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SEPARATE-TYPE HEAT PIPE SOLAR RECEIVERS FOR CONCENTRATING SOLAR POWER

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ABSTRACT

This paper introduces separate-type heat pipe (STHP) based solar receiver systems that enable more efficient operation of concentrated solar power plants without relying on a heat transfer fluid. The solar receiver system may consist of a number of STHP modules that receive concentrated solar flux from a solar collector system, spread the high concentrated solar flux to a low heat flux level, and effectively transfer the received heat to the working fluid of a heat engine to enable a higher working temperature and higher plant efficiency. In general, the introduced STHP solar receiver has characteristics of high heat transfer capacity, high heat transfer coefficient in the evaporator to handle a high concentrated solar flux, non-condensable gas release mechanism, and lower costs. The STHP receiver in a solar plant may also integrate the hot/cold tank based thermal energy storage system without using a heat transfer fluid.

Keywords: *High-temperature separate-type heat pipe (STHP), solar receiver, concentrated solar power plant*

1. INTRODUCTION

The concentrated solar power (CSP) generation system is a viable technology for conversion of solar energy into mechanical power and then into electricity through a generator. The solar power system generally includes a collector system that redirects or concentrates solar radiation from the sun onto a solar receiver system, which converts the received solar radiation into thermal energy. The thermal energy is then used as a heat source in a heat engine that converts the received thermal energy from the receiver system into mechanical power. Various CSP systems have been developed with additional advantages of integrating thermal energy storage (TES) mechanisms (Mancini et al., 1994; Stine and Diver, 1994; Lovegrove et al., 2007, Yang et al., 2009; Nithyanandam and Pitchumani, 2012a and 2012b; and Tao and Rayegan, 2011).

Because of economies of scale to compete with traditional fossil-fuel based utility industries, large-scale solar thermal power plants, such as solar tower and trough plants, have been gaining momentum in both governmental and private investments and developments (Tao and Rayegan, 2011). For large-scale CSP plants, a central solar receiver system is often employed in conjunction with a heat engine such as a steam turbine engine or gas turbine engine. Currently, a heat transfer fluid (HTF), such as a molten salt, is employed to receive solar energy and transport it from the receiver to the engine for the conversion of the received thermal energy to mechanical work. However, the maximum operating temperature of commonly used heat transfer fluids may be limited. At a high operating temperature, the heat transfer fluid may be chemically broken down. This limited operating temperature may inevitably affect the performance of the power plant as the power output and thermal efficiency of the related heat engine are highly dependent upon the temperature level of the heat source. Because of the relatively low operating temperature, the successful demonstrations of large-scale solar thermal power plants are primarily based on steam-turbine power plants that produce power at lower peak temperatures. In addition to the lower energy efficiency due to the limited peak working temperature of a heat transfer fluid, the amount of water consumption associated with wet cooling condenser may be unacceptable for the construction of a solar power plant in a desert. Alternative power systems, such as the advanced supercritical carbon dioxide power cycles, have been proposed (Ma, et al., 2012) to mitigate these problems.

The difficulties associated with the heat transfer fluid and water consumption of wet cooling may favor a gas turbine based power plant without relying on the use of a heat transfer fluid. First, without the constraint imposed by the heat transfer fluid, the gas turbine may work at an increased peak temperature to attain a substantially improved thermal efficiency as compared to the counterpart of a steam-turbine based power plant. Second, the gas turbine power plant may remove the need for wet cooling and be constructed in a location where water resource is limited. Additionally, the gas turbine power plant may incorporate compressor inter-cooling, reheat, and energy recovery from the exhaust stream, and could attain an overall thermal efficiency well above 50% even under a moderately high peak working temperature.

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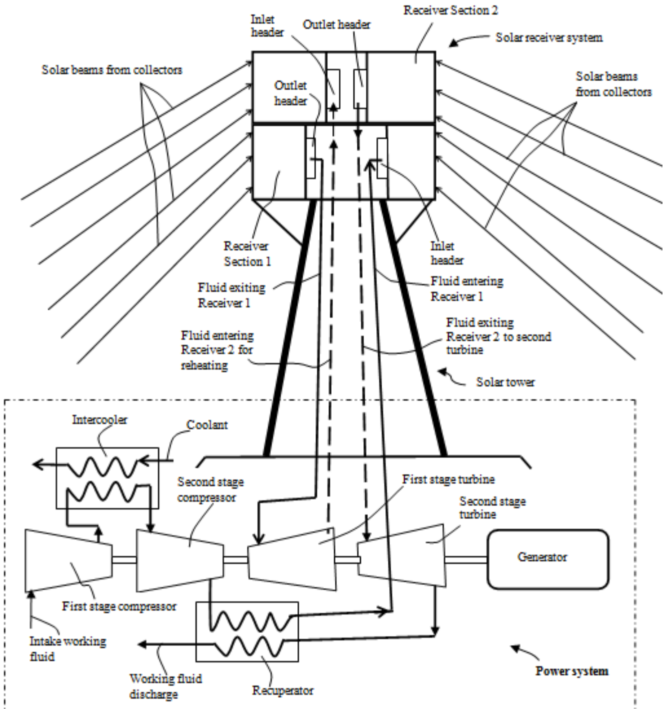

Fig. 1 Schematic of a solar tower power plant integrating compression intercooling, reheat, and recuperative techniques

Figure 1 illustrates schematically a concentrated solar tower power plant based on a gas turbine engine system. The power plant includes a central solar receiver that may be divided into two sections (sections 1 and 2), which are mounted on a solar tower. The receiver receives concentrated sunlight and coverts it into heat from a collector system (not shown) that redirects solar radiation from the sun onto the solar receiver. The gas turbine system includes compression intercooling, reheat, and recuperative techniques to increase its thermal efficiency. As illustrated of Figure 1, a working fluid enters a first stage compressor and exits the compressor at an outlet and then enters an intercooler. The working fluid flows through heat exchanger and transfers an amount of heat to a coolant stream. With a reduced temperature, the working fluid stream exits the intercooler and enters a second stage compressor. Upon exiting the second stage compressor, the further compressed stream enters a recuperator where it receives an amount of heat from an exhaust stream before the exhaust stream is discharged out of the recuperator. Having received the heat from the recuperator and raised its temperature, the working fluid stream exits the recuperator and flows up the tower and enters an inlet header of receiver 1. After acquiring an amount of heat from the receiver 1, the stream exits the receiver 1 through an outlet header and enters a first stage expander or turbine. The stream expands through the turbine and produces mechanical power. The produced mechanical power may be used to drive the compressors or to generate electricity through a generator. After expansion through the first stage turbine, the stream leaves the turbine and flows up the tower again into a second solar receiver through an inlet header, where it is reheated. The reheated stream exits the second receiver through an outlet header and flows down the tower and then enters a second stage turbine. The gas stream expands through the second turbine and produces mechanical power. After expansion through the second turbine, the stream exits the second turbine and becomes an exhaust stream. The exhaust stream then flows through the recuperator, transferring an amount of heat to the engine working fluid, and finally it is discharged out of the recuperator. For the gas turbine working in an open cycle, the exhaust stream is eventually released into the ambient. However, for closed cycles, such as a CO_2 or steam cycle, the discharged stream may return to the first stage compressor after a heat exchanger (not shown).

The solar tower power plant as illustrated above may represent one of the most attractive solar power generation systems from the view point of thermodynamics, as it can reach a thermal efficiency above 50% without requiring an extremely high peak working temperature (Moran, et al., 2009). However, two critical issues must be resolved before the system could be practically built. The first issue is that the receiver must be able to work at a high temperature and handle a high solar flux.

To attain a higher receiver efficiency of a solar receiver, the fraction of radiation loss from the receiver wall surface to the ambient should be reduced, particularly when the receiver surface temperature is high. The solar receiver efficiency may be approximately expressed by the following relation for a flat receiver:

$$\eta = 1 - \frac{\varepsilon_s \sigma (T_s^4 - T_{surr}^4)}{\alpha q_c''} \tag{1}$$

where ε_s is the emissivity of the receiver surface, σ is the Stefan-Boltzmann constant, T_s is the receiver surface temperature, T_{surr} is the ambient temperature, α is the absorptivity to the solar flux and q_c'' is the concentrated solar flux reaching the receiver surface. From the above equation, it may be seen that for given emissivity and absorptivity, a higher surface temperature may significantly increase the radiation loss, resulting in lower receiver efficiency. However, a higher surface temperature is generally desirable for a higher energy efficiency of the power plant. To maintain the receiver efficiency at an acceptably high level, the concentrated solar flux q_c'' must be increased. However, this increased solar flux (In some cases on the order of 50 to 100 W/cm^2) in conjunction with the high working temperature may cause safety problem of the solar receiver for a power system of a gaseous working fluid. It is believed by the author of this paper that the high heat flux must be spread to a sufficiently low heat flux level before the received heat is transferred to the gaseous working fluid of the engine.

The second critical issue is the integration of thermal energy storage (TES) mechanisms with the power plant. The successful integration of the thermal energy storage in a heat transfer fluid based power plant is a bright spot among all solar power plants including PV and CSP power plants. The question is how the thermal energy storage mechanism may be integrated to power plant without the use of a heat transfer fluid.

2. SOLAR RECEIVERS BASED ON SEPARATE-TYPE HEAT PIPES

The heat pipe, particularly high-temperature heat pipe using a liquid metal as the working fluid, is a unique and simple heat transfer device that could work at a high temperature and handle a relatively high heat flux (Faghri, 1995; Cao and Faghri, 1991, 1992, and 1993; Ling and Cao, 2001; and Cao and Wang, 1995a and 1995b). It is a passive heat transfer device that could have an effective thermal conductance hundreds of times higher than the conductivity of copper. Therefore, intensive heat flux can be spread to a much larger surface area having a heat flux commensurate with that an engine working fluid could tolerate. Although a capillary effect may be employed to return the condensate from the condenser section to the evaporator section, the gravitational force should be employed in the present application to reduce the cost and increase the reliability. However, the use of traditional tubular heat pipes as a solar receiver may be impractical for large-scale power plants. In this case, hundreds or thousands of individual heat pipes may be needed, which could increase the power plant costs to an unacceptably high level. Additionally, the countercurrent liquid/vapor flows may significantly limit the heat transfer capacity of the traditional heat pipes. For these reasons, the concept of low-temperature separate-type heat pipes (Jin and Chen, 1986 and Chen et al., 1986) that are used for industrial waste heat recovery may be borrowed and adapted for the present high-temperature solar applications (Cao, 2012a).

It is a common practice in a waste energy recovery field that a separate-type heat pipe (STHP) or gravity-assisted loop heat pipe may be employed as a large-scale heat exchanger unit (Jin and Chen, 1986 and Chen et al., 1986). Likewise, a receiver module based on high-temperature separate-type heat pipes (STHPs) or gravity-driven loop heat pipes may be employed, which may have the benefits of receiving and transferring a large amount heat at high temperatures with only a few heat-pipe modules required. In a particular situation, a receiver wall may constitute a single STHP module even for a relatively large power plant. Figure 2 schematically illustrates a proposed STHP based receiver module that includes a evaporator having a liquid drum at the bottom and a vapor drum at the top, and a finned condenser heat exchanger having a liquid drum at the bottom and a vapor drum at the top (Cao, 2012a). The vapor drum of the condenser is connected with the vapor drum of the evaporator through a vapor line while the liquid drum of the condenser is connected to the liquid drum of the evaporator through a liquid line to form a looped heat pipe system. The condenser heat exchanger is positioned at a sufficiently higher elevation as compared to the elevation of the evaporator to provide a sufficiently large gravitational driving force for the operation of the heat pipe.

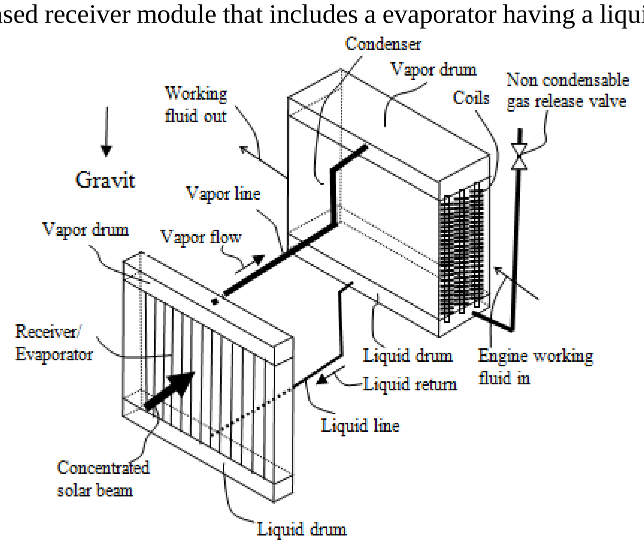

Fig. 2 Schematic of a high-temperature separate-type heat pipe module as a solar receiver unit (Cao, 2012a).

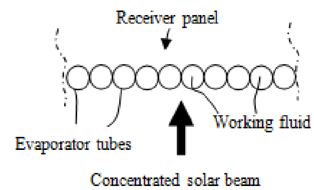

Fig. 3 Schematic of a receiver panel consisting of evaporator tubes between the liquid and vapor drums.

During the operation, the evaporator receives concentrated solar flux and converts it into heat that is then transferred to the working fluid inside the evaporator through boiling/evaporation/forced convection. The vapor generated inside the evaporator flows upward to the vapor drum and then through the vapor line into the vapor drum of the condenser, where it is redistributed to the interior spaces of finned tubes (the fins are optional depending on specific applications). The vapor flows downwardly inside the tubes and is condensed by transferring the heat to the working fluid of an engine system (not shown). The condensate is accumulated in the liquid drum and flows downwardly under the influence of gravity through the liquid line to the liquid drum of the evaporator, to supply liquid for boiling/evaporation/forced convection in the evaporator, and the aforementioned cycle is repeated.

Although the construction of the evaporator may have a number of options, for a large solar receiver, an evaporator that comprises a number of evaporator tubes between the vapor drum and the liquid drum may be employed. Figure 3 shows the cross-section of a tubes panel that forms the evaporator in Fig. 2. The panel consists of an array of tubes that are connected to the vapor drum at the top and to the liquid drum at the bottom and filled with an amount of heat pipe working fluid. For high-temperature operations, established working fluids are sodium that has a working temperature range of 900-1450 K and lithium that has a working temperature range of 1300-2000 K. These two working fluids have been employed and extensively tested in conventional heat pipes. The filling amount of sodium or lithium into the STHP is also very small as compared to the entire size of the STHP module, primarily a fraction of the interior volume of the evaporator.

Some of the biggest advantages of the separate-type heat pipe (STHP) system as compared to other types of heat pipes are large heat transfer loads and the flexibility of deploying the evaporator or condenser. In this case, the evaporator may be attached to a frame structure of a solar tower (not shown) and forms the outmost surface of the tower to receive concentrated solar flux from a collector system, while the condenser may be anchored to an inner structure of the tower (not shown) to facilitate the heat transfer from the condenser surface to the working fluid of a heat engine.

Although the evaporator of a single STHP module may provide a relatively large solar receiver wall and receive a large amount solar energy over 1000 kW, in some cases the size of a single evaporator may be still unable to accommodate the size of the collector or to meet the requirement for the heat load of the solar receiver system. For these reasons, a solar receiver wall in a receiver system may be constructed by a number of STHP solar receiver modules, as shown in Fig. 4 (Cao, 2012a). Individual

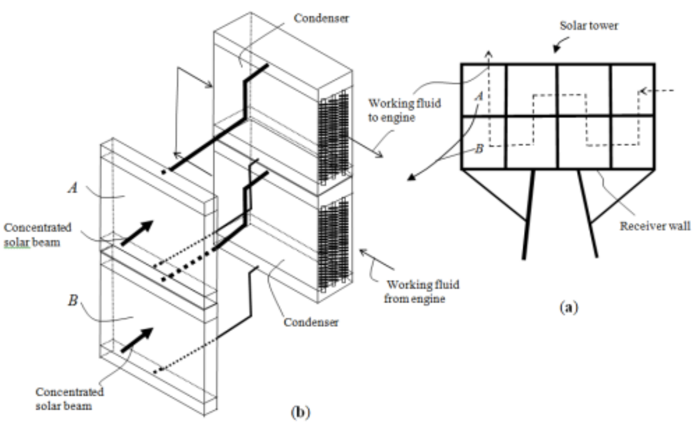

Fig. 4 Stacking arrangement of individual STHP modules (4a: A schematic frontal view of a receiver wall consisting of a number of STHP evaporators; 4b: Two STHP receiver modules with their evaporators being stacked one on top of the other)

Individual modules may be simply assembled side by side to form a wider receiver wall. Additionally, the evaporators of individual heat pipe modules may be stacked one on top of another, such as the stacking arrangement of evaporators *A* and *B* shown in Fig. 4b, to form a taller receiver wall. It may also be important that the gaps or inter-space in Fig. 4a and Fig. 4b between adjacent evaporators be minimized to form an integral receiver wall. Although the concentrated solar flux striking the receiver wall may be substantially uniform, the wall surface temperature may vary substantially, as the wall temperature is largely controlled by the engine working fluid temperature in the condenser. For this reason, the schematic serpentine arrangement of the engine working fluid in Fig. 4a could result in peak temperatures only over the last few heat pipe evaporators, which could substantially reduce the demand for high-temperature alloy materials for the construction of the receiver wall.

It should be noted that the vapor drum of an individual evaporator, as shown in Fig. 2, may be subjected to intensive concentrated solar flux once it is integrated with other evaporators to form a larger receiver wall, such as that in Fig. 4a. In this case, certain working fluid overcharging in the evaporator may be needed to avoid overheating the evaporator drum. However, this overcharging may result in a vapor-liquid two-phase mixture within the vapor drum, which may affect the heat transfer efficiency of the heat pipe system. To mitigate this problem, the vapor drum may be spaced back away from the evaporator and connected to the evaporator through a transition region as shown in Fig. 5 (Cao, 2012a). With this arrangement, the stacking of individual evaporators will not be affected and potential overheating of the vapor drum is avoided.

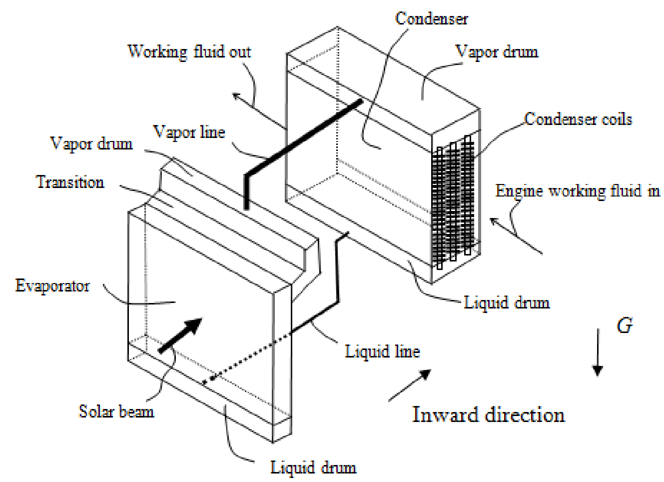

Fig. 5 A schematic view of a separate-type heat pipe receiver module with the vapor drum of the heat pipe evaporator being spaced away from the evaporator

The adoption of the separate-type heat pipe as the solar receiver has the following distinct advantages:

- **High heat transfer capacity**. In a conventional longitude heat pipe that combines the evaporator and condenser into a single pipe, the vapor flow and liquid return are countercurrent. When the heat transfer load is high accompanied by a high vapor velocity from the evaporator to condenser, this high vapor velocity may retard the condensate return from the condenser to the evaporator and cause a so-called entrainment limit. In a separate-type heat pipe, however, the vapor flow and liquid return are concurrent and the entrainment limit mentioned above no longer exists. As a result, the heat transfer capacity can be significantly increased. In low-temperature waste-heat recovery industrial applications using water as the working fluid, a heat transfer rate of more than 2000 kW per STHP module is not unusual.
- **High heat transfer coefficient in the evaporator.** In a STHP evaporator, the heat transfer is the combination of boiling and forced convection. While the boiling heat transfer coefficient may not be exceptionally high, having a similar magnitude to that of water pool boiling, the forced convection component may be significantly enhanced, which may be many times higher than the boiling component. The combined high heat transfer coefficient is essential to handle a high solar flux for increased receiver efficiency. Additionally, the heat transfer limitation using sodium as the working fluid is related to the critical heat flux of sodium pool boiling that may be above 200 W/cm^2 (Noyes, 1963).

- **Non-condensable gas release valve.** One problem that may be encountered by a high-temperature heat pipe is the non-condensable gas generation inside the heat pipe. For a conventional heat pipe, the gas generation over time may reduce the heat transfer capability substantially. For a STHP, a non-condensable gas release valve may be installed, as shown in Fig. 2. The non-condensable gas generated may be periodically discharged out of the separate-type heat pipe, so that its performance can be maintained over a long period of time, which significantly reduces the maintenance costs.
- **Simple structure and no pumping power requirement**. The STHP has a very simple structure without the need for a wick structure. Additionally, the heat pipe is driven by gravity, which eliminates the need for a pumping system and related power consumption for the receiver. By using a tube panel structure as shown in Fig. 3, a receiver wall of 6×6 m^2 may be constructed by using a few STHP modules. The amount of the high-temperature metal used to construct the STHP receiver should be close to that required in a heat transfer fluid based solar receiver, because of similar receiver size and the heat transfer surface areas in the heat exchanger between the heat transfer fluid and the engine working fluid.
- **High heat flux spreading.** The condenser inside surface area may be 10 times higher than that of the evaporator. With the installation of the fins outside of the condenser tubes (Fig. 2), the condenser heat transfer surface area may be 100 times higher than that of the evaporator. Therefore, the 100 W/cm^2 heat flux in the receiver may be spread to 1 W/cm^2 in the condenser. With this low heat flux, the flow velocity of the engine working fluid in the condenser may be sufficiently low to minimize the loss of pressure head. This reduced pressure loss is essential to a gas turbine engine, as its thermal efficiency is directly related to the gas pressure before entering the turbine [Moran, et al., 2009]. The liquid-metal working fluid charge into the heat pipe is also very small, as it may be a fraction of the evaporator volume that is less than 10 percent of the entire heat pipe interior volume.

3. INTEGRATION OF THERMAL ENERGY STORAGE (TES) MECHANISM WITH THE SOLAR POWER PLANT

The above introduced STHP based solar receiver does not employ a heat transfer fluid. However, the hot/cold tanks mechanism that has been successfully used in heat transfer fluid based solar power plants may also be employed for a solar power plant without the use of a heat transfer fluid. In this case, the working fluid of the heat engine may replace the functionalities of the heat transfer fluid to carry energy from the receiver to the storage tank or carry energy from the storage tank to the heat engine. Figure 6 illustrates schematically a solar tower power plant based on a simple gas turbine cycle and integrating a hot/cold tank TES system. In this case, before entering the turbine, the hot engine working fluid that carries thermal energy

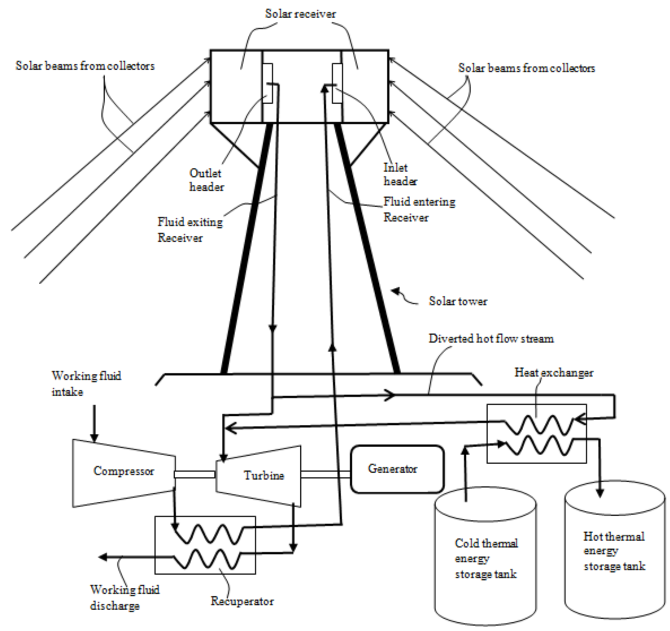

Fig. 6 Schematic illustration of the thermal charge process of a hot/cold tank TES system integrated with a solar tower power plant without employing a heat transfer fluid (Cao, 2012a)

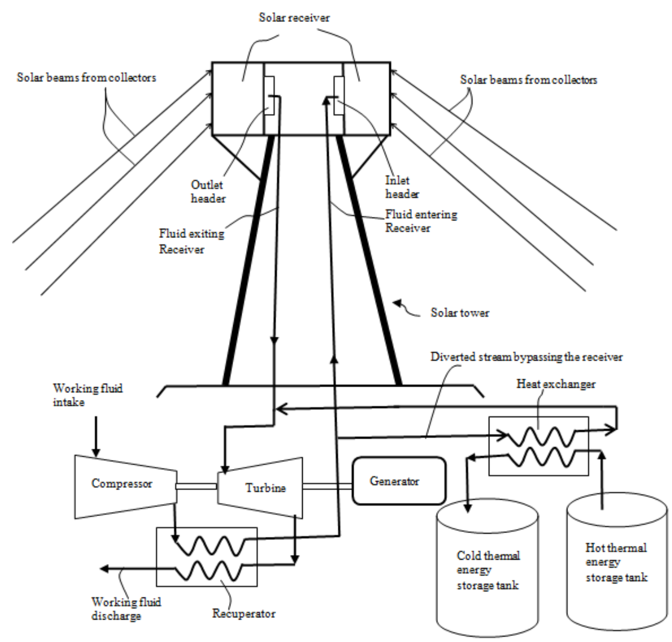

Fig. 7 Energy release of the hot/cold tank TES system integrated with a solar tower power plant without employing a heat transfer fluid (Cao, 2012a)

from the solar receiver is partially or completely diverted to a heat exchanger associated with the hot and cold TES tanks. Through the heat exchanger, thermal energy is transferred from the hot engine working fluid to the TES material that is pumped from the cold TES material tank to the hot TES material tank to store the thermal energy therewithin. After passing an amount of thermal energy to the TES system, the working fluid is routed back to the turbine where some remained energy is converted into power. If the working fluid temperature at the exit of the turbine is too low, it may bypass the recuperator (not shown).

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The above process is the thermal energy charge of the TES system. Figure 7 shows schematically the energy release process of the TES system. In this case, before entering solar receiver to acquire heat, the engine working fluid exiting the recuperator may be partially or completely diverted to the heat exchanger to receiver heat from the TES system, while the hot TES material is pumped from the hot tank to the heat exchanger. After transferring an amount of heat to the engine working fluid, the TES material with a reduced temperature returns back to the cold tank. At the same time, after acquiring an amount of heat from the TES system, the working fluid by-passes the solar receiver and enters the turbine to convert thermal energy into mechanical power. Alternatively, the engine working fluid exiting the heat exchanger may be routed back to the solar receiver (not shown). The above TES systems are based on the well-established hot/cold tank systems. Other options of TES systems as well as different heat pipe receiver configurations may also be possible (Cao, 2012a), which are not the topics of this paper.

4. EVALUATION OF THE HEAT TRANSFER COEFFICIENT IN THE EVAPORATOR OF THE STHP RECEIVER

As discussed earlier in this paper, for a STHP to be successfully employed as a solar receiver of a CSP system, it must be able to handle a high solar flux on the order of 50 to 100 W/cm^2, which would demand a very high heat transfer coefficient between the liquid metal working fluid and interior surface of the evaporator tube. Despite a very high thermal conductivity of the liquid metal, its boiling heat transfer coefficient is generally not exceptionally high, which may be a concern for high-temperature heat pipes to achieve a high flux. However, this difficulty may be resolved in a STHP, as the heat transfer in the evaporator of a STHP is the combination of boiling and forced convection.

The combined heat transfer coefficient for convective boiling in an upward vertical tube may be evaluated by the following correlation (Chen, 1963 and Faghri and Zhang, 2006):

$$h = Fh_l + Sh_b \tag{2}$$

where h_l and h_b are, respectively, the heat transfer coefficients for single-phase convection of the liquid and nucleate boiling. F and S are dynamic factors that modify the contribution of single-phase liquid convection and nucleate boiling, respectively, and may be evaluated by the following relations:

$$F = 2.35(0.213 + X_{tt}^{-1})^{0.736}, \text{ for } X_{tt}^{-1} > 1 \text{ ; and } F = 1 \text{ for } X_{tt}^{-1} \le 1 \tag{3}$$

$$S = 1/(1 + 2.53\times 10^{-6}\,\mathrm{Re}_{TP}^{1.17}) \tag{4}$$

where $\mathrm{Re}_{TP} = \mathrm{Re}_l F^{1.25}$ and X_{tt} is the Lockhart-Martinelli parameter defined by the following equation:

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{5}$$

The single-phase heat transfer coefficient for liquid may be calculated by using the following correlation for liquid metals with a constant heat flux boundary condition (Bergman et al., 2011):

$$Nu_D = 4.82 + 0.0185(\mathrm{Re}_D\,\mathrm{Pr})^{0.827} \tag{6}$$

The boiling heat transfer coefficient may be approximated by the following relation:

$$h_b = 4.0(q'')^{2/3} \tag{7}$$

Considering a sodium STHP solar receiver that has an overall receiver wall surface area of 6×6 m^2, the length of evaporator tube as shown in Fig. 2 and Fig. 3 is taken to be 6 m. In the following representative calculations, the tube outer diameter and inner diameter are taken to be 43 mm and 40 mm, respectively. A concentrated solar flux of 100 W/cm^2 and a working temperature of 1200 K are considered, and the average heat transfer coefficient with an average vapor quality of $x_v = 0.5$ is calculated.

By assuming saturated conditions in both liquid and vapor drums, the mass flow rate of the working fluid in a single tube is:

$$\dot{m} = q''HD_o / h_{fg} = 10\times 10^6 \times 6 \times 0.043 / 3.829\times 10^6$$
$$= 0.0674 \quad \text{kg/s}$$

Then,

$$\mathrm{Re}_l = \frac{4\dot{m}(1-x_v)}{\pi D_i \mu_l} = \frac{4\times 0.0674\times(1-0.5)}{\pi\times 0.04\times 1.514\times 10^{-4}} = 7085.0$$

Therefore

$$h_l = \frac{k_l}{D_i} Nu_D = \frac{k_l}{D_i}(4.82 + 0.0185(\mathrm{Re}_D\,\mathrm{Pr})^{0.827})$$
$$= \frac{47}{0.04}(4.82 + 0.0185(7085\times 0.00415)^{0.827})$$
$$= 6019.6 \quad \text{W/m}^2\text{-K}$$

According to Eq. (7),

$$h_b = 4.0\times(10^6)^{(2/3)} = 4.0\times(1\times 10^6)^{(2/3)} = 40000 \quad \text{W/m}^2\text{-K}$$

According to Eq. (5),

$$X_{tt} = \left(\frac{1-x_v}{x_v}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}$$
$$= \left(\frac{1-0.5}{0.5}\right)^{0.9}\left(\frac{0.379}{730}\right)^{0.5}\left(\frac{1.514\times 10^{-4}}{0.2257\times 10^{-4}}\right)^{0.1} = 0.0282$$

Therefore,

$$F = 2.35(0.213 + X_{tt}^{-1})^{0.736} = 2.35 \times (0.213 + (0.0282)^{-1})^{0.736}$$
$$= 32.63$$

Also,

$$\mathrm{Re}_{TP} = \mathrm{Re}_l F^{1.25} = 7085 \times 32.63^{1.25} = 5.53 \times 10^5$$

$$S = 1/(1 + 2.53 \times 10^{-6} \times \mathrm{Re}_{TP}^{1.17}) = 1/(1 + 2.53 \times 10^{-6} \times (5.53 \times 10^5)^{1.17})$$
$$=0.07$$

Finally,

$$h = Fh_l + Sh_b = 32.63 \times 6019.6 + 0.071 \times 40000$$
$$= 1.96 \times 10^5 + 2840 = 1.99 \times 10^5 \quad \mathrm{W/m^2\text{-}K}$$

According to the above result, the direct contribution of boiling to the overall heat transfer coefficient is very small. However, its agitation and enhancement on the forced liquid convective heat transfer has enabled a very high convective heat transfer coefficient. The temperature drop between inner surface of the evaporator tube and the heat pipe working fluid may be calculated as follows,

$$\Delta T = q''/h = 10^6 / 1.99 \times 10^5 = 5.0^o C$$

The above result may indicate that the use of a STHP may be able to overcome the difficulty of relatively low heat flux of other high-temperature heat pipes and handle a high heat flux needed for CSP applications. It should be pointed out that the above temperature drop is only associated with that between the inner surface and the sodium working fluid, not including the temperature drop across the pipe wall that is related to the conductivity of the tube wall material. Depending upon a complete design of a STHP, other temperature drops over other sections of the system may occur, which is beyond scope of this paper. It should also be pointed out that the sodium filling in a SHTP is generally very small, corresponding to one third or one half of the inner volume of the evaporator, but its filling may significantly affect the performance of the STHP.

5. CONCLUSIONS

In this paper, the concept of low-temperature STHP in waste heat recovery industry is adapted for CSP solar receiver systems. The STHP based solar receiver has the advantages of high heat transfer capacity, high heat transfer coefficient in the evaporator to handle a high concentrated solar flux, non-condensable release mechanism, and lower costs. The STHP has a very simple structure, driven by gravity, and does not increase the use of high-temperature alloy materials. A preliminary calculation indicates that the heat transfer in the evaporator is exceptionally high due to the drastically enhanced convective heat transfer. This high heat transfer coefficient may allow a STHP solar receiver to handle a solar flux above 100 W/cm^2. It is believed that the introduced STHP receiver is an enabling method for a gas turbine based solar power plant as the heat flux at the receiver must be spread to a much lower level for heat transfer to the gaseous engine working fluid.

Although the illustration of the STHP solar receiver is based on a solar tower power plant, the concept can also be employed for other solar power plants, including parabolic dish, Stirling engine, and Fresnel plants. The introduced STHP may also be employed for solar reactors of thermochemical storage systems (Cao, 2012b).

REFERENCES

Bergman, T.L, Lavine, A.S., Incropera, F.P., and Dewitt, D.P., 2011, *Fundamentals of Heat and Mass Transfer*, 7th Edition, John Wiley & Son.

Cao, Y. and Faghri, A., 1991, "Transient Multidimensional Analysis of Nonconventional Heat Pipes with Uniform and Nonuniform Heat Distributions," *ASME J. Heat Transfer*, Vol. 113, pp. 995-1002.
http://dx.doi.org/10.1115/1.2911233

Cao, Y. and Faghri, A., 1992, "Closed-Form Analytical Solutions of High-Temperature Heat Pipe Startup and Frozen Startup Limitation," *ASME J. Heat Transfer*, Vol. 114, pp. 1028-1035.
http://dx.doi.org/10.1115/1.2911873

Cao, Y. and Faghri, A., 1993, "A Numerical Analysis of High-Temperature Heat Pipe Startup from the Frozen State," *ASME J. Heat Transfer*, Vol. 115, pp. 247-254.
http://dx.doi.org/10.1115/1.2910657

Cao, Y. and Wang, Q., 1995a, "Reciprocating Heat Pipes and Their Applications," *ASME J. of Heat Transfer*, Vol. 117, 1094-1096.
http://dx.doi.org/10.1115/1.2836293

Cao, Y. and Wang, Q., 1995b, "Thermal Analysis of a Piston Cooling System with Reciprocating Heat Pipes," *Journal of Heat Transfer Engineering*, Vol. 16, pp. 50 - 57.
http://dx.doi.org/10.1080/01457639508939853

Cao, Y., 2012a, Heat Pipe Receivers for Concentrated Solar Power, US patent pending.

Cao, Y., 2012b, Heat Pipe Solar Thermochemical Reactor Units, US patent pending.

Chen, Y.G., Gao, M., and Xin, M.D., 1986, "Experiments of Heat Transfer Performance of Separate-Type Thermosyphon," *Int. Heat Pipe Symposium*, Osaka, Japan.

Chen, I.C., 1963, "A correlation for boiling heat transfer to saturated fluids in convective flow," ASME preprint 63-HT-34, Presented at the 6th National Heat Transfer Conference, Boston, MA.
http://dx.doi.org/10.2172/4636459

DOI: 10.5098/fhp.6.1
ISSN: 2155-658X

Faghri, A., 1995, Heat Pipe Science and Technology, Taylor and Francis.

Faghri, A. and Zhang, Y., 2006, Transport Phenomena in Multiphase Systems, Elsevier, New York.

Jin, M.C. and Chen, Y.G, 1986, Heat Pipes and Heat Pipe Exchangers, Chongqing University Press, Chongqing, China.

Noyes, R.C., 1963, "An Experimental Study of Sodium Pool Boiling Heat Transfer," J. Heat Transfer -- May 1963 -- Volume 85, Issue 2, 125.
http://dx.doi.org/10.1115/1.3686026

Ling, J. and Cao, Y., 2001, "Experimental Investigations of a Radially Rotating Miniature High-Temperature Heat Pipe for Turbine Blade Cooling," *ASME Journal of Heat Transfer*, Vol. 113, pp. 113-119.
http://dx.doi.org/10.1115/1.1332777

Lovegrove, K., Taumoefolau, T., Paitoonsurikarn, S., Siangsukone, P., Burgess, G., Luzzi, A., Johnston, G., Becker, O., Joe, W., and Major, G., 2007, "Paraboloidal dish solar concentrators for multi-megawatt power generation," In: Solar World Congress 2007, Beijing, China.

Ma, D., Tuchi, C., Neises, T., and Wagner, M., 2012, "Thermodynamic Study of Advanced Supercritical Carbon Dioxide Power Cycles for High Performance Concentrating Solar Power Systems," *ASME 6th Conference on Energy Sustainability*, ES-FuelCell2012-91179.
http://dx.doi.org/10.1115/es2012-91179

Mancini, T.R., Kolb, G.J., and Chavez, J. M., 1994, "Solar thermal power today and tomorrow," *Mechanical Engineering* 1994; 116(8): 74-79.

Moran, M.J., Shapiro, H.N., and Boettner, D.B., 2009, *Fundamentals of Engineering Thermodynamics*, 7th edition, John Wiley & Sons.

Nithyanandam, K. and Pitchumani, R., 2012a, "Analysis of a Latent Thermocline Energy Storage System for Concentrating Solar Power Plants," *ASME 6th Conference on Energy Sustainability*, ES-FuelCell2012 – 91389.
http://dx.doi.org/10.1115/es2012-91389

Nithyanandam, K. and Pitchumani, R., 2012b, "Numerical Analysis of Latent Thermal Energy Storage System with Embedded Thermosyphons," *ASME 6th Conference on Energy Sustainability*, ES-FuelCell2012 - 91416.
http://dx.doi.org/10.1115/es2012-91416

Noyes, R.C., 1963, "An Experimental Study of Sodium Pool Boiling Heat Transfer," J. Heat Transfer -- May 1963 -- Volume 85, Issue 2, 125.
http://dx.doi.org/10.1115/1.3686026

Stine, W.B, and Diver, R.B., 1994, "A compendium of solar dish/stirling technology," Sandia National Laboratories Report, SAND 93-7026.
http://dx.doi.org/10.2172/10130410

Tao, Y. and Rayegan, R., 2011, "SOLAR ENERGY APPLICATIONS AND COMPARISONS," in Energy and Power Generation Handbook - ASME, Editor: K. R. Rao.
http://dx.doi.org/10.1115/1.859551

Yang, M., Yang, X., and Ding, J., 2009, "Heat transfer enhancement and performance of the molten salt receiver of a solar power tower." *Applied Energy*, 87(9): 2808-2811.
http://dx.doi.org/10.1016/j.apenergy.2009.04.042