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SOLAR THERMAL ENERGY CONVERSION SYSTEM WITH THERMO CLINE STORAGE FOR PROCESS STEAM GENERATION IN MILK DAIRIES

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Abstract- Energy has major role to play in processing industries like milk and dairies in India. Andhra Pradesh and Telangana States are at the third place in the country with respect to milk production. Milk and dairy industry has a potential thermal energy requirement of 120^{0}C or more in the operations such as cleaning & washing, pasteurization, sterilization,evaporation and chilling. Thermal energy storage with concentrated solar power using Parabolic trough Collectors (PTC)can be a betteralternative solution to supply the process steam to milk processing units for extended hours of plant operation. The present work estimates the energy requirement to produce around40 kg/hr steamfor every 1000 litres of milk requirement. A 32m^{2}PTC with a thermocline tank is modelled and analysed for supplying the required amount of steam for t+2, t+4 and t+6 hours of operation, where t is number of sunshine hours in that particular location. Daily Process steam generation through solar thermal energy conversion is modelled for various favourable sites in south India for year duration.Design procedure for calculating the size of PTC, collector field array and thermal energy storage is presented. The equivalent electrical energy units generated during the same plant hours is also presented. Anantapur is found out to have maximum steam generation with minimum number of collectors for the year-round operation of the plant. This work elucidates the use of solar thermal power with energy storage for process steam requirements in medium scale milk dairies.

Keywords: Thermal Energy Storage, Thermocline, parabolic trough, molten salt, Stratification.

I. INTRODUCTION

India is endowed with rich solar energy resource. The average solar radiation intensity received on India is 200 MW/km square (megawatt per kilometer square) [1]. With a total geographical area of 3.287 million km square, this amounts to 657.4 million MW. With the increase in awareness among the people about renewable energy, governments across the world are introducing new policies and providing incentives to industries as well as individuals who are employing renewable energy technologies. As a result, solar energy is becoming more economically attractive. It is estimated that one third of the energy used by the dairy industry is actually in the processing operations [2]. So, dairy industries have a huge scope to substitute their conventional energy sources with PTC based solar energy systems. Steam is consumed at various stages in a dairy processing plant such as milk pasteurization, cream pasteurization, cleaning process for tanks etc. Steam generated through conventional methods is costlier therefore a PTC based solar plant with thermocline Thermal Energy Storage (TES) is designed and modeled for small scale milk processing plant. For this purpose, solar data for ten different locations of Andhra Pradesh and Telanganais analyzed. At these locations, in actual practice, milk processing units are functioning with ample rooftop space. The adoption of solar thermal energy-based plants to supply the process steam, is a prospective option for many of the existing dairy plants at the given locations.

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II. DESIGN, MODELING AND ANALYSIS OF SOLAR THERMAL BASED STEAM GENERATION SYSTEM

Solar energy is abundantly available at all the commercial dairy units that are located in south India. The rooftop area of these commercial plants can be utilized to install solar thermal energy equipment to generate process steam required for the units. Presently majority of the pasteurization units are dependent on coal and other fossil fuels for heating the feed water to generate process steam for various processes. Solar thermal power plant with molten salt-based storage will save the operating costs of the fuel followed by the benefit of reducing hazardous emissions. Dairy milk units are majorly are classified as industrial milk processing units and fluid milk processing units. Former will have a capacity of 7 Lakh liters per day of milk and the later may have up to a few thousands [5]. Every milk processing unit needs process steam at a pressure of 4-5 bar where pasteurization is taking place at 70^0C. Pasteurization, sterilization, washing, cleaning, evaporation and spray drying are a few operations in milk processing units that consume around 500 kWh of electricity per tonne of milk produced [6]. Solar thermal energy conversion cycle will be a sustainable solution for the milk processing unit with attractive payback option. In this work, an energy conversion system based on concentrated solar thermal power is designed, modeled and analysed for process steam application of fluid milk processing units. The procedure starts with solar resources estimation at selected locations followed by designing the solar thermal collector array, storage system and ends with performance analysis of the cycle.

A.*Solar resource data and site analysis*

Solar irradiance data of ten different locations to design a suitable steam generation plant for dairy plants modeled by taking the reference data [3]& [4]. Fig. 1 denotes the average annual solar irradiance received in the respective locations.

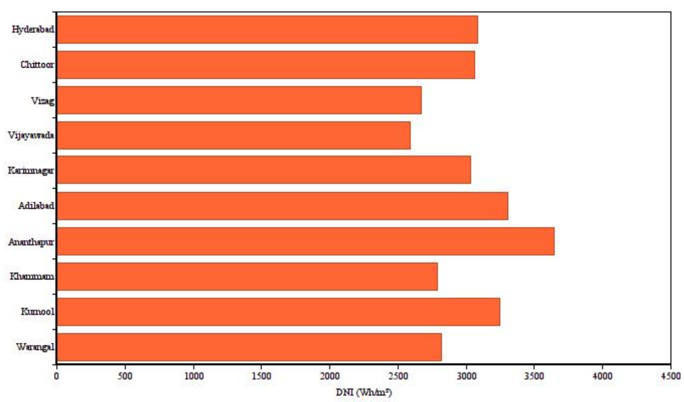

Figure 1:DirectNormal Irradiance (DNI) at different locations

Ananthapuramu location receives the highest amount of annual solar irradiance of about 3645Wh/m² while it is followed by Adilabad, Kurnool with average solar irradiance of 3303Wh/m² & 3245Wh/m².

B.*Process steam requirement in dairy industry*

A medium scale milk processing unit will have a continuous steam requirement for majority of the processes given in the table 1. When milk is arrived at the unit, it is cooled down to a temperature below 5^oC and stored. This stored milk is then sent via tubes to the pasteurization chamber where it is heated to 72^oC and maintained at that temperature for duration of 15 seconds. Then it is further cooled down to 5^oC and is then sent to the packaging unit where it is packed and shipped for distribution.The steam requirement for various operations performed in the milk processing dairy plant are specified in table 1.

Heat absorbed by the milk is given by

$$Q_{milk} = W_{milk} * \rho * C_p * (T_2 - T_1) \qquad [5]$$

Note: The usual value of density of the milk is found out to be 1032 kg/m^3. The specific heat of milk is 3.93 kJ/kg-K.The steam requirement is given by the equation below.

Table 1: Steam and Energy consumption in a commercial dairy plant [6]

Particulars	**Steam consumed (kg /1000 l milk)**	**Thermal energy (GJ/1000 l milk)**
Milk Pasteurization	**37.26**	**0.100**
Cream Pasteurization	**97.57**	**0.262**
Particulars	**Steam consumed (kg/day)**	**Thermal energy (GJ/day)**
Milk Pasteurization cleaning	**802.2**	**2.156**
Cream Pasteurization cleaning	**584.85**	**1.571**
Raw milk storage tank cleaning	**251.05**	**0.675**
Pre pack cleaning	**556.02**	**1.494**
Aging tank cleaning	**43.66**	**0.177**

C.*Design and modeling of solar thermal energy conversion cycle*

A schematic design of the PTC plant is shown in the Fig. 2. It consists of a solar field array, thermal energy storage tank and a steam generator. The solar field array consists of parabolic trough collectors with receiver tubes for the flow of Heat Transfer Fluid (HTF) throughout the field. Each collector has a reflector trough, receiver tube, support structure. This receiver tube will run along the loops till it reaches the heat exchanger and storage area. The Thermocline tank is used to maintain the stratification in the zone where the hot and cold fluids meet.The steam generator is a shell and tube heat exchanger that has molten salt flowing in a tube and feed water transferring into the steam in the shell.

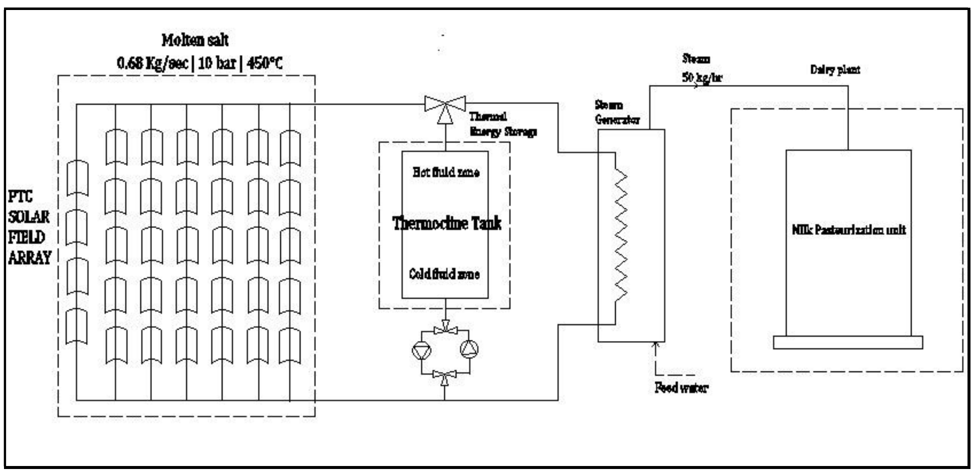

Figure 2: Schematic of the Plant

i) *Design of Parabolic Trough Collector*

The PTC module is modeled and designed for the small-scale process heat requirements. Initially some assumptions are required in the procedure. The assumptions for aperture width, length of the collector and rim angle of the trough collector were made keeping in mind the optimal performance of the module. Aperture width and length of the collector were fixed as 4 meters and 8 meters respectively. The rim angle is taken as 90°. The gap between inner and outer diameter of receiver is taken as 4 mm and in glass cover is taken as 6 mm. The steps below illustrate the design terminology and procedure for a standard PTC.

Focal length of the collector (f):

$$W_a = 4 * f * \tan\left(\frac{\theta_r}{2}\right) \quad [7]$$

Height of the collector (h_c):

$$16 * h_c * f = W_a^2 \quad [7]$$

Aperture Area (A):

$$A = W_a * l_c$$

Concentration Ratio (CR):

$$CR = \frac{effective\ aperture\ area}{reciever\ tube\ area} = \frac{(W_a - D_o) * l_c}{\pi * D_o * l_c} \quad [8]$$

Optical Efficiency of collector (η_o):

$$\eta_o = \rho * \tau * \alpha * \gamma * (1 - A_f * tan(\theta_i) * cos(\theta_i)) \quad [9]$$

In optical efficiency the incidence angle (θ_i) = 0° because the parameters are evaluated at normal to the incidence of solar beam radiation.

Hence

$$\eta_o = \rho * \tau * \alpha * \gamma$$

Thermal efficiency of collector (η_t):

$$\eta_t = Fr\left(\eta_o - \frac{U_l}{CR}\left(\frac{T_i - T_a}{S}\right)\right) \quad [10]$$

Radiation absorbed by the receiver (S):

$$S = I_b * \rho * \tau * \alpha * \gamma * K(\theta) \quad [11]$$

Where K(θ) is the incidence angle modifier which is equal to unity at direct tracking

Collector Heat Removal factor (Fr):

$$Fr = \frac{m * C_p}{\pi * D_o * l_c * U_l} * \left(1 - e^{-\left(\frac{F' * U_l * \pi * D_o * l_c}{m * C_p}\right)}\right) \quad [12]$$

Collector fin efficiency (F'):

$$F' = \frac{1}{U_l * \left(\frac{1}{U_l} + \frac{D_o}{D_i * h_f} + \left(\frac{D_o}{2 * K_f} ln\left(\frac{D_o}{D_i}\right)\right)\right)} \quad [12]$$

Heat transfer coefficient from receiver wall surface to working fluid (h_f):

$$h_f = \frac{K_f * Nu}{D_i} \quad [12]$$

Nusselt Number for the receiver (Nu):

$$Nu = 0.023 * Re^{0.8} * Pr^{0.4} \quad [12]$$

Reynolds Number (Re):

$$Re = \frac{4 * m}{\pi * D_i * \mu} \quad [13]$$

Prandtl Number (Pr):

$$Pr = \frac{\mu * C_p}{K_f} \quad [13]$$

Overall heat transfer coefficient (U_l):

$$U_l = \frac{Q_{loss}}{A_r * (T_r - T_a)} \quad [13]$$

Where $Q_{loss} = K_2 * (T_r^{\,4} - T_a^{\,4})$

$$K_2 = A_{ro} * \varepsilon_r^{\,*} * \sigma * \left(1 + \frac{4 * T_a^{\,3} * A_{ro} * \varepsilon_r^{\,*} * \sigma}{K_1}\right)$$

$$K_1 = \left(A_{co} * \varepsilon_c * \sigma * 4 * T_a^{\,3}\right) + A_{co} * h_f$$

$$\varepsilon_r^{\,*} = \left(\frac{1}{\varepsilon_r} + \frac{1 - \varepsilon_c}{\varepsilon_c} * \frac{A_{ro}}{A_{ci}}\right)^{-1}$$

With the above design parameters the PTC is modelled and designed. The outcome of the above procedure is the specifications table 2 with design parameters of the trough.

Table 2: Specifications of the designed PTC collector

Aperture width	**4 metres**	**Receiver tube material**	**Stainless steel with cermet coating**
Length	**8 meters**	**Receiver cover material**	**Glass**
Aperture area	**32 m²**	**Reflectance**	**0.83**
Concentration ratio	**17.87**	**Absorptivity**	**0.96**
Optical efficiency	**69.64%**	**Transmittance**	**0.95**
Thermal efficiency	**54.79%**	**Emissivity of collector**	**0.9**

ii) *Performance of designed PTC Collector*

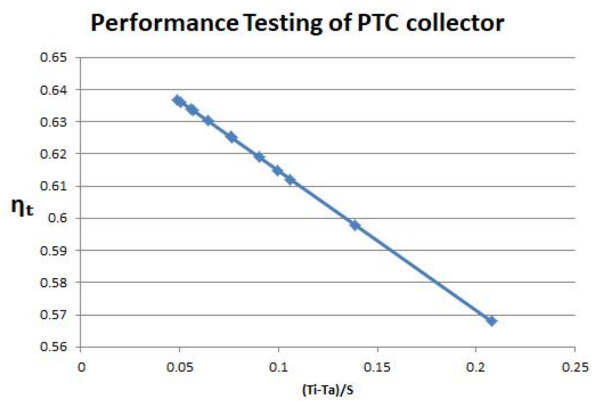

Figure 3: Performance testing of PTC collector

A graph is plotted between thermal efficiency of the designed PTC collector against the ratio of $\frac{T_i - T_a}{S}$.This relation shows that the designed PTC collector has optimum performance with the lowest value of$\frac{T_i - T_a}{S}$.

iii) Field array design

The solar field length here can be calculated by taking $L_c$as the solar field length in the above formula.
Number of Trough collectors required:

$$N = \frac{Total\ power\ required}{Power\ \ generated\ per\ trough}$$

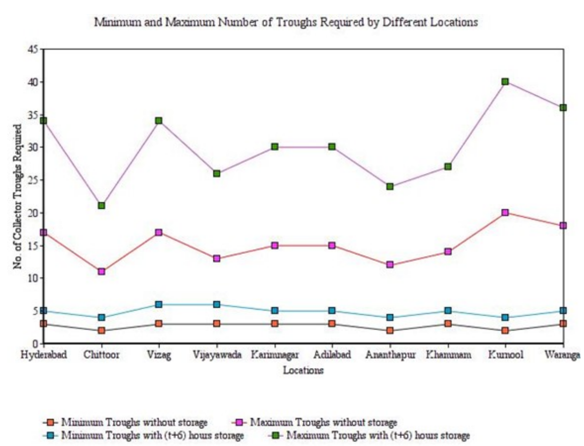

Figure 7: Number of trough collectors required for different locations

The Fig. 7 above explains the maximum and minimum number of troughs required for different cities to fulfil the steam requirement.

Mass flow rate of Heat Transfer Fluid:

$$Q = m_s C_{pw}(T_{max} - T_{min}) + (m_l * L_f) + m_s C_{ps}(T_{max} - T_{min})$$

Taking heat exchanger efficiency into account

$$\frac{Q}{0.9} = mC_p \Delta T$$

Mass flow rate heat is found from the 'm' term.

iv) *Process steam calculations*

Heat absorbed by the molten salt:

$$Q_u = mC_p(T_f - T_i) = F_i(W_u - D_o)L_c\left[S - \frac{U_l}{CR}(T_f - T_a)\right] \qquad [14]$$

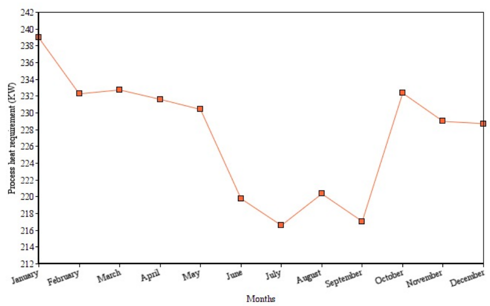

Figure 4: Monthly Process heat requirement for Ananthapuramu

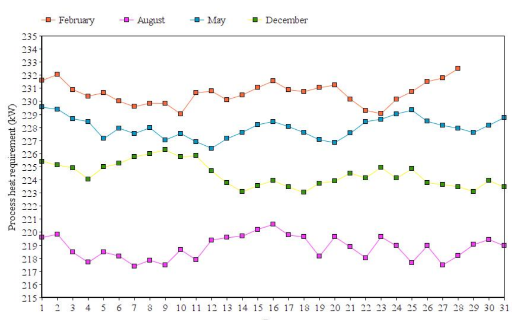

Figure 5: Comparison of daily process heat requirement for Ananthapuramu

The figures 4 & 5 illustrate the heat carried by the heat transfer fluid which is molten salt. This heat is used to generate steam in the steam generator and further this steam is used as a source for the processing operations in the dairy plant.

v) *Thermal power cycle calculations*

Assuming that steam runs a conventional Rankine cycle, the number of energy units generated with and with out storage facility for the selected locations is also calculated and represented in a graph. Figure 6 shows that Ananthapuramu will have maximum electrical energy units generated by assuming the regular Rankine cycle parameters and efficiencies.

Power generated by the plant is calculated as follows:

$$PG_{total} = \frac{I_{bn} * A_p * \eta_p * t_{plant}}{1000} \qquad [14]$$

Note: For power generated with storage the efficiency of plant value is multiplied by the storage efficiency which is taken as 0.75. The Fig. 6 below explains the average annual power generation in ten different locations based on the solar irradiance data for the sites. The power generation is analysed for four different cases including with and without storage scenarios.

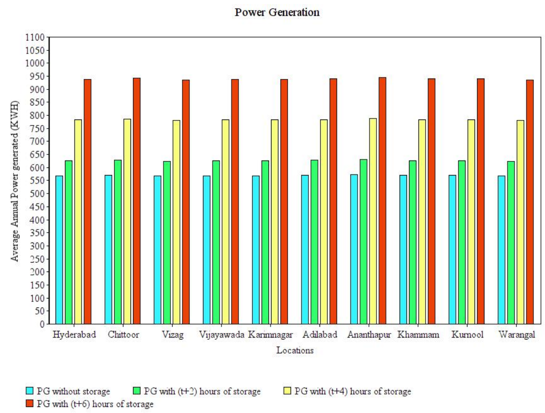

Figure 6: Power generation with CSP across different locations

vi) *Design and modelling of thermal energy storage*

A lot of research efforts have been recently focused on the integration of solar thermal energy storage (TES) as a viable means to enhance dispatchability, increase the value of concentrated solar energy and make the plant more reliable. There are a number of viable candidates for TES systems that might be developed and applied on a commercial scale for CSP plants. Presently, sensible molten salt TES systems including two-tank systems and one-tank thermocline systems are widely applied or under development worldwide , as molten salt used as the storage medium and direct heat transfer fluid can offer the best balance of capacity, cost, efficiency and usability at high temperatures . Design and analysis of a thermal energy storage tank was performed using ANSYS Fluent software [16]. The volume of the thermocline tank was calculated for t+2, t+4 and t+6 storage time.

Thermal Energy Storage calculations: [15]

The total heat stored in TES is calculated as

$$Q_{total} = \frac{mC_p\left(T_f - T_i\right) * t * 3600}{\eta_T}$$

The ideal and real volumes of the thermocline tanks are found as follows

$$V_{ideal} = \frac{Q_{total}}{\rho * C_p * (T_f - T_i)}$$

$$V_{real} = \frac{\rho * C_p * V_{ideal}}{\left[\rho_{filler} * C_{p_{filler}} * (1 - \varepsilon) + \rho * C_p * \varepsilon\right]}$$

$$V_{real} = \pi * \left(\frac{D}{2}\right)^2 * H$$

Note: (H/D) ratio is taken as 2.3948 for better efficiency.

Design specifications for different storage capacities:

Table 3: Different designs according to required storage capacities for thermocline tank

Storage time (hours)	Height (meters)	Diameter (meters)
t+2	3.386	1.414
t+4	4.266	1.78

t+6	**4.884**	**2.039**

C.SIMULATION OF THERMOCLINE TANK

The Transient flow analysis was carried out to check the formation of stratification zone of the molten salt in the thermal energy storage system. The simulation was performed for three different cases of storage capacities (t+2, t+4, and t+6 hours).The K-epsilon realizable model is used in solving the simulation. Type of solver used is SIMPLE [16]. The boundary conditions given to the model were wall based to give the storage walls insulation function. Numbers of time steps were taken as 6000 and the time step size as 0.01 and maximum iterations per time step are 10.

The results of the simulation for the three cases are shown in figure 8.

The Temperature contours of the thermocline tank are depicted for each case as shown in the Fig. 8. A graph is plotted with abscissa as vertical distance in the thermocline tank and ordinate as the temperature values corresponding to that point in the tank as shown in the Fig. 9. The required stratification in the thermocline tanks was formed in all the three cases. Better thermocline zone was formed in the latter cases than the former.

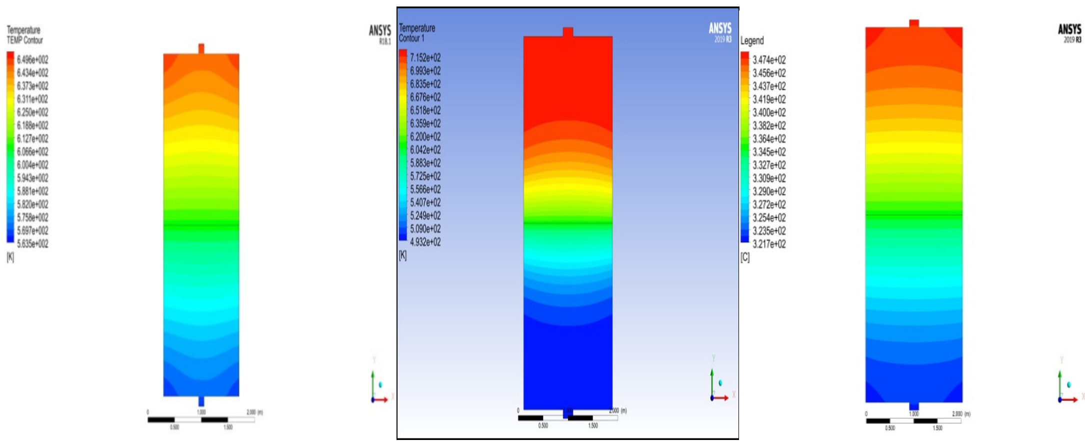

Figure 8: Stratification zones formed in t+2, t+4 & t+6 cases (from left to right)

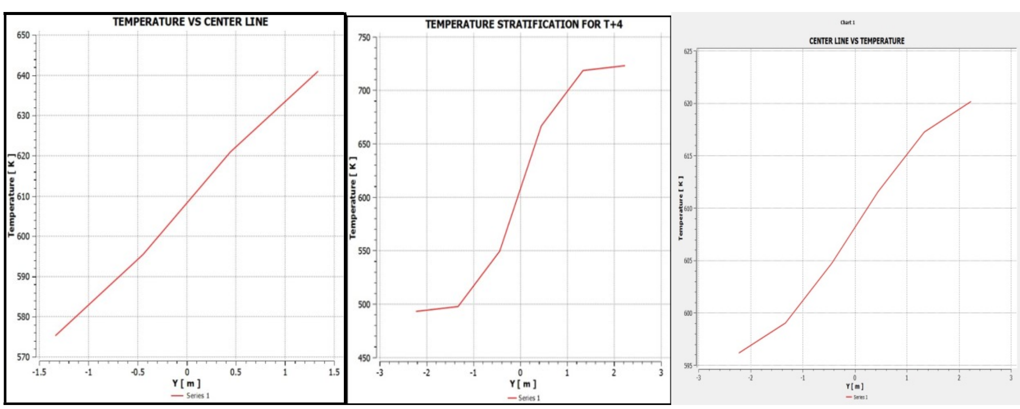

Figure 9: Temperature against height graph for t+2, t+4 & t+6 cases (from left to right)

IV. CONCLUSION

Process heating industry majorly involves the stake holders of milk processing units, textile industries and paper making factories. Solar thermal collector system with suitable energy storage is an ideal option for these process heat consumers. In this work, parabolic trough collector of 32m² has been modeled and designed. Transient CFD simulations of Thermocline tank for different storage durations have been carried out to investigate the formation of stratification zone. The temperature distribution across the Thermocline tank is observed. The key parameters considered for the design of the parabolic trough are solar radiation data, aperture width and height. A steam generation unit using parabolic trough collectors along with Thermocline tank is designed. The important aspects considered for the design and modeling of the plant are steam flow rate, mass flow rate of molten salt and location of the plant. The number of collectors required for each location is precisely calculated.

After fixing the number of collectors for a particular location, the process heat requirement so as to generate fixed amount of steam in kg/hour is estimated. Steam generation capacity of the designed plant is continuous and highest during February month and lowest during September month. Ananthapur and Adilabad turn out to be favorable locations with better process heat generation using solar thermal technology. This work gives procedure and guidelines to design a rooftop solar thermal energy conversion system with thermocline storage for fluid milk industries located in Indian urban cities. The present work successfully illustrated the feasibility of solar thermal energy conversion system to generate process steam for extended hours of plant cycle operation.

Nomenclature:

A - Collector Aperture Area (m²)

A_p - Total Aperture area of the plant (m²)

C.R. - concentration ratio

C_{pm} - Specific heat of milk (kcal/kg ⁰C)

C_{pw} - Specific heat of water (kJ/kg-K)

C_{ps} - Specific heat of steam (kJ/kg-K)

D_o - Outer diameter of receiver (m)

D_i - Inner diameter of receiver (m)

f - Focal length of collector (m)

Fr - Collector heat removal factor

F'- Collector fin efficiency

h_c - Height of the collector (m)

h_f - Heat transfer coefficient (W/m² K)

I_b - Solar irradiation (Watt/m²)

K(θ) - incident angle modifier

K_f - Thermal conductivity of fluid (W/m-K)

l_c - Length of collector (m)

m_s - Mass flow rate of molten salt (kg/sec)

N – No. of collectors required

Nu – Nusselt number

PG_{total} - Total power generated (kWh)

Pr – Prandtl number

Q_{milk} - Heat absorbed by milk (Watt)

Q_u - Heat absorbed my molten salt (Watt)

Re – Reynolds number

S – Radiation absorbed by receiver (Watt/m²)

t_p - Operating duration of the plant (hours)

U_l - Overall heat transfer coefficient (W/m² K)

W_{milk} – Mass flow rate of milk (m^3/hour)

W_{steam} – Mass flow rate of steam (kg/hour)

W_a - Aperture width of collector (m)

θ_r – Rim Angle of collector

T_1 - Lower temperature of milk (K)

T_2 - Temperature to which milk is heated (K)

T_i – Inlet temperature of molten salt (K)

T_f – outlet temperature of molten salt (K)

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