# Performance of Compressor Driven Metal Hydride Cooling Systems Under Different Operating Conditions

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# PERFORMANCE OF COMPRESSOR DRIVEN METAL HYDRIDE COOLING SYSTEMS UNDER DIFFERENT OPERATING CONDITIONS

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## ABSTRACT

The performance of a compressor driven metal hydride cooling system is analyzed using a mathematical model which takes into account the transient nature of the compressor, conditioned space and the external fins in the reactor tubes. The design is first optimized to satisfy a given comfort and pulldown criterion at an ambient temperature of 32$^0$C. Results are obtained to study the performance of the system at higher ambient temperatures. It is observed that the coefficient of performance of the system decreases with an increase in ambient temperature whereas the reverse trend is exhibited for average specific power. It is also noted that though reasonable comfort temperatures are obtained in the conditioned space with rise in ambient temperatures, there is a drastic rise in the operating pressures of the system. The study also predicts the effect of change in compressor speed, speed of the blowers and cycle time on the performance of the system under extreme operating temperatures. It is observed that controlling speed of the blowers is the most effective technique to enhance the performance.

## 1. INTRODUCTION

Solid sorption systems are being developed and explored all over the world (Meunier *et al*., 1996) to compete with, besides providing an environment friendly alternative to, vapour compression systems (VCS) (Klein and Groll). Theoretical studies (Lloyd, 1998) and experiments (Park *et al*., 2001) towards the development of a compressor driven metal hydride heat pump (CDMHHP) systems is a step towards this.

Early theoretical studies on CDMHHPs have shown coefficient of performances (COPs) comparable to that of VCS (Kim *et al*., 1997; 1998) and efforts are currently underway to develop CDMHHP system as a commercial air conditioner (Park *et al*., 2002). Hence a realistic theoretical estimate of the performance of these systems under different operating conditions will be of immense help, since obtaining such results experimentally will be time consuming, costly and difficult.

The present study gives a brief description of a mathematical model that incorporates the transient nature of both the compressor and the conditioned space. The transient nature of external fins on the reactor tube to enhance external heat transfer, which has a major influence on the heat and mass transfer of the metal hydride beds, has also been accounted for in the model. The effect of tube and fin dimensions has been incorporated in the model to calculate the external heat transfer coefficient. The model can effectively predict performance parameters during initial pull-down and stable cycling periods. The optimized design parameters are fixed for a cooling system having a load of 1 kW using $Zr_{0.9}Ti_{0.1}Cr_{0.55}Fe_{1.45}$ hydride. The optimized system satisfies a given comfort and pulldown criteria with the external ambient at 32$^0$C (305 K). Performance of the above cooling system is obtained with the external ambient at 42$^0$C and 52$^0$C. It is seen that even if the system is operated at 52$^0$C (325 K), the final temperature of the conditioned space is around 27.5$^0$C. It is also observed that the COP of the system decreases and the specific power (SP) increases with an increase in ambient temperatures. Since compressor speed, the speed of the blowers and cycle time can be varied in such systems, the study also examines the effect of controlling the above parameters on the performance of the system.

## 2. MATHEMATICAL MODELING

### Physical model and description of the system

Figure 1(a) shows the schematic of a compressor driven metal hydride based cooling system. It consists of two identical metal hydride reactor assemblies connected through a compressor. Each reactor assembly consists of a number of reactor tubes. One of the reactor assemblies desorbs hydrogen while the other absorbs hydrogen at a given instant of time. The desorbing one is in contact with the room to be cooled, while the absorbing one is kept in outside ambient. The compressor pumps the hydrogen from the desorbing reactor to the absorbing reactor. Hence, the suction and discharge sides of the compressor are connected to the desorbing and absorbing reactor assembly respectively. A reciprocating compressor is chosen for subsequent analyses. The roles of the reactors are reversed after a given half cycle time, since the hydrogen desorption/absorption capacity of the metal hydride is limited. This is done with the help of intermediate three-way or four-way valves, connected between the reactor assembly and the compressor, and air-side dampers switch position and reverse the roles of the reactor assemblies. The cooling of the room is achieved by blowing room air over the desorbing reactor assembly since desorption of hydrogen is an endothermic reaction. In the absorbing reactor assembly, the work of compression and the heat of exothermic absorption are rejected to the outside ambient. Fin-and-tube type heat exchangers are used to enhance external heat transfer in the absorbing and desorbing reactor assemblies. The type of fin analyzed in the present study is annular in nature. A central hollow core is provided in the reactor tubes to facilitate the flow of hydrogen to the metal hydride beds.

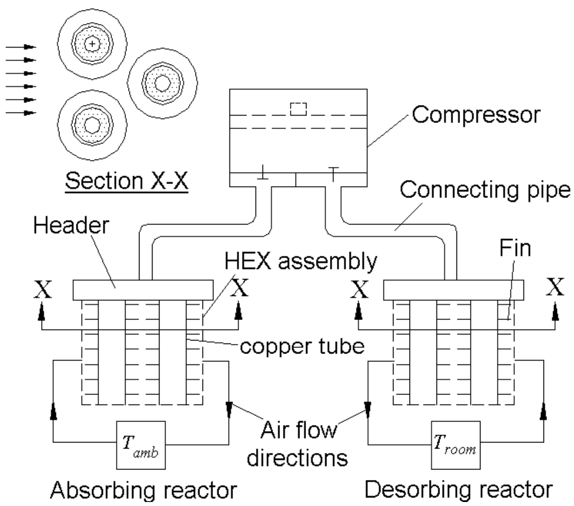


Figure 1(a): Schematic of a compressor driven metal hydride the cooling system.

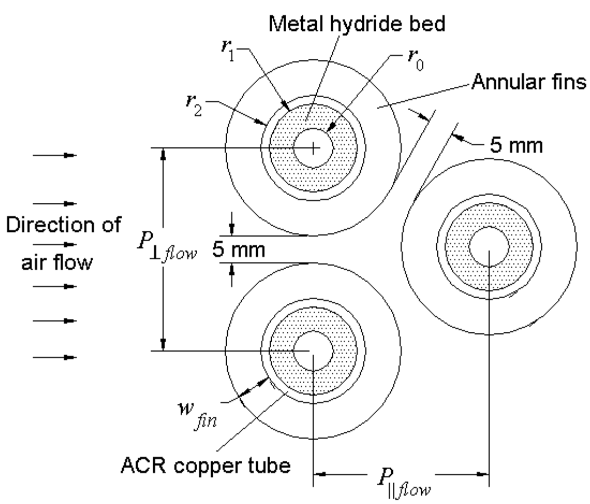


Figure 1(b): Sectional view of tube arrangement in metal hydride heat exchanger.

## Assumptions

The following assumptions have been made while developing the mathematical model of the system:

a) Heat transfer through the bed is by conduction only. Local thermal equilibrium is assumed between the gas and solid particles. The validity of assuming local thermal equilibrium and negligence of the convective term can be analytically justified from the criteria proposed by Kuznetsov and Vafai (1995).

b) Heat conduction through hydride bed is one-dimensional since the bed thickness is very small compared to the length of each reactor.

c) The equilibrium pressures are calculated using static P-C-T relations (Groll and Isselhorst, 1989).

d) Compression processes are isentropic.

e) Power consumed by the blower is neglected while calculating the COP.

f) Thermal properties of hydride beds, the room and the outside ambient are constant during the process.

g) The temperature distribution in the conditioned space is considered uniform at a given time.

h) The suction and discharge valves are assumed to be ideal, i.e. the valves are weightless and the opening and closing are instantaneous.

i) The piston in the cylinder is assumed to be weightless and frictionless and it sweeps equal volumes at equal intervals of time.

j) The frictional pressure drop through the piping, airflow ducts and across the valves is considered negligible.

## Governing Equations

The detailed mathematical model of the above system has already been reported in a previous study (Mazumdar *et al*., 2004a); here a brief description of the model and the resulting governing equations are presented. The energy equation for the hydride bed is given by:

$$k_{eff}\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T}{\partial r})-(1-\phi)\rho_s\Delta H\beta=(1-\phi)\rho_s c_s\frac{\partial T}{\partial t} \tag{1}$$

where $\beta$, the hydrogen absorption/desorption rate, is:

$$\beta=\frac{1000N_{mol}}{2M_{mol}}\frac{\partial X}{\partial t} \tag{2}$$

The reaction rate is obtained from the general kinetic expression suggested by Sun and Den (1989):

$$\frac{\partial X}{\partial t}=\sigma\frac{(P-P_{eq})}{P_r}\frac{(X-X_f)}{(X_i-X_f)}\exp(-\frac{E_a}{RT}) \tag{3}$$

The equilibrium pressure, $P_{eq}$, of the bed can be obtained from the van't Hoff equation (Lloyd, 1998). The energy equation for annular fins in the reactor tubes can be written as:

$$k_{fin}\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T_{fin}}{\partial r})-\frac{2h_{air}}{t_{fin}}\left(T_{fin}-T_{air}\right)=\left(\rho c_p\right)_{fin}\frac{\partial T_{fin}}{\partial t} \tag{4}$$

where $T_{air}$ is taken as the arithmetic mean of the temperatures of air entering ($T_{air}\big|^{in}$) and leaving ($T_{air}\big|^{out}$) the heat exchanger (HEX) assembly. In the model examined in the present analysis the effect of tube and fin dimensions and its orientation in the HEX has been incorporated to calculate the external heat transfer coefficient Thus, the sensible heat transfer coefficient on the airside, $h_{air}$, is calculated using the following correlation (Hewitt *et al.*, 1994):

$$h_{air}=\frac{0.121k_{air}}{r_2}\mathrm{Re}^{0.658}\left(\frac{P_{fin}-t_{fin}}{w_{fin}}\right)^{0.297}\left(\frac{P_{\perp flow}}{P_{\| flow}}\right)^{-0.091}\mathrm{Pr}^{1/3}F_1^{"}F_2^{"} \tag{5}$$

Figure 1(b) illustrates the configuration of the heat exchanger geometry used in the present study. A gap of 5 mm is kept between the tubes for ease of fabrication. The reactor tubes are assumed to be made of standard ACR copper tubes. Since copper is a material of high thermal conductivity, the temperature of the reactor at any given instant of time can be assumed to be uniform ($T_{r_1}=T_{r_2}=T_R$). The energy balance on the copper reactor gives the boundary condition of the metal hydride bed at $r=r_1$:

$$-\left(k_{eff}A\frac{\partial T}{\partial r}\right)_{r_1}+k_{fin}A_{fin}\frac{\partial T_{fin}}{\partial r}\bigg|_{r_2}-h_{air}A_{open}(T_{r_2}-T_{air})==\left(mc_p\right)_R\frac{\partial T_R}{\partial t} \tag{6}$$

where $A_{fin}$ and $A_{open}$ are given by:

$$A_{fin}=2\pi r_2 t_{fin}n_{fin}\ \&\ A_{open}=2\pi r_2 L_R-A_{fin} \tag{7}$$

The room is cooled by blowing air of the conditioned space over the desorbing reactors. Similarly, the heat of absorption and compression is rejected to the external ambient by blowing air over the absorbing reactors. The mass flow rate of air, $\dot{m}_{air}$, over the reactor assembly can be obtained in terms of density, velocity of air and the frontal area of the heat exchanger ($A_{fron}$) using the relation:

$$\dot{m}_{air}=\rho_{air}A_{fron}v_{air} \tag{8}$$

The rate of heat transfer ($\dot{Q}$) from the desorbing/absorbing reactor assembly can be obtained by:

$$\dot{Q}=n_{fin}h_{air}\left[2\pi(r_2+w_{fin})t_{fin}(T_{fin}\big|_{r_2+w_{fin}}-T_{air})+2\int_{r_2}^{r_2+w_{fin}}\left(T_{fin}-T_{air}\right)2\pi r dr\right]+h_{air}A_{open}\left(T_{r_2}-T_{air}\right) \tag{9}$$

This is equal to the energy gained by the air flowing over the tubes, hence:

$$\dot{Q} = \dot{m}_{air} c_{p/air} \left( T_{air}\Big|^{out} - T_{air}\Big|^{in} \right) \tag{10}$$

The energy balance for air inside the room gives us the temperature of the conditioned space ( $T_{room}$ ):

$$\dot{m}_{air} c_{p/air} \left( T_{air}\Big|_D^{out} - T_{air}\Big|_D^{in} \right) + Q_{load} = \left( mc_p \right)_{\text{air in room}} \frac{\partial T_{room}}{\partial t} \tag{11}$$

The total cooling ( $Q_{cold}$ ) achieved over a given time can be evaluated using:

$$Q_{cold} = \int \dot{m}_{air} c_{p/air} \left( T_{air}\Big|_D^{out} - T_{air}\Big|_D^{in} \right) dt \tag{12}$$

*Operating pressure estimation:*

When the suction and discharge valves are closed, gas pressure inside the compressor is given by (Chlumsky, 1965):

$$P_{comp} V_{comp}{}^{m} = C \tag{13}$$

where $m$ is the polytropic exponent and $C$ is a constant. The volume $V_{comp}$ is expressed as:

$$V_{comp} = V_{CL} + V_{swept}(t) \tag{14}$$

where $V_{swept}(t)$ is the swept volume at time $t$. Since the opening and closing of valves are assumed instantaneous, the suction valve remains closed as long as $P_{comp} \geq P_D$ and the reverse is true for the discharge valve. At the instant of valve opening temperature of the gas in the reactor core can be evaluated using the property of mixing of gases assuming that the gases mix instantaneously provided the number of free moles before valve opening in the reactor core, connecting pipes and the header, $n_{AR/DR}\Big|_{free}^{core}$, and $n_{comp}$ are known at that instant along with their respective volumes. Ideal gas law has been used to evaluate the operating pressures in the reactors. The number of free moles at any given time instant is obtained from:

$$\left. n_{AR/DR}\Big|_{free} \right]_{t+dt} = \left. n_{AR/DR}\Big|_{free} \right]_{t} + \frac{\partial n_{AR/DR}}{\partial t} dt \tag{15}$$

where $\dfrac{\partial n_{AR/DR}}{\partial t}$ is estimated employing equation (3).

*System performance parameters:*

COP and specific power (SP) are the two major parameters that define the performance of the above system. The average COP/SP are obtained over the total period of operation of the above system. Hence average COP/SP are defined as:

$$\text{Average COP} = Q_{cold}\Big|_0^t \Big/ W_{comp}\Big|_0^t \tag{16}$$

$$\text{Average SP} = Q_{cold}\Big|_0^t \Big/ \left( \text{mass of metal hydride}_{\text{both reactors}} \right) t \tag{17}$$

## 3. RESULTS AND DISCUSSIONS

Results have been obtained for a system operating with $Zr_{0.9}Ti_{0.1}Cr_{0.55}Fe_{1.45}$ as metal hydride, whose properties are evaluated from the P-C-T curves and data reported by Park *et al.* (2002). Calculations have been carried out for a room measuring 4 m X 4 m X 2.5 m and having a constant load ( $Q_{load}$ ) of 1 kW. While fixing the initial concentrations it is assumed that only 75 percent of the storage capacity in the plateau region is usable. The reactors are assumed to be at their equilibrium pressures at the beginning of the process while the temperature of the room is same as that of external ambient. The bed void fraction is taken as 0.5 (Park *et al*., 2002). Standard ACR copper tube reactors of outer diameter 19 mm (0.75 inch) has been selected with the inner core radius of 3 mm, based on optimisation studies reported previously by Mazumdar *et al.* (2004b). The effective thermal conductivity of the bed ( $k_{eff}$ ) is taken as 5 W/mK as any further improvement in thermal conductivity does not result in a drastic improvement in performance (Lloyd, 1998). Based on optimisation studies of Lloyd (1998), the face velocity of air entering the heat exchangers (HEXs) is taken to be 2.5 m/s. The width of the fins made of aluminium is taken as 20 mm increasing the area available for heat transfer by a factor of 27 compared to bare copper tube. Fin spacing is chosen to be 316 fins/m with a fin thickness of 0.15 mm (Jones, 2001). The optimal design is fixed for a system that can achieve a pulldown from $32^0$C to an average comfort temperature of $22^0$C within an hour. A double row tube heat exchanger is considered and the speed of the compressor is taken to be 50 rps.

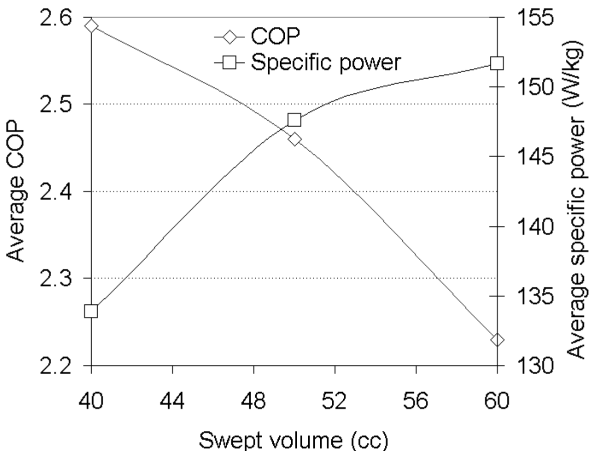


Figure 2: Average COP & SP with swept volume.

Compressor optimization was carried out for this set of input parameters. Figure 2 shows the variation of average COP and SP for an hour of system operation with compressor swept volume for systems which satisfy the pulldown criterion stated above. As is evident from the figure, a 50 cc compressor is the best choice. From 40 cc to 50 cc the deterioration in COP is about 5% while the gain in SP is about 10.2%; hence the rise in SP compensates for the drop in COP. But from 50 cc to 60 cc the loss in COP is 9.3% while the gain in SP is only 6.1%. Hence we loose out on COP by selecting a compressor having a swept volume larger than 50 cc. Henceforth all analyses have been done with a 50 cc compressor unless stated otherwise. The mass of metal hydride required per reactor assembly is around 3.85 kg. The size of the HEX comes out to be 0.38 m X 0.38 m X 0.11 m and the number of tubes per row is 6. The half cycle time is selected as 3 minutes as it corresponds to the time at which maximum SP is obtained (Park *et al*., 2002).

Figure 3(a) shows the variation of room temperature with time at different ambient temperatures. The time-temperature trace reveals that the comfort criterion of $22^0$C (295K) is achieved well within the specified pulldown criterion for an ambient temperature of $32^0$C. With increase in ambient temperature, the final temperature achieved in the conditioned space also increases, as is evident for the case of an extreme ambient temperature of $52^0$C where the average room temperature settles around $27.5^0$C. But the major problem seems to be the increase in operating pressures with increase in ambient temperatures as is evident from Fig. 3(b). It can be seen that operating pressures are higher during initial pulldown than during stable cycling. The maximum operating pressure is 12 atm. at $32^0$C whereas it is 32 atm. at $52^0$C, i.e. a rise of around 150%. This is expected, as equilibrium pressures in the reactors are higher at higher temperatures. In fact this is also the reason why we obtain a higher cooling power at a higher temperature as exhibited in Fig. 3(c). Since $P_{eq}$ increases with increase in temperature, hence $(P_{eq} - P)$ increases as well. Moreover, the exponential term in the kinetic equation (Eq. 3) increases with the increase of temperature hence leading to higher desorption rates at higher temperatures. The same logic holds while explaining why the cooling powers are higher during initial pulldown. However, the compressor power requirement also increases with ambient temperatures, as is shown in Fig. 3(d). It is the highest at the beginning of each cycle when mass flow rates are higher. Since mass flow rates are greater during initial pulldown than during stable cycling, power requirement during pulldown is also higher. The maximum power requirement is around 0.75 kW at $32^0$C while it is 1.4 kW at $52^0$C, i.e. a rise of about 90%. The maximum power requirement during stable cycling is 0.7 kW at $32^0$C and 1.2 kW at $52^0$C. It is worth a mention that the average COP & SP obtained is 2.46 & 147.5 W/kg at $32^0$C while it is 1.85 & 172.8 W/kg at $52^0$C in an hour of operation. Hence, we loose valuable COP with increase in ambient temperature. Also the percentage of reversible hydrogen utilized is about 41% at $32^0$C while it is 50% at $52^0$C.

Control on parameters such as compressor speed, the speed of the blowers and cycle time can improve the performance at higher temperatures. Hence, the effect of these parameters on performance is studied at an extreme ambient temperature of $52^0$C. Figure 4(a) shows the variation in room temperature with increase of compressor speed, which can be varied, as needed, using a frequency modulator. As is shown in the figure the benefit is only of $0.8^0$C if the compressor speed is increased from 50 rps to 70 rps while the average COP is seen to drop from 1.85 to 1.54. Hence this technique is not preferable. Figure 4(b) shows the effect on room temperature if the speed of the blowers is increased. It can be seen that room temperatures achieved when the air velocity is 5 m/s is $3^0$C cooler than what is attained at 2.5 m/s. In addition, the maximum discharge side pressure falls from 32 atm. to 27 atm. on doubling the blower speed. The average COP also increases from 1.85 to 1.99. However, increasing the blower speed does lead to additional power requirement and heat addition to the room, which are not considered in the above analysis. But the enhancement of air velocities in the desorbing reactors, equivalent to evaporators in VCS, is limited as at faster airflow rates the probability of carry over of condensate is higher and the disturbance due to the noise of the blower also increases. The effect of varying the speed of the external blower keeping the speed of the blower connected to the desorbing reactor at 2.5 m/s (Jones, 2001) was studied and is shown in Fig.4(c). Here the benefit in final room temperature is less than $0.9^0$C, with reduction in maximum operating pressure to 27 atm. and increase in average COP to 1.97. Increasing the speed of the desorbing blower to 5 m/s keeping the speed of outside blower at 2.5 m/s, cools the room by more than $2^0$C, but with insignificant effect on COP and operating pressures. Hence, performance enhancement by changing the airflow rate in such systems will be achieved if speed of both the blowers can be increased, i.e. to say that it is better suited for dry climates where control of humidity is not a major factor. Figure 4(d) shows the benefit obtained by changing the cycle time. It can be seen that with higher ambient temperatures the optimum cycle time also changes. At $52^0$C it is around 10 minutes. A final temperature achieved at cycle time of 10 minutes is $1^0$C less that that achieved with a cycle time of 6 minutes. The average COP also increases to 1.98. Finally, among all the control options reviewed, speed modulation of both the blowers appears to be the most beneficial.

## 4. CONCLUSIONS

Performance analysis of a compressor driven metal hydride cooling system has been carried out at different ambient temperatures with an optimized system capable of pulling down the temperature of the room from $32^0$C to $22^0$C within an hour. It is seen that maximum operating pressure rise by about 150% if the temperature of the ambient is increased from $32^0$C to $52^0$C leading to an excess power requirement of about 90%. The final room temperature achieved at $52^0$C is $27.5^0$C, which can be improved by changing compressor speed, speed of the blowers and cycle time. Enhancing the compressor speed will lead to a major loss of average COP, hence the other options are preferable. It is seen that changing the speed of the blowers gives the best results in terms of performance enhancement provided control of humidity is not a factor. It is expected that such transient analyses and its reflection on system performance will lead to a better engineering of CDMH cooling systems.

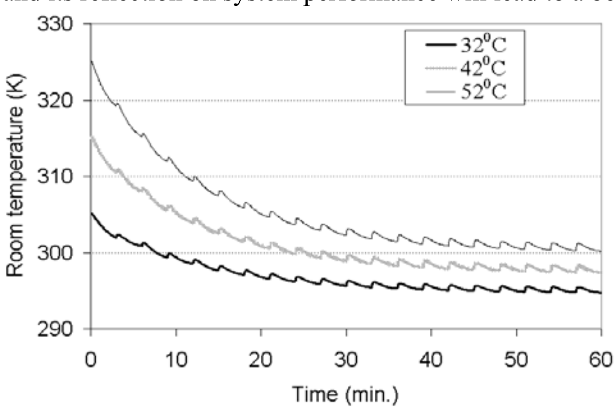

Figure 3(a): Temperature signature of the room for different ambient temperatures.

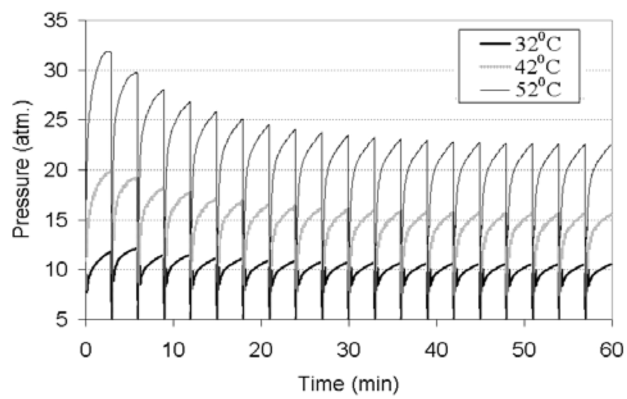

Figure 3(b): Variation of discharge operating pressure with time for different ambient temperatures.

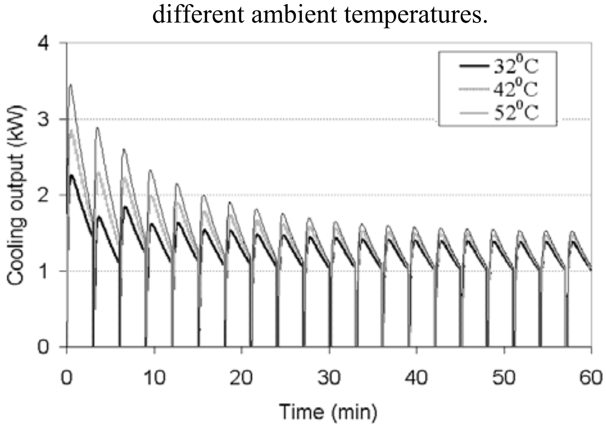

Figure 3(c): Variation of cooling output with time for different ambient temperatures.

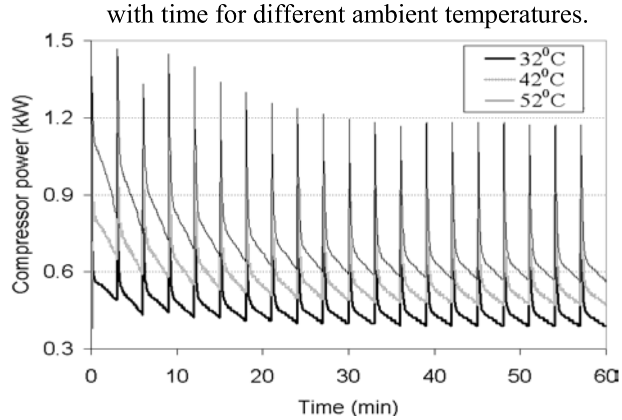

Figure 3(d): Variation of compressor power with time for different ambient temperatures.

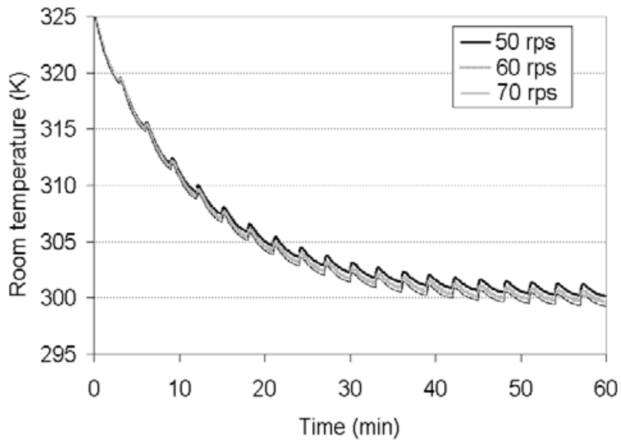


Figure 4(a): Temperature signature of the room for different compressor speeds.

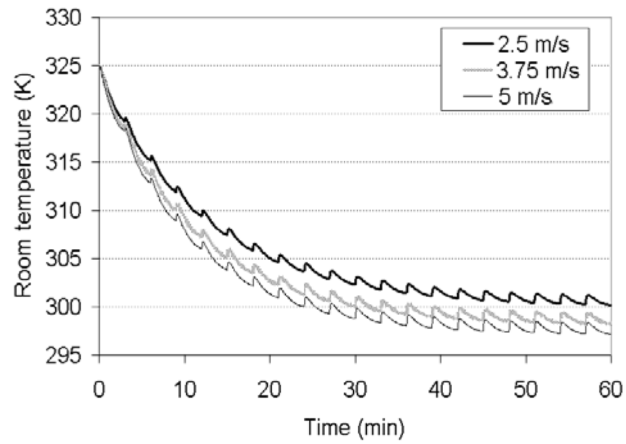


Figure 4(b): Temperature signature of the room for different air flow rates.

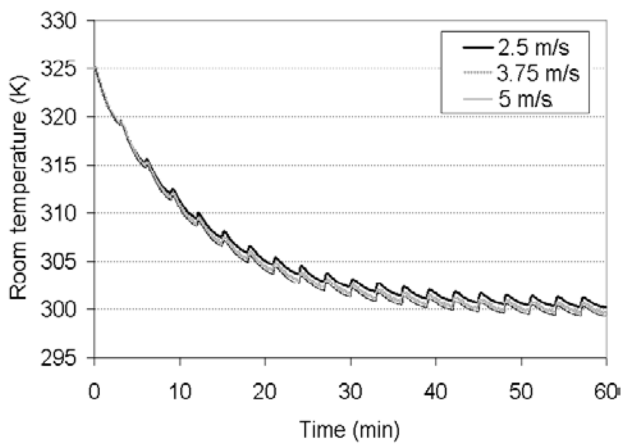


Figure 4(c): Temperature signature of the room for different air flow rates over absorbing reactors.

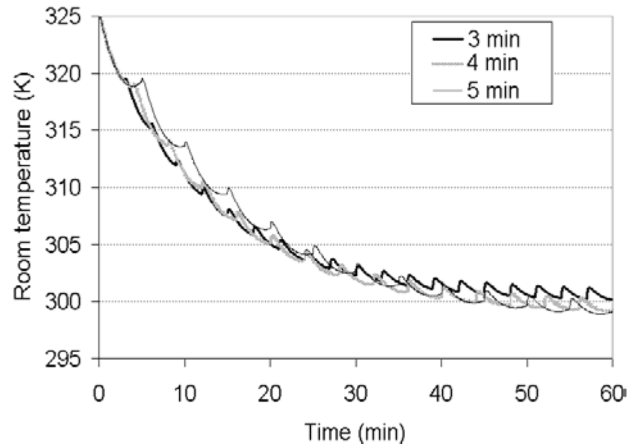


Figure 4(d): Temperature signature of the room for different half cycle time.

## NOMENCLATURE

| | |
|---|---|
| $A$ | Area ($m^2$) |
| $c$, $c_p$ | Specific heat ($Jkg^{-1}K^{-1}$) |
| $E_a$ | Activation energy ($J\ mol^{-1}$) |
| $F_1^{"}$ | Factor accounting for the variation in physical properties of the fluid being heated or cooled, for air cooled heat changers, $F_1^{"}=1$ |
| $F_2^{"}$ | Factor accounting for the number of tube rows |
| $\Delta H$ | Heat of formation (J mol of $H_2^{-1}$) |
| $h$ | Heat transfer coefficient ($Wm^{-2}K^{-1}$) |
| $k$ | Thermal conductivity ($Wm^{-1}K^{-1}$) |
| $L$ | Length (m) |
| $m$ | Mass; unless otherwise specified (kg) |
| $\dot{m}_{air}$ | Mass flow rate of air ($kgs^{-1}$) |
| $M_{mol}$ | Molecular weight (kg $(kmol)^{-1}$ of alloy) |
| $N_{mol}$ | Number of metal atoms per mole of alloy |
| $n$ | Number of moles (mol.) |
| $n_{fin}$ | Number of fins |
| $P$ | Pressure ($Nm^{-2}$ or atm.) |
| $P_r$ | Reference pressure (1 atm.) |
| $P_{fin}$ | Pitch of fins in reactor tube (m) |
| $Pr$ | Prandtl number |
| $Q$ | Energy (J) |
| $Re$ | Reynolds number |
| $r$ | Radial distance (m) |
| SP | Specific power (W kg-alloy$^{-1}$) |
| $T$ | Temperature (K) |
| $t$ | Time (s) |
| $t_{fin}$ | Thickness of fin (m) |
| $V$ | Volume ($m^3$) |
| $V_{CL}$ | Clearance volume, 5% of total $V_{swept}$ ($m^3$) |

$v_{air}$ Velocity of air ($ms^{-1}$)
$W_{comp}$ Work done by the compressor (J)
$w_{fin}$ Width of fin (m)
$X$ Concentration (atom of $H_2$/atom of alloy)
$\phi$ Void fraction
$\rho$ Density ($kgm^{-3}$)
$\beta$ Reaction rate (mol of $H_2$ (kg hydride)$^{-1}$ $s^{-1}$)
$\sigma$ Reaction rate constant ($s^{-1}$)

**Subscripts/Superscripts**
*a, A* Absorption or absorbing
*amb* Ambient
*AR* Absorbing reactor
*Comp* Compressor
*CL* Clearance
*d, D* Desorption or desorbing
*DR* Desorbing reactor
*eff* Effective
*eq* Equilibrium
*f* Final
*fron* Frontal
$H_2$ Hydrogen
*g* Gas
*i* Initial
*R* Reactor
*s* Solid phase (metal hydride)

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## ACKNOWLEDGEMENT

The financial support extended by Ministry of Non-Conventional Energy Sources (MNES), Government of India toward this study is gratefully acknowledged.