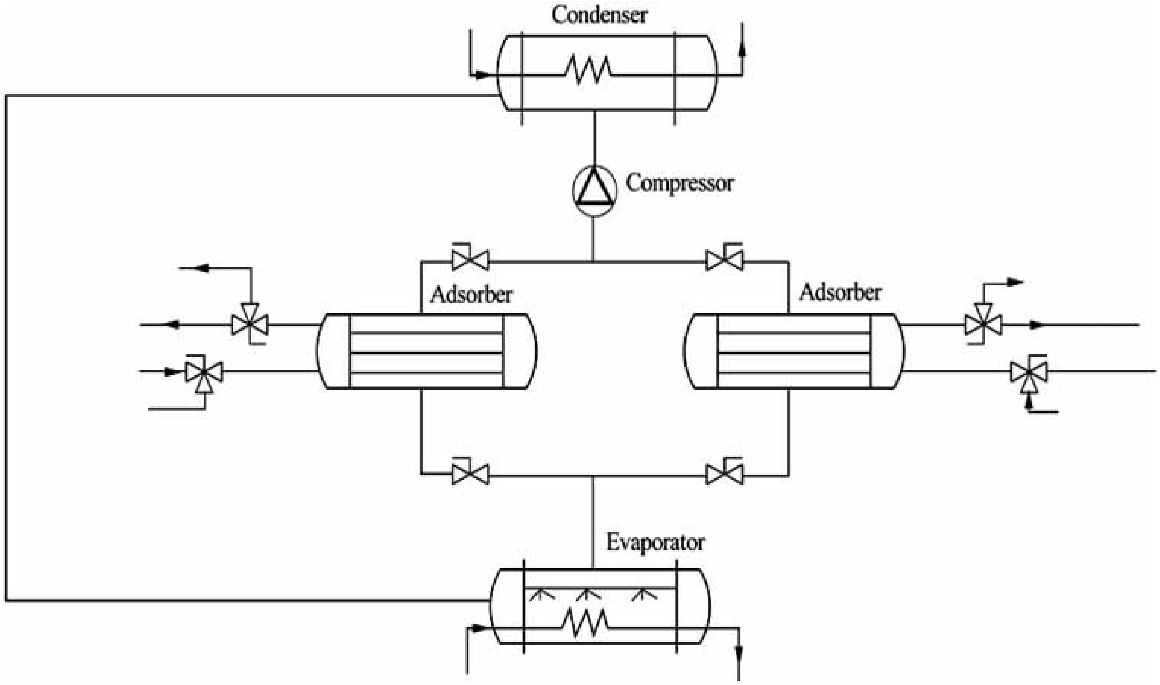


Fig. 2.8: An adsorption heat pump with steam pressure machine, Kakiuchi (2009).

A refrigerating unit which combined adsorption cycle with compressive cycle was disclosed by Inoue and Honda (1999) at Denso. As illustrated in Fig. 2.9, the refrigerating unit is provided with a compressor, a condenser, an evaporator to cool the refrigerant and an adsorbent bed which has an adsorbent that generates heat during the adsorption of moisture and absorbs heat during the releasing of moisture. During the cooling, the refrigerant discharged from the compressor flows to the adsorbent bed instead of the condenser, where the adsorbate is released from the adsorbent to be condensed. During the non-cooling, the adsorbent bed is cooled by the refrigerant condensed by condenser to make the adsorbent adsorb refrigerant. By efficiently combining a steam compression type refrigeration cycle with the adsorption cycle, a higher cooling capacity can be obtained with a limited power. At the same year, Honda et al. (1999), suggested another hybrid system by compression refrigerating cycle and an adsorption type refrigerating cycle in a different form. This system is used to cool a heat generating apparatus. The heat of adsorption is absorbed by a low-temperature refrigerant of a vapor compression type refrigerating cycle.

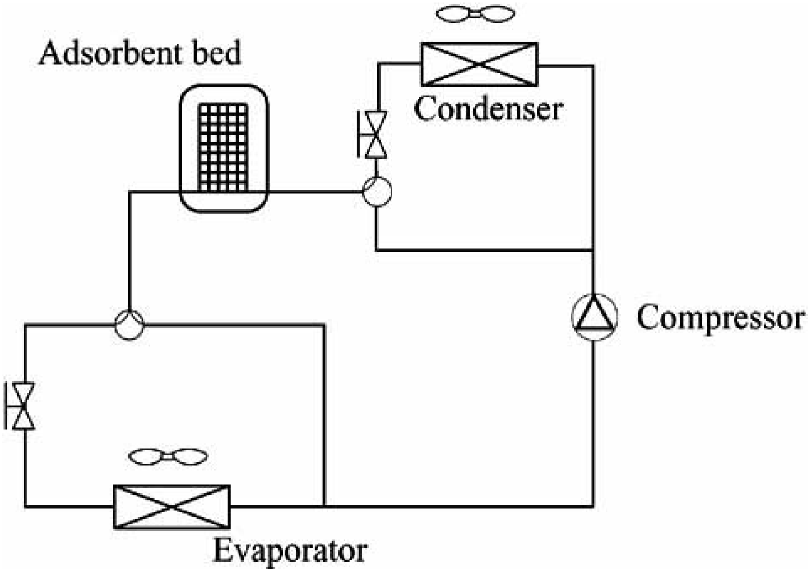


Fig. 2.9: A refrigeration unit combining adsorption cycle with compressive cycle, Inoue and Honda (1999).

Kakiuchi et al. (2005), designed a compressive heat pump hybrid with an adsorption unit to improve the system COP. As shown in Fig. 2.10, in this heat pump, a main body has a compressor, a condenser condensing the refrigerant by heat exchange with a heated fluid such as outside air, and an evaporator evaporating the refrigerant condensed in the condenser by heat exchange with a cooled fluid such as indoor air. An adsorption type cooling device is included in the system. The adsorbent bed makes use the condense heat of compression system for desorption.

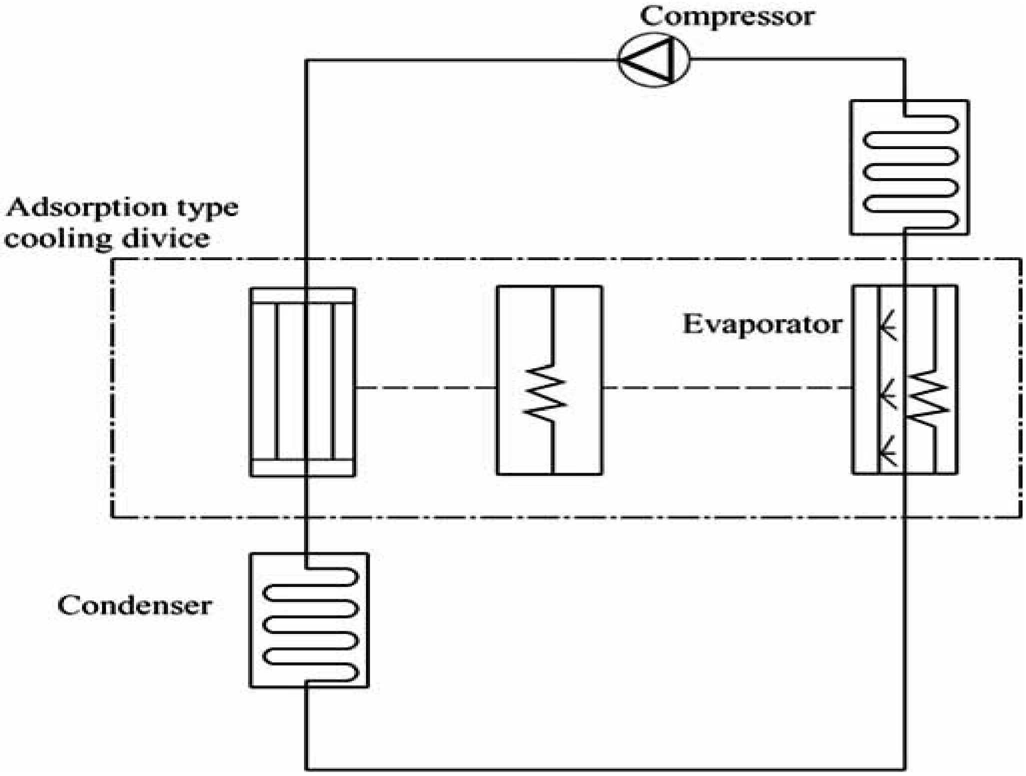


Fig. 2.10: Hybrid heat-pump. Kakiuchi (2005).

Maier-Laxhuber et al. (1996), described an adsorbent bed coating process for use on a metallic surface. The powder-like solid adsorbent with a grain diameter between 1 and 50 μm is bounded by a finely dispersed binder agent with a mass component of less than 50% with respect to the powder component. The thickness of the adsorbent bed coating on the metallic surface is less than 4mm such that the adsorption speed of water vapor at a steam pressure of less than 20 mbar results in a weight increase of at least 5% (with respect to the adsorbent bed) within 30 minutes. In order to obtain a durable adsorbent coating structure, it is suggested that the binder can be an adhesive and that the thickness of the adsorbent coating can be dipped, painted or sprayed with a drying step comprising heating the layer at temperatures higher than 150℃.

Miyairi et al. (1996), at Mitsubishi developed a coating method to attain excellent adhesiveness and to prevent the generation of stripping or crack even after repeating a heat cycle and an absorbing and desorbing cycle in a film of a coated silica gel. This coating method consists of following steps: 1) forming an oxidized film on a roughened heat transfer surface of a base material; 2) spraying a colloidal silica on the heat transfer surface heated at 150 – 200 °C to stick it thinly to the whole surface; 3) applying a silica gel wetted by the colloidal silica thereon; 4) repeated spraying of the silica colloids afterwards.

Similarly, to obtain an adsorber excellent in adsorption/desorption performance in which adsorbent can be filled well. Inoue and Sato (1999), described five steps for coating adsorbent on the heat exchanger: 1) the surface of a heat exchanger is coated with an adhesive solution; 2) solvent is dried in order to eliminate adhesion of adhesive temporarily; 3) under that state, an adsorbent comprising multiple granules is filled from the face side of the heat exchanger; 4) adhesion is recovered by wetting the adhesive with the solvent; then, 5) the adsorbent is bonded to the face of the heat exchanger through heating.

In addition to the problem of heat transfer resistance on the heat exchanger surface, a second potential problem arises when clean, uncoated aluminum is exposed to water vapor under vacuum conditions. This is the problem of corrosion of the aluminum surface and formation of $AlOH$ radicals on the surface. This reaction liberates hydrogen gas and is a cause for the loss of vacuum under some conditions that may influence the adsorber/generator performance of a sorption cooler or heat pump. Stainless steel could be used to solve this deficiency, but the low conductivity of stainless steel increases the heat transfer resistance. This makes adsorber/generators made from stainless steel incapable of transferring the required heat and can result in structures that are much more costly and only slightly more efficient than packed bed systems.

Nagashima et al. (2006), developed a method to prevent generation of a hydrogen gas in a casing that the heat exchanger is made of aluminum. The adsorber comprises a case made of a stainless steel (SUS 304) for sealing a refrigerant (water) in the state for holding substantially vacuum therein. A first heat exchanger (evaporator/condenser) is used for exchanging heat between a heat transfer fluid and the refrigerant (water). In the case, there is a second heat exchanger (adsorber) for containing the adsorbent (silica gel), and the like. The case is made of the stainless steel, and SiO_2 films are formed on the surfaces of the exchangers. Thus, a chemical reaction of the aluminum with the water (refrigerant) can be prevented and generation of a hydrogen gas can be avoided.

Adsorption refrigeration which uses ammonia as refrigerant is attractive for making ice on fishing boat which needs large amounts of ice for keeping fish fresh. Since there is enough waste heat from the exhausted fume gases of diesel engine, it is desirable to use fume gas as heat source. Also, seawater is an ideal cooling medium for adsorption systems, which could make the system more compact and powerful comparing with the systems cooled by air.

However, ammonia which is used as refrigerant is corrosive to copper, while seawater is corrosive to steel. Hence, ammonia systems usually cannot be cooled by seawater directly. To solve this problem, Xia et al. (2005) at SJTU devised three types adsorbers based on heat pipe technology. Fig. 2.11; shows a compound alternative heat pipe type absorbing bed invented by Xia et al. (2004). The seawater circuit is separated from the section for the adsorbent and refrigerant, the copper cooler (the section in the dotted frame), which is corrosion resistant to seawater and is not compatible for ammonia, is used for the seawater circuit. Seawater cools the adsorber directly through the copper cooler. The evaporator section of the heat pipes absorbs the waste heat from exhaust gas of engine. The heat pipes at the adsorbent section serves as the condenser in the process of heating and desorption and serve as the evaporator section in the process of cooling and adsorption. The working fluid in the heat pipe is water. There is a valve 1(valve for heat pipes) on the tube between the absorbing working medium section and the heat pipe evaporating section, the copper condenser is set on the top of the absorbing bed and fixed on the outer wall of the absorbing bed, and the second valve 2(water valve) is welded on the sea water inlet pipeline of the copper condenser. In the heating and desorption process, valve 1 is open and valve 2 is closed. The working fluid in the heat pipes is heated by the exhausted gas circuit and evaporates. The thermal energy is used to desorb the adsorbent, and the working fluid in the heat pipes condenses at the section for the adsorbent. In the cooling and adsorption process, valve 1 is closed and valve 2 is opened. The working media of heat pipe is heated by the adsorption heat and evaporates at the section for the adsorbent. Then, the vapor is cooled by the water circuit and condensed in the copper cooler. This system uses seawater to cool the adsorber directly. The intermediate fresh water heat exchanger and related pump is omitted in comparison with traditional absorber using ammonia as refrigerant. Thus, the system structure can be simplified and electrical energy can be saved.

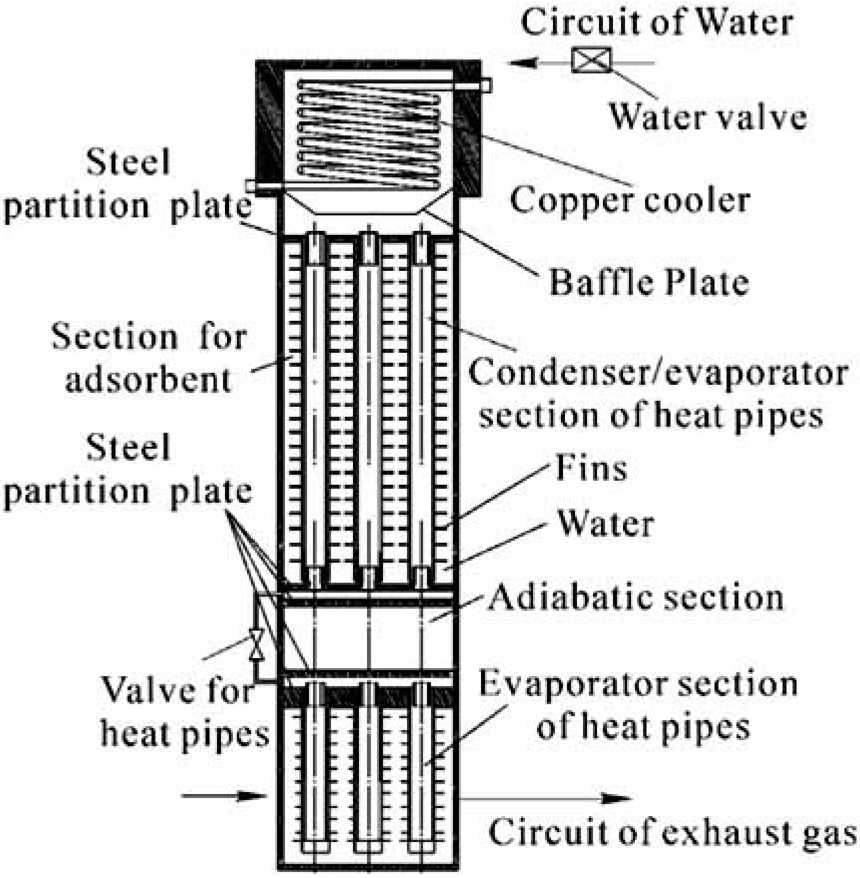
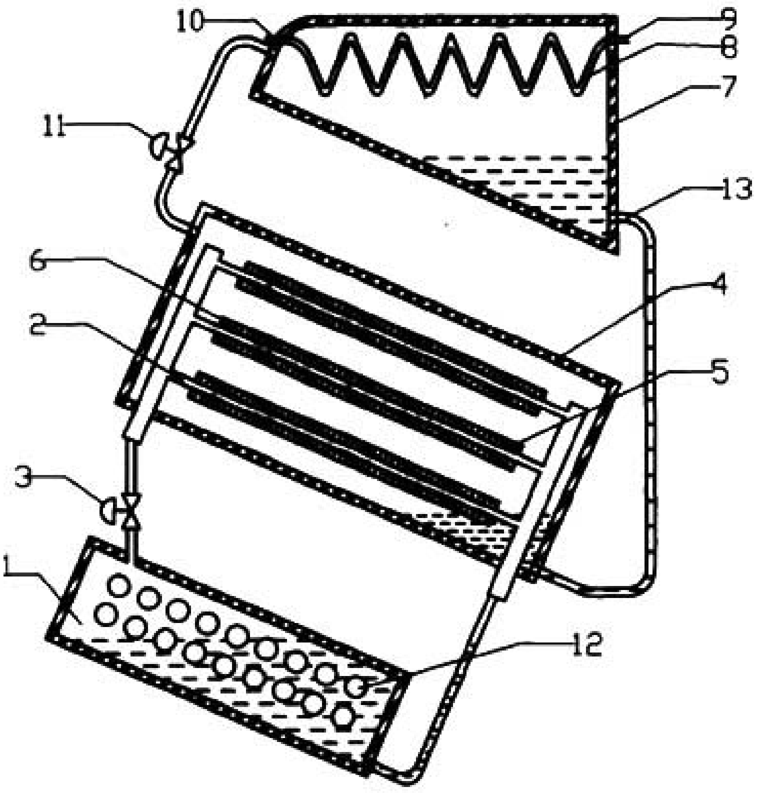


Fig. 2.11: A compound alternative heat pipe type adsorbing bed invented by Xia et al. (2004, 2005).

Xia et al. (2005), designed a double heat pipe generator driven by waste heat of fishing boat for ice making. As shown in Fig. 2.12, the adsorber comprises: fume heating steam generator, fume heat pipe, heat pipe pneumatic valve, adsorption bed, adsorption unit pipe, absorbent, sea water condenser, cooling valve, sea water pipe, sea water inlet and sea water outlet fume heating pipeline is installed on the upper part of fume heating steam boiler, its space outside is heat generator with water as medium; heating zone upper part of heat pump is connected to heating channel which is set in adsorption unit pipe with absorbent inside by heat pipe air operated valve; sea water condenser whose two ends are linked to sea water circuit is set on top of generator, and sea water inlet and outlet are linked separately to sea water pipeline and sea water condenser. Condensing zone of heat pipe is consisted of the outside space of adsorption unit pipe in adsorption bed and lower part space. The fume heat pipe uses water as working media, while seawater condenser uses methanol as intermediate heat transfer fluid. Hence, fume gas corrosion problem can be solved and reliability of the system is increased by reducing the amount of valves.



1-steam boiler, 2-heating pipeline, 3-valve, 4-adsorbent bed, 5-adsorbent module, 6-adsorbent, 7-condenser, 8-sea water pipeline, 9-seawater inlet, 10-seawater outlet, 11-valve, 12-fume line

Fig. 2.12: A double heat pipe adsorbent bed driven by waste heat of fishing boat for ice making; Xia et al. (2005).

Anyanwu and Ogueke (2001), proposed a solar refrigeration system, based on solid adsorption effect. The basic principles of operation of solar adsorption cooling are similar to those of the absorption coolers. They do, however, differ in two respects. First, the process of refrigerant accumulation on the sorbents in which solid adsorbents act like a sponge, soaking and trapping on their surfaces and inside their micro - pores the refrigerant. Second, an absorption cycle may use either liquid or solid absorbent and can be operated continuously or intermittently, while adsorption cycles use only solid adsorbents and, therefore, are characteristically intermittent in operation. The system is comprised of a container of adsorbents, which also serves as a solar collector, a condenser and an evaporator. In such units, the adsorbents perform the same function as a compressor in the conventional vapour compression refrigerator but without any moving parts.

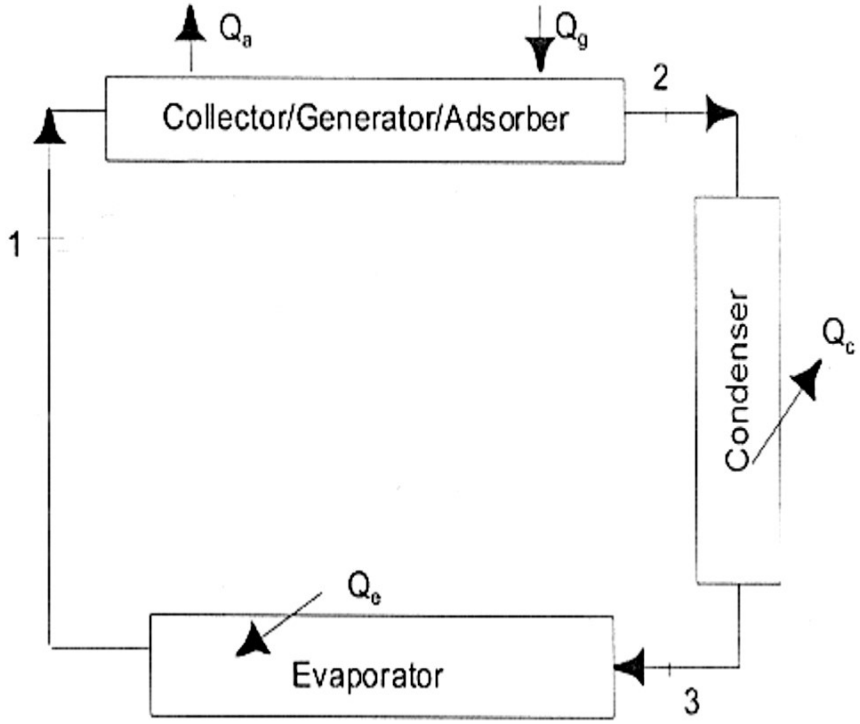


Fig. 2.13: A Simple Adsorption Solar Refrigerator, Anyanwu and Ogueke (2001)

The cycle starts at sunrise with the solar collector/generator/adsorber at ambient temperature Ta and the maximum possible concentration of refrigerant contained in the adsorbents. Solar heating of the collector results in the increase of pressure with the adsorbed mass constant, at which value the refrigerant begins to be expelled, endothermically, and be condensed. As the refrigerant begins to be released, the pressure remains approximately constant at Pc (condensing pressure), and the adsorbent becomes progressively dilute until the maximum cycle temperature is attained at point. Desorption - condensation continues even after the peak cycle temperature is reached, until the collector/generator/adsorber temperature begins to decrease due to the progressive decrease in solar radiation harvested. When the system pressure equals the saturation vapour pressure of the refrigerant at point F, the refrigerant in the evaporator begins to boil, causing heat to be extracted from the immediate environment. This results in the production of cold, thus completing the cycle.

Adsorbent module is a small heat exchange element which contains adsorbent. Usually, its structure is simple and can be produced in large scale with low cost. Adsorption refrigeration system can be built by assembling a number of adsorption modules. The capacity of system can be different by varying the number of adsorption modules. This increases the flexibility in building the adsorption system and reduces the total initial cost for system manufacturing. Further, using multi-adsorbent modules can increase the heat recovery and achieve continuous cooling since they are actually multi-bed system. Hence, adsorbent module is another important subject in developing high performance refrigeration systems. Some adsorbent module only includes the adsorbent section, Wang et al. (2005). Others include also the condensation and evaporation sections, Xia et al. (2005), which can be looked as mini adsorption system. Fig. 2.13; shows the schematic diagram of the tubular adsorption module proposed by Critoph (2001). The tube is preferably made of aluminum or stainless steel with a wall thickness of around 0.5 mm for good thermal conduction. The module contains an adsorbate fluid, preferably ammonia. The inner surface of the adsorbent tube is lined with adsorbent which serves as a sorption generator (adsorbent bed) at one end and the other end of the tube contains a combined evaporator as well as a condenser. In order to achieve heat transfer enhancement, the tube is provided with fins around its surface. Preferably the adsorbent material is an active carbon. However, alternative material may be used such as zeolite, silica gel, metal hydrides and chemical adsorbents such as calcium chloride and liquid absorbents such as water or lithium bromide. An alternative of adsorbent module comprises three sections: adsorbent section, condenser section and separated evaporator section separately, Critoph and Metcalf (2004).

Apart from the above mentioned technologies, other ideas were disclosed by inventors to enhance the performance of adsorption systems as well as reduce its size or weight.

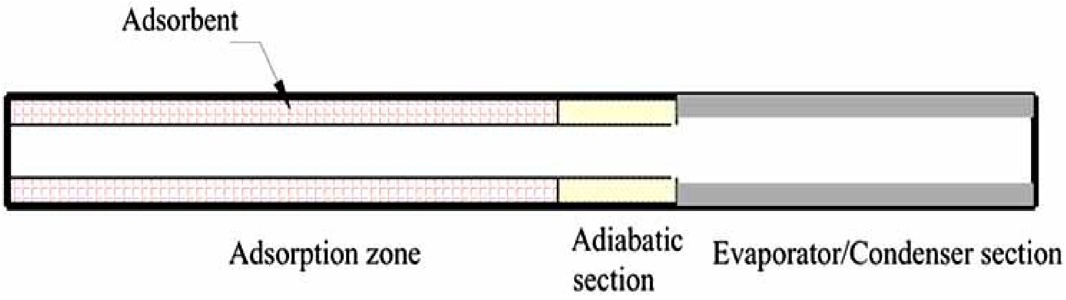


Fig. 2.14: Schematic cross section of an adsorbent module in accordance with Critoph's concept (2001).

To provide an adsorber of an adsorption refrigerator for surely preventing a deviation of filling quantity of an adsorbent, and improving adsorption efficiency by facilitating filling of the adsorbent, and securing strength.

Mochizuki et al. (2004), designed an adsorber which is composed of a single or a number of laminated heat exchange plate bodies, and a heat transfer tube inserted into these heat exchange plate bodies, and conducting a heating medium for heating/cooling the adsorbent. The heat exchange plate bodies have a pan-like plate having bored parts for inserting the heat transfer tube, the adsorbent placed on this plate, and a ventilating fixing plate installed on and fixed to the plate so as to fill and store the adsorbent between the plates having the bored parts for inserting the heat transfer tube.

2.2 ENERGY AND EXERGY ANALYSIS.

Energy analysis is the traditional method of assessing the way energy is used in an operation involving the physical or chemical processing of materials and the transfer and/or conversion of energy. This usually entails performing energy balances, which are based on the First Law of Thermodynamics, and evaluating energy efficiencies. This balance is employed to determine and reduce waste exergy like heat losses and sometimes to enhance waste and heat recovery, Ibrahim and Marc (2007).

However, an energy balance provides no information on the degradation of energy or resources during a process and does not quantify the usefulness or quality of the various energy and material streams flowing through a system and exiting as products and wastes.

On the other hand Exergy Analysis is a thermodynamic analysis technique based on the second law of thermodynamics which provides an alternative and illuminating means of assessing and comparing processes and systems rationally and meaningfully. In particular, exergy analysis yields efficiencies which provide a true measure of how nearly actual performance approaches the ideal, and identifies more clearly than energy analysis the causes and locations of thermodynamic losses. Consequently, exergy analysis can assist in improving and optimizing designs.

In order to appreciate the term Exergy, it is imperative to spell out its characteristics and come to terms with all its facets.

- A system in complete equilibrium with its environment does not have any exergy. No difference appears in temperature, pressure, concentration, etc. so there is no driving force for any process.
- The exergy of a system increases the more it deviates from the environment. For instance, a specified quantity of hot water has higher exergy content during the winter than on a hot summer day. A block of ice carries little exergy in winter while it can have significant exergy in summer.

- When Exergy loses its quality, it is destroyed. Exergy is the part of energy which is useful and therefore has economic value and is worth managing carefully.

- When energy loses its quality, exergy is destroyed. Exergy is the part of energy which is useful and therefore has economic value and is worth managing carefully.

- Exergy efficiencies are a measure of how close the system performance is to reality. This is not necessarily true for energy efficiencies, which are often misleading.

- Exergy can generally be considered a valuable resource. There are both energy or non-energy resources and exergy is observed to be a measure of value for both.

- Energy forms with high exergy contents are typically more valued and useful than energy forms with low exergy. Fossil fuels, for instance, have high energy and exergy contents. Waste heat at a near environmental condition, on the other hand, has little exergy, even though it may contain much energy, and thus is of limited value. Solar energy, which is thermal radiation emitted at the temperature of the sun (approximately 5800 K), contains much energy and exergy.

- A concentrated mineral deposit ‘contrasts’ with the environment and thus has exergy. This contrast and exergy increase with the concentration of the mineral. When the mineral is mined the exergy content of the mineral is retained, and if it is enriched or purified the exergy content increases. A poor quality mineral deposit contains less exergy and can accordingly be utilized only through a larger input of external exergy. Today this substitution of exergy often comes from exergy forms such as coal and oil. When a concentrated mineral is dispersed the exergy content is decreased, Ibrahim and Marc (2007).

2.2.1 Why Energy and Exergy Analysis.

Thermodynamics permits the behavior, performance and efficiency to be described for systems for the conversion of energy from one form to another. Conventional thermodynamic analysis is based primarily on the first law of thermodynamics, which states the principle of conservation of energy. An energy analysis of an energy-conversion system is essentially an accounting of the energies entering and exiting. The exiting energy can be broken down into products and wastes. Efficiencies are often evaluated as ratios of energy quantities, and are often used to assess and compare various systems. Power plants, heaters, refrigerators and thermal storages, for example, are often compared based on energy efficiencies or energy-based measures of merit.

However, energy efficiencies are often misleading in that they do not always provide a measure of how nearly the performance of a system approaches ideality. Further, the thermodynamic losses which occur within a system (i.e., those factors which cause performance to deviate from ideality) often are not accurately identified and assessed with energy analysis. The results of energy analysis can indicate the main inefficiencies to be within the wrong sections of the system, and a state of technological efficiency different than actually exists, Ibrahim and Marc (2007).

Exergy analysis permits many of the shortcomings of energy analysis to be overcome. Exergy analysis is based on the second law of thermodynamics, and is useful in identifying the causes, locations and magnitudes of process inefficiencies. The exergy associated with an energy quantity is a quantitative assessment of its usefulness or quality. Exergy analysis acknowledges that, although energy cannot be created or destroyed, it can be degraded in quality, eventually reaching a state in which it is in complete equilibrium with the surroundings and hence of no further use for performing tasks. For energy storage systems, for example, exergy analysis allows one to determine the maximum potential associated with the incoming energy. This maximum is retained and recovered only if the energy undergoes processes in a reversible manner. Losses in the potential for exergy recovery occur in the real world because actual processes are always irreversible.

The exergy flow rate of a flowing commodity is the maximum rate that work may be obtained from it as it passes reversibly to the environmental state, exchanging heat and materials only with the surroundings. In essence, exergy analysis states the theoretical limitations imposed on a system, clearly pointing out that no real system can conserve exergy and that only a portion of the input exergy can be recovered. Also, exergy analysis quantitatively specifies practical limitations by providing losses in a form in which they are a direct measure of lost exergy.

2.3 THE REFERENCE ENVIRONMENT IN EXERGY ANALYSIS.

The reference environment is in stable equilibrium with all parts at rest relative to one another. No chemical reactions can occur between the environmental components. The reference environment acts as an infinite system, and is a sink and source for heat and materials. It experiences only internally reversible processes in which its intensive state remains unaltered (i.e., its temperature T_0, pressure P_0 and the chemical potentials μ_{i00} for each of the i components present remain constant). The exergy of the reference environment is zero. The exergy of a stream or system is zero when it is in equilibrium with the reference environment.

The natural environment does not have the theoretical characteristics of a reference environment, since it is not in equilibrium. Its intensive properties exhibit spatial and temporal variations. Many chemical reactions in the natural environment are blocked because the transport mechanisms necessary to reach equilibrium are too slow at ambient conditions. Thus, the exergy of the natural environment is not zero; work could be obtained if it were to come to equilibrium. Consequently, models for the reference environment are used which try to achieve a compromise between the theoretical requirements of the reference environment and the actual behavior of the natural environment.

Several facets of the reference environment has been discovered and developed over the years and they are listed below

2.3.1 Natural – environment - subsystem models: these models attempt to simulate realistically subsystems of the natural environment.

2.3.2 Reference-substance models: Here, a ‘reference substance’ is selected and assigned zero exergy for every chemical element. One such model in which the reference substances were selected as the most valueless substances found in abundance in the natural environment was proposed by Szargut (1967). The criteria for selecting such reference substances is consistent with the notion of simulating the natural environment, but is primarily economic in nature, and is vague and arbitrary with respect to the selection of reference substances.

2.3.3 Equilibrium models: A model in which all the materials present in the atmosphere, oceans and a layer of the crust of the earth are pooled together and an equilibrium composition is calculated for a given temperature.

2.3.4 Constrained - equilibrium models: Ahrendts (1980), proposed a modified version of his equilibrium environment in which the calculation of an equilibrium composition excludes the possibility of the formation of nitric acid and its compounds.

2.3.5 Process - dependent models: A model which contains only components that participate in the process being examined in a stable equilibrium composition at the temperature and total pressure of the natural environment. This model is dependent on the process examined, and is not general. Exergies evaluated for a specific process-dependent model are relevant only to the process; they cannot rationally be compared with exergies evaluated for other process-dependent models. Many researchers and more researches still being conducted have examined and are still examining the characteristics of and models for reference environments e.g., Ahrendts (1980), Wepfer and Gaggioli (1980) and the sensitivities of exergy values to different reference environment models; Ibrahim and Marc (2007).

2.4 CONSIDERATIONS OF ENERGY AND EXERGY ANALYSIS.

The exergy method of analysis overcomes the limitations of the First Law of Thermodynamics. The concept of exergy is based on both the First and the Second Law of Thermodynamics. Exergy analysis clearly indicates the locations of energy degradation in a process and can therefore lead to improved operation or technology.

Exergy analysis can also quantify the quality of heat in a waste stream. A main aim of exergy analysis is to identify meaningful (exergy) efficiencies and the causes and true magnitudes of exergy losses. It is important to distinguish between exergy and energy in order to avoid confusion with traditional energy-based methods of thermal system analysis and design. Energy flows into and out of a system with mass flows, heat transfers and work interactions (e.g., work associated with shafts and piston rods). Energy is conserved, in line with the First Law of Thermodynamics. Exergy, although similar in some respects, is different. It loosely represents a quantitative measure of the usefulness or quality of an energy or material substance. More rigorously, exergy is a measure of the ability to do work (or the work potential) of the great variety of streams (mass, heat, work) that flow through a system.

A key attribute of exergy is that it makes it possible to compare on a common basis interactions (inputs, outputs) that are quite different in a physical sense. Another benefit is that by accounting for all the exergy streams of the system it is possible to determine the extent to which the system destroys exergy. The destroyed exergy is proportional to the generated entropy. Exergy is always destroyed in real processes, partially or totally, in line with the Second Law of Thermodynamics. The destroyed exergy, or the generated entropy, is responsible for the less-than-ideal efficiencies of systems or processes.

- Energy is dependent on properties of only a matter or energy flow and independent of environmental properties; Exergy on the other hand is dependent on both the properties of both a matter or energy flow and the environment.

- Energy has values different from zero when in equilibrium with the environment (including being equal to mc^2 in accordance with Einstein's equation); Exergy is equal to zero when in the dead state by virtue of being in complete equilibrium with the environment.

- Energy is conserved for all processes, based on the First Law of Thermodynamics; Exergy is conserved for reversible processes and not for real processes (where it is partly or completely destroyed due to irreversibilities), based on the Second Law of Thermodynamics.

- Energy can be neither destroyed nor produced; Exergy can be neither destroyed nor produced in a reversible process, but is always destroyed (consumed) in an irreversible process.

- Energy appears in many forms (e.g., kinetic energy, potential energy, work, heat) and is measured in that form; Exergy on the other hand appears in many forms (e.g., kinetic exergy, potential exergy, work, thermal exergy), and is measured on the basis of work or the ability to produce work.

- Energy is a measure of quantity only; while Exergy is a measure of quantity and quality, Ibrahim and Marc (2007).

2.5 USEFULNESS OF ENERGY AND EXERGY ANALYSIS RESULTS.

Besides the use of Energy and Exergy analysis results discussed in 2.2.1, results of exergy analyses of processes and systems have direct implications on application decisions and on research and development directions in that exergy analyses more than energy analyses provide insights into the ‘best’ directions for research and development effort. Here, ‘best’ is loosely taken to mean ‘most promising for significant efficiency gains’; two reasons abound:

- Exergy losses represent true losses of the potential that exists to generate the desired product from the given driving input. This is not true in general for energy losses. Thus, if the objective is to increase efficiency, focusing on exergy losses permits research and development to focus on reducing losses that will affect the objective; and

- Exergy efficiencies always provide a measure of how nearly the operation of a system approaches the actual (true) efficiency value. This is not in general true for energy efficiencies. By focusing Research and Development effort on those plant sections or processes with the lowest exergy efficiencies, the effort is being directed to those areas which inherently have the largest margins for efficiency improvement. By focusing on energy efficiencies, on the other hand, one can expend research and development effort on topics for which little margins for improvement, even theoretically, exist.

CHAPTER THREE

RESEARCH METHODOLOGY

3.1 THEORETICAL BACKGROUND OF THE SOLID ADSORPTION SOLAR REFRIGERATOR.

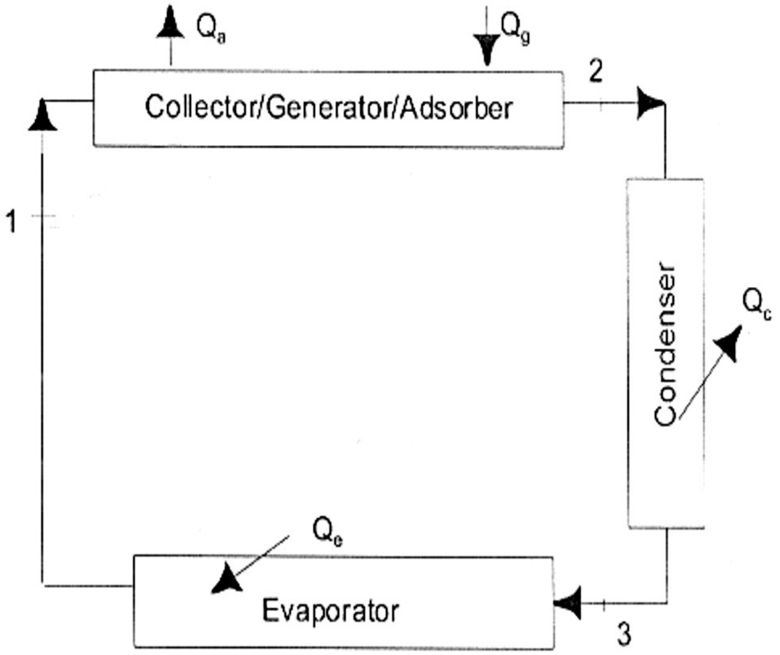


FIG. 3.1: A simple Solar Adsorption Refrigerator, Anyanwu and Ogueke (2001).

Fig. 3.1 shows the schematic representation of the solid adsorption solar refrigerator under consideration, its basic principle of operation is similar to others already researched and reviewed adsorption coolers, noticeably it differs in two basic ways: its solid adsorbent acts like sponge, soaking and trapping the refrigerant on the surface and inside the micro-pores; secondly the adsorption cycle uses solid adsorbents and can be operated continuously or intermittently.

The system described is composed of a container of adsorbents, which can also serve as a solar collector; a condenser and an evaporator. The adsorbents perform the same function as a compressor in the conventional vapour compression refrigerator but without any moving parts.

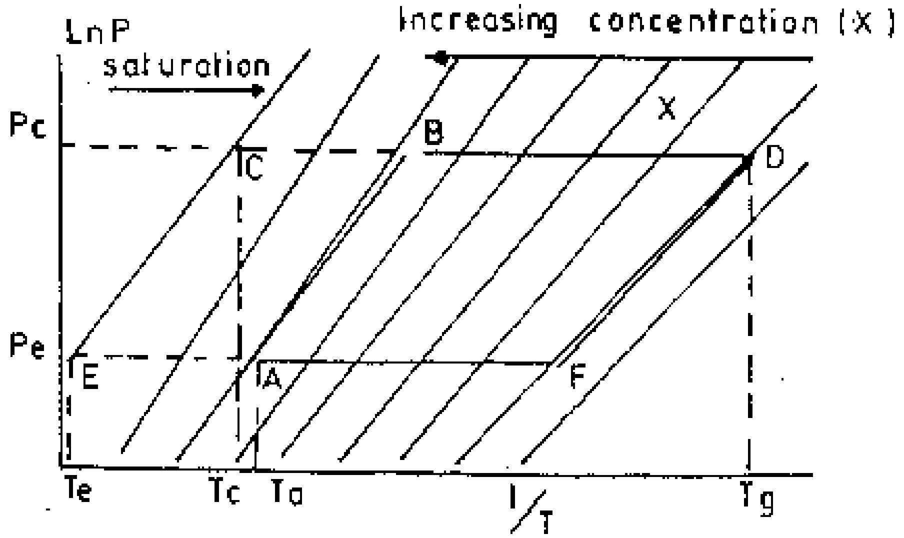


FIG. 3.2: P-T-X or Clapeyron diagram of adsorption refrigeration cycle.

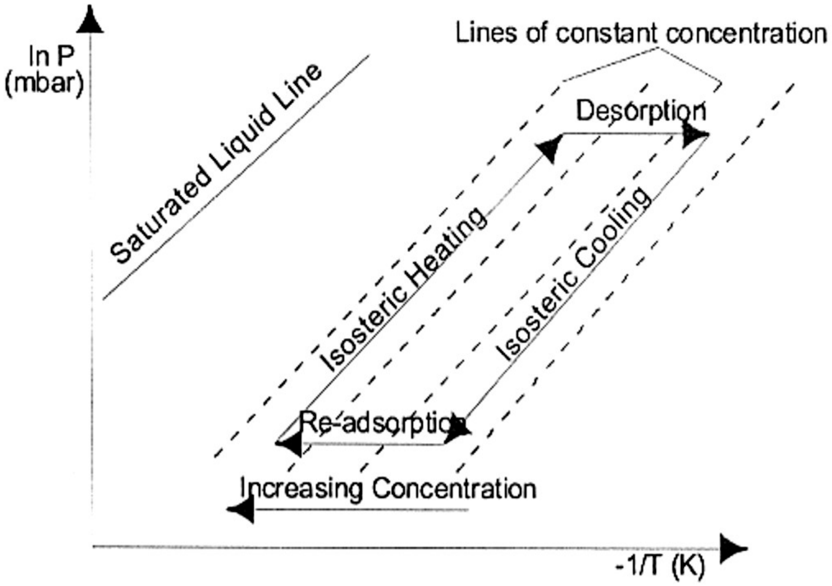


Fig. 3.3: P-T-X diagram of an adsorption system showing the theoretical refrigeration cycles.

The typical cycle is best explained with reference to the Pressure-Temperature-Concentration (P-T-X) diagram or Clapeyron's diagram, see Figs 3.2 and 3.3 above.

The cycle starts at sunrise which is at point A in Fig. 3.2; with the solar collector/generator/adsorber at ambient temperature T_a and the concentration of the refrigerant contained in the adsorber at its maximum possible concentration.

Solar heating of the collector from point A to B increases the pressure from P_e to P_c (with the mass of adsorbent constant), at which value the refrigerant begins to be expelled endothermically and be condensed. During the release of the refrigerant, the pressure remains constant at P_c (condensing pressure) and the adsorbent becomes progressively dilute until a maximum cycle temperature T_g is attained at point D. Desorption – Condensation continues even after peak temperature is reached and the system pressure increases from P_e to P_c. When this pressure equals the saturation vapour pressure at point F, it boils causing the heat to be extracted from the immediate environment. These results in the production of cold, completing the cycle at point A. Details of the principles of operation and theory of this sort of refrigerator are also presented in Anyanwu (2004).

3.2 DEVELOPMENT OF ENERGY AND EXERGY EQUATIONS.

3.2.1 Energy Equations for various Components of the Solar Refrigerator

The Energy and Exergy analysis of the Solid Adsorption Refrigeration System is carried out with the following assumptions

- The gaseous refrigerant behaves as an ideal gas; ideal or perfect gases are gases that obey the equation $pv = RT$ at temperatures that are considerably in excess of the critical temperature of the fluid, and also at very low pressures. Under such conditions, the vapour of the fluid tends to obey the said equation above, see Eastop and McConkey (1993).
- Pressure is uniform inside the adsorbent beds; in analyzing the sub-processes of the refrigeration cycle, the pressure inside the adsorbent is assumed constant, allowing for a steady flow of refrigerant in the gaseous phase.
- The mass transfer inside the adsorbent beds occurs only in the vapour phase; this is so because the focus of the analysis is on the refrigerant mass generated and less on the adsorbent. Also since the adsorbate alone is desorbed and trapped in gaseous and liquid phases, needed mass transfer in the adsorbent bed centered on the vapour phase.
- Potential, Kinetic and Chemical effects are neglected; since all the potential, kinetic and chemical effects on the interval irreversibility of the adsorbent/adsorbate pair, is translated to temperature increase as the case applies, the thermodynamics variations well covers the analysis involving any physical alterations in terms of temperature increase.

- The mass flow rate of the refrigerant is constant;
- The bed porosity is constant; allowing for even flow across beds
- All processes are at steady state,
- Losses in the ducts and walls of member components of the system are neglected.

3.2.1.1 Collector/Generator/Adsorber.

During the generation process, the quantity of energy used to generate refrigerant vapour from the adsorbent is given by Anyanwu and Ogueke (2005)

$$Q_g = m_a C_{pa} \Delta T_a + \left(m_{r,i} - m_{r,g}\right) C_{pr} \Delta T_r + m_{r,g}(h_2 - h_1) + m_{r,g} h_{sg} \quad \langle 3.1 \rangle$$

The total mass of refrigerant $m_{r,i}$ charged into the system is given by

$$m_{r,i} = m_a X_i \quad \langle 3.2 \rangle$$

The Concentration X, of the adsorbate in the adsorbent is obtained from the equation of state of the bivariant solid-vapour equilibrium given in functional form as in Guilleminot et al (1987)

$$X = \Gamma(In\, P, T) \quad \langle 3.3 \rangle$$

The Dubinin-Astakhov (D-A) eq. 3.4 relates Concentration, Pressure and Temperature in the form

$$X = \rho_r(T) W_0 exp\left[-D\left(T In \frac{P_s(T)}{P}\right)^n\right] \quad \langle 3.4 \rangle$$

The initial concentration of adsorbate in adsorbent, X_i is given by the equilibrium state equation

$$X_i = \Gamma(In P_e, T_e) \quad \langle 3.5 \rangle$$

During adsorption/refrigeration phase, $P_s(T_i) = P_e$; thus

$$X_i = \rho_r(T_i)W_0 \qquad \langle 3.6 \rangle$$

The mass of refrigerant generated $m_{r,g}$ is thus obtained from

$$m_{r,g} = \left(m_{r,i} - m_{r,f}\right) \qquad \langle 3.7 \rangle$$

Where

$$m_{r,f} = m_a X_f \qquad \langle 3.8 \rangle$$

And X_f, the concentration of adsorbate in the adsorbent at the end of the generation is obtained from

$$X_f = \Gamma\left(InP_c, T_f\right) \qquad \langle 3.9 \rangle$$

P_c is obtained from Anyanwu and Ogueke (2001) as

$$P_c = exp[17.75 - (3800/T_c)] \qquad \langle 3.10 \rangle$$

3.2.1.2 Evaporator.

The quantity of heat transferred in the evaporator during the refrigeration process is expressed as

$$Q_e = m_{r,g}(h_1 - h_3) \qquad \langle 3.11 \rangle$$

3.2.1.3 Condenser.

The quantity of heat dissipated by the hot refrigerant vapour to change to fluid form is expressed as

$$Q_c = m_{r,g}(h_2 - h_3) \qquad \langle 3.12 \rangle$$

3.2.2 System Performance.

The coefficient of performance for a heat operated refrigerating system, which relates the net cooling obtained to the heat input, is given as

$$COP = \frac{Q_e}{Q_g} \tag{3.13}$$

Where Q_e and Q_g are the heat transferred during refrigeration and the heat used to generate refrigerant during generation respectively.

Since a solar powered system is being considered, the net solar coefficient of performance is expressed as

$$COP_N = \frac{Q_e}{Q_s} = \frac{Q_e}{Q_g} \cdot \frac{Q_g}{Q_s} \tag{3.14}$$

The ratio Q_g / Q_s gives the solar collector efficiency.

Duffie and Beckman (1991) give the best solar efficiency for a conventional flat plate collector as 0.5, thus the COP_N is expressed as

$$COP_N = 0.5\frac{Q_e}{Q_g} \tag{3.15}$$

The Carnot coefficient of performance COP_{ct} gives the best possible coefficient of performance for the adsorption cycle and it is expressed as

$$COP_{ct} \leq \frac{\left(1 - (T_a / T_g)\right)}{\left((T_a / T_e) - 1\right)} \tag{3.16}$$

The thermodynamic efficiency of the system under consideration relates the actual coefficient of the performance of the system to its expected maximum performance as determined from its ideal Carnot cycle. It is obtained from a relation proposed by Pons et al. (1999),

$$\eta_{th} = \frac{COP}{COP_{ct}} \equiv \frac{Q_e\left((1/T_e)-(1/T_a)\right)}{Q_g\left((1/T_a)-(1/T_g)\right)} \qquad \langle 3.17 \rangle$$

Anyanwu and Ogueke (2005), provide the coefficient of performance/thermodynamic efficiency plots against different generator temperature and for various adsorbate/adsorbent pairs at diverse evaporation temperatures.

3.3 EXERGY EQUATIONS DEVELOPMENT.

The exergy of a thermal system considering a control volume with reference to a dead state, being heated or cooled at constant volume and in an interaction with other systems can be expressed as in Ibrahim and Marc (2007) as

$$Ex_Q = \int_i^f (1 - T_0/T)\,\delta Q \qquad \langle 3.18 \rangle$$

The dimensionless quantity in parenthesis in eq. 3.18 is called the "exegetic temperature factor'' and denoted as τ.

Estimating the exergy destroyed in each sub - process of the refrigeration cycle and in all components of the refrigerator, as a method of calculating the exergy efficiency, Gouy-Stodola developed a theorem for estimating exergy destruction in a closed flow system, expressed as in Elena et al (2007),

$$\Delta U - T_0 \Delta S = \sum\left(1 - \frac{T_0}{T_i}\right)Q_i + \sum_i m(h - T_0 s) - \sum_f m(h - T_0 s) - T_0 \Delta S_{gen} \qquad \langle 3.19 \rangle$$

Where the entropy S_{gen} is that produced (generated) in the irreversible process.

Applying exergy balance equations to each sub - process and components of the refrigeration machine, the exergy destroyed is obtained an in Elena et al (2007), for the transient regime as:

$$\Delta U - T_0 \Delta S = \sum \left(1 - \frac{T_0}{T_i}\right) Q_i + \sum_i m(h - T_0 s) - \sum_f m(h - T_0 s) - T_0 \Delta S_{gen} \quad (3.20)$$

The quantity $\Delta U - T_0 \Delta S = W_{max}$ is the maximum work potential and $T_0 \Delta S_{gen}$ is the exergy destroyed during the closed flow process.

The summation

$$\sum \left(1 - \frac{T_0}{T_i}\right) Q_i = \sum Ex\, Q_i \quad (3.21)$$

This denotes heat exergy or the transfer rate of Q_i to the environment, while

$$\sum_i m(h - T_0 s) - \sum_f m(h - T_0 s) = \sum \varepsilon_i - \sum \varepsilon_f \quad (3.22)$$

is the variation of the mass exergy between the entrance (initial state) and the exit (final state) of the system, ΔU is change in internal energy of the process being considered, T_0 is the dead state reference temperature and ΔS is the entropy change in the process.

Exergy loss $T_0 \Delta S_{gen}$ can thus be fully expressed as

$$\Pi = \sum Ex\, Q_i + \sum \varepsilon_i - \sum \varepsilon_f - (\Delta U - T_0 \Delta S) \quad (3.23)$$

It is worthy to note that eq. 3.20 is equivalent to eq. 3.23.

For a detailed Exergy analysis of the refrigeration machine under consideration, the refrigeration cycle was subdivided into the following sub - processes making up the refrigeration cycle

- Process 1 - 2 Isosteric heating phase**:** during which the adsorbent mass remains unchanged as the temperature increases;
- Process 2 - 3 Desorption phase: during which the mass of the adsorbate in the adsorbent changes as heat is added from the collector. The mass of the adsorbate in the adsorbent changes, as a result of desorption of the adsorbate from the surface of the adsorbent, from liquid to gaseous phase sets in. The adsorbent temperature continues to increase.
- Process 3 - 4 Isosteric cool down phase: during which the mass of the refrigerant produced remains unchanged and the adsorbent temperature decreases.
- Process 4 - 1 Re-adsorption phase: during which the refrigerant produced is re-adsorbed by the adsorbent. There is therefore an increase of adsorbate mass in the generator.
- Exergy loss in the Condenser in a quasi - static regime of operation and
- Exergy loss in the Evaporator particularly that expended in refrigeration in a quasi - static regime of operation.

Applying eq. 3.23 into each subdivision of the solid adsorption machine for detailed exergy analysis, the following exergy equations were developed:

3.3.1. Exergy Analysis of the Isosteric heating (Heat-Up) phase

The exergy analysis of Process 1 - 2 entails the Isosteric heating of the adsorbent/adsorbate mass during which, the mass of the adsorbent $\mathrm{m_a}$ and $\mathrm{m_{ri}}$ is constant. This is achieved by the heat energy $\mathrm{Q_{12}}$ received from the sun

Recalling eq. 3.23 with $\sum \varepsilon_i - \sum \varepsilon_f = 0$ since $\mathrm{m_a} = \mathrm{constant}$, the exergy loss Π_{12} is thus expressed as

$$\Pi_{12} = Ex(Q_{12}) - (\Delta U_{12} - T_0 \Delta S_{12}) \qquad (3.24)$$

Introducing the exegetic temperature factor into eq. 3.24, and noting that the reference is to the incremental temperature of the Generator/Adsorber during the isosteric heat-up phase, we see that the exergy loss during the desorption process as

$$\Pi_{12} = \left(1 - \frac{T_0}{T_{i,ih}}\right)(Q_{12}) - (\Delta U_{12} - T_0 \Delta S_{12}) \tag{3.25}$$

NOTE: The ‘‘exegetic temperature factor’’ defined as $\left(1 - \frac{T_0}{T_i}\right)$ there T_0 is the dead state/reference temperature, and T_i is the temperature of the component being analyzed. Since the work is transient, it is important to recall that T_i changes with time. The variation of T_i in the Collector/Generator/Adsorber, throughout the refrigeration cycle is such that during the isosteric heat - up phase, it rises till enough pressure is built where desorption of adsorbate from the adsorbent sets. During desorption, it continues to rise till the point where no noticeable refrigerant is generated in the generator. During isosteric cooling, the T_i value drop as the remaining adsorbent/adsorbate mass cools down at constant mass. Finally for the re-adsorption phase, the T_i in the Collector/ Generator/Adsorber drops slightly as refrigeration occurs in the evaporator, the value of which flattens out to the temperature at inception of the refrigeration cycle, during which re-adsorption occurs. The T_i, for the water cooled condenser, relates to the ambient temperature as expressed in eq. 3.67b and that for the evaporator measured and varies as shown in Table B 4 of Appendix B.

Picking out the components of eq. 3.25 for detailed analysis, from eq. 3.1 the heat responsible for Isosteric heating of the refrigerant mass since no refrigerant mass is here produced; is expressed as

$$dQ_{12} = m_a C_{pa} \, dT_a + m_{r,i} C_{pr} dT_r \tag{3.26}$$

The integral value of Q_{12} is as expressed

$$Q_{12} = \int_{T_1}^{T_2} m_a C_{pa} dT_a + \int_{T_1}^{T_2} m_{r,i} C_{pr} dT_r \tag{3.27}$$

It is worthy to note here that from the First Law of Thermodynamics,

$$dU = dQ + dW \qquad (3.28)$$

Since the containment vessel is of fixed volume, increase in pressure does not alter the volume. Hence we may assume that

$$dW = pdv = 0 \qquad (3.29a)$$

Substituting eq. 3.29a into eq. 3.28, gives eq. 3.29 as

$$dU = dQ + 0 \Leftrightarrow dQ \cong dU \qquad (3.29)$$

Eq. 3.29 applies to all the sub-processes of the refrigeration cycle. This is so because from the assumption preceding eq. 3.29a, increase or decrease in temperature, and increase or decrease in pressure in the containment vessel amounts to no work done on the system since $\mathrm{dW} = \mathrm{pdv} = 0$.

The difference in value of the internal energy of the system between Processes 1 - 2 is therefore expressed as an integral value of the heat energy between Processes 1 – 2 as

$$U_{12} = \left(\int_{T_1}^{T_2} m_a\, C_{pa}\, dT_a + \int_{T_1}^{T_2} m_{r,i}\, C_{pr} dT_r \right) \qquad (3.30)$$

From the Second Law of Thermodynamics,

$$dS = dQ/T \qquad (3.31)$$

Thus the entropy generated in the isosteric Process 1 - 2 is expressed as

$$S_{12} = \left(\int_{T_1}^{T_2} m_a\, C_{pa} \frac{dT_a}{T_a} + \int_{T_1}^{T_2} m_{r,i}\, C_{pr} \frac{dT_r}{T_r} \right) \qquad (3.32)$$

The exergy loss for during the isosteric heat up process is obtained by substituting equations. 3.27, 3.30 and 3.32 into eq. 3.25 to give

$$\Pi_{12} = \left(1 - \frac{T_0}{T_{i,ih}}\right)\left[\int_{T_1}^{T_2} m_a C_{pa}\, dT_a + \int_{T_1}^{T_2} m_{r,i}\, C_{pr} dT_r\right] - \left[\left(\int_{T_1}^{T_2} m_a\, C_{pa} dT_a + \int_{T_1}^{T_2} m_{r,i}\, C_{pr}\, dT_r\right) - T_0\left(\int_{T_1}^{T_2} m_a\, C_{pa}\,\frac{dT_a}{T_a} + \int_{T_1}^{T_2} m_{r,i}\, C_{pr}\,\frac{dT_r}{T_r}\right)\right] \qquad \langle 3.33 \rangle$$

The exergy loss Π_{12} during the isosteric heat up phase as expressed in eq. 3.33 above further reduces to

$$\Pi_{12} = -\frac{T_0}{T_{i,ih}}\left[\int_{T_1}^{T_2} m_a C_{pa}\, dT_a + \int_{T_1}^{T_2} m_{r,i}\, C_{pr}\, dT_r\right] + T_0\left[\int_{T_1}^{T_2} m_a\, C_{pa}\,\frac{dT_a}{T_a} + \int_{T_1}^{T_2} m_{r,i}\, C_{pr}\,\frac{dT_r}{T_r}\right] \qquad \langle 3.34 \rangle$$

3.3.2. Exergy Analysis of the Desorption Phase

This phase of analysis Process 2 – 3 of the refrigeration cycle captures the thermodynamic variations associated with refrigerant mass generation. As desorption of the adsorbate from the walls of the adsorbent occurs, the rate of adsorption reduces. Though some heat is added from heating of the adsorbent during this process, the sensible heat during this phase is experienced more from the changing mass of adsorbent, resulting in the generation of refrigerant mass. Heat expended during desorption is expressed as

$$dQ_{23} = m_{r,g} C_{pr} dT_r + m_{r,g} h_{sg} \qquad \langle 3.35 \rangle$$

The integral value of Q_{23} is expressed as

$$Q_{23} = \int_{T_2}^{T_3} m_{r,g} C_{pr} dT_r + m_{r,g} h_{sg} \qquad (3.36)$$

Recalling eq. 3.23, the exergy loss for Process 2 - 3 can be expressed as

$$\Pi_{23} = Ex(Q_{23}) - (\varepsilon_f - \varepsilon_i) - (\Delta U_{23} - T_0 \Delta S_{23}) \qquad (3.37)$$

Introducing the exegetic temperature factor into eq. 3.37 and noting again that the reference is to the temperature at which desorption occurs along the desorption process, eq. 3.37 transforms to

$$\Pi_{23} = \left(1 - \frac{T_0}{T_{i,d}}\right)(Q_{23}) - (\varepsilon_f - \varepsilon_i) - (\Delta U_{23} - T_0 \Delta S_{23}) \qquad (3.38)$$

The components of eq. 3.38 can be broken down thus…

The variation of mass exergy as

$$\varepsilon_f - \varepsilon_i = m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,d}}\right) \qquad (3.39)$$

The internal energy U_{23} in eq. 3.38 is expressed as

$$U_{23} = \left(\int_{T_2}^{T_3} m_{r,g} C_{pr} dT_r + m_{rg} h_{sg}\right) \qquad (3.40)$$

And the entropy change S_{23} in eq. 3.38 is obtained from the Second Law of Thermodynamics as

$$S_{23} = \left(\int_{T_2}^{T_3} m_{r,g} C_{pr} \frac{dT_r}{T} + \frac{m_{rg} h_{sg}}{T_{i,d}}\right) \qquad (3.41)$$

Substituting equations 3.36, 3.39, 3.40 and 3.41 into eq. 3.38 we have the exergy loss for Process 2 - 3 the desorption process as

$$
\begin{aligned}
\Pi_{23} = \left(1 - \frac{T_0}{T_{i,d}}\right)\left[\int_{T_2}^{T_3} m_{r,g} C_{pr} dT_r + m_{r,g} h_{sg}\right] - \left[m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,d}}\right)\right] \\
- \left[\left(\int_{T_2}^{T_3} m_{r,g}\, C_{pr} dT_r + m_{rg} h_{sg}\right)\right. \\
\left. - T_0\left(\int_{T_2}^{T_3} m_{r,g}\, C_{pr} \frac{dT_r}{T} + \frac{m_{rg} h_{sg}}{T_{i,d}}\right)\right] \qquad \langle 3.42 \rangle
\end{aligned}
$$

Opening up compound brackets and eliminating like terms, the exergy loss equation during desorption Π_{23} reduces to

$$
\begin{aligned}
\Pi_{23} = -\frac{T_0}{T_{i,d}}\left[\int_{T_2}^{T_3} m_{r,g} C_{pr} dT_r\right] - \left[m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,d}}\right)\right] \\
+ T_0\left[\int_{T_2}^{T_3} m_{r,g}\, C_{pr} \frac{dT_r}{T} + \frac{m_{rg} h_{sg}}{T_{i,d}}\right] \qquad \langle 3.43 \rangle
\end{aligned}
$$

3.3.3. Exergy Analysis of the Isosteric Cooling (Cool - down) Phase

During the Isosteric cool down phase, the mass of refrigerant remaining in the Collector - Generator - Adsorber is assumed as constant after desorption. The generator thus yields Q_{34} to the environment till it cools down to the dead state temperature T_0.

Recalling eq. 3.23 which is the general exergy loss, the variation of mass exergy for Process 3 - 4

$$
\sum \varepsilon_i - \sum \varepsilon_f = 0 \qquad \langle 3.44 \rangle
$$

since the mass of adsorbent remaining after desorption is assumed constant. Thus rewriting eq. 3.23 for the exergy analysis for Process 3 - 4, we have that the exergy loss Π_{34} can be expressed as

$$
\Pi_{34} = Ex(Q_{34}) - (\Delta U_{34} - T_0 \Delta S_{34}) \qquad \langle 3.45 \rangle
$$

But the exergy variation of Q_{34} is expressed as

$$Ex(Q_{34}) = \left(1 - \frac{T_0}{T_{i,ic}}\right) Q_{34} \qquad (3.46)$$

Since the remaining adsorbent mass will cool down to the referred environment temperature as cooling progresses, the exergy loss due to the heat content Q_{34} in eq. 3.46 to the environment over the isosteric cooling process will tend to zero.

The heat content Q_{34} associated with the isosteric cooling of the remaining adsorbent/adsorbate mass after desorption is expressed as in eq. 3.47 below

$$Q_{34} = \int_{T_3}^{T_4} m_a C_{pa} dT_a + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r \qquad (3.47)$$

The internal energy change for Process 3 - 4 from the First Law of Thermodynamics can thus be expressed as

$$U_{34} = \int_{T_3}^{T_4} m_a C_{pa} dT_a + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r \qquad (3.48)$$

The entropy change for the Process 3 - 4 can be obtained from the SLT analysis as

$$S_{34} = \int_{T_3}^{T_4} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr} \frac{dT_r}{T_r} \qquad (3.49)$$

Substituting equations 3.46, 3.47, 3.48 and 3.49 into eq. 3.45, the exergy loss for Process 3 - 4 is expressed as

$$\begin{aligned} \Pi_{34} = \left(1 - \frac{T_0}{T_{i,ic}}\right) & \left[\int_{T_3}^{T_4} m_a C_{pa} dT_a + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r\right] \\ & - \left[\left(\int_{T_3}^{T_4} m_a C_{pa} dT_a + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r\right)\right. \\ & \left. - T_0 \left(\int_{T_3}^{T_4} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr} \frac{dT_r}{T_r}\right)\right] \qquad (3.50) \end{aligned}$$

Eq. 3.50 will reduce to

$$\Pi_{34} = -\frac{T_0}{T_{i,ic}}\left[\int_{T_3}^{T_4} m_a C_{pa}\, dT_a + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r\right] + T_0\left[\int_{T_3}^{T_4} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_3}^{T_4} (m_{r,i} - m_{r,g}) C_{pr} \frac{dT_r}{T_r}\right] \qquad (3.51)$$

3.3.4. Exergy Analysis of the Re-adsorption Phase

In this phase, the refrigerant generated is re-adsorbed by the adsorbent in the generator after the generated refrigerant fluid has been vaporized by the heat Q_{Ev} extracted from the source to be kept at low temperature T_e. The generator yields Q_{41} to the environment and over time equalizes with the ambient temperature T_0 .

The exergy loss for Process 4 – 1, Π_{41} from eq. 3.23 is expressed as

$$\Pi_{41} = Ex(Q_{41}) - (\varepsilon_f - \varepsilon_i) - (\Delta U_{41} - T_0 \Delta S_{41}) \qquad (3.52)$$

It is worthy to note that the exergy value $\mathrm{Ex}(Q_{41})$as in eq. 3.52 for Q_{41} tends to zero as the generator equalizes with the dead state reference temperature. The adsorbent/adsorbate pair remaining goes on to adsorb the refrigerant vapour emanating from the evaporator. This occurs deep into the night and over relatively equal temperatures with ambient environment temperature.

The amount of heat expended during re - adsorption is expressed as

$$Q_{41} = \int_{T_4}^{T_1} m_a C_{pa} dT_a + \int_{T_4}^{T_1} (m_{r,i} - m_{r,g}) C_{pr}\, dT_r + m_{r,g} h_{sg} \qquad (3.53)$$

Over the re - adsorption process, the generator temperature equalizes with the ambient temperature, making the specific heat components of the adsorbent and refrigerant of less importance, as compared to the heat of re - adsorption, which occurs at equal or uniform temperatures.

$$\begin{aligned} Ex(Q_{41}) &= \left(1-\frac{T_0}{T_{i,r}}\right)Q_{41} \\ &= \left(1-\frac{T_0}{T_1}\right)\left[\int_{T_4}^{T_1} m_a C_{pa}\, dT_a + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr}\, dT_r \right. \\ &\quad \left. + m_{r,g} h_{sg} \right] \qquad (3.54) \end{aligned}$$

The variation of mass exergy in Process 4 - 1 of eq. 3.53 is expressed as

$$\varepsilon_f - \varepsilon_i = m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,r}}\right) \qquad (3.55)$$

The change in internal energy in eq. 3.53 can be obtained from the First Law of Thermodynamics and expressed as

$$U_{41} = \int_{T_4}^{T_1} m_a C_{pa}\, dT_a + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr}\, dT_r + m_{r,g} h_{sg} \qquad (3.56)$$

The entropy generated during the re - adsorption process for eq. 3.51, from the Second Law of Thermodynamics is expressed as

$$S_{41} = \int_{T_4}^{T_1} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr} \frac{dT_r}{T_r} + m_{r,g} \frac{h_{sg}}{T_{i,r}} \qquad (3.57)$$

Substituting equations 3.54, 3.55, 3.56 and 3.57 into the exergy loss equation for the re - adsorption phase Process 4 - 1: eq. 3.52, we have that

$$
\begin{aligned}
\Pi_{41} = \left(1-\frac{T_0}{T_{i,r}}\right)&\left[\int_{T_4}^{T_1} m_a C_{pa}\, dT_a + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr}\, dT_r + m_{r,g} h_{sg}\right] \\
&- \left[m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,r}}\right)\right] \\
&- \left[\left(\int_{T_4}^{T_1} m_a C_{pa}\, dT_a + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr}\, dT_r + m_{r,g} h_{sg}\right)\right. \\
&\left. - T_0 \left(\int_{T_4}^{T_1} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr} \frac{dT_r}{T_r} + m_{r,g} \frac{h_{sg}}{T_{i,r}}\right)\right]
\end{aligned}
$$

$$\langle 3.58 \rangle$$

Opening compound brackets and cancelling out similar components of eq. 3.58, the exergy loss equation during the re - adsorption process reduces to

$$
\begin{aligned}
\Pi_{41} = -\frac{T_0}{T_{i,r}}&\left[\int_{T_4}^{T_1} m_a C_{pa}\, dT_a + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr}\, dT_r + m_{r,g} h_{sg}\right] \\
&- \left[m_{rg}\left(h_{sg} - T_0 \frac{h_{sg}}{T_{i,r}}\right)\right] \\
&+ T_0 \left[\int_{T_4}^{T_1} m_a C_{pa} \frac{dT_a}{T_a} + \int_{T_4}^{T_1} (m_{r,i}-m_{r,g}) C_{pr} \frac{dT_r}{T_r} + m_{r,g} \frac{h_{sg}}{T_{i,r}}\right]
\end{aligned}
$$

$$\langle 3.59 \rangle$$

3.3.5. Exergy Analysis for the Condenser

In estimating the exergy losses in the condenser, a simplified method of analysis is assumed by considering a quasi - static regime of operation in the condenser without internal energy and entropy generation, Elena et al. (2007).

Thus the exergy destruction Π_c in the condenser; recalling eq. 3.23 where the internal energy and entropy components $(\Delta U - T_0 \Delta S) = 0$ can be expressed as

$$\Pi_c = Ex(Q_c) - (\varepsilon_f - \varepsilon_i) \qquad (3.60)$$

Assuming all the refrigerant vapour generated is cooled to liquid before entering the evaporator, the quantity of heat expended in cooling the refrigerant from the vapour to liquid is expressed as

$$Q_c = m_{r,g} L_{c,e} \qquad (3.61)$$

The heat exergy in the condenser $Ex(Q_c)$ tends to zero as cooling of the refrigerant occur over the condensation phase and the condenser equalizes with the ambient environment temperature T_0.

$$Ex(Q_c) = \left(1 - \frac{T_0}{T_{i,c}}\right) Q_c = \left(1 - \frac{T_0}{T_{i,c}}\right) m_{r,g} L_{c,e} \qquad (3.62)$$

The variation of the mass exergy in the condenser is expressed as

$$(\varepsilon_f - \varepsilon_i) = m_{rg}\left[L_e(T_c) - T_0 \frac{L_e(T_c)}{T_{i,c}}\right] \qquad (3.63)$$

Thus the exergy loss in the condenser Π_c when equations 3.62 and 3.63 are substituted into eq. 3.60 is expressed as

$$\Pi_c = \left(1 - \frac{T_0}{T_{i,c}}\right) m_{r,g} L_{c,e} - m_{rg}\left[L_{c,e}(T_c) - T_0 \frac{L_{c,e}(T_c)}{T_{i,c}}\right] \qquad (3.64)$$

In theory, both functions of the right hand side of the equation cancel out because the heat exergy and the mass variation both depend on the latent heat of condensation of the refrigerant at the condenser temperature. In practice though, the exergy loss during gaseous refrigerant condensation is equivalent in value to the heat exergy during the condensation process as well as the mass variation of the gaseous refrigerant. Thus, eq. 3.64 becomes

$$\Pi_c = \left(1 - \frac{T_0}{T_{i,c}}\right) m_{r,g} L_{c,e} \approx m_{rg}\left[L_{c,e}(T_c) - T_0 \frac{L_{c,e}(T_c)}{T_{i,c}}\right] \qquad \langle * 3.64\rangle$$

3.3.6. Exergy Analysis for the Evaporator;

Considering the destruction of exergy in the evaporator in the quasi-static regime: no change in internal energy and no entropy generation, the general exergy destroyed $\Pi_{\mathbf{ref}}$ can be expressed as

$$\Pi_{Ev} = Ex(Q_{Ev}) - \left(\varepsilon_f - \varepsilon_i\right) \qquad \langle 3.65\rangle$$

Assuming the entire refrigerant generated is used up during the refrigeration process and vaporized, the quantity of heat received by the refrigerant can be expressed as

$$Q_{Ev} = \left[m_{rg} L_e(T_e)\right](1 - x) \qquad \langle 3.66\rangle$$

NOTE that during the refrigeration process in the evaporator, the heat drawn from the refrigeration chamber will first boil and evaporate the refrigerant- which enters the evaporator in the liquid phase and leaves after being vaporized. The x in the equation denotes the quality of the refrigerant humid vapour at the entrance of the evaporator, which is calculated from the enthalpy equation of the refrigerant at the entrance of the evaporator, expressed as

$$h_{iEv} = h'(T_c) = m_{rg} C_{p,sl} \Delta T_c \equiv h'(T_v) + x[h''(T_v) - h'(T_v)] \qquad \langle 3.67a\rangle$$

Where h' is the enthalpy of the saturated refrigerant and h'' is the enthalpy of the saturated vapour. The temperature of the condenser and the ambient environment are related by the equation below

$$T_c = 159.6 + 0.4574T_0 \tag{3.67b}$$

The heat exergy in the evaporator $Ex(Q_e)$ as in eq. 3.65 is expressed as

$$Ex(Q_{Ev}) = \left(1 - \frac{T_0}{T_{i,e}}\right)[m_{rg}L_e(T_e)](1 - x) \tag{3.68}$$

The mass variation of gaseous refrigerant in the evaporator is expressed as

$$(\varepsilon_f - \varepsilon_i) = m_{rg}\left[(1 - x)L_e(T_e) - T_0(1 - x)\frac{L_e(T_e)}{T_{i,e}}\right] \tag{3.69}$$

Substituting equations 3.68 and 3.69 into eq. 3.65, we get the exergy destroyed during the refrigeration process in the evaporator as

$$\begin{aligned} \Pi_{Ev} = {} & \left(1 - \frac{T_0}{T_{i,e}}\right)[m_{rg}L_e(T_e)](1 - x) \\ & - m_{rg}\left[(1 - x)L_e(T_e) - T_0(1 - x)\frac{L_e(T_e)}{T_{i,e}}\right] \end{aligned} \tag{3.70}$$

In theory eq. 3.70 is equal to zero since the quantity of heat and the mass variation of the refrigerant is based on the latent heat of refrigerant. In practice, the exergy destroyed in the evaporator during refrigeration is equivalent to the quantity of heat destroyed during refrigeration, which is also equivalent to the mass variation of the refrigerant during refrigeration. Thus eq. 3.70 becomes

$$\begin{aligned} \Pi_{Ev} = {} & \left(1 - \frac{T_0}{T_{i,e}}\right)[m_{rg}L_e(T_e)](1 - x) \\ & \approx m_{rg}\left[(1 - x)L_e(T_e) - T_0(1 - x)\frac{L_e(T_e)}{T_{i,e}}\right] \end{aligned} \tag{3.70}$$

Applying eq. 3.67a, noting the condenser pressure calculated from eq. 3.10, and the following equations applies for calculating the quality of refrigerant humid vapour at the entrance of the evaporator x, this is needed to determine the quality of the vapour (dry) generated as the refrigerant leaves the condenser into the evaporator.

For Day I, $\mathrm{P_c = 161.447\ mmHg \equiv 0.34\ bars,}$ and the following enthalpies were determined from the respective equations

$$h_{iEv} = h'(T_c) = m_{rg} C_p \Delta T_{c-e} = 0.28862\ kJ/kg \tag{3.71}$$

$$h'(T_v) = m_{rg} C_p \Delta T_{c-e} = 0.28599\ kJ/kg \tag{3.72}$$

$$h''(T_v) = m_{rg} C_p \Delta T_{c-e} = 0.28602\ kJ/kg \tag{3.73}$$

Applying equations 3.71, 3.72 and 3.73 to eq. 3.67a, the quality of refrigerant humid vapour at the entrance of the evaporator x is 0.877 for Day I. This connotes that the refrigerant stream entering the evaporator is 0.123 rich in flash gases due to flashing off of the refrigerant from the condenser, which is at a higher temperature and pressure, as compared to the evaporator of lesser temperature and pressure.

For Day II, $\mathrm{P_c = 150.269\ mmHg \equiv 0.319\ bars}$, applying equations 3.71 and 3.72, values of $\mathrm{h'(T_c) \cong h'(T_v) = 0.05179\ kJ/kg}$ were obtained. Substituting into eq. 3.67a, the quality of refrigerant humid vapour $\mathrm{x = 0}$. implying that during the second test day, the refrigerant fluid stream has negligible vapour traces as it leaves the condenser and enters into the evaporator.

For Day III, $\mathrm{P_c = 154.75\ mmHg \equiv 0.328\ bars}$, from eq. 3.71, $\mathrm{h_{iEv} = h'(T_c) = 0.1004\ kJ/kg}$, from eq. 3.72, $\mathrm{h'(T_v) = 0.10006\ kJ/kg}$, and from eq. 3.73, $\mathrm{h''(T_v) = 0.10007\ kJ/kg}$. Substituting these values into eq. 3.67a, the quality of refrigerant humid vapour $x = 0.34$. A refrigerant humid vapour value of 0.34 connotes that the refrigerant fluid stream entering the evaporator from the condenser during the third test day, is 0.66 rich in flash gases, as a result of variable pressure and temperature.

It is worthy to note that eq. 3.67a holds where the condenser is throttled. Neglecting the fractional losses of the refrigerant fluid at the entrance of the evaporator, and assuming all the condensed fluid from the Condenser transferred to the evaporator, is used up during the refrigeration process, the exergy loss during the refrigeration process for all test days as in eq. 3.70, translates to

$$\Pi_{Ev} = \left(1 - \frac{T_0}{T_{i,e}}\right)\left[m_{rg} L_e(T_e)\right] \qquad \langle 3.70\ {*}{*}{*}\rangle$$

3.4 Total Exergy Destruction and Efficiency Calculation.

The total exergy loss Π can be expressed as

$$\Pi = \Pi_{12} + \Pi_{23} + \Pi_{34} + \Pi_{41} + \Pi_c + \Pi_{Ev} \qquad \langle 3.74\rangle$$

Exergy balance across the system as applied Elena et al. (2007) is expressed as

$$Ex(Q_d) \equiv \Pi_{Ev} + \Pi \qquad \langle 3.75\rangle$$

The exergy desorbed $\mathbf{Ex(Q_d)}$ being the sum of the exergies expended in the evaporator plus the entire exergy destroyed in a complete cycle.

The exergy efficiency ψ_{ex} is defined as

$$\psi_{ex} = \frac{\text{exergy output in system}}{\text{exergy input}} = 1 - \left[\frac{\text{exergy loss}}{\text{exergy input}}\right] \qquad \langle 3.76\rangle$$

The exergy efficiency ψ_{ex} is expressed as

$$\psi_{ex} = \frac{\Pi_{Ev}}{Ex(Q_d)} = \frac{\Pi_{Ev}}{\Pi_{Ev} + \Pi} \equiv \frac{Ex(Q_d) - \Pi}{Ex(Q_d)} \qquad \langle 3.77\rangle$$

The exergy efficiency of the refrigerant machine can thus be written overall as

$$\psi_{ex} = \frac{\Pi_{Ev}}{Ex(Q_d)} = \frac{Ex(Q_d) - \sum_k \Pi_k}{Ex(Q_d)} = 1 - \sum_k \overline{\Pi}_k \qquad (3.78)$$

Where $\sum_k \overline{\Pi}_k$ denotes the ratio of total exergy loss to that desorbed for a complete refrigeration cycle.

3.5 METHOD OF SOLUTION.

The simple procedure adopted in performing the exergy analysis for the system under consideration entails the following steps

- The system was sub-divided into six sections as developed from the processes described in the P - T - X diagram. See fig. 3.3, which are Process 1 - 2 Isosteric Heating, Process 2 - 3 Desorption/Generation Phase, Process 3 - 4 Isosteric cool down phase and Process 4 - 1 Re-adsorption phase. Other exergy destruction phases considered are the Exergy Destruction in the Condenser as well as in the Evaporator quasi - static regimes of operations.
- The general exergy analysis equation for the refrigeration process was developed as in eq. 3.23 and applied to various processes.
- The exergy analysis for every process was carried out with respect to the constrain characteristic of each process considered.
- A variable reference environment model, corresponding to the ambient environment temperature, measured at 30mins interval was chosen. This gave a clear cut description of the exergy profiles and quantity of exergy lost during each process. An average reference temperature analysis was also carried out, with daily ambient temperature of 301K.
- Exergy values were evaluated relative to the reference environment models chosen.
- Results obtained for all mentioned sub - processes of the refrigeration cycle, were interpreted from exergy variation profiles presented in

Figures 4.1 through 4.12, for variable and constant reference temperature analysis.

CHAPTER FOUR

RESULTS AND DISCUSSIONS

4.1 PRESENTATION OF RESULTS

Daily average wind velocities, and measured ambient temperatures at 30mins interval for test days I, II and III of the Solid Adsorption Solar Refrigerator were recorded. See Appendix A, Anyanwu and Ogueke 2001. This serve as a value base to which the reference environment model temperatures were obtained, for varying (corresponding to ambient environment temperature for the duration of each sub - process considered) and constant (corresponding to the average ambient temperature of the day, taken as 301 K) reference temperature analysis. The measured collector plate temperatures (℃), measured generator outer tube surface temperatures in (℃) (adsorbent temperature in the generator/adsorber), measured condensate yield (kg) and the measured evaporator water temperatures (℃) for the said days are as presented in Tables B 1 - 4 of Appendix B.

The measured ambient temperatures for the three test days are recorded from 08:00hrs to 05:30hrs, values range from about 296 K (23 ℃) peaks at about 14:30hrs at some 306.4 K (33.4 ℃), and drops to 293.1 K (20.1 ℃) at 05:30 hrs early the next morning for day I. See Appendix A 1.

The measured collector plate temperature starts out from about 305.4 K (32.4 ℃) at 08:00hrs, peaks at 381.8 K (108.8 ℃) at 13:30 hrs and falls gradually to some 293.4 K (20.4 ℃) for day I. See details of collector plate variations in Appendix B 1 for test days II and III

The refrigeration cycle of the solid adsorption solar refrigerator is sub - divided into six major sub-processes like earlier mentioned, of the six sub - process, four occur in the generator/adsorber, these four processes, although within the daily allotted time, are demarcated by keen observance in temperature variations in the generator/adsorber for all test days, as is explained in detail below. The measured outer tube surface of the generator/adsorber marking the beginning of the isosteric heating process for test day I starts at 292.9 K (19.9 ℃), peaks at 368.5 K (63.1 ℃) at 11:00 hrs, during this constant mass heating of the adsorbent/adsorbate pair, adsorption of the adsorbate occur till enough pressure is built in the generator/adsorber, for desorption of the adsorbed refrigerant to occur. Refrigerant mass accumulation in the condenser, marks the beginning of desorption. This is captured from 11:30 hrs at 71.7 ℃ (344.7 K) when the refrigerant has attained the needed heat of desorption for test day I, this continues well over the peak daily temperature of 95.5 ℃ (368.5 K) and down to 15:30 hrs at 90 ℃ (363 K). During the desorption phase, the mass of condensate yield- marking the Condensation phase, is seen to rise from 0.079 kg to 0.838 kg from 11:30 hrs to 15:30 hrs, the end of increase which marks the end of desorption and the commencement of isosteric cooling of the mass of the heated adsorbent in the generator/adsorber. Isosteric Cooling of the generator/adsorber for test day I begin at 16:00 hrs and ends at 19:30 hrs, which is from 82.6 ℃ (355.6 K) to 27.7 ℃ (300.7 K). During this cool - down process, the condenser is throttled to allow for condensed refrigerant in the condenser to flow into the evaporator, for cooling to occur. When the vapour pressure of the refrigerant generated equals the atmospheric vapour pressure, it boils and vaporizes as a result of its taking away the heat from the source to be cooled, the noticeable drop in the temperature of the evaporator water indicates that cooling has effectively commenced. See Table B 4 in Appendix B. The refrigeration process thus occur in the evaporator and re - adsorption of the used refrigerant in the generator/adsorber from the beginning of refrigeration, which is at 20:00 hrs, well into the early hours of the second test day, sunrise of which starts out another cycle.

The same sub - processes are repeated for test days II and III, but at different time frames within the cycle. For test day II, isosteric heating of the adsorbent/adsorbate starts out at 08:00 hrs, with temperature at 28.4 ℃ (301.4 K) to 12:00 hrs, with temperature at 77.5 ℃ (350.5 K). Desorption starts out at 12:30 hrs, with temperature of 78.5 ℃ (351.5 K) through peak temperatures and stops at 16:00 hrs, at a temperature of 79.7 ℃ (352.7 K). Condensation of refrigerant during the desorption phase sees the measured condensate yield rise in the condenser, from 0.040 kg to 0.91 kg, between 12:30 hrs and 16:00hrs. Isosteric Cooling occur from 16:30 hrs to 17:30 hrs, where the temperature of the adsorbent/adsorbate mass remaining in the generator/adsorber is seen to reduce from 57.5 ℃ (330.5 K) to 43.4 ℃ (316.4 K). Refrigeration and re - adsorption commence at 18:00 hrs till the next test day.

For test day III, isosteric heating of the adsorbent/adsorbate starts out at 08:00 hrs, with temperature at 24.6 ℃ (297.6 K) to 11:00 hrs, with temperature at 66.5 ℃ (339.5 K). Desorption starts out at 11:30 hrs, with temperature of 76.8 ℃ (349.8 K) through peak temperatures and stops at 16:00 hrs, at a temperature of 76.2 ℃ (349.2 K). Condensation of refrigerant during the desorption phase sees the measured condensate yield rise in the condenser, from 0.158 kg to 0.811 kg, between 11:30 hrs and 16:00hrs. Isosteric Cooling occur from 16:30 hrs to 18:30 hrs, where the temperature of the adsorbent/adsorbate mass remaining in the generator/adsorber is seen to reduce from 58 ℃ (331 K) to 30 ℃ (303 K). Refrigeration and re - adsorption commence at 19:00 hrs late into the night.

In calculating the exergy destruction in each sub-process of the adsorption refrigerator considered, detailed attention was given to the commencement and end of all processes and the values carefully picked from appropriate tables, considering the process analyzed, time of occurrence and the test day in question..

For Exergy destruction in the Process 1 – 2, Π_{12}: Isosteric Heat up Phase, eq. 3.34 was employed in calculating the exergy values. Calculated values obtained corresponded with values generated from MATLAB when the generated equations were ran with it. Table D 1 in Appendix D, shows the exergy values obtained for test days I, II and III for varying reference temperatures exergy analysis corresponding to the ambient environment temperature, measured at 30 minutes intervals. Table D 7 in Appendix D, shows the exergy values at a constant (daily average) reference temperature taken as 301 K. The reference environment is one in which the exergy is zero, thus posing as a sure reference point to which exergy variations can be quantified from (forming a datum). See Section 2.3.

For Exergy destruction in the Process 2 – 3, Π_{23}: Desorption Phase, eq. 3.43 was applied for calculating exergy values during this process. Table D 2 in Appendix D, shows the exergy values obtained for days I, II and III for varying dead state reference temperatures corresponding to the ambient environment temperature, measured at 30 minutes intervals. Table D 8 in Appendix D, shows the exergy values at a constant reference dead state temperature taken as 301 K.

For exergy destruction in the Process 3 - 4, Π_{34}: Isosteric Cool down Phase, eq. 3.51 was employed in calculating the exergy destroyed. Table D 3 in Appendix D, shows the exergy values obtained for days I, II and III for varying dead state reference temperatures corresponding to the ambient environment temperature, measured at intervals. Table D 9 in Appendix D, the exergy values at constant reference dead state temperature taken as 301 K.

For Exergy destruction in the Process 4 – 1, Π_{41}: Re-adsorption Phase, eq. 3.59 was used in calculating the exergy destruction in this process. Table D 4 in Appendix D, shows the exergy values obtained for days I, II and III for varying dead state reference temperatures corresponding to the ambient environment temperature, measured at 30 minutes intervals. Table D 10 in Appendix D, shows the exergy values at a constant reference dead state temperature taken as 301 K.

For Exergy Destruction in the Condenser, eq. 3.64 was employed in calculating the exergy destruction in a quasi-static regime of operation. Results calculated for test days I, II and III were tabulated in Table D 5 in Appendix D, for varying dead state reference temperatures corresponding to the ambient environment temperature, measured at 30 minutes intervals. Table D 11 of

Appendix D, shows the exergy values obtained at a constant reference dead state temperature taken as 301 K.

For Exergy Destroyed in the Refrigeration Process, eq. 3.70 was employed in calculating the exergy destruction in a quasi-static regime of operation. Results gathered in days I, II and III were tabulated in Table D 6 of Appendix D, for varying dead state reference temperatures corresponding to the ambient environment, measured at 30 minutes intervals. Table D 12 in Appendix D, shows the exergy values obtained at a constant reference dead state temperature taken as 301 K.

Tables D V - RT and D C - RT in Appendix D, shows the values of exergy destroyed in the generator/adsorber during the entire refrigeration cycle for test days I, II and III, for varying and constant reference temperature analysis.

Table E 13 and 14 in Appendix E, shows the Accrued Exergy Values for all sub - processes of the refrigeration cycle for test days I, II and III for varying and constant reference temperature analysis respectively. The exergy efficiency values obtained are shown at the bottom row of the Tables for each test day, which connotes the exergy efficiency of the entire refrigeration cycle.

The variations of Exergy destruction for all sub-processes of the refrigeration cycle, measured at varying and constant (average) reference temperature models, are presented in Figures 4.1 - 4.12.

The variations of Exergy destruction with time for each sub-process, for all three test days, are presented on a figure to outline the actual behavior of each sub-process with time over the process period.

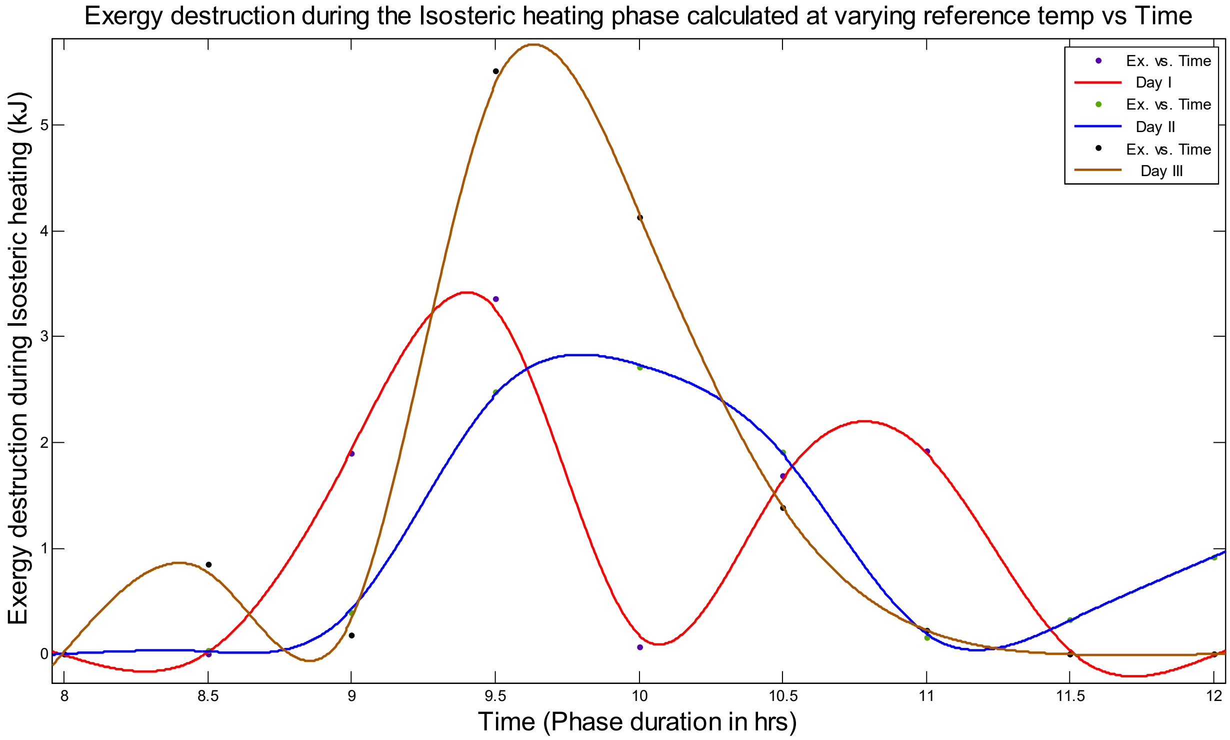


Fig 4.1 Variations of Exergy destroyed during the heat up phase with time of occurrence, for varying reference temperature analysis.

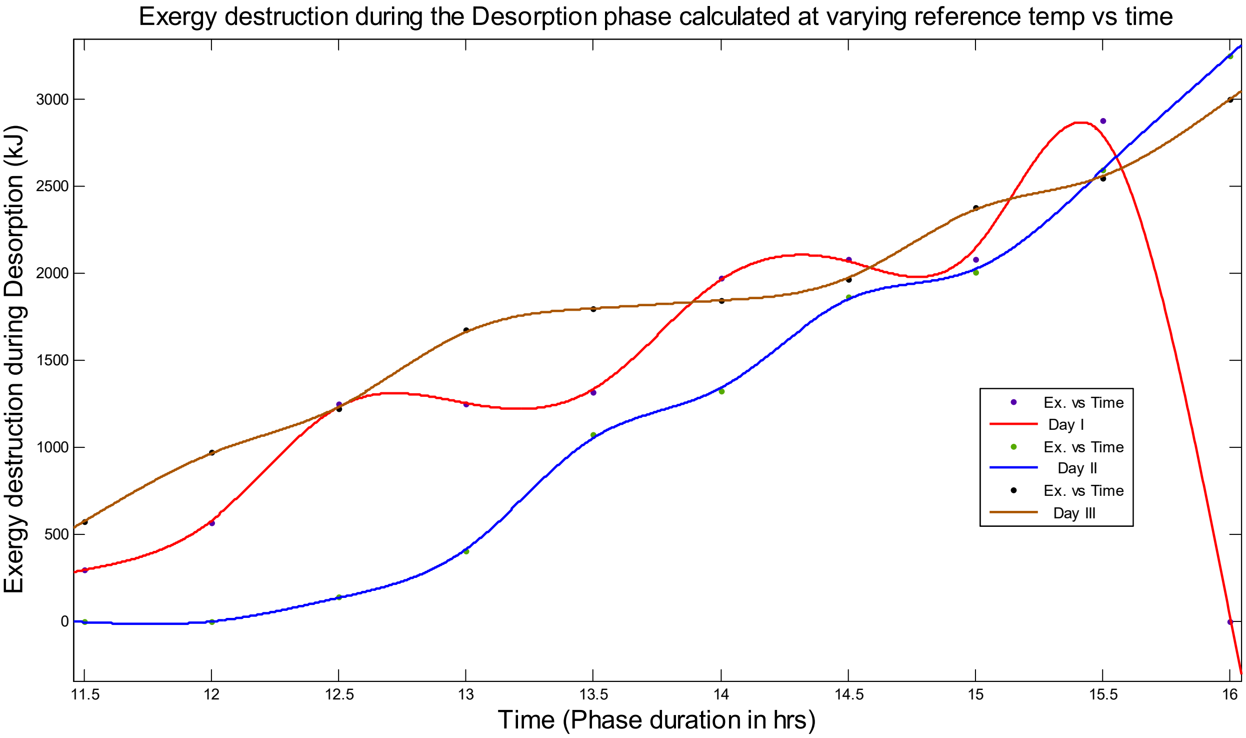


Fig 4.2 Variations of Exergy destroyed during the Desorption phase with time of occurrence, for varying reference temperature analysis.

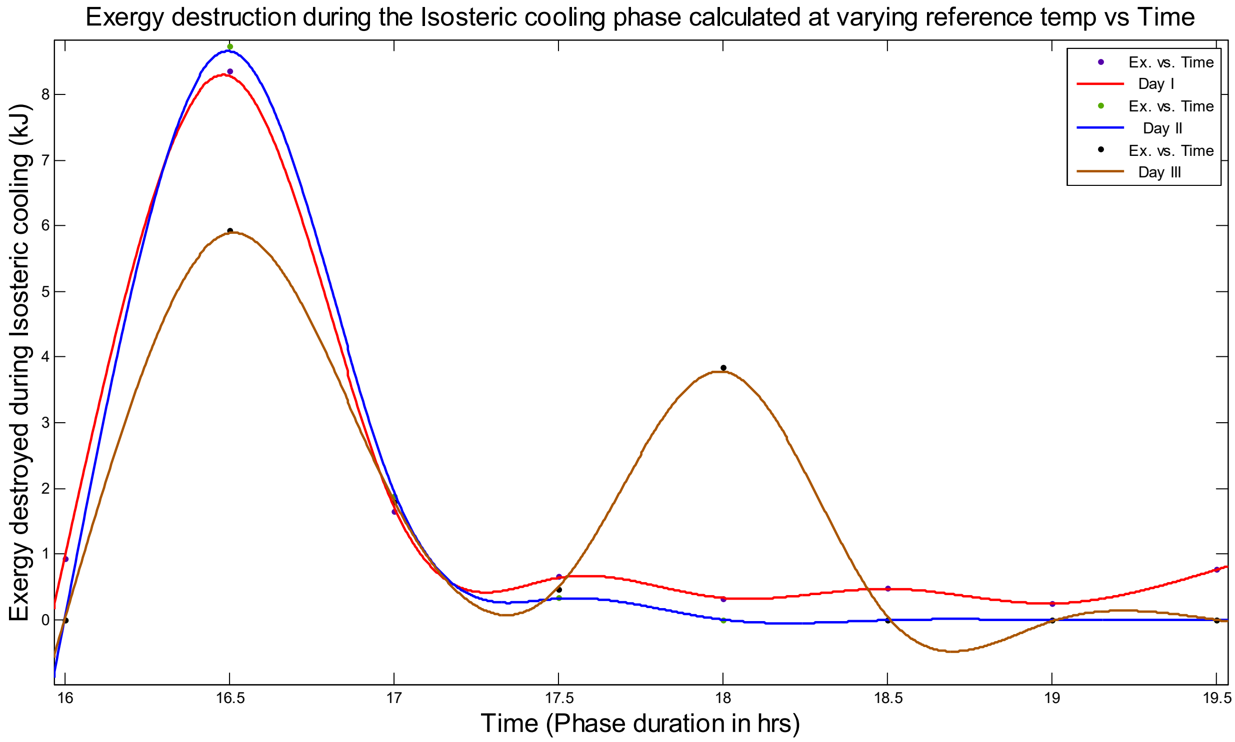


Fig 4.3 Plots of Exergy destroyed during the isosteric cool down phase with time of occurrence, for varying reference temperature analysis.

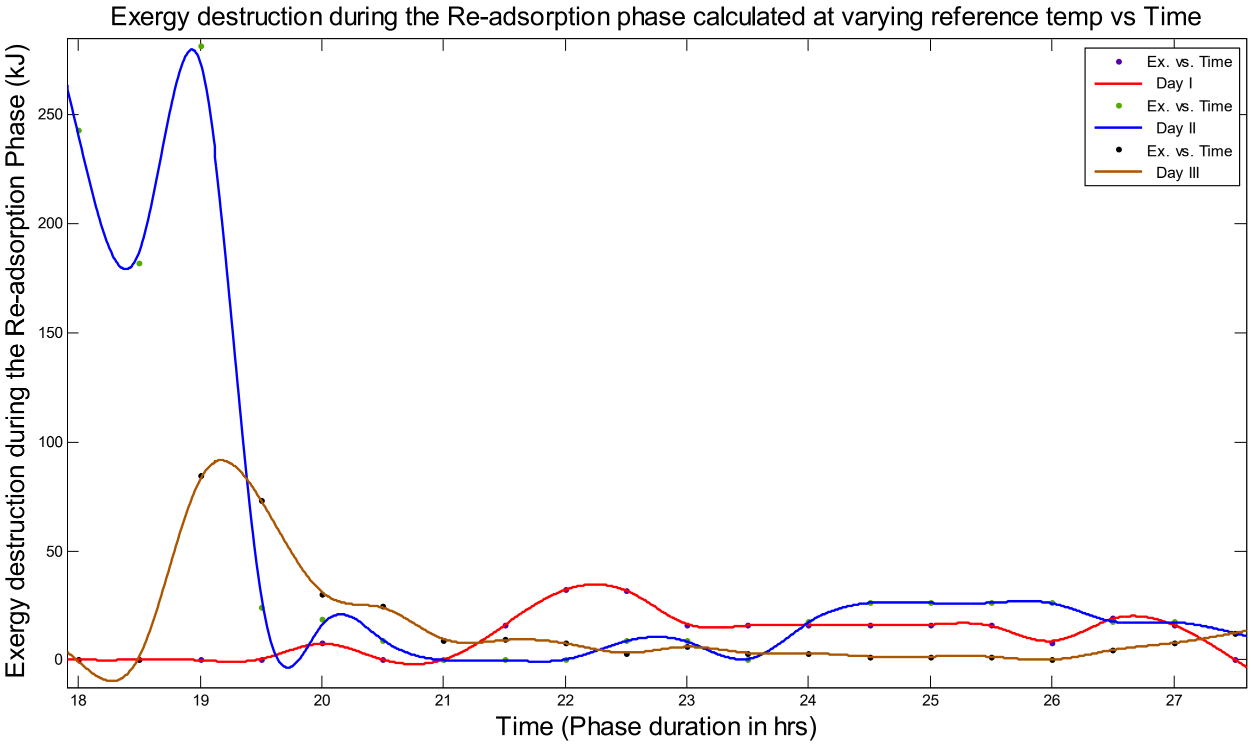


Fig 4.4 Variations of Exergy destroyed during the re-adsorption phase with time of occurrence, for varying reference temperature analysis.

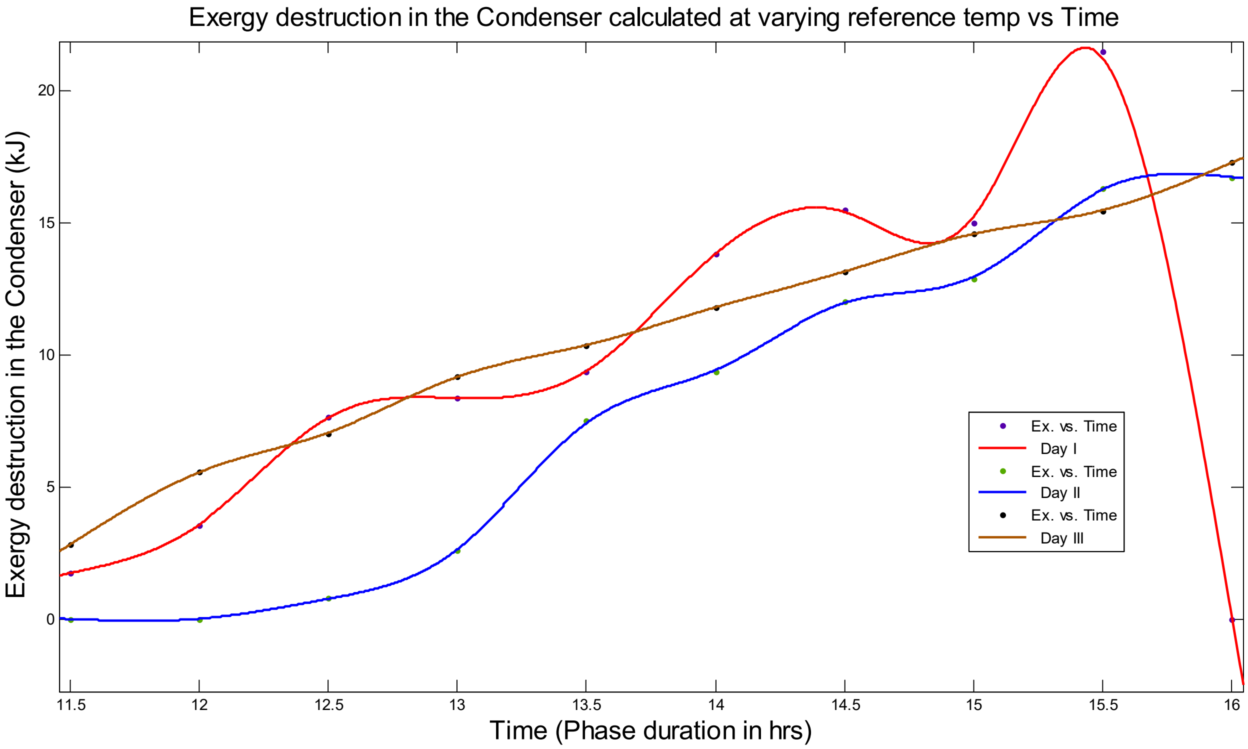


Fig 4.5 Variations of Exergy destroyed in the condenser with time of occurrence, for varying reference temperature analysis.

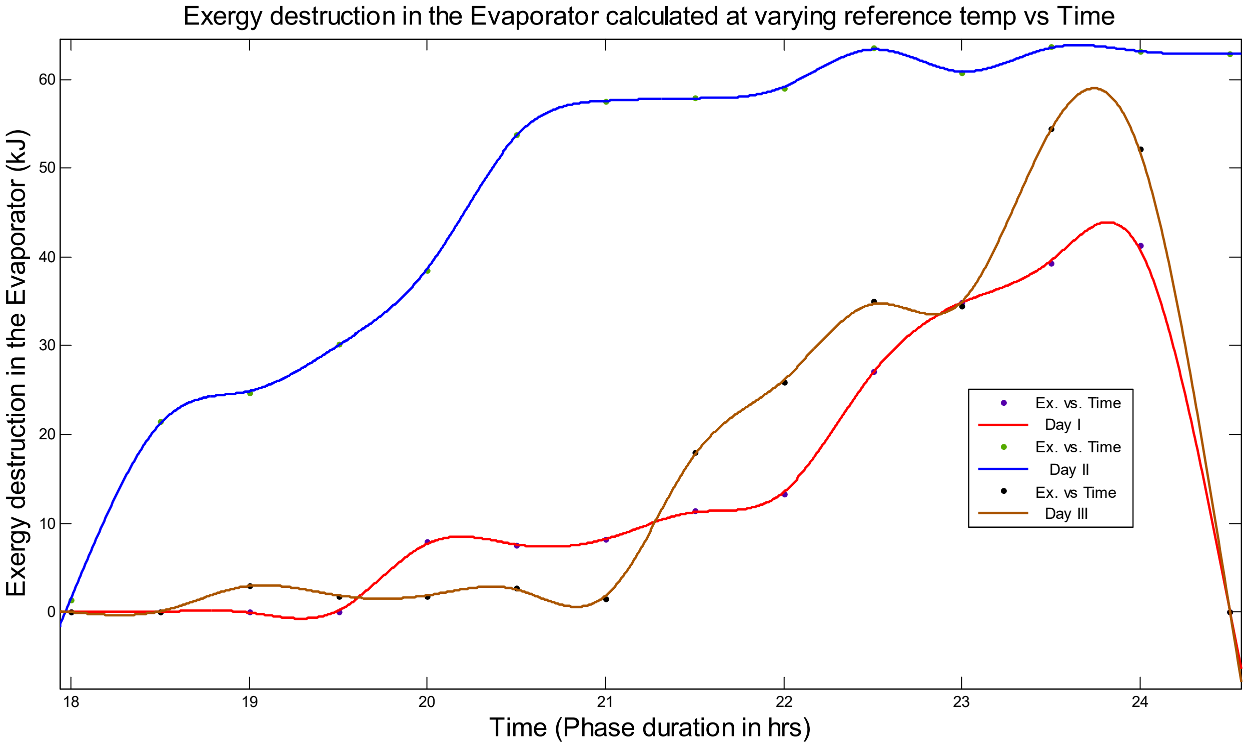


Fig 4.6 Variations of Exergy destroyed during the refrigeration phase with time of occurrence, for varying reference temperature analysis.

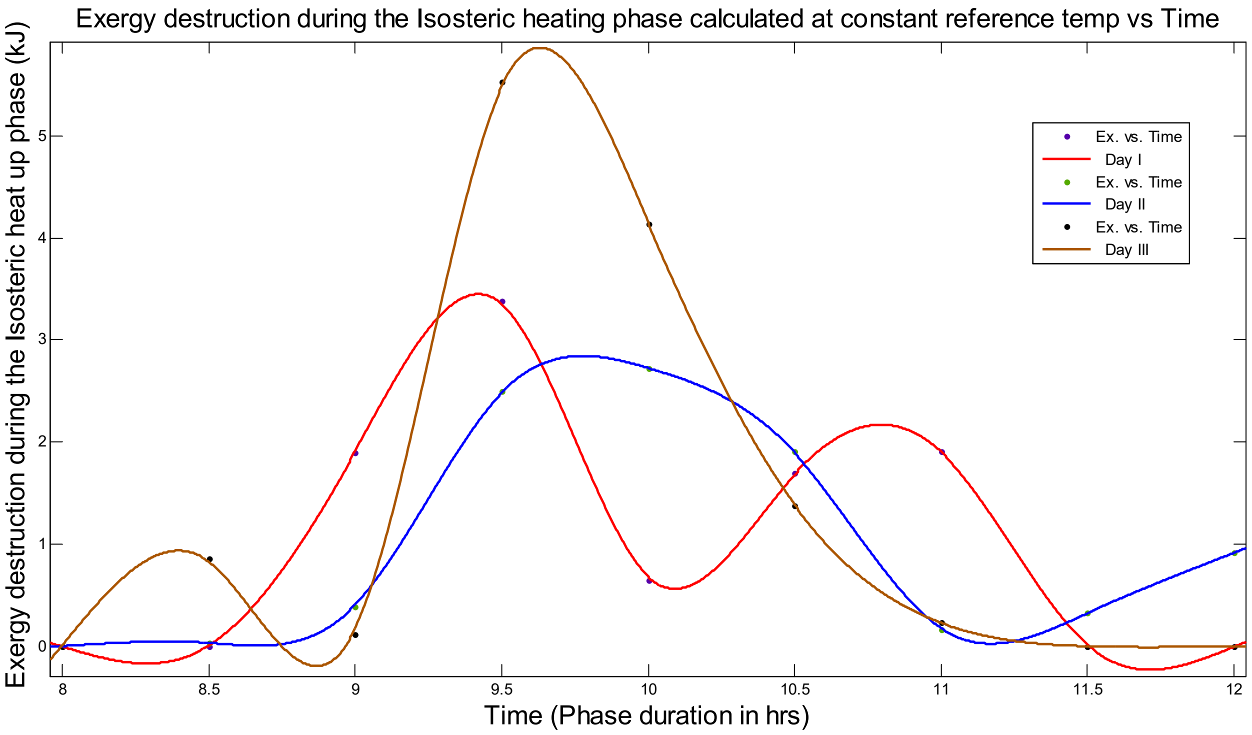


Fig 4.7 Variations of Exergy destroyed during the heat up phase with time of occurrence, for constant reference temperature analysis, at 301K.

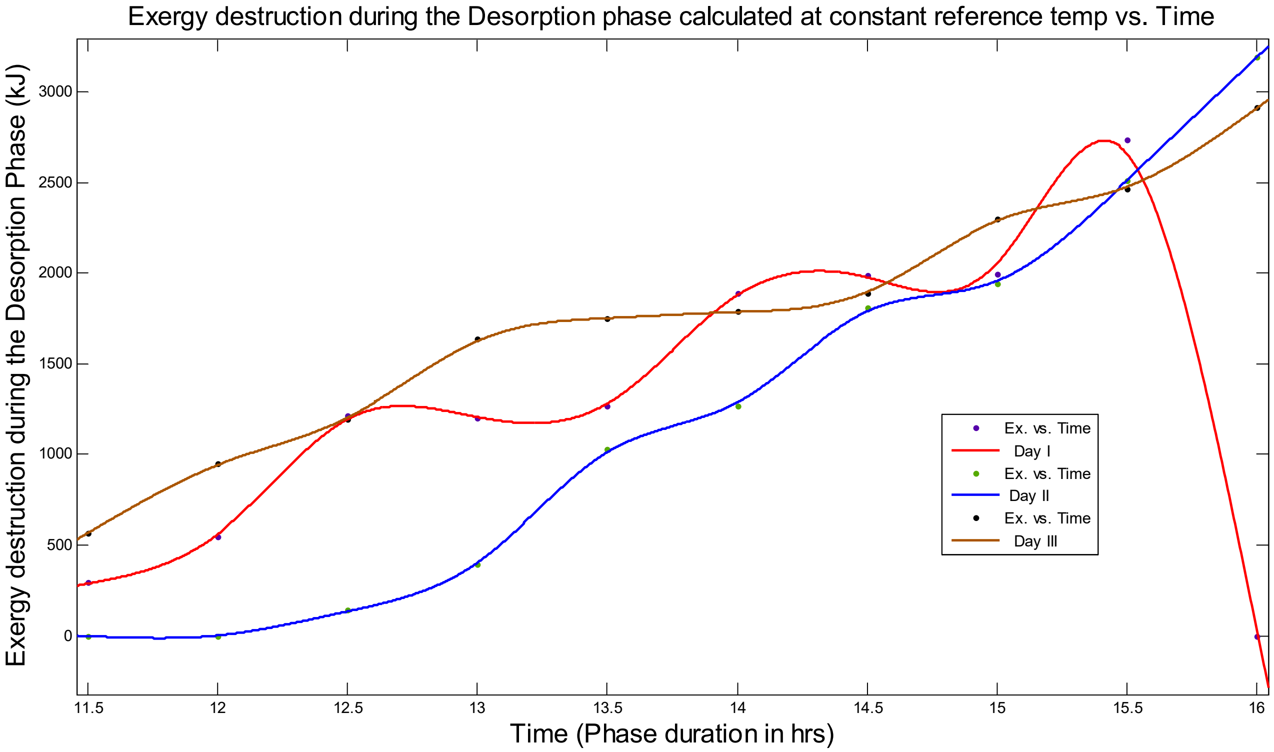


Fig 4.8 Variations of Exergy destroyed during the desorption phase with time of occurrence, for constant reference temperature analysis, at 301K.

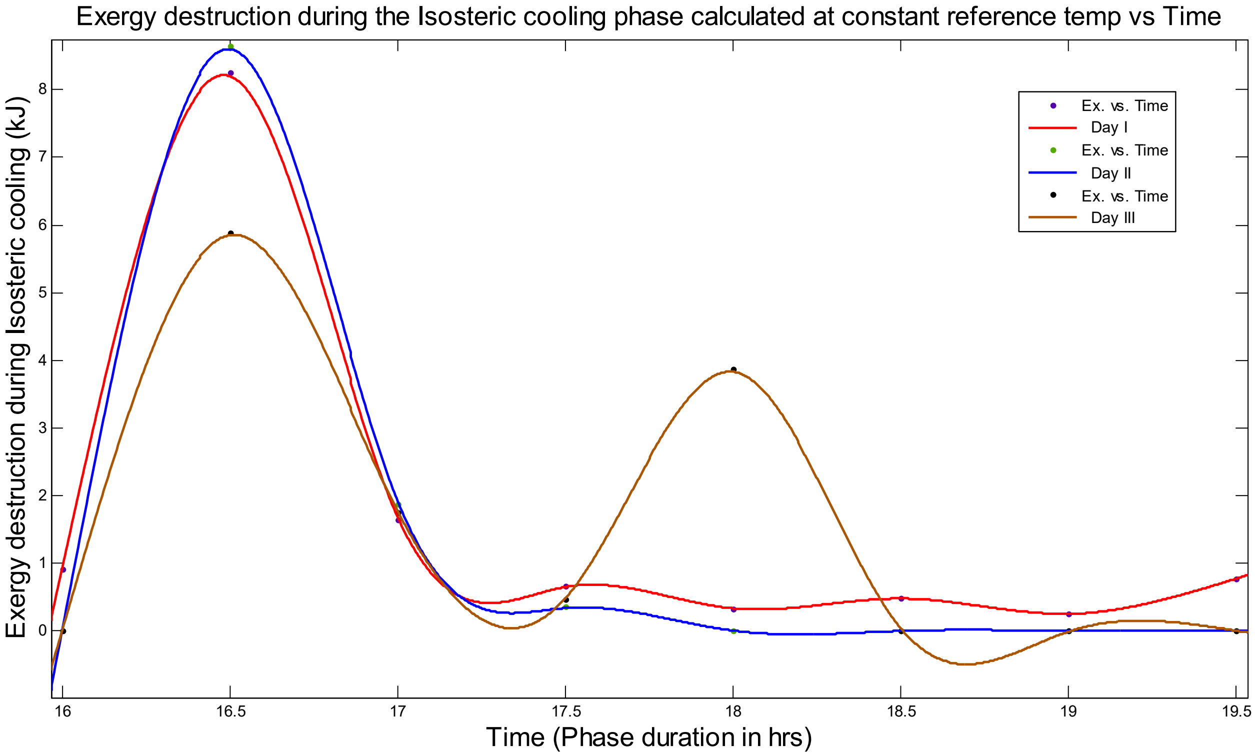


Fig 4.9 Variations of Exergy destroyed during the isosteric cool down phase with time of occurrence, for constant reference temperature analysis, at 301K.

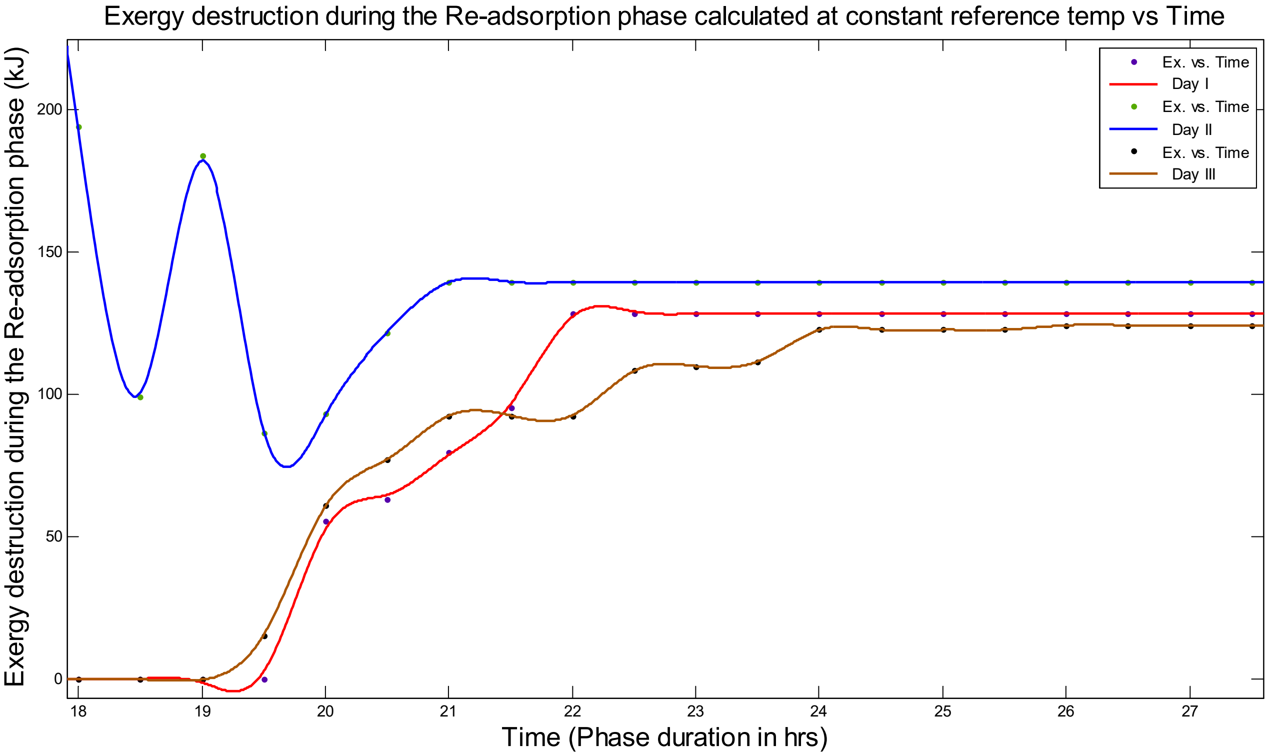


Fig 4.10 Variations of Exergy destroyed during the re-adsorption phase with time of occurrence, for constant reference temperature analysis, at 301K.

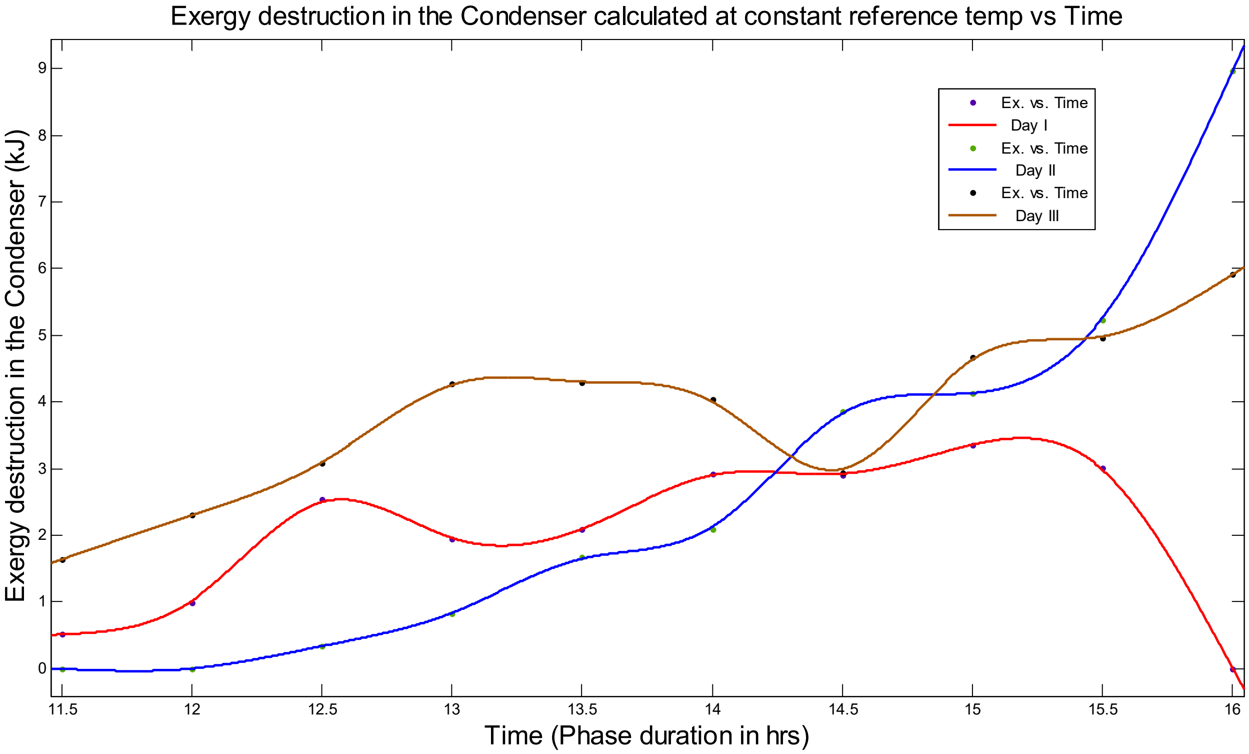


Fig 4.11 Variations of Exergy destroyed in the condenser with time of occurrence, for constant reference temperature analysis, at 301K.

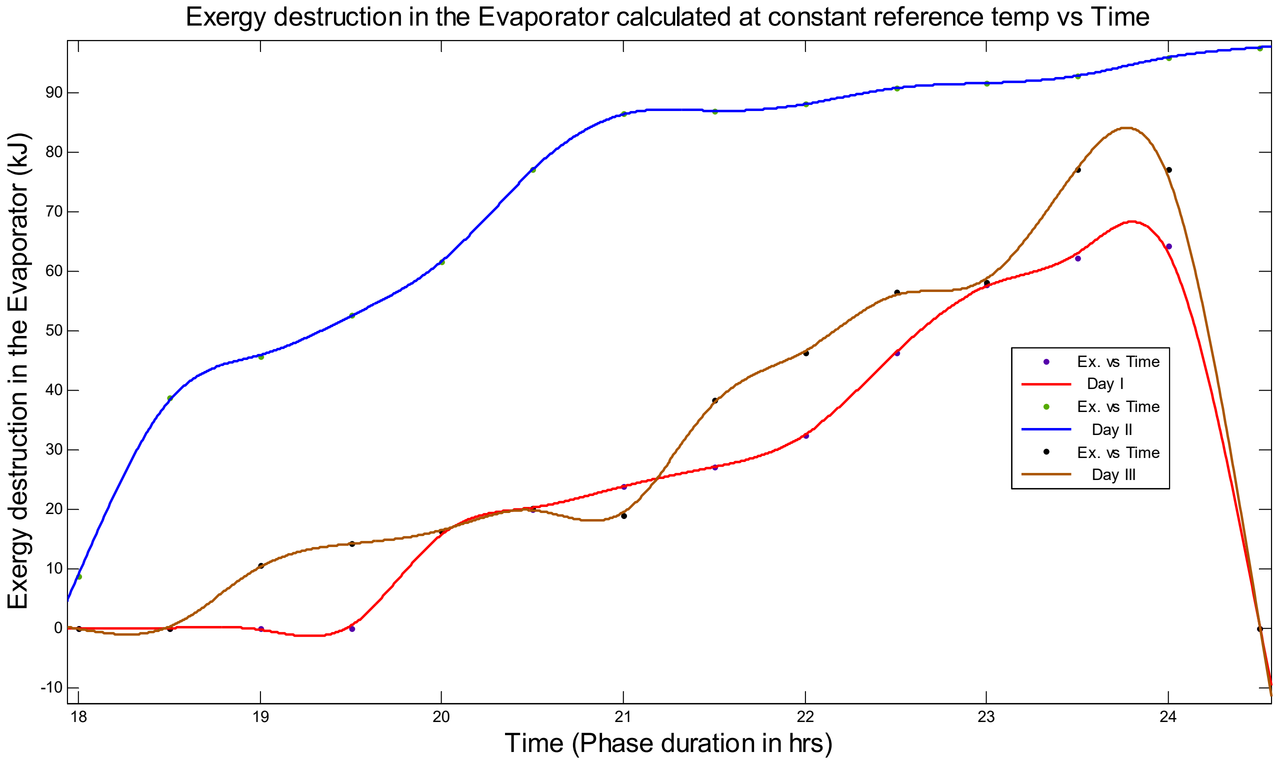


Fig 4.12 Variations of Exergy destroyed during the refrigeration phase with time of occurrence, for constant reference temperature analysis, at 301K.

4.2 DISCUSSION OF RESULTS.

For detailed analysis and discussions, the Energy-Exergy based Performance Analysis of the Solid Adsorption Solar Refrigerator was categorized into the sub-processes making up the refrigeration cycle, which are Isosteric heating of the adsorbent/adsorbent mass; Desorption of the adsorbed refrigerant from the outer pores of the adsorbent; Isosteric cooling of the remaining adsorbent/adsorbent mass in the generator/adsorber that do not take part in the desorption process; Re-adsorption of the used refrigerant mass and for the Condenser and Evaporator, a quasi-static regime of operation was considered (with no internal energy and entropy generation). These sub-processes are discussed below from the results of the analysis made:

4.2.1 Isosteric Heat-up Phase

The estimated Exergy destroyed over the isosteric heat-up phase are presented for varying and constant reference temperatures respectively in Tables 4.1 and 4.7. Their variations with time are shown in Figures 4.1 and 4.7 for varying and constant reference temperature results obtained. From the commencement of the heat-up of the generator/adsorber, exergy destructions of these range are experienced: 0.000 – 3.362 kJ; 0.000 – 2.720 kJ and 0.000 – 5.516 kJ for test days I, II and III at varying reference temperature analysis and 0.000 – 3.328 kJ; 0.000 – 2.720 kJ and 0.000 – 5.533 kJ for test days I, II and III at constant reference temperature analysis.

From Figs 4.1 and 4.7, the exergy destroyed is seen to increase as the heat source temperature increases due to high irreversibility associated with adsorption. The heat from the heat source is responsible for heating up the adsorbent/adsorbate pair, making the molecules of the adsorbate volatile enough to be adsorbed onto the pores of the adsorbent, all these occurring more at the liquid phase of the adsorbate. From the values of Exergy obtained during the Isosteric heat up of the adsorbent/adsorbate pair over the three test days, at both reference temperature analysis models chosen, noticeable increase in value of heat lost is experienced as the change in temperature over the time intervals recorded increases, peaking between 09:30 and 10:00 hrs for all test days.

4.2.2 Desorption Phase

The estimated Exergy destroyed over the desorption phase are presented in Tables 4.2 and 4.8. Their variations with time are also shown in Figures 4.2 and 4.8 for varying and constant reference temperature values respectively. The effect of the isosteric heat of desorption - being heat needed to vaporize the adsorbate from the pores of the adsorbent; obtained from the heat source, is responsible for breaking off the adsorbed molecules of the adsorbate from the adsorbent. Desorption occurs when the adsorbate attains the needed heat of desorption, during which the exergy loss rises and peaks at the highest ambient temperature. This Exergy value drops from the peak over the desorption period and for the sake of the refrigeration cycle, focus shifts from the desorption process when refrigerant mass accumulation in the condenser ceases. The high exergy losses which are experienced from the point where peak daily temperatures sets in for days I, II and III are seen to range from: 302.059 – 2874.119 kJ; 144.778 – 3249.376 kJ and 574.044 – 2999.723 kJ, for test days I, II and III for analysis based on varying reference temperatures and 292.510 – 2736.713 kJ; 141.392 – 3190.508 kJ and 565.080 – 2912.175 kJ for test days I, II and III for analysis based on constant reference temperature of 301K.

It is evident that much heat is lost during desorption, resulting in the formation of the refrigerant gas. The results indicate that the operational parameters and the material properties of the collector/generator/adsorber be optimized to reduce exergy destruction during refrigerant mass formation and improve exergy efficiency of the entire system in turn. This is so because the magnitude of exergy destruction for the entire system is highest during the Desorption phase.

4.2.3 Isosteric Cool-down Phase

The estimated Exergy destroyed over the Isosteric cool-down phase are presented in Tables 4.3 and 4.9. Their variations with time are also shown in Figures 4.3 and 4.9 for varying and constant reference temperature analysis respectively. The ranges of exergy destruction during the isosteric cool-down phase for varying reference temperature analysis are: 0.249 – 8.357 kJ; 0.350 – 8.737 kJ and 0.007 – 5.983 kJ, for test days I, II and III respectively, and for constant reference temperature analysis as: 0.248 – 8.248 kJ; 0.352 – 8.651 kJ and 0.008 – 5.880 kJ, for test days I, II and III respectively.

The Exergy destruction during the Isosteric cooling of the adsorbent/adsorbate mass remaining after desorption is here expressed, this value should be optimized to allow for effective re-adsorption of the used up refrigerant during refrigeration, as high heat content over the Isosteric cool-down process, will lead to irreversibilities during the re-adsorption phase.

4.2.4 Re-adsorption Phase

The estimated Exergy destroyed over the Re-adsorption phase are presented in Tables 4.4 and 4.10. Their variations with time are also shown in Figures 4.4 and 4.10 for varying and constant reference temperature values respectively. The ranges of exergy destroyed over the re-adsorption phase, for varying reference temperature analysis are 0.000 – 32.183 kJ; 0.000 – 281.807 kJ and 0.000 – 84.715 kJ for test days I, II and III and that for constant reference temperature analysis as 55.327 – 128.443 kJ; 86.251 – 193.905 kJ and 0.000 – 124.224 kJ for test days I, II and III.

The destruction of Exergy is as a result of irreversibilities associated with the Re-adsorption of the adsorbate by the adsorbent after the adsorbate has been vaporized by the heat from the chamber which needed to be cooled. It is worthy to note that heat is expelled when the adsorbent adsorbs molecules of the adsorbate into its micro-pores – isosteric heat of re-adsorption.

4.2.5 Condenser

The estimated Exergy destroyed over the condensation phase of the generated refrigerant are presented in Tables 4.5 and 4.11. Their variations with time are also shown in Figures 4.5 and 4.11, for varying and constant reference temperature results respectively. The ranges of the values of exergy destruction over this phase are 1.746 – 21.458 kJ; 0.790 – 16.706 kJ and 2.809 – 17.275 kJ for analysis based on varying reference temperature for test days I, II and III. For constant reference temperature analysis the ranges of values for test days I, II and III are 0.524 – 3.353 kJ; 0.347 – 8.975 kJ and 1.641 – 5.918 kJ respectively.

The exergy destruction in the condenser depends upon several parameters such as the cooling medium- water in this case, the cooling water temperature, the condensation temperature and the internal irreversibilities during the condensation of the refrigerant vapour. The source of the internal irreversibility in the condenser is the de-superheating of the refrigerant vapour from the desorption temperature to the condenser temperature. As the heat source temperature increases, it increases the amount of the refrigerant being desorbed from the generator/adsorber. The condensation of more refrigerant releases more heat, which leaves the condenser needing more cold water to cool it down. Thus, the higher amount of heat transfer, as well as the increased temperature difference between the cooling water and the refrigerant vapour, leads to increased irreversibility in the condenser. Hence for a fixed heat transfer area and known condensation temperatures, the optimization of cooling water temperatures is necessary to minimize the exergy destruction and increase the exergy efficiency of the condenser and the system at large.

4.2.6 Evaporator

The estimated Exergy destroyed in the evaporator are presented in Tables 4.6 and 4.12. Their variations with time are also shown in Figures 4.6 and 4.12, for varying and constant reference temperature analysis respectively. The ranges of the values of exergy destruction during this phase are 6.868 – 41.350 kJ; 1.344 – 62.880 kJ and 1.515 – 52.250 kJ for analysis based on variable reference temperature for test days I, II and III. For constant reference temperature analysis the ranges of values for test days I, II and III are 16.234 – 64.322 kJ; 8.738 – 97.613 kJ and 10.514 – 77.084 kJ.

From Figs 4.6 and 4.12 the exergy destroyed during refrigeration increases gradually as the already condensed refrigerant fluid vaporizes, since it gains heat from the chamber to be cooled. The residual heat (withdrawn late into the hours of the day from the cooled chamber) results in irreversibilities as a result of the phase change process of the refrigerant in the evaporator- from liquid to gaseous. The vicious pressure drop in the coils of the evaporator causes pressure variations throughout the evaporator, leading to exergy destructions.

4.3 Overall System Performance

The calculated exergy efficiency of the system obtained at varying reference temperature analysis for test days I, II and III are 0.01331, 0.04380 and 0.01226 respectively. That obtained for constant reference temperature analysis for test days I, II and III are 0.02275, 0.05933 and 0.02162. It is worthy to note here that the exergy destroyed during desorption and re-adsorption phases are of significant considerations when discussing the system performance, since exergy destructions of the highest magnitudes are experienced during these phases.

Although refrigerant mass generation starts out late for day II at 12:30 hrs, unlike for days I and III that starts out about 11:30 hrs, the highest condensate yield is achieved during the day II test run and in turn refrigeration starts out earlier at 18:00 hrs unlike for days I and III which starts out at about 20:00 hrs and 19:00 hrs respectively. In addition, since the temperature at the entrance of the evaporator during test day II is almost equivalent to the temperature of the refrigerant at the exit of the condenser, no flash off is experienced implying that the refrigerant fluid stream enters the evaporator, and effective cooling of the refrigeration chamber is experienced, since the lowest refrigerator water temperature is documented for the second test day. Unlike during test days I and II where due to difference in the condenser exit and evaporator entry temperatures, flashing of the refrigerant stream occur, which no doubt reduces the exergy efficiency of the system, since the calculated exergy efficiency value of the refrigeration system is hinged on the performance of the evaporator.

In the case of the Solid Adsorption Solar Refrigerator analyzed, the COP depends strongly on the adsorption and condensation temperatures but weakly on the evaporator temperature. Anyanwu and Ogueke 2001. From the exergy analysis carried out for the Solid Adsorption Solar Refrigerator, the values shown in Appendix E 1 and 2, indicate clearly that exergy destruction is of its highest magnitude during the Desorption and Re-adsorption phases. Therefore an optimization of the desorption and optimization temperatures of the generator/adsorber or a choice of generator/adsorber with less heat emission during desorption and re-adsorption, will improve the system performance of the solar refrigerator from the Exergy perspective.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

The Energy-Exergy Based Performance Analysis of the Solid Adsorption Solar Refrigerator has been undertaken in this study. The exergy tool was found preferable to the energy analysis, as losses were identified and properly quantified for all the sub-processes which made up the refrigeration cycle, and applied in estimating the exergy efficiency of the refrigeration system for test days I, II and III.

The exergy efficiencies calculated from the analysis based varying reference temperature are 0.01331, 0.04380 and 0.01226 for test days I, II and III respectively. Exergy efficiency values for constant reference temperature analysis are 0.02275, 0.05933 and 0.02162 for test days I, II and III respectively.

The exergy efficiency- which is always of a lower value when compared to the energy efficiency value (COP) for any energy system- provides the efficiency value which is most reliable in evaluating and upgrading the performance of the system considered. It is evident that the true picture of efficiency is obtained by exergy analysis, as it overcomes the limitations of the First Law of Thermodynamics on which the energy efficiency is based.

The exergy performance analysis is based both on the First and Second Law of Thermodynamics and its phase by phase analysis clearly indicates the location of exergy destruction, quantifying and qualifying losses, providing a sure basis for improvement of the system and for innovations as well on the technology, since the areas of exergy losses are easily identified with the generated exergy equations.

Research and development efforts on the improvement of the overall efficiency of the system are best done using the exergy efficiency as a basis for judgment, since heat losses unaccounted for by energy analysis are identified, quantified, qualified and utilized as the basis for exergy analysis and exergy efficiency calculations. Research works geared towards the reduction of exergy losses during desorption and re-adsorption phases will well improve the overall efficiency of the system. A feat energy analysis cannot address.

5.2 RECOMMENDATIONS

For future upgrades of this model of Solid Adsorption Solar Refrigerator, efforts should be centered at making a choice of collector/generator/adsorber that does not give off such magnitude of heat during desorption and re-adsorption of the adsorbate. Tables 4.13 and 4.14 indicate values of total exergy destructions for all processes as they occur. It is obvious from these tables that the desorption phase - for varying and constant reference temperature analysis, pose the highest process of exergy destruction. Adsorbers of better designs that would utilize the heat generated from the collector in more desorption of the adsorbate would suffice.

5.3 ACHIEVEMMENTS/ADDITION TO KNOWLEDGE

- The exergy efficiency equation of the refrigeration cycle which comprised the various sub-processes of the system considered was generated, taking into account the detailed exergy destructions.
- The values of Exergy destroyed, for all sub-processes which make up the complete refrigeration cycle, has been deduced and the values tabulated in Tables 4.13 for analysis bases on varying dead state reference temperature and Table 4.14 for analysis based on a fixed reference temperature value.
- The Exergy Profiles for all analyzed sub-processes are presented in Figs 4.1 through 4.12 for both varying and constant reference temperatures.

- The exergy efficiency, which is the closest efficiency value to the actual efficiency of the system has been deduced,

- The effects of adsorption and re-adsorption temperatures on the performance of the adsorption refrigerator, has been underscored as the optimal performance parameters on which improvements carried upon will improve the exergy efficiency of the system.

- This work is based on results obtained from a real system, unlike other analysis of solid adsorption refrigeration, which are purely empirical.

- The varying reference temperature model chosen as the reference temperature for the Exergy Analysis is novel, as the values of exergy destruction, obtained at every point considered with reference to the ambient environmental temperature at that particular time, gave exergy destructions closest to the actual value of exergy loss at that point of analysis. This goes to show that exergy efficiency values thus calculated, for all three test days, were closest to the actual exergy efficiency of the system. Unlike other research works done on this area of study, where the constant reference (average) reference environmental temperature was adopted, which gave noticeable deviations from the actual value sought. Compare Exergy Efficiency values in Table E 1 and E 2, of Appendix E.

REFERENCES

Ahrendts, Joachim. (1980). Reference States. Energy. Volume 5, Issues 8-9. August-September 1989, pages 667-668. Second Law Analysis of Energy Devices and Processes. Elsevier Publishers.

Aittomaecki, A. and Haerkoenen, M. (1992). Internal Regeneration of the Adsorption Process. The Solid Sorption Refrigeration Symposium, Paris, France Nov. 18-20, 1992.

Anyanwu, E. E. and Ogueke, N. V. (2001). Activated Carbon/Refrigerant Combinations for Solar Refrigeration Applications: Nigerian Journal of Technology, Vol. 20. No. 1.

Anyanwu, E. E. (2004). Review of Solid Adsorption Solar Refrigeration II: an Overview of the Principles and Theory. Energy Converse Manage 45; 1279-1295.

Anyanwu, E. E.; Oteh, U. U. and Ogueke, N. V. (2001). Simulation of a Solid Adsorption Solar Refrigerator, Using Activated Carbon/Methanol adsorbent/adsorbate pair. Energy Converse Manage 2001; 42: 899-915.

Anyanwu, E. E. and Ogueke, N. V. (2005). Thermodynamic Design Procedure for Solid Adsorption Solar Refrigeration; Renewable Energy 30, (2005), 81-96.

Bogomolov, P. A.; Prokof, E. A. A.; Jachishin, D. A. S.; Chalaev, H. D. M.; Tolstykhigor, P. and Shavrin, V. S. (2001). Jiao Tong University, Shanghai, China. Renewable Sustainable Energy. Rev 2001; 5:1-37.

Bratton, G. J.; Buck, K. D. and Naylor, T. (2001). Patent Reports/Microporous and Mesoporous Materials 42 (2001) 355 – 368.

Cengel, Y. A. and Boles, M. A. (2008). Thermodynamic, an Engineering Approach; (6 Ed.) p. 445. ISBN 0-07-125771-3.

Chua, H. T.; Ng, K. C.; Malek, A.; Kashiwagi, T.; Akisawa, A. and Saha, B. B. (2001). Multi-bed regenerative adsorption chiller - improving the utilization of waste heat and reducing the chilled water outlet temperature fluctuation.

International Journal of Refrigeration. Volume 24, Issue 2. March, 2001. p. 124-136

Critoph, R. E. and Thorpe, R. (1996). Momentum and Heat Transfer by Forced Convection in fixed beds of granular active carbon. Applied Thermal Engineering. Vol. 16, Issue 5. May, 1996. p. 419 - 427.

Critoph, R. E. (2001). Simulation of a Continuous multiple-bed Regenerative Adsorption Cycle. International Journal of Refrigeration. Vol. 24, Issue 5, August, 2001. p. 428-437

Critoph R. E. (2002). Multiple bed Regenerative Adsorption Cycle Using the Monolithic Carbon–Ammonia Pair. Applied Thermal Engineering. Vol. 22. Issue 6. April, 2002. p. 667 - 677

Critoph, R. E. and Tamainot-Telto, Z. (2003). Advanced Solid Sorption Air Conditioning Modules Using Monolithic Carbon - Ammonia Pair. Applied Thermal Engineering. Vol. 23. Issue 6. April, 2003. p. 659-674

Critoph, R. E. and Metcalf, S. J. (2004) – Specific cooling power intensification limits in ammonia-carbon adsorption refrigeration systems. Applied Thermal Engineering. Elsevier publications, volume 24, issue 5-6. April 2004, pages 661 – 678.

Duffie, J. A. and Beckman, W. A. (1999). Solar Energy Thermal Processes; NY. John Wiley. 1991.

Elena, Eugenia Vasilescu; Rahal, Boussehain; Michel Feidt and Alexandru, Dobrovicescu. (2007). “Energy and Exergy Optimization of the Adsorption Refrigeration Machine with Simple and Double Effect’’. TERMOTEHNICA Jan. 2007.

Endo, K. and Nishikawa, K. (2003). Adsorption Refrigeration Machine. Published by SumoBrain Solution Company. June 2003. Assignee: DENSO CORPORATION. Japanese Patent: JP2003166770.

Endo, A. and Komori, A. (2005). Adsorption Type Refrigeration Machine and its Operating Method. Published by SumoBrain Solution Company. May

2005. Assignee: National Institute of Advanced Industry and Technology - Kobe Steel. Japanese Patent: JP2005127614.

Ferrari E.; Rispoli, F. and Sciubba, E. (2010). ‘’Interactions of cement model systems with super plasticizers investigated by atomic force microscopy, zeta potential and adsorption measurements’’. J. Colloid Interface Sci. 347(1): 15-24.

Gibbs, J. W. (1873). “A Method of geometrical representation of thermodynamic properties of substances by means of surfaces”. Reprinted in Gibbs; Collected works, ed. W. R. Longley and R. G. Van Name (New York;

Guilleminot, J. J.; Meunier, F. and Rakleza, J. (1987). Heat and Mass Transfer in a non-isothermal fixed bed Solid Adsorbent Reactor, a uniform pressure-non uniform temperature case. Int. J. Heat and Mass Transfer, 1987; 30(8): 1595- 606.

Harper, Douglas. (2007). “Energy defined’’. Urban - Online Etymology Dictionary. First published Dec. 12, 2007. By Douglas Harper Corp.

Honda, S.; Ishii, K.; Tomoari, K. and Suzuki, T. (1999). Refrigerating Equipment and Air Conditioner. Published by SumoBrain Solution Company. Mar. 1999. Assignee: DENSO CORP. Japanese Patent: JP11083235

Hidaka, H.; Kakiuchi, H.; Iwade, Y.; Takewaki, T.; Yamazaki, M. and Watanabe, N. (2005). Adsorption type Cooler. Published by SumoBrain Solution Company. Apr. 2005. Assignee: MITSUBISHI CHEMICAL CORPORATION. Japanese Patent: JP2005098647.

Ibrahim D. and Marc, A. R. (2007). Exergy: Energy, Environment and Sustainable Development. Elsevier publishers 2007. First edition.

Inaoka, Hiroya. (2005). Adsorption Type Heat Pump. Published by SumoBrain Solution Company. May 2005. Assignee: TOYOTA MOTOR CORPORATION. Japanese Patent: JP2005127632.

Inoue, S. and Honda, S. (1999). Refrigerating Unit. Published by SumoBrain Solution Company. Aug. 1999. Assignee: DENSO CORPORATION. Japanese Patent: JP11211261.

Inoue, S. and Sato, H. (1999). Production of Adsorption core for Adsorption Refrigeration Machine. Published by SumoBrain Solution Company. June 1999. Assignee: DENSO CORP. Japanese Patent: JP11166772.

Kakiuchi, H.; Yamamoto, H.; Takewaki, T.; Yamazaki, M. and Watanabe, N.; Watanabe, K. (2005): Heat Pump with Adsorption type Heating Device. Published by SumoBrain Solution Company. June 2005. Assignee: MITSUBISHI CHEMICALS CORP.; TOKYO ELECTRIC POWER CO. INC: THE. Japanese Patent: JP2005147550.

Kakiuchi, H.; Takewaki, T.; Fujii, M.; Yamazaki, M.; Takumi, H.; Watanabe, H; Inagaki, K.; Kosaka, A.; Inoue, S. and Inoue, S. (2009). Adsorption Heat pump and use of Adsorbent as Adsorbent for Adsorption Heat Pump. Published by FPO IP Research and Communities. Mar. 2009. Assignee: Mitsubishi Chemical Corporation (Tokyo, JP), Denso Corp (Kariya-shi, JP). United State Patent: US7497089.

Kan, C. A. A.; Akahira, A., Hamamoto, Y., Akisawa, A. and Kashiwagi, T. (2005). Adsorption Refrigerator. Published by SumoBrain Solution Company; Apr. 2005. Assignee: AKISAWA ATSUSHI HAMAMOTO YOSHINORI. Japanese Patent: JP2004116812.

Klotz I. and Rosenberg, R. (2008). Chemical Thermodynamics - Basic Concepts and Methods, Seventh Ed., Wiley (2008), p.39.

Li, Yong and Ruzhu Z. Wang (2007). Adsorption Refrigeration: A Survey of Novel Technologies. Recent Patents on Engineering: 1; 1-21.

Lofts, G. and O’ Keffe, D. (2004). Mechanical Interactions. Jacaranda Physics 1 (Second Ed.). Milton; Queensland, Australia: John Wiley & Sons Australia ltd. Pg 286, ISBN 0-7016-37773.

Longmans, Green. (1931). Transactions of the Connecticut Academy of Arts and Sciences 2: 382- 404.

Maier-Laxhuber, P.; Becky, A.; Schmitt, R.; Woerz, R. and Weinzierl, N. (2004). Adsorption Cooling Device with Buffer Material Storage Device and Method for Operating Adsorption Cooling Devices. Published by SumoBrain Solution Company. May 2004. Assignee: ZEO-TECH ZEOLITH-TECHNOLOGIE GMBH. Japanese Patent: JP2004150792.

Maier-Laxhuber, P. and Engelhardt, R. (1996). Adsorbent Bed Coating on Metals and Processing for making the same. Published by the United State Patent and Trademark Office. Dec. 1996. Assignee: Zeo-Tech GMBH. United States Patent: US5585145.

McMurry, John (2003). Fundamentals of Organic Chemistry (Fifth ed.). Agnus McDonald, pp. 409; ISBN 0-534-39573-2.

Mochizuki, T. and Yamazaki, K.; Hongo, I.; (2004). Adsorber of Adsorption Refrigerator. Published by SumoBrain Solution Company. Nov. 2004. Assignee: TOSHIBA KYARIA KK. Japanese Patent: JP2004317011.

Monma, T. and Mizota, T. (2005). Adsorption type Refrigerator. Published by SumoBrain Solution Company. Oct. 2005. Assignee: SHARP CORP. AND YAMAGUCHI TECHNOLOGY LICENSING ORGANISATION LTD. Japanese Patent JP 2005299974.

Miyairi, Yoshio; Hattori, Toshio; Tsubaki, Yasuhiro and Katajima Kazuo. (1996). Coating of absorbing and desorbing agent, absorbing and desorbing agent element and adsorption type refrigerator unit. Published by SOOPAT: 1996-05-21. Assignee Mitsubishi Heavy Industries Limited. Japanese patent JP8126841.

Nagashima, H., Otobe, J. and Ishii, K. (2006). Method to prevent hydrogen production in a sorption refrigeration machine. Published by SumoBrain Solution Company. June (2006). Assignee: Denso Coporation.

Nic. W.; Elan, G. and Brand, F. (2006). ''Ionization Energy''. IUPAC Compendium of Chemical Terminology (Online Ed.). ISBN 0-9678550-9-8.

Okumura, H.; Kumura, H.; Tanaka, Y.; Kamiya, Y. and Inoue, S. (2004). Temperature Control System in Adsorption Refrigeration Machines.

Published by SumoBrain Solution Company; Aug. 2004. Assignee; DENSO CORP. Japanese Patent: JP2004239593.

Patzner, N. (2001). Refrigeration method and device – Evaporative Coolers. Published by Directory Patents. Nov. 2001. Assignee: Firma H+P Technologie; Gese Uschaft fur Kuhlsysteme. Copyright 2012.

Perrot, Pierre (1998). A to Z of Thermodynamics. Oxford University Press. ISBN 0-19-856552-6.

Pons, M. and Guilleminot, J. J. (1986). Design of an experimental solar-powered solid adsorption ice maker. Research organization: Laboratoire de Thermodynamique des Fluides, Campus Universitaire, Bat. 502 TER. 91405 Orsay Cedex. Journal name: J. Sol. Energy (United States); Vol. 108:4. Pgs 332-337. Reviewed 2009.

Pons, M.; Meunier, F.; Cacciola, G.; Critoph, R. E.; Groll, M. and Puigjaner, L. (1999). Thermodynamic based Comparison of Sorption Systems for Cooling and Heat Pumping. Int. J. Refrig. 1999; 22: 5-17.

Rant, Z. (1956). ‘’Exergie, ein neues wort fur’ Technische Arbeitsfahigkeit’’ (Exergy, a new word for ‘’technical available work’’). Forschung auf dem Gebiete des Ingenieurwesens 22: 36 - 37.

Rosser, Jr. F. and Dunne, S. R. (1998). Process for Heat Recovery in a Sorption Refrigeration System. Published by the United State Patent and Trademark Office; Oct. 1998. Assignee: UOP LLC.

Sato, H.; Tanaka, K.; Honda, S.; Fujiwara, K. and Tanaka, Y. (1998). Adsorptive-type regenerative apparatus. Posted by the United States Patent Trademark Office (USPTO). Jul. 1998. Assignee: DENSO CORPORATION. United States Patent: US5775126.

Sato, H.; Ishii, K.; Nagashima, H.; Tanaka, Y. and Inoue, S. (2001). Adsorption type Refrigerator. Published by SumoBrain Solution Company. Aug. 2001. Assignee: DENSO CORPORATION. Japanese Patent: JP2001215068.

Sato, Mki.; J. Hansen; D. Koc; A. Lacis; R. Ruedy; O. Dubovik; B. Holben; M. Chin, and T. Novakov (2003). Global atmospheric black carbon inferred from AERONET. *Proc. Natl. Acad. Sci.*, 100, 6319-6324, doi:10.1073/pnas.0731897100.

Shelton, S. V. and Miles, D. J. (1996). Design and testing of a Solid - Sorption Heat-Pump System. Applied Thermal Engineering. Vol. 16. Issue 5. May, 1996. p. 389-394

Shinoda, T.; Kawakubo, M. and Matsuo, H. (1999). Adsorption Refrigeration Machine. Published by SumoBrain Solution Company; Jul. 1999. Assignee DENSOR CORPORATION. Japanese Patent: JP11201581.

Szargut, J. (1967). Exergy analysis of thermal processes. Journal of Power Technologies – 1996 – papers.itc.pw.edu.pl. Egzergia jest wielkością termodynamiczną, nie zaś ekonomiczną (Szargut 1967). ... Brennstoff--Wärme-Kraft 19 (1967) nr 7 - 8. [21] J. Szargut: Standard Chemical Exergy of some Elements and their Compounds, Based upon the Concentration in Earth's Crust. Bull. Pol. Acad.

Tchernev, D. I. (1989). Regenerative zeolite heat pump. Studies in Surface Science and Catalysis. Volume 49. 1989. Pg 519 - 526.

Wang, L. W.; Wang, R. Z.; Wu, J. Y.; Xia, Z. Z. and Wang, K. (2005). A new type adsorber for adsorption ice maker on fishing boats. Energy Conversion and Management. Vol. 46, Issue 13 - 14. Aug. 2005. p. 2301 - 2316

Wang R. Z. and R. G. Oliviera; (2005). Adsorption Refrigeration, an Efficient Way to Make Good Use of Waste Heat and Solar Energy. International Sorption Heat Pump Conference: 101K.

Wepfer, J. William and Gaggioli A. Richard. (1980). Thermodynamics: Second Law Analysis. Chapter 5, pp 77 – 92. ACS Symposium Series, vol. 122. ISBN 13: 9780841205413, eISBN: 9780841206878. Publication date (Print): May 28, 1980.

Xia, Z. Z.; Wang, D. C.; Wu, J. Y.; Wang, R. Z.; Zhai, H. and Dou, W. D. (2005). Study of a novel silica gel-water adsorption chiller. Design and Performance Prediction. Part I. International Journal of Refrigeration. Vol. 28. Issue 7. November, 2005. p. 1073 - 1083.

Xia, Z. Z.; Wang, D. C.; Wu, J. Y.; Wang, R. Z.; Zhai, H. and Dou, W. D. (2005). Study of a novel silica gel–water adsorption chiller. Part II. Design and Performance Prediction. International Journal of Refrigeration. Vol. 28. Issue 7. November, 2005. p. 1084 - 1091.

APPENDIX A

MEASURED AMBIENT TEMPERATURE VALUES AND THEIR CORRESPONDING DAILY AVERAGE WIND VELOCITY VALUES

Table A 1: Measured ambient temperatures and wind velocity

	Day I	Day II	Day III
Daily Average wind velocity (m/s)	1.3	0.8	1.1
Local Time (hrs)	Ambient Temperature (°C)	Ambient Temperature (°C)	Ambient Temperature (°C)
08:00	23	24	23.6
08:30	23	24.2	24.8
09:00	23.8	25	25
09:30	26	26	26
10:00	27.8	28	27.5
10:30	28	29.6	28
11:00	29	29.4	28.8
11:30	32.2	30	30
12:00	32.4	30	31.2
12:30	31.9	31	31
13:00	32.9	32	30.8
13:30	33	33	31.2
14:00	33.2	33	31.8
14:30	33.4	32	33
15:00	33	32	32
15:30	34	32	32
16:00	32	30.3	31.8
16:30	32	31	31
17:00	31	30.7	30.2
17:30	30	28.3	29
18:00	29	25	26
18:30	28	23	24
19:00	27	22	23
19:30	26	21.6	22.7
20:00	25	21.5	22
20:30	24	21.5	21.4
21:00	23	20	21.4

21:30	23	20	21.4
22:00	22	20	21.5
22:30	22	20.5	21.2
23:00	21	19.5	20.5
23:30	21	20	21
00:00	21	19	20.3
00:30	21	18.5	20
01:00	21	18.5	20
01:30	21	18.5	20
02:00	20.5	18.5	20
02:30	21.2	19	20.3
03:00	21	19	20.5
03:30	20	19.3	20.8
04:00	20.9	19.5	20.4
04:30	20.7	19.2	20.5
05:00	20	19	20.1
05:30	20.1	19.6	20.8

APPENDIX B

MEASURED REFRIGERATOR PERFORMANCE

Table B 1: Measured Collector Plate Temperatures

	Day I	Day II	Day III
Local time (hrs)	Plate Temperature (°C)	Plate Temperature (°C)	Plate Temperature (°C)
08:00	32.4	29.7	28.9
08:30	32.4	37.4	42.4
09:00	36.9	44.7	46.1
09:30	45.3	56.9	63.5
10:00	59.1	58.4	74
10:30	71.9	81.6	79.4
11:00	81.7	81.4	85.4
11:30	91.1	89.5	94.2
12:00	99.8	94.1	98.8
12:30	93.8	97.7	99.4
13:00	102.5	98.3	92.6
13:30	108.8	99.2	94
14:00	107.2	99.8	102.4
14:30	101.2	99.7	93.3
15:00	98.7	90.2	91.4
15:30	98.4	90.4	86.2
16:00	81	78.6	79.7
16:30	57	48.2	52.8
17:00	49.1	42.5	46.4
17:30	45.1	31.5	41.2
18:00	40.9	30.2	27.1
18:30	35.8	30.8	25.4
19:00	31.2	22	24.6
19:30	25.8	22	23.2
20:00	23.4	22.1	23
20:30	23	21	22.6
21:00	22.3	20	22
21:30	22.3	20	22
22:00	22.3	20	22
22:30	22	20	21.8
23:00	22	20	21.6
23:30	20.2	20	21
00:00	20.2	20	20.7
00:30	20.2	20	20.6

01:00	20.2	20	20.6
01:30	20.2	20	20
02:00	20.2	20	20
02:30	20.2	20	20
03:00	20.3	20	20
03:30	20.4	20	20

Table B 2: Measured Outer Tube Surface Temperature.

	Day I	Day II	Day III
Local Time (hrs)	Outer tube Temperatures (°C)	Outer tube Temperatures (°C)	Outer tube Temperatures (°C)
08:00	19.9	28.4	24.6
08:30	19.9	29.6	30.3
09:00	28.3	33.5	33
09:30	39.9	43.6	48.2
10:00	45.1	54.5	55.2
10:30	53.7	63.9	63.2
11:00	63.1	66.7	66.5
11:30	71.7	70.7	76.8
12:00	81.6	77.5	83.1
12:30	83.7	78.5	86.1
13:00	85	88.9	82.4
13:30	92.7	90.1	84.1
14:00	93.5	92.5	91.6
14:30	95.5	90.2	84
15:00	95.3	89.6	83.2
15:30	90	86.5	81.5
16:00	82.6	79.7	76.2
16:30	60.9	57.5	58
17:00	51.6	47.6	48.8
17:30	45.8	43.4	44
18:00	41.8	39.9	30.6
18:30	37	34	30
19:00	33.6	24	28
19:30	27.7	23	27
20:00	24.2	22.6	24
20:30	24	21	23
21:00	23	20	22
21:30	22	20	22
22:00	20	20	22
22:30	20	20	21
23:00	20	20	20.9
23:30	20	20	20.8
00:00	20	20	20.1
00:30	20	20	20.1
01:00	20	20	20.1
01:30	20	20	20.1
02:00	20	20	20
02:30	20	20	20
03:00	20	20	20
03:30	20	20	20

Table B 3: Measured Condensate Yield

	Day I	Day II	Day III
Local Time (hrs)	Condensate Yield (kg)	Condensate Yield (kg)	Condensate Yield (kg)
08:00	0	0	0
08:30	0	0	0
09:00	0	0	0
09:30	0	0	0
10:00	0	0	0
10:30	0	0	0
11:00	0	0	0
11:30	0.079	0	0.158
12:00	0.158	0	0.277
12:30	0.356	0.040	0.356
13:00	0.356	0.119	0.475
13:30	0.395	0.316	0.514
14:00	0.593	0.395	0.554
14:30	0.633	0.554	0.554
15:00	0.633	0.593	0.672
15:30	0.838	0.751	0.712
16:00		0.910	0.811

Table B 4: Measured Evaporator Water Temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Evaporator Water Temperature (°C)	Evaporator Water Temperature (°C)	Evaporator Water Temperature (°C)
18:00		25.4	
18:30		16.8	
19:00		14.9	24.5
19:30		13	23.3
20:00	22.8	10.6	22.6
20:30	21.6	6.5	22.3
21:00	20.4	4.1	21.9
21:30	19.4	4	15.6
22:00	17.8	3.7	13.2
22:30	13.6	3	10.1
23:00	10.3	2.8	9.6
23:30	9	2.5	4.1
00:00	8.4	1.7	4.1
00:30		1.3	
01:00			
01:30			

APPENDIX C

PHYSICAL CHARACTERISTICS OF THE REFRIGERATOR

COLLECTOR DIMENSION

Length (m)	1.6
Width (m)	1.0
Depth (m)	0.19

GLAZING (Glass)

Number	1.0
Thickness (m)	0.005
Extinction Coefficient (m^{-1})	4.0
Refractive Index	1.526
Emmitance	0.88

THE PLATE

Thermal Conductivity (W/mK)	54
Density (kg/m^3)	7833
Specific Heat Capacity (KJ/kgK)	0.465
Thickness (m)	0.00076 (22G)
Absorptance	0.85
Emmitance	0.15

THE BOND

Thermal Conductivity (W/mK)	0.33
Specific Heat Capacity (KJ/kgK)	0.465
Density (kg/m^3)	7850
Bond Thickness (m)	0.004

THE PIPES

Outer Pipe Outer Diameter (m)	0.06
Outer Pipe Inside Diameter (m)	0.052

Inner Pipe Outside Diameter (m)	0.021
Inner Pipe Inside Diameter (m)	0.015
Thermal Conductivity (W/mK)	54
Density (kg/m^3)	7833
Specific Heat Capacity (KJ/kgK)	0.465
Number	6

THE ADSORBENT

Thermal Conductivity (W/mK)	0.22
Specific Heat Capacity (KJ/kgK)	0.92
Packing Density (kg/m^3)	472
Maximum Capacity (m^3 /kg)	0.000425
Heat Transfer Coefficient at Adsorbent/Tube Interface (W/m^2K)	18

THE REFRIGERANT

Specific Heat Capacity (KJ/kgK)	2.91
Isosteric Heat of Desorption (KJ/kg)	4960
Isosteric Heat of Adsorption(KJ/kg)	5610
Latent Heat of Vaporization (KJ/kg)	1102

THE INSULATION

Thickness (m)	0.1
Thermal Conductivity (W/mK)	0.038

THE EVAPORATOR WATER

Mass (kg)	3.0

APPENDIX D

CALCULATED EXERGY DESTROYED FOR EACH SUB - PROCESS OF THE REFRIGERATION CYCLE.

Table D 1. Exergy Destroyed in the Generator/Adsorber during the Isosteric Heat - up Phase. Calculated at varying reference temperature.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{12} (kJ)	Exergy Destroyed Π_{12} (kJ)	Exergy Destroyed Π_{12} (kJ)
08:00	0.000	0.000	0.000
08:30	0.000	0.038	0.850
09:00	1.904	0.388	0.186
09:30	3.362	2.481	5.516
10:00	0.648	2.720	4.129
10:30	1.693	1.915	1.382
11:00	1.920	0.164	0.228
11:30		0.330	
12:00		0.922	

Table D 2. Exergy Destroyed in the Generator/Adsorber during the Desorption Phase. Calculated at varying reference temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{23} (kJ)	Exergy Destroyed Π_{23} (kJ)	Exergy Destroyed Π_{23} (kJ)
11:30	302.059		574.044
12:00	566.268		973.473
12:30	1252.917	144.778	1223.898
13:00	1251.459	4044.682	1671.895
13:30	1319.605	1074.405	1794.109
14:00	1973.428	1321.328	1846.569
14:30	2081.484	1867.210	1962.853
15:00	2077.492	2006.820	2374.920
15:30	2874.119	2595.581	2545.291
16:00		3249.376	2999.723

Table D 3. Exergy destroyed in the Generator/Adsorber during the Isosteric Cool down phase. Calculated at varying reference temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{34} (kJ)	Exergy Destroyed Π_{34} (kJ)	Exergy Destroyed Π_{34} (kJ)
16:00	0.932		
16:30	8.357	8.737	5.933
17:00	1.655	1.888	1.793
17:30	0.670	0.350	0.466
18:00	0.327		3.841
18:30	0.482		0.007
19:00	0.247		
19:30	0.768		

Table D 4. Exergy destroyed in the Generator/Adsorber during the Re-adsorption phase. Calculated at varying reference temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{41} (kJ)	Exergy Destroyed Π_{41} (kJ)	Exergy Destroyed Π_{41} (kJ)
18:00		242.854	
18:30		182.201	
19:00		281.807	84.715
19:30		24.124	73.151
20:00	7.920	18.994	30.435
20:30	0.000	8.739	24.570
21:00	0.023	0.000	9.231
21:30	15.959	0.000	9.254
22:00	32.183	0.000	7.711
22:30	32.090	8.712	3.118
23:00	16.050	8.712	6.192
23:30	16.050	0.000	3.097
00:00	16.050	17.424	3.116
00:30	16.050	26.135	1.552
01:00	16.050	26.135	1.552
01:30	16.050	26.135	1.552
02:00	8.028	26.135	0.000
02:30	19.259	17.424	4.658
03:00	16.050	17.424	7.764
03:30	0.000	12.196	12.422

Table D V - RT. Exergy destruction in the Generator/Adsorber for all processes, for test days I, II and III complete refrigeration cycles.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed kJ	Exergy Destroyed kJ	Exergy Destroyed kJ
08:00	0.000	0.000	0.000
08:30	0.000	0.038	0.850
09:00	1.904	0.388	0.186
09:30	3.362	2.481	5.516
10:00	0.648	2.720	4.129
10:30	1.693	1.915	1.382
11:00	1.920	0.164	0.228
11:30	**302.059**	0.330	**574.044**
12:00	**556.268**	0.922	**973.473**
12:30	**1252.917**	**144.778**	**1223.898**
13:00	**1251.459**	**404.682**	**1671.895**
13:30	**1319.605**	**1074.405**	**1794.109**
14:00	**1973.428**	**1321.328**	**1846.569**
14:30	**2081.484**	**1867.210**	**1962.853**
15:00	**2077.492**	**2006.820**	**2374.920**
15:30	**2874.119**	**2595.581**	**2545.291**
16:00	0.932	**3249.376**	**2999.723**
16:30	8.357	8.737	5.933
17:00	1.655	1.888	1.793
17:30	0.670	0.350	0.466
18:00	0.327	**242.854**	3.841
18:30	0.482	**182.201**	0.007
19:00	0.247	**281.807**	**84.715**
19:30	0.768	**24.124**	**73.151**
20:00	**7.920**	**18.994**	**30.435**
20:30	**0.000**	**8.739**	**24.570**
21:00	**0.023**	**0.000**	**9.231**
21:30	**15.959**	**0.000**	**9.254**
22:00	**32.183**	**0.000**	**7.711**
22:30	**32.090**	**8.712**	**3.118**
23:00	**16.050**	**8.712**	**6.192**
23:30	**16.050**	**0.000**	**3.097**
00:00	**16.050**	**17.424**	**3.116**
00:30	**16.050**	**26.135**	**1.552**
01:00	**16.050**	**26.135**	**1.552**
01:30	**16.050**	**26.135**	**1.552**
02:00	**8.028**	**26.135**	**0.000**
02:30	**19.259**	**17.424**	**4.658**
03:00	**16.050**	**17.424**	**7.764**
03:30	**0.000**	**12.196**	**12.422**

Table D 5. Exergy destroyed in the Condenser during refrigerant mass condensation. Calculated at varying reference temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_c (kJ)	Exergy Destroyed Π_c (kJ)	Exergy Destroyed Π_c (kJ)
11:30	1.746		2.809
12:00	3.555		5.579
12:30	7.661	0.790	7.028
13:00	8.357	2.582	9.187
13:30	9.358	7.486	10.352
14:00	13.826	9.358	11.801
14:30	15.476	12.022	13.135
15:00	14.996	12.868	14.583
15:30	21.485	16.297	15.451
16:00		16.706	17.275

Table D 6. Exergy destroyed in the Evaporator during Refrigeration. Calculated at varying reference temperatures.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{Ev} (kJ)	Exergy Destroyed Π_{Ev} (kJ)	Exergy Destroyed Π_{Ev} (kJ)
18:00	-	1.344	-
18:30	-	21.450	-
19:00	-	24.731	3.009
19:30	-	30.150	1.810
20:00	6.868	38.543	1.814
20:30	7.523	53.819	2.724
21:00	8.138	57.540	1.515
21:30	11.370	57.920	17.961
22:00	13.338	59.075	25.919
22:30	27.066	63.580	35.042
23:00	34.879	60.720	34.471
23:30	39.297	63.700	54.507
00:00	41.350	63.150	52.250
00:30	-	62.880	-

Table D 7. Exergy destroyed in the Generator/Adsorber during the Isosteric Heat-up phase. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{12} (kJ)	Exergy Destroyed Π_{12} (kJ)	Exergy Destroyed Π_{12} (kJ)
08:00	0.000	0.000	0.000
08:30	0.000	0.038	0.859
09:00	1.902	0.392	0.188
09:30	3.384	2.497	5.533
10:00	0.649	2.720	4.136
10:30	1.693	1.905	1.382
11:00	1.914	0.165	0.229
11:30		0.328	
12:00		0.916	

Table D 8. Exergy destroyed in the Generator/Adsorber during the Desorption phase. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{23} (kJ)	Exergy Destroyed Π_{23} (kJ)	Exergy Destroyed Π_{23} (kJ)
11:30	292.510		565.081
12:00	546.819		948.781
12:30	1214.305	141.392	1194.395
13:00	1203.303	391.635	1634.770
13:30	1266.030	1031.244	1748.417
14:00	1889.965	1267.725	1789.290
14:30	1989.466	1806.684	1885.880
15:00	1992.244	1941.926	2300.058
15:30	2736.713	2512.870	2465.596
16:00		3190.508	2912.175

Table D 9. Exergy destroyed in the Generator/Adsorber during the Isosteric cool - down phase. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{34} (kJ)	Exergy Destroyed Π_{34} (kJ)	Exergy Destroyed Π_{34} (kJ)
16:00	0.919		
16:30	8.248	8.651	5.880
17:00	1.638	1.875	1.754
17:30	0.666	0.352	0.462
18:00	0.326		3.866
18:30	0.482		0.008
19:00	0.248		
19:30	0.773		

Table D 10. Exergy destroyed in the Generator/Adsorber during the Re - adsorption phase. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{41} (kJ)	Exergy Destroyed Π_{41} (kJ)	Exergy Destroyed Π_{41} (kJ)
18:00		193.905	
18:30		99.043	
19:00		183.706	0.000
19:30		86.251	15.282
20:00	55.327	93.260	61.057
20:30	63.316	121.607	76.939
21:00	79.501	139.388	92.547
21:30	95.501	139.388	92.536
22:00	128.443	139.388	92.536
22:30	128.365	139.388	108.315
23:00	128.365	139.388	109.898
23:30	128.365	139.388	111.463
00:00	128.365	139.388	122.643
00:30	128.365	139.388	122.630
01:00	128.365	139.388	122.630
01:30	128.365	139.388	122.630
02:00	128.365	139.388	124.224
02:30	128.365	139.388	124.224
03:00	128.365	139.388	124.224
03:30	128.365	139.388	124.224

Table D C - RT. Exergy destruction in the Generator/Adsorber for all processes, for test days I, II and III complete refrigeration cycles.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed kJ	Exergy Destroyed kJ	Exergy Destroyed kJ
08:00	0.000	0.000	0.000
08:30	0.000	0.038	0.859
09:00	1.902	0.392	0.188
09:30	3.384	2.497	5.533
10:00	0.649	2.720	4.136
10:30	1.693	1.905	1.382
11:00	1.914	0.165	0.229
11:30	**292.510**	0.328	**565.081**
12:00	**546.819**	0.916	**948.781**
12:30	**1214.305**	**141.392**	**1194.395**
13:00	**1203.303**	**391.635**	**1634.770**
13:30	**1266.030**	**1031.244**	**1748.417**
14:00	**1889.965**	**1267.725**	**1789.290**
14:30	**1989.466**	**1806.684**	**1885.880**
15:00	**1992.244**	**1941.926**	**2300.058**
15:30	**2736.713**	**2512.870**	**2465.596**
16:00	0.919	**3190.508**	**2912.175**
16:30	8.248	8.651	5.880
17:00	1.638	1.875	1.754
17:30	0.666	0.352	0.462
18:00	0.326	**193.905**	3.866
18:30	0.482	**99.043**	0.008
19:00	0.248	**183.706**	**0.000**
19:30	0.773	**86.251**	**15.282**
20:00	**55.327**	**93.260**	**61.057**
20:30	**63.316**	**121.607**	**76.939**
21:00	**79.501**	**139.388**	**92.547**
21:30	**95.501**	**139.388**	**92.536**
22:00	**128.443**	**139.388**	**92.536**
22:30	**128.365**	**139.388**	**108.315**
23:00	**128.365**	**139.388**	**109.898**
23:30	**128.365**	**139.388**	**111.463**
00:00	**128.365**	**139.388**	**122.643**
00:30	**128.365**	**139.388**	**122.630**
01:00	**128.365**	**139.388**	**122.630**
01:30	**128.365**	**139.388**	**122.630**
02:00	**128.365**	**139.388**	**124.224**
02:30	**128.365**	**139.388**	**124.224**
03:00	**128.365**	**139.388**	**124.224**
03:30	**128.365**	**139.388**	**124.224**

Table D 11. Exergy destroyed in the Condenser during refrigerant mass condensation. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_c (kJ)	Exergy Destroyed Π_c (kJ)	Exergy Destroyed Π_c (kJ)
11:30	0.524		1.641
12:00	0.994		2.309
12:30	2.545	0.347	3.087
13:00	1.939	0.829	4.278
13:30	2.092	1.674	4.285
14:00	2.992	2.092	4.043
14:30	2.909	3.858	2.935
15:00	3.353	4.129	4.679
15:30	3.016	5.229	4.958
16:00		8.975	5.918

Table D 12. Exergy destroyed in the Evaporator during Refrigeration. Calculated at a constant (average) reference temperature of 301 K.

	Day I	Day II	Day III
Local Time (hrs)	Exergy Destroyed Π_{Ev} (kJ)	Exergy Destroyed Π_{Ev} (kJ)	Exergy Destroyed Π_{Ev} (kJ)
18:00	-	8.738	-
18:30	-	38.756	-
19:00	-	45.630	10.514
19:30	-	52.595	14.176
20:00	16.234	61.527	16.326
20:30	20.062	77.140	20.277
21:00	23.921	86.494	18.867
21:30	27.161	86.887	38.400
22:00	32.392	88.068	46.216
22:30	46.399	90.835	56.509
23:00	57.697	91.628	58.190
23:30	62.220	92.820	77.084
00:00	64.322	96.011	77.084
00:30	-	97.613	-

APPENDIX E

ACCRUED EXERGY LOSSES FOR ALL SUB – PROCESSES OF THE REFRIGERATION CYCLE

Table E 1: Sum of Exergy Values for all test days Calculated at Varying Reference Temperatures.

Phases of Exergy Analysis	Day I	Day II	Day III
Isosteric Heating Π_{12}	9.527	8.958	12.291
Desorption Π_{23}	13698.831	12664.180	17966.775
Isosteric Cooling Π_{34}	13.438	10.975	12.040
Re-adsorption Π_{41}	227.812	945.151	284.090
Condenser Π_c	96.460	78.109	107.200
Refrigeration Π_{Ev}	189.874	658.602	231.022
Total loss Π	14069.423	14365.975	18615.825
Exergy efficiency ψ_{ex}	**0.01331 $\equiv$ 1.331%**	**0.04380 $\equiv$ 4.380%**	**0.01226 $\equiv$ 1.226%**

Table E 2: Sum of Exergy Values for all test days Calculated at constant (average) Reference Temperature of 301 K.

Phases of Exergy Analysis	Day I	Day II	Day III
Heat up Π_{12}	9.542	8.961	12.327
Desorption Π_{23}	13131.355	12283.984	17444.443
Isosteric Cooling Π_{34}	13.300	10.878	11.97
Re-adsorption Π_{41}	1834.103	2729.204	1747.822
Condenser Π_{c}	20.294	27.133	38.133
Refrigeration Π_{Ev}	350.408	1013.742	433.643
Total loss Π	15051.695	16071.902	19626.191
Exergy efficiency ψ_{ex}	**0.02275 ≡ 2.275%**	**0.05933 ≡ 5.933%**	**0.02162 ≡ 2.162%**

APPENDIX F