Numerical Investigation of Active Magnetic Regenerative Refrigeration Systems

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Abstract

In this paper, one dimensional simulation of a bed of magnetic refrigeration system is performed. The numerical model assumes a stationary magnetic porous bed inside a rotating magnetic field. The porous bed consists of fluid part and solid part, so it is necessary to solve two energy equations which are coupled to find temperature distribution. The coefficient of performance is also predicted to compare with other refrigeration systems.

Keywords: magnetic refrigeration-numerical simulation -temperature distribution-coefficient of performance

Introduction

Nowadays energy crisis has become a real problem all around the world so human look for higher efficiencies, and more pure devices and systems to recover energy. In refrigeration systems magnetic systems is a new born technology that appears recently [1]. For historical review from 1933 when Giauque used a magnetic cooling system [2] up to the most recent project 2002 where Zimm has produced new generation of magnetic cooling [3].The basic of such systems is temperature lift of a magnetic material which is caused by moving from a lower to a higher magnetic filed. This phenomenon is called magnetocalloric effect (MCE).This effect depends on the initial temperature of the material and a high MCE occurs only over a relatively limited temperature span. The temperature at which the maximum MCE occurs is referred to as the Curie point of the material. Magnetic cooling is an alternative refrigeration technology that may be competitive with vapor compression systems. Different thermodynamic cycles can be assumed for such systems because MCE materials can be forced to undergo various types of processes that cause entropy move from a low temperature to a high temperature due to a magnetic work. A version of magnetic cooling is active magnetic regenerative refrigeration (AMRR) which its main part is a porous packed bed of magnetic material exposed to a time varying field and flow [3,4]. AMRR cycle has 4 main processes which begins with magnetization process, causes the bed temperature to increase. This process follows by an outer flow process that gains the added energy of the bed to come back to its initial profile temperature. Demagnetization in the third part of the cycle decreases the profile below its initial profile. Finally an outer flow (in the inverse direction of the second process) catch cooling load from the bed and the bed returns to its original profile.

Governing Equations

In order to simplify the basic model according to the geometrical factors porous bed is separated to solid and fluid part. The temperature variations fluid and solid part of bed over a steady state cycle are ultimately the output of this model ($T_f(x,t)$ and $T_r(x,t)$).These variations ,coupled with prescribed mass flow rate will allow other output of the model that show the performance of the cycle like refrigeration load and magnetic power required. These temperature variations are obtained by solving a set of coupled, partial differential equations that are obtained from 1st law consideration on fluid and solid. For a differential element (its length is dx) of each media (solid and fluid) various forms of energy transferring can be shown in (Fig.3) and (Fig.4). The following equation is the governing equation of the fluid part of the bed:

$$\dot{m}(t)\frac{\partial}{\partial x}\left[c_f(T_f).T_f\right]+\frac{Nu(\mathrm{Re}_f,\mathrm{Pr}_f).K_f(T_f)}{d_h}.a_s$$
$$.A_c(T_f-T_r)+\rho_f.A_c.\varepsilon\frac{\partial}{\partial t}\left[c_f(T_f).T_f\right]=$$
$$\left|\frac{\partial p}{\partial x}\frac{\dot{m}(t)}{\rho_f}\right| \qquad (1)$$

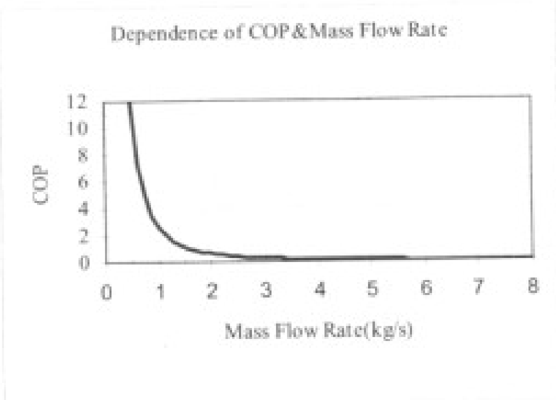
And the energy balance for the solid part is as follow:

$$\frac{Nu(\mathrm{Re}_f,\mathrm{Pr}_f)k_f(T_f)}{d_h}a_s.A_c(T_f-T_r)+$$
$$k_{eff}.A_c\frac{\partial^2 T_r}{\partial x^2}=A_c.(1-\varepsilon)\rho_r.\left[\frac{\partial u_r}{\partial t}-\mu_0 H\frac{\partial(v_r M)}{\partial t}\right]$$
$$\rho_f.A_c.\varepsilon\frac{\partial}{\partial t}\left[c_f(T_r)T_r\right] \qquad (2)$$

These equations are solved by using a numerical method over a numerical 2D grid. One dimension belongs to space and the other to time. In the two previous equations f refers to fluid and r refers to solid part of porous bed. Other parameters can be defined as: A_c (total cross section), a_s (surface area density), ε (Porosity), $\mu_0 H$ (magnetic field), $v_r M$ (magnetic moment).

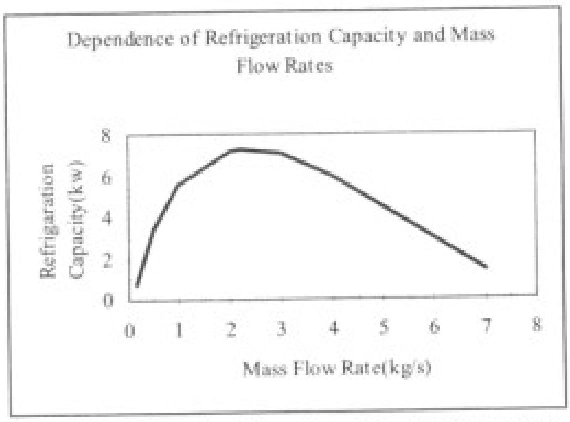
Results and Discussion

By fixing the bed length and its volume a function for mass flow rate and COP have been found. The (Fig.1) shows such dependence for a bed of fixed volume and aspect ratio of 0.36.



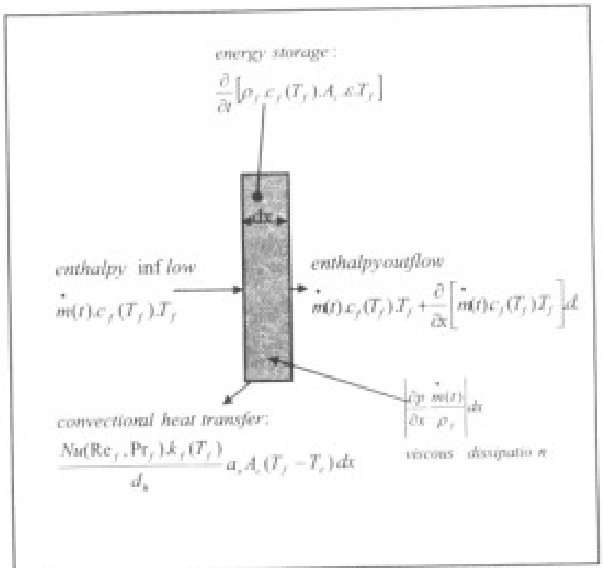
Fig(1): COP of bed of 25L volume and aspect ratio of 0.36

For each mass flow rates there is a corresponding coefficient of performance and it can be seen that a higher COP correspond to the lower mass flow rate.

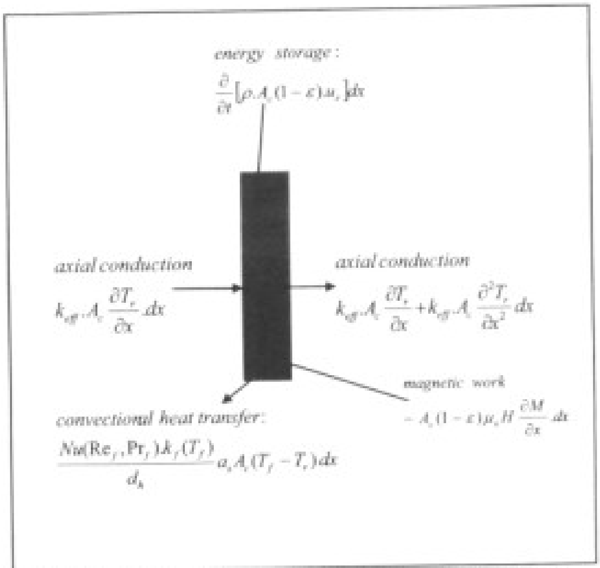


Fig(2): Refrigeration Capacity of bed of 25L volume and aspect ratio of 0.36

Also from (Fig.2) it can be seen that there are two mass flow rate for one cooling capacity therefore, by assuming (Fig.1) a specific cooling capacity has two corresponding mass flow rates which the smaller one has a higher COP, because higher mass flow rate results in more losses. At last we have to choose higher COP by selection of lower mass flow rate. This design leads to have a pure and more efficient device which is one of main purposes of this paper.



Fig(3): Differential element of fluid part of the bed with various related forms of energy



Fig(4): Differential element of solid part of the bed with various related forms of energy

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Abstract
Commonly, magnetic refrigeration is based on the entropy invoked by the application of an external magnetic field on a paramagnetic or magnetic compound. In this paper, a one dimensional model for a bed of active magnetic regenerative refrigeration systems (AMRRs) has been developed. The physical case is a wheel which consist of some porous beds that one of these beds has been considered. In order to simplify, geometry of the model, the porous bed separated to fluid and solid part. The numerical model consists of two energy equations which are coupled. An iterative scheme has been used to solve finite difference equations to find temperature distribution. With this model the refrigeration capacity and the coefficient of performance (COP) of a real AMRR system can be predicted; the numerical results indicate that the maximum of COP and cooling capacity occurs at different values of mass flow rate of heat transfer fluid.
Keywords: magnetic refrigeration-numerical-temperature distribution-COP - cooling capacity

Introduction

Nowadays energy crisis has become a real problem all around the world so human look for higher efficiencies, and more pure devices and systems to recover energy. In refrigeration systems magnetic systems is a new born technology. The recent achievements on magnetic refrigeration, especially near room temperature region, largely rest upon the active magnetic regenerator (AMR) concept. By employing AMRs in magnetic refrigeration system, two active magnetic regenerative refrigerators (AMRRs) which were designed and built by the Astronautics Cooperation in the USA [1,2] demonstrated a competitive efficiency near room temperature and made the commercial AMRRs feasible in the near. The research on the analysis of the AMRR cycle has been conducted and the investigation of AMRR prototypes has been carried out in multinational scope [3].Before building an AMRR, it is desirable to predict the performance of the refrigerator. However, previous works relating to the analysis of AMRRs have mainly focused on thermodynamic investigation of regenerators [4e6] and magnetocaloric materials [7], lack of applicable models that considers the details in a practical refrigerator, such as the calculation of the cooling capacity, coefficient of performance and Selection of heat transfer fluids, etc. Keeping most of these practical problems in mind, this paper is not confined to analyze regenerators but to establish a practical and applicable method to predict temperature distribution, cooling capacity and COP of an integrated magnetic refrigeration system.

The thermal and magnetic properties of some substances are highly coupled over certain, typically limited operating ranges, allowing them to be used within energy conversion systems. A thermodynamic substance can change its internal energy (U) as a result of either a work transfer or a heat transfer, leading to the differential energy balance:

$$dU = T \cdot dS + dW \tag{1}$$

The 1st term in Equation (1) corresponds to an inflow of heat (T.dS) and the 2nd to an inflow of work (dW).Temperature (T) and entropy (S) form what in thermodynamics is called a canonical conjugate property pair that, together, defines the transfer of heat. Pressure (P) and volume (V) form another such property pair defining the transfer of mechanical work. If hysteresis is ignored, then it is possible to define a similar pair of properties to describe the transfer of magnetic work: applied field (μ0H) and magnetic moment (VM), Guggenheim (1967).

$$dU = T \cdot dS + \mu_0 \cdot H \cdot d(V.M) \tag{2}$$

The above equation reveals that the applied field is analogous to pressure and magnetic moment is analogous to (the inverse of) volume. This analogy is physically revealing; increasing the applied field tends to align the magnetic dipoles, and the result is an increase in the magnetization. The process of aligning the magnetic dipoles requires work and reduces entropy. In a compressible substance, increasing the pressure reduces the space between molecules, which is analogous to compressing small linear springs. Reducing the volume requires work and reduces entropy. Using the analogy described above; it becomes possible to apply all of the typical thermodynamic results and identities ordinarily used with a pure compressible substance to a magnetic material; including Maxwell's relations that describe relationships between partial derivatives of properties and the idea of an equation of state that describes the magnetization as a function of temperature and applied field. A temperature-entropy diagram for a magnetic material will therefore include lines of constant applied magnetic field rather than pressure but is otherwise analogous to a similar diagram for air or any other compressible working fluid. The basic of refrigeration systems is temperature lift of a magnetic material which is caused by moving from a lower to a higher magnetic filed. This phenomenon is called magnetocalloric effect (MCE).This effect depends on the initial temperature of the material and a high MCE occurs only over a relatively limited temperature span. The temperature at which the maximum MCE occurs is referred to as the Curie point of the material. Magnetic cooling is an alternative refrigeration technology that may be competitive with vapor compression systems. Different thermodynamic cycles can be assumed for such systems because MCE materials can be forced to undergo various types of processes that cause entropy move from a low temperature to a high temperature due to a magnetic work. A version of magnetic cooling is active magnetic regenerative refrigeration (AMRR) which its main part is a porous packed bed of magnetic material exposed to a time varying field and flow [8,9]. AMRR cycle has 4 main processes which begins with magnetization process, causes the bed temperature to increase. This process follows by an outer flow process that gains the added energy of the bed to come back to its initial profile temperature. Demagnetization in the third part of the cycle decreases the profile below its initial profile

Finally an outer flow (in the inverse direction of the second process) catch cooling load from the bed and the bed returns to its original profile. The schematic temperature profile variation has been illustrated by the following figure numbered from 1 to 4 to 1.This figure shows a complete cycle. (Fig.1)

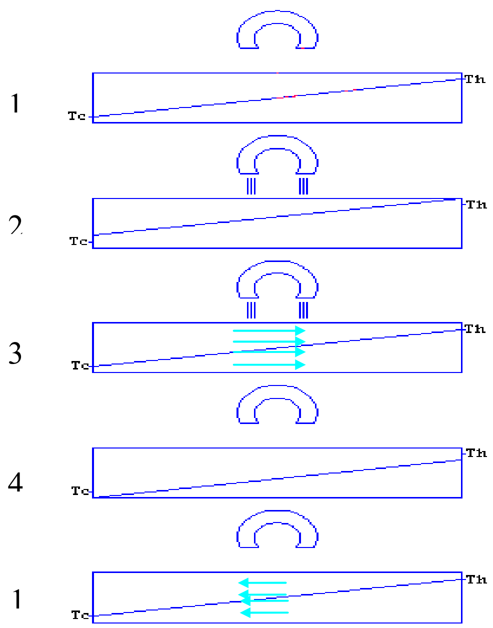

Fig (1): schematic temperature profile variation for different time during one AMRR cycle

Physical Model

The physical model is a wheel which contains some porous beds that one of these beds has been modeled using a one dimensional approximation. The equipment outside of the bed, including the pumps, heat exchangers, and permanent magnets, are not explicitly modeled. Rather, their effect on the bed is felt through an imposed time variation of the mass flow rate $\dot{m}(t)$ and the variation of the magnetic field in time and space μ0H(x,t); these variations must somehow be related to the fluid-mechanical-magnetic configuration that is employed to operate the bed. The interface between these boundary conditions for this regenerator model and these auxiliary pieces of equipment is handled by a system-level model that interacts with this component level model. When the fluid mass flow rate is positive, flow is in the positive x direction and therefore enters the hot end of the regenerator bed; when it is negative it enters at the cold end.

The fluid is assumed to be incompressible and therefore there can be no time variation in the mass of fluid entrained in the bed; consequently , the mass flow rate must be spatially uniform. The flow entering the bed is assumed to have the temperature of the adjacent thermal reservoir, TH or TC depending on whether the flow rate is positive or negative, respectively.

The required fluid properties include the density (ρf), specific heat capacity (cf), viscosity (μf), and thermal conductivity (kf). The specific heat capacity, viscosity, and thermal conductivity are assumed to be some function of temperature but not pressure. The fluid is assumed to be incompressible and so its density is unaffected by either temperature or pressure. .The material is assumed to be incompressible and therefore its density is assumed to be only a function of its Curie temperature ρr(Tcurie) and not of the local temperature or applied field. The geometry of the matrix must consist of many small passages that allow the fluid to flow in intimate thermal contact with the regenerator material. Regenerator geometries ranging from packed beds of spheres to screens to perforated plates may all be considered by this model. In order to maintain this flexibility, the regenerator geometry is characterized by a hydraulic diameter (dh), porosity (ε), and surface area density (as). The Nusselt number of the matrix is assumed to be a function of the local Reynolds number and Prandtl number of the fluid (Nu (Ref, Prf)). The friction factor is assumed to be a function of the local Reynolds number (f (Ref)). The matrix is also assumed to be characterized by an effective thermal conductivity (keff) that relates the actual, axial conduction heat transfer to the heat transfer through a comparable solid piece of material. The above constraints will be specified more completely based on the particular geometry and materials that are selected. The overall size of the regenerator is specified according to its length (L) and total cross-sectional area (Ac).

Governing Equations

In order to simplify the basic model according to the geometrical factors porous bed is separated to solid and fluid part. The temperature variations in fluid and solid part of the bed over a steady state cycle are ultimately And the governing equation with respect to solid part of the bed is as follow:

the output of this model (Tf(x,t) and Tr(x,t)).These variations ,coupled with prescribed mass flow rate will allow other output of the model that show the performance of the cycle like refrigeration load and magnetic power required. These temperature variations are obtained by solving a set of coupled, partial differential equations that are obtained from 1st law consideration on fluid and solid. For a differential element (dx) of each media (solid and fluid) various forms of energy transferring are illustrated in (Fig.2) and (Fig.3) [10].

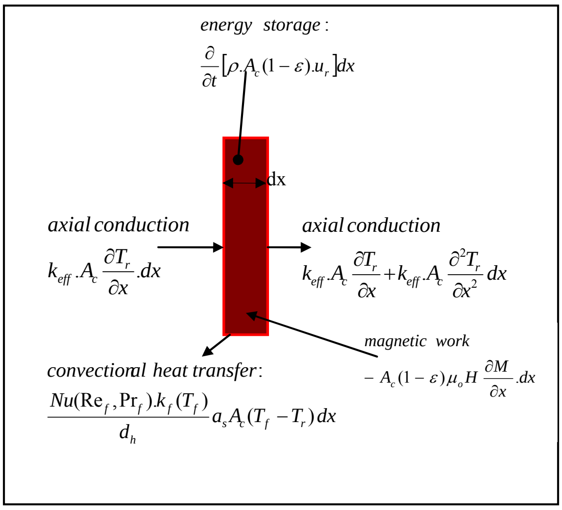

Fig (2): Differential element of fluid part of the bed with various related forms of energy[10]

The energy balance for the fluid part of the bed is as follow:

$$\overset{\bullet}{m}(t)\frac{\partial}{\partial x}\left[c_f(T_f).T_f\right]+\frac{Nu(\mathrm{Re}_f,\mathrm{Pr}_f).K_f(T_f)}{d_h}.a_s$$
$$.A_c(T_f-T_r)+\rho_f.A_c.\varepsilon\frac{\partial}{\partial t}\left[c_f(T_f).T_f\right]=$$
$$\left|\frac{\partial p}{\partial x}\frac{\overset{\bullet}{m}(t)}{\rho_f}\right| \qquad (3)$$

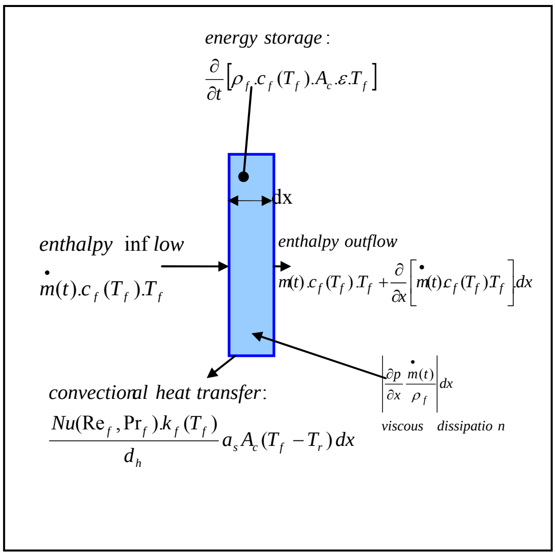

Fig (3): Differential element of solid part of the bed with various related forms of energy[10]

$$\frac{Nu(\mathrm{Re}_f,\mathrm{Pr}_f)k_f(T_f)}{d_h}.a_s.A_c(T_f-T_r)+$$
$$k_{eff}.A_c.\frac{\partial^2 T_r}{\partial x^2}=A_c.(1-\varepsilon).\rho_r.\left[\frac{\partial u_r}{\partial t}-\mu_0 H\frac{\partial(v_r M)}{\partial t}\right]+$$
$$\rho_f.A_c.\varepsilon\frac{\partial}{\partial t}\left[c_f(T_r).T_r\right] \qquad (4)$$

In these equations f and r refers to fluid and solid part of porous bed respectively. And A_c is the total cross section, a_s is the surface area density, ε is the Porosity, $\mu_0 H$ is the magnetic field and $v_r M$ is the magnetic moment.

Numerical Method

Equations described above and boundary and initial conditions form a system of linear equations in each of the nodal temperatures grid. These equations have been solved by using a numerical method over a numerical 2d grid. One dimension belongs to space and the other to time. These equations are solved using Jacobi iteration method, running the program on parallel processor so all the equations have been solved. The absolute value of the maximum error between the guess values of the regenerator and fluid temperatures and the calculated values is determined at each iteration. If the error is less than a specific value then the iteration process is complete.

Results and Discussion

Temperature variation is the ultimate and desirable output of this model because all other quantities such as COP or cooling capacity can be derived by the use of these profiles and physical assumptions. As shown in schematic temperature profile (Fig.1), the model output for temperature variation during one cycle of refrigeration (Fig.4), shows such changes in temperature. The numbers 1 to 4 in the legend are the same as in the schematic (Fig.1) for comparing and illustrating cyclic form of the solution. Because the cycle has four main processes, cycle time has been divided by four and in each time the temperature profile has been plotted. It means that there are many profiles between upper and lower profile. The length of the bed is 10cm and other data are given in table (1).

By fixing the bed length and its volume a function for mass flow rate and COP has been found. The (Fig.5) shows such dependence for a bed of fixed volume and aspect ratio of 4. For each mass flow rates there is a corresponding coefficient of performance. It can be seen that the maximum of COP occurs at 0.04 kg/s of mass flow rate of heat transfer fluid.

Fig.6 exhibits the variation of refrigeration capacity of an AMRR system consist of six beds with Mass flow rate of heat Transfer Fluid. It can be seen that there are two mass flow rate for one cooling capacity therefore, by considering (Fig.5) a specific cooling capacity has two corresponding mass flow rates which the smaller one has a higher COP.

Table 1: input data of the model

Parameter	Value
Maximum magnetic field	1.5 T
Number of AMR bed	6
AMR diameter(D)	2.5 cm
Aspect ratio (L/D)	4
Particle Size	200 µm
Porosity	0.36
TH&TC	308K & 299K
Heat transfer fluid	water

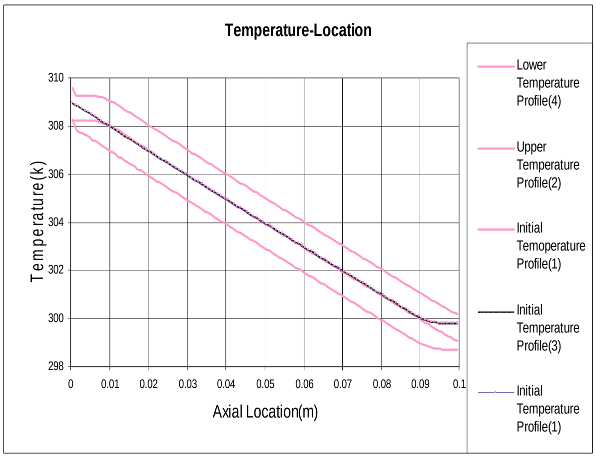

Fig (4): Model output for temperature variation with length of the bed during one cycle of refrigeration

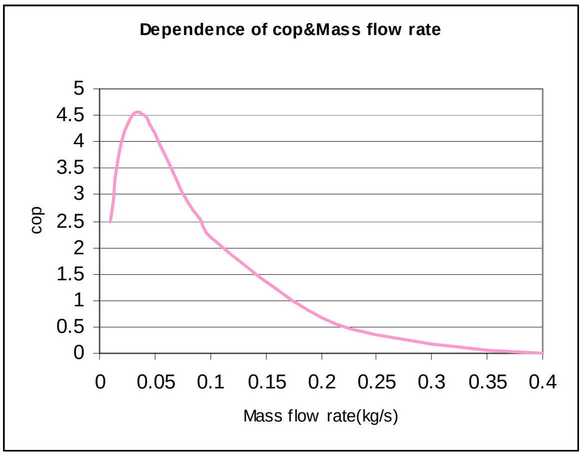

Fig (5): Variation of COP with Mass flow rate of Heat Transfer Fluid

Conclusions

It is evident that the maximum of COP and cooling capacity occurs at different values of mass flow rates so practical systems can be designed on the basis of maximum COP or maximum cooling capacity. System on the basis of maximum COP would be expensive in initial investment, on the other hand if the system is designed on the basis of maximum cooling, its running costs would be more compared to former therefore, it requires economic considerations while deciding the AMRR systems.

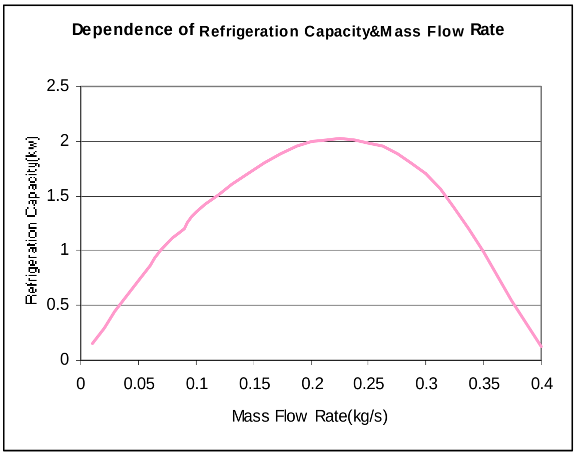

Fig (6): Refrigeration Capacity of an AMRR system consist of six bed with Mass flow rate of Heat Transfer Fluid

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