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A Prototype Steam Storage System for Power Production

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Abstract— ***Use of solar energy on a large scale is mainly limited to the sun duration hours, weather conditions and adequate solar thermal storage technology. A means of addressing this problem using local materials is provided. A prototype pressure storage system was designed using auto cad 2010 and fabricated using locally available materials. The steam storage system was tested using ASME 2000b guidelines for boiler and pressure vessels at a small scale. The maximum continuous rating for the storage system was 60 kgh^{-1}, maximum instantaneous demand 40 kgh^{-1}, storage capacity 50 kgh^{-1} and mean off peak load of 100 W was realized. The maximum test temperatures of operation obtained using seamless galvanized iron pipe was 264.7 ^{o} C and 140 ^{o} C using polypropylene pipes. The steam storage system operated between 1.0 $\times 10^5$ Pa and 1.0$\times 10^6$ Pa. Its charging duration was twenty minutes and would discharge at the rate of 50 lh^{-1}continuously after full charge with solar irradiance between 800 W/m^2 and 1020 W/m^2. The length of complete discharge for the heat transfer fluids was found to range between 4.4 hours. and 6.9 hours. The power output for the heat transfer fluids were in the range of 251.8 W and 486.9W. The steam storage system was found to have an efficiency of 93.5 % and a thermal capacity of 4.54$\times 10^3$ k J. This system presents a means of storing solar energy in form of steam during low energy demand and its conversion to power during high peak demand twenty four hours a day depending on the size of the storage and application and hence addressing the problem of variability and low density of solar energy.***

Keywords— Solar irradiance, Steam storage, Heat transfer fluid, Thermal capacity.

INTRODUCTION

Design of components to enhance solar thermal collection efficiency is an important step in solar thermal power cost reduction (Baker, *et al*, 1996). The sun provides an abundant, clean and safe source of energy. However the supply of this energy is periodic following yearly and diurnal cycles, intermittent, unpredictable and it is often diffused. Its density is low compared to the energy flux densities found in convectional fossil energy sources like coal or oil making storage necessary. Technical use of solar energy presently poses problems primarily because of inefficient collection and storage (Bayon, 2010). Enhancement of technologies that generate power from renewable energy and alternative sources are a dependable way of cutting down on dependency on carbon based fuels and negative environmental impacts of acid rain, reduced ozone layer and global warming. (Ercan, 2006). Solar technologies have the potential of lowering the cost of power since the sun is the most abundant source of renewable energy over the centuries maintaining almost a constant energy supply.

METHODOLOGY

DESIGN AND FABRICATION

The pressure storage system was designed and fabricated to operate at a pressure of 1.20×10^6 Pa. Mild steel sheets of thickness 0.015 m for inner casing and the same for the outer surface were laminated and welded together before cutting out using the auto cad design lay out. The dimensions of the storage system were external radius of 0.14 m, internal radius of 0.11 m, length of storage system of 0.48 m, inlet hole diameter of 0.03 m, internal hole diameter of 0.02m, Outlet hole diameter of 0.26 m. mass of 84.5 kg and a volume capacity of 0.024 m^3.

The mild steel plates were laid out, bent and cut along cutting lines that were marked using the auto cad figures. The metal plates were shaped using shears and drill presses. The metal plates were cut into desired sizes using oxy fuel torch and its parts were ground to shape for fitting together. The various pieces of storage tank were fitted and tack welded together using electric arc welding equipment. Some cutting was also done using chiseling and special band saws. The band saws had hardened blades. Bending was done using manual hammering. The whole assembly was done by welding, riveting, threading, fastening and bending inform of crimped seam. To prevent warping, the welded parts were covered with sand during cooling and straightening operations which were carried out after welding. Straightening was carried out using oxy-acetylene flame. The heat was selectively applied to the plates in a slow, linear sweep.

Annealing was carried out to reveal residual stress. Storage system breather value joints and flow curves were prepared. Slots for temperature sensors and pressure gauges were prepared by drilling and sealing after protection of the probes.

The required thickness of the mild steel plate was determined by elastic analysis considering allowable stress of σ = 17462.5 Pa, pressure of 5.588×10^6 Pa, radius, a = 0.48 m, modulus of elasticity of E = 689.5 kPa and Poisson's constant of μ = 0.5. (Maan, 2011)

Thickness of the plate was obtained from Maan, 2011 as shown in equation 1:-

$$t = \sqrt{\frac{6M}{\sigma}} = \quad \ldots \quad (1)$$

The required thickness, t of the curved shell estimated to be spherical was obtained from equation 2:-

$$t = \frac{P_r}{2\sigma} \quad \ldots \quad (2)$$

The longitudinal bending, P_b stress of the curved storage system was obtained from equation 3:-

$$P_b = \frac{6M_o}{t^2} \quad \ldots \quad (3)$$

The longitudinal stress for the hemispherical part, was obtained from Gould, 1988 as shown in equation 4

$$P_s = \frac{p_r}{2t} + \frac{6M_o}{t^2} \quad \ldots\ldots (4)$$

To find the buckling of cylindrical structure under lateral and axial pressure in elastic range Equation 5 by Kollar, et al, 1984 was used as shown

$$P_{cr} = \frac{2.42E}{(1-\mu^2)^{\frac{3}{4}}} \frac{(t/2r)^{2.5}}{[L/2r - 0.45(t/2r)^{1/2}]} \quad \ldots\ldots (5)$$

The optimum insulation was determined from Jawaad, et al, 2003 as shown in equation (6):-

$$\frac{dQ}{dr_o} = \frac{T_i - T_\infty}{\left(\frac{1}{2\pi r_o h} + \ln\frac{(r_o/r_i)}{2\pi K}\right)^2} \left(-\frac{1}{2\pi r_o^2 h} + \frac{1}{2\pi K r_o}\right) = 0 \quad \ldots\ldots (6)$$

Insulation beyond critical value increases the heat loss instead of reducing it. Solving the above equation for $r_o = r_{crit}$ at which R_t is minimum, obtain

$B_i = 1 = \frac{h r_{crit}}{K}$ at maximum heat flux: Where

$R_{crit} -$ Critical radius of insulation, $h -$ average heat transfer coefficient, $K -$ thermal conductivity, $dQ -$ heat flux, $B_i -$ Biot number, $r_o -$ outside area, $r_i -$ inside area, $U -$ heat loss coefficient.

SIZING OF STEAM STORE

The sizing of the steam store was undertaken by consideration of the following design parameters which were, Maximum continuous rating was 60 kgh^{-1}. , Normal working pressure ranged from 1.0×10^6 Pa to 1.0×10^6 Pa,

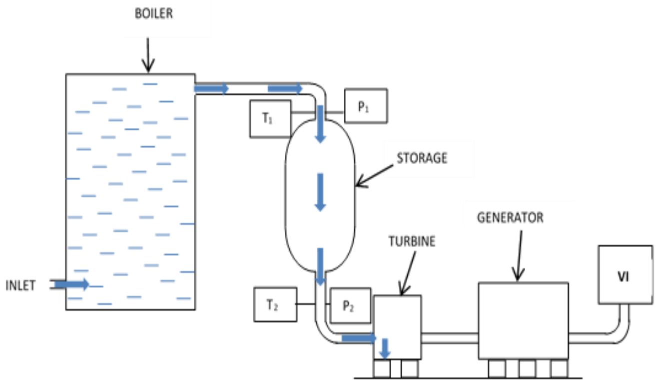

Figure 1: Steam storage testing loop

maximum instantaneous demand was 60 kgh^{-1}. The mean value of overload used to size the steam storage system was 40 kgh^{-1}. . The mean off peak load was approximated as 20 kgh^{-1}. and was used in off peak sizing calculation. The off peak load was less than maximum continuous rating. The steam storage capacity was obtained from Duffie, *et al*, 1991as shown in Equation 9 as 30 kg/h of flashing steam.

$$STC = \frac{\text{difference in enthalpy of water xmass of water}}{\text{enthalpy of evaporation at lower pressure}} \quad \ldots (9)$$

Where STC is the steam storage capacity

TESTING OF THE STEAM STORE

Testing of steam storage system was carried out in two stages. In stage one the piping work was done using polyproperelyne pipes and operated at a maximum temperature of 140.9 °C and at pressures of 7.0×10^6 Pa and in the second stage the piping work was done using seamless galvanized iron pipes. The temperatures of operation for this were a maximum of 264.7 °C and operational pressures of 1.0×10^6 Pa. The steam storage system was half way filled with water. The steam from the boiler was fed into the steam storage system. The supply steam valve would be set at different pressures and the rate of steam supply, corresponding power output from the generator and the inlet and outlet temperatures were recorded. The steam pressure setting valve was set to allow accumulation of pressure in the steam storage system during charging. During discharge, exhaust valve would be opened to release the steam to the turbine. The steam pressure control regulated the amount of steam entering the storage system. The non-return valve allowed the steam to proceed out of the steam store but would prevent back flow of the steam when pressure in the store dropped. The feed water was manually controlled. The mass flow rate was read from the automatic flow meter and amount of condensed steam was determined by trapping steam in cold water and measuring the mass of condensed steam using a beam balance. The heat emitted and heat absorbed by the heat transfer fluid was then calculated from the measurements made.

The steam storage efficiency was determined from finding the heat in steam output by use of the steam tables and finding the heat in electrical output expressed as a percent.

The evaporation ratio was determined by measuring the quantity of steam generated within a time interval as a ratio of electrical energy input which was expressed as a percent. Discharge rate was determined by measuring the amount of condensed steam in ten minutes.

Load levelling was effected by the help of a steam control valve which was fitted on the boiler outlet, which would discharge more steam during extra demand i.e. when steam outlet valve was open to discharge maximum steam. The exhaust pressures and the corresponding power output values were noted.

RESULTS AND DATA ANLYSIS

STEAM STORAGE

It was observed that when pressure in the boiler increased beyond 3.0×10^5 Pa after 20 minutes the boiler pressure remained constant and the supply of steam was constant. This was attributed to the fact that the water in the boiler was under steam pressure and pressure drop was prevented by the flashing of steam once there was discharge.

The mild steel plates thickness used were 0.006 m, maximum moment for the system was obtained as 47.62 N/m, required thickness of the curved shell was 0.0028 m, longitudinal bending stress was obtained as 79.3 kPa, the longitudinal bending stress for the hemispherical part was obtained as 71.8 kPa, optimum insulation was obtained as 0.03 m and a safety factor of 4 was used. Lagging of the storage system was done using cotton wool. It was observed that the steam storage system increased the efficiency of power production in that the quantity of sensible heat in the heat transfer fluid was increased by pressurization and consequently super-heated steam would be produced. The assurance of power provision was achieved by performing load sizing that ensured the steam store fabricated size met the demand. The steam storage system was operated between $1.0\times10^5\,Pa$ and $1.0\times10^6\,Pa$. When in operation and there was discharge of steam to the turbine the super-heated water in the steam storage system flashed and the pressure in the steam store remained constant. Higher discharge rates were obtained when the heat transfer fluids at higher temperatures continuously supplied heat for steam formation and storage in the steam storage system. During the charging process, the mass in the volume of heat exchanger increased as a result of more incoming steam. The warm water exiting from the turbine was fed into the reservoir of cold water and then passed to the boiler after its inlet temperature was recorded. Lower inlet temperatures were observed to enhance efficiency of the steam storage system as was observed from figure 2.

The charging duration of the steam storage system was twenty minutes and would discharge at the rate of 60 $l\,h^{-1}$continuously after full charge, when there was over 800 W/m^2 solar irradiance during the testing. Length of complete discharge for the heat transfer fluids each in turn was: sunflower oil was 4.4 hours, Rina vegetable oil was 4.6 hours, water was 5.0 hours, unused engine oil was 5.3 hours. Used engine oil was 4.8 hours, 2 M sodium chloride solution was 5.8 hours, 4 M sodium chloride solution was 6.3 hours and 6 M sodium chloride solution was 6.9 hours. The power output for the heat transfer fluids was as follows, sunflower oil was 87.9 W, Rina vegetable oil was 89.7 W, unused engine oil was 95.4 W, used engine oil was 92.8 W, 2 M sodium chloride solution was 104.1 W, 4 M sodium chloride solution was 116.3W and 6 M sodium chloride was 121.7 W. Figure 2 shows the efficiency for the steam storage system against the inlet temperatures that were obtained for the storage system. Increase in temperature of operation of the steam storage system reduced the efficiency of the steam storage system. Lower temperatures of inlet for the heat transfer fluid led to higher efficiency of operation of the steam storage system.

Table 1: Quantity of heat absorbed by water at following conditions

Pressure (Pa)	Q(kJ)	Mass flow rate kgs^{-1}	Mass of steam(kg)
2	14630.9	3.5	6.5
4	16720.4	4.0	7.3
6	18810.1	4.5	8.2
8	20900.5	5.0	9.0
10	22990.3	5.5	10.2

During the direct power production water was used as the heat transfer fluid and as the storage medium. Table 1 shows the heat that was absorbed by water at various pressures of operation. It was observed that the higher the pressure of operation, the higher the amount of heat that was absorbed by the heat transfer fluid.

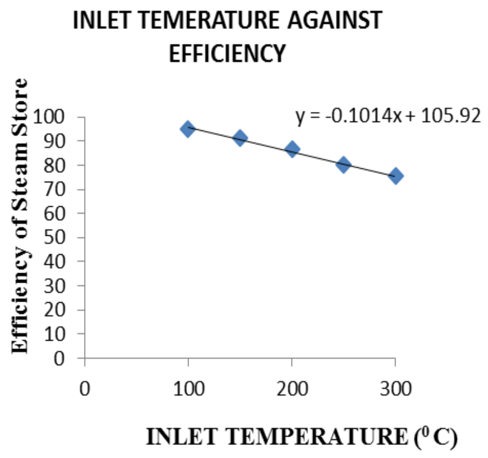

Figure 2: Relation between steam storage system efficiency against inlet temperature

The steam storage system provided approximately 21.4 kg saturated steam per cubic meter of saturated steam storage volume. The boiling point of water rose to a maximum temperature of 254.3 ^{0}C at 9.0 $\times 10^5$ Pa pressure.

Figure 3 shows the power output obtained for the heat transfer liquids for the operating pressures shown. Increase in pressure increased the boiling point at which the water absorbing heat from the heat transfer fluid turned into steam. The lower the output pressure of the turbine the higher the amount of power output that was obtained. The more pressure fall at the turbine exhaust the more power that was obtained from the generator.
The steam storage system efficiency was determined and obtained as 93.5 % using equation 10:-

$$\eta = \frac{Heat\ out\ put}{Heat\ input} \times 100 \ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

Evaporation ratio was obtained from equation 11 as:-

$$E.R = \frac{Heat\ in\ steam\ out\ put}{Heat\ in\ electrical\ input} \times 100 \ldots\ldots\ldots\ldots \quad (11)$$

Where E.R is the evaporation ratio.

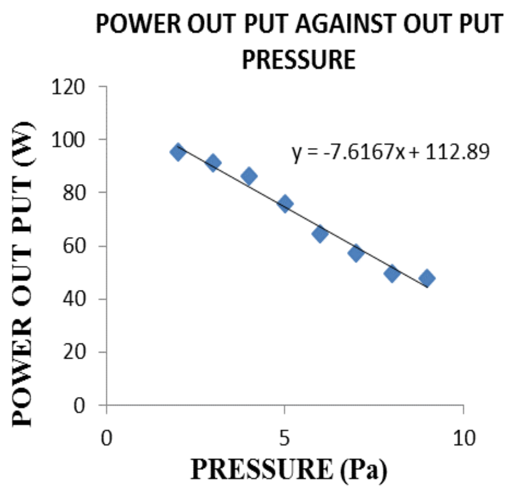

Figure 3: Power output against output pressure for the steam store

The 6M salt solution had the highest rate of transmitting the heat it carried to the water to produce steam for power production. Storage of steam as a gas is not practical because the storage volume required at normal boiler pressures is expensive in terms of the huge volume that needs to be provided.

CONCLUSION

The steam store was designed to operate at 1.2×10^6 Pa, at an inlet temperature of 22 ^{o}C and an outlet temperature of 250 ^{o} C. Two mild steel sheets 0.015 m were laminated and the system parts were cut out according to the auto cad design. The efficiencies available are for industrial applications whose efficiency range between 20 % and 90 %. Most of the solar thermal energy storage medium available is in form of water storage. Johnson and Johnson energy project has thermal capacity of 1.33 GJ and operates at a maximum temperature of 363 ^{o}C. Invapah solar generating facility that uses molten salt has a storage efficiency of 98 %. (Sukhatme, 1999). Solar power generation can be used to produce industrial power by use of appropriate technology. The heat conduction of water and the heat transfer fluids requires improvement to get more power generation. This would also enhance the efficiency of absorption of solar thermal power.

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