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EVALUATION OF SNAKING ISSUES IN 250KWe POWER PLANT BASED ON DIRECT STEAM GENERATION USING PARABOLIC TROUGH COLLECTOR

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Abstract: A Direct Steam Generation (DSG) 256 KWe plant based on parabolic trough solar collector at Shive, Pune, and India is studied. During operation of the plant the snaking of Heat Collection Element (HCE) is observed resulting in plant instability and damage of glass cover. The thermo-hydraulic analytical model has been developed for the performance prediction of DSG process under actual operating conditions. Various controlling parameters causing the snaking problem and the effects of the Flow variation, heat input, internal working pressure and amount of sub-cooling has been simulated in the model. The analysis shows that the higher pressure and lower sub-cooling at the constant mass flow rate and heat input will lead to the stable system.
Keywords: Direct Steam Generation (DSG), Heat Collection Element (HCE), Instabilities, parabolic trough collector (PTC), Solar Thermal Concentrators

INTRODUCTION

A solar thermal power generation demonstration plant was commissioned in 2011 under grant from Department of Science and Technology, Government of India to explore as renewable energy generation in rural areas. The plant uses parabolic trough collector (PTC) to generate low pressure saturated steam at low pressure to drive turbine. A unique phenomenon of bending of receiver tube was observed after commissioning of the plant. It is observed that at noon the wave is passing through the series of HCE from one end to the other, the bending of the HCE is more than 12.5mm on either side. From outside, the HCE looked like a moving snake and hence local operators called it as snaking effect and making system unstable and unsafe. Over 10 % of the glass tubes are broken due to the movement of receiver tube or HCE in the glass envelopes. Maximum bending observed in past is 50mm, [1-3]. The detailed study to understand this bending process is taken up which is explained in this paper through a Thermo-hydraulic model. Based on the understanding of the system and the model, corrective action was taken and the issue got resolved. All the DSG plants as per the literature available are operating at DSG systems operating at pressures minimum at 30 bar, 34 bar, 100 bar, 112 bar are analyzed for prediction of its behavior and also for the lab scale models minimum 30 bar pressure. However the plant under study is a low pressure low temperature plant. It is operating in the range of 14-17 bar and saturation temperature. Complex phenomenon of flow interactions, non-uniform circumferential heating of HCE and transient temperature distribution causing snaking is addressed in this paper. It is observed experimentally that there exists a temperature difference of maximum 80°C [4-5] and the gradient on the wall tubes up to 30 °C [4].

The state of art has less predicted the thermo-hydraulic behavior of the DSG process. In this context, the prediction of the thermal performance of parabolic trough solar collectors used for hot water and steam generation is essential for the observation, operation, safety monitoring and certification of these solar collectors

Different flow patterns such as: bubbly flow, plug flow, slug flow, stratified flow and annular flow are observed inside the HCE [6]. It is the result of the interaction between phases the instabilities occurring during the DSG process [2]. In practice, the stratified flow, at low liquid and gas velocities, complete separation of the two phases occurs. Causes the overheating in the dry section causes highest thermal stress risk due to the which may lead to the bending of the absorber tube [1-2,7-8]. The annular flow represents the most favorable flow pattern as it maintains the contact between water and the absorber pipe.

Despite the numerous studies by various authors about DSG systems, mathematical modeling of water-steam two-phase flow in solar receivers still has considerable challenges. Basic, numerical and experimental research is still needed

Thermo-hydraulic behavior inside a DSG collector has been analyzed using in several software [7,9-10], commercial computational fluid dynamic CFD models were used to solve the superheated steam section [10-11] and for the whole DSG loop, other DSG models were based on numerical tools such as Modelica language [12], RELAP , ATHLET [7,10] and TRNSYS [13] for dealing with the dynamic simulation of DSG parabolic trough solar collectors. Hydrodynamic model for Once though system based direct steam generation is developed [14], Elsafi presented a complete flow pattern analysis along the DSG absorber tube and discussed the flow transitions in different sections of the DSG loop considering a constant heat flux distribution around the receiver. In his work, the flow pattern map proposed by Wotjan [15] was used to predict the flow pattern in the DSG loops. Despite the various numerical studies to predict the thermo-hydraulic behavior of the DSG process, the modeling of water-steam flow in parabolic trough collectors is still challenging [9]. According to the state of the art of DSG modeling, no previous work has been conducted to predict the thermo-hydraulic behavior and performance of the DSG process under the conditions of the Shive plant i. e for the low pressure and mass flow rates. It becomes obvious from the aforementioned literatures that modeling the thermo-hydraulic process of DSG at lower temperature and pressure operating parameters to understand the stability of the 250 KWe direct steam generation plant facility under study.

Current numerical model is heat input based proposed to solve the two-phase flow under uniform heat flux, for heat transfer calculation using the correlation of Gunger and Winterton model [16]. This model was based on the correlation of Friedel [6] for the two-phase pressure drop calculation and the correlation of Gunger and Winterton [16].

PLANT CONFIGURATION

The configuration of the plant for the analysis presented here has main steam parameters of 250°C and 17 bar. The following sections cover the plant's layout as well as models and assumptions of the solar field and the power block.

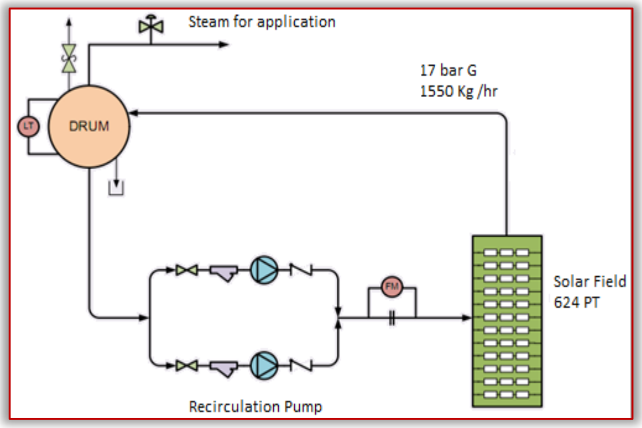


Figure 1. Direct steam generation Demonstration plant, Shive, Maharashtra, India

The power plant is designed for a gross electricity output of 250 KWe. The solar field consists of 624 parabolic troughs E-W or N-S orientated based on the space availability. The plant was operated in recirculation mode. The solar heat collectors are non-evacuated type, though it drops over the evacuated receiver the cost of the project reduces and collector field reliability would increase [17]. Tracking mechanism is so designed that it tracks 2 modules of 8 collector each at a time [18]. The Solar field has 39 parallel loops. Parabolic trough focal length is 0.5 m with the width 1.8m and the length is 3.8m. The diameter of absorber tube is 25.4mm. Water passes through 16 collectors in series. Such 39 loops are connected in parallel. The heated water and steam mixture is then fed to the drum. The saturated steam from the drum is passed through the turbine to generate electricity, which is fed to the grid depending upon the requirements.

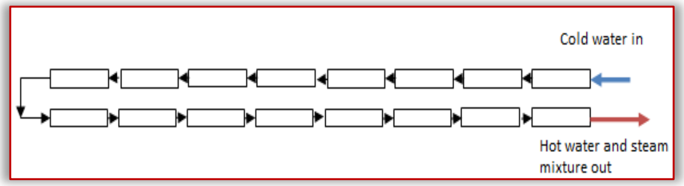


Figure 2. Solar HCE loop at site

ANALYTICAL MODEL

The analytical model based on separated flow model, is simplistic method for calculating the frictional pressure drop, understanding the flow pattern that may be causing the bending of the heat collection element of the parabolic trough. The flow properties are calculated using REFPROP based on the NIST properties. The model assumes, across the entire parallel channel, mass flow rate is equal, but in reality due to cloud cover the flow rate in parallel channels will vary, varying the pressure drop that needs more sophisticated approach and modeling.

The analytical model to evaluate the Flow pattern is based on the following working parameters. Operating pressure, heat input, mass flow rate and amount of subcooling are the variables used to simulate the effect of each parameter.

Table 1. Cases under Study

Mass flow rate kg/hr	Inlet pressure effect Pressure in bar	Heat input effect (Collector length is 3.8 m.) Heat input: KW/collector	Amount of Sub cooling °C
100–1200kg/hr with increments of 50	2, 3, 4, 5, 8, 11, 14, 17	0.25 ,0.50 0.75 , 1	0, 10 20

The total length of the complete segment of 16 collectors is 61.2m i.e. each segments of 8 collectors is divided into three sections, a section where the sub cooled liquid enters ($x<0$), second where the liquid is evaporates ($0<x<=1$) and third where the saturated vapor is superheated ($x=1$). The steam quality more than 1 is not expected with the considered mass flow rate and the input heat flux. The exit enthalpy of each segment is calculated to confirm the quality of the flow. The exit flow of first segment forms the inlet flow of the second segment.

The piping, valves and connections from pump outlet to the economizer (i.e. first segment) is considered while calculating the pressure drop due to single phase mass flow. Also the pressure drop in the piping, valves and connections from the segment 2, outlet to the solar boiler is considered while calculating the total pressure drop due to two phase flow.

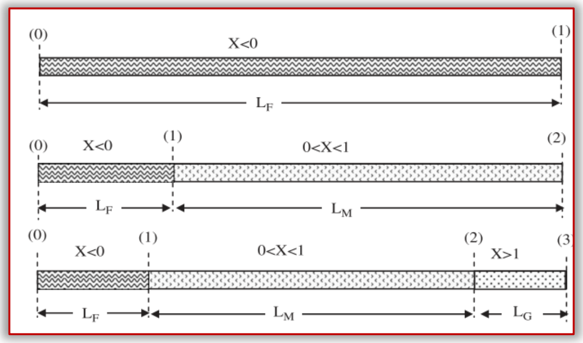


Figure 3. The three possible flow sections for the evaporation in a single pipe [19]

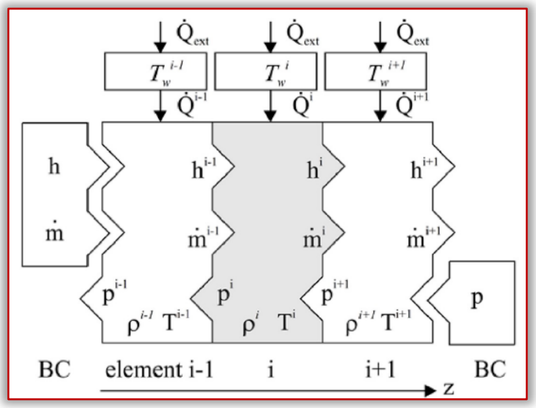


Figure 4. Axial discretization of fluid and pipe sections [20]

Separate pressure drop equations are used with the economizer, evaporator, piping and valves. The first step of the analysis is to get mass flow rate vs. pressure drop curve for the system. As the 39 different entries are parallel to each other and connected to common manifolds, at the entry and exit of the segments. The pressure drop is considered in only 1 section of 2 segments of 8 collectors each. It is assumed that all the parallel channels observe equal flow.

A one dimensional step by step approach has given to determine the outlet enthalpy as follows,

$$h_{out} = h_o + \frac{qL}{\dot{m}} \quad (1)$$

where q is the total average heat rate per unit length absorbed by the fluid, L is the length of the pipe and $\dot{m}$ is the mass flow rate in the pipe. After calculating the enthalpy the quality of the flow x_{out}can be calculated as,

$$x_{out} = \frac{h_{out}-h_{F,sat}}{h_{G,sat}-h_{F,sat}} \quad (2)$$

For sub-cooled region $x_{out} < 0$, for the two phase flow during the evaporation $0 < x_{out} < 1$, after the evaporation is complete the quality of exit vapor increases more than 1 ie for the superheated region $x_{out} > 1$.

The total heat input is assumed after the heat loss to the surrounding. Heat flux is taken to be uniform along and around the pipe.

- **Finalization of the pressure drop equation to be used effectively for the DSG application**

The pressure drop of the DSG process is dominated by the water/steam two-phase flow pressure drop. Different models known from literature predicting the pressure drop of a two phase flow that fulfill the specific requirements of the DSG process have been identified [6].

- **For Single phase flow modeling**

The pressure drop is estimated using experimental correlation system under consideration has not been designed for superheating; hence the pressure dropΔP_{1ph} in preheating section is only evaluated using Blasius equation.

$$\Delta P_{1ph} = 4f_{1ph}\left(\frac{L}{d_i}\right)\dot{m}_{total}^2\left(\frac{1}{2}\rho_{1ph}\right) \quad (3)$$

where f_{1ph} is the friction factor for turbulent flow, which is determined by using Moody's friction factor correlation considering the relative surface roughness $(\frac{\varepsilon}{d_i})$:

$$f_{1ph} = 0.0055\left[1 + \left(20000\frac{\varepsilon}{d_i} + \frac{10^6}{R_{e1ph}}\right)^{\frac{1}{3}}\right] \quad (4)$$

- **For two phase flow modeling**

Friedel correlation has been used (6) for calculating the two phase flow, it was the most accurate model used in European DISS Project (9). The correlation method of Friedel (1979) utilizes a two-phase multiplier

$$\Delta P_{2ph} = \Phi_{fr}^2\,\Delta P_{1ph} \quad (5)$$

Using the liquid dynamic viscosity μL, two-phase multiplier is

$$\Phi_{fr}^2 = E + \frac{3.24\,FH}{Fr_H^{0.045}\,We_L^{0.035}} \quad (6)$$

The dimensionless factors Fr_H, E, F and H are as follows:

$$Fr_H = \frac{\dot{m}_{total}^2}{g d_i \rho_{1ph}^2} \quad (7)$$

$$E = (1-x^2) + x^2\frac{\rho_L f_G}{\rho_G f_L} \quad (8)$$

$$F = x^{0.78}(1-x)^{0.224} \quad (9)$$

$$H = \left(\frac{\rho_L}{\rho_G}\right)^{0.91}\left(\frac{\mu_G}{\mu_L}\right)^{0.19}\left(1-\frac{\mu_G}{\mu_L}\right)^{0.7} \quad (10)$$

The liquid Weber We_L is defined as:

$$We_L = \frac{d_i\dot{m}_{total}^2}{\sigma\rho_H} \quad (11)$$

Homogeneous density ρ_H is based on the vapor quality.

$$\rho_H = \left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L}\right) \quad (12)$$

— **Finalization of the Heat Transfer coefficient equation to be used effectively for the Direct Steam Generation application.**

The heat transfer coefficient under the single-phase flow condition can be obtained from the Nusselt number as $h_{c1ph} = \frac{N_u K}{d}$. Odeh [21] developed the map based on which the heat transfer coefficient for two phase flow is evaluated. The flow transition is determined using the dimensionless Froud number. Froud number is the ratio of the flow inertia to the gravity force.

$$F_r = \frac{G^2}{\rho_{1ph}^2 \, g \, d_i} \quad (13)$$

If Fr < 0.04, the flow is stratified. If Fr > 0.04, annular flow occurs inside the absorber tube. The heat transfer coefficient for two phase flow can be evaluated using appropriate co-relations depending upon the type of flow.

In the thermo-hydraulic model, the properties of the heat transfer fluid (water/steam) are determined using the NIST Properties. A spreadsheet-based parabolic trough performance model. The model has been developed in Microsoft Excel® spreadsheet program was developed during the study. The spreadsheet is used for data input and output. The model uses the Visual Basic for applications language built into Excel for programming the hourly performance simulation.

It is observed that the simulation results are in good agreement with the literature.

RESULTS

— **Effect of operating pressure and mass flow rate:**

As shown in figure 5, at low mass flow rates, low pressure and constant heat input the boiling zone is observing higher pressure drop due to two phase flow. Higher pressure drops due to flow transition may lead to instability such as Ledinegg instability (or flow excursion) and the flow-pattern transition instability as defined by various researchers, [22], H T Liu et al.[23], Naik and Vijayan [24]. The Current plant under study is observing snaking in the 80% of the collectors in segment 2 of each loop.

— **Flow Pattern map:**

In this section, the present flow distribution model is compared against experimental data and numerical results for single and two-phase flow manifold systems reported in the technical literature. At higher pressures the flow inside the HCE will be intermittent. State of the art has suggested various transition models for two phase flow. Taitle [25] has summarized the models. Unified model for flow transition prediction used by Odeh [21] is used in this study. In the current analysis flow pattern maps are generated. The flow regimes considered are annular flow, stratified, bubbly dispersed and intermittent flow. Flow pattern maps for 25.4 mm tube at 17 bar and 3 bar are plotted as shown in the figure 6.

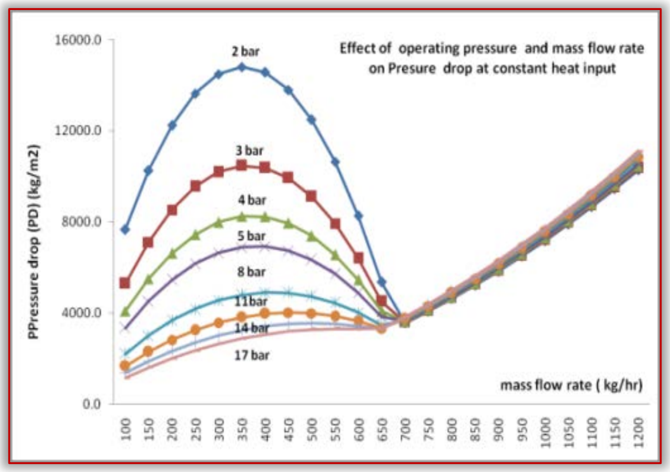


Figure 5. Two Phase Flow pressure drop in the HCE, this work

— **Effect of operating condition on pressure drop**

In DSG feed water flow rate, operating pressure, absorber tube diameter and tube inclinations are the major factors affecting the pressure drop. This collector would generate the steam when feed water flow rate is.

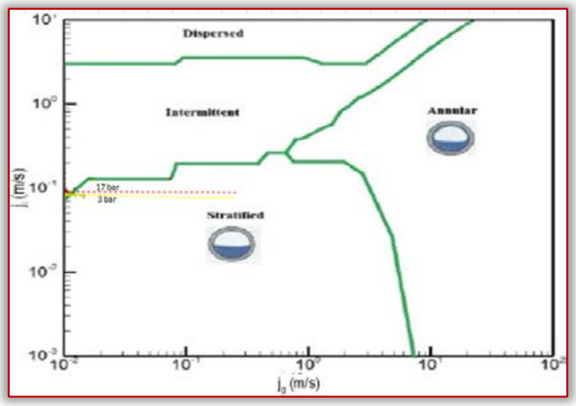
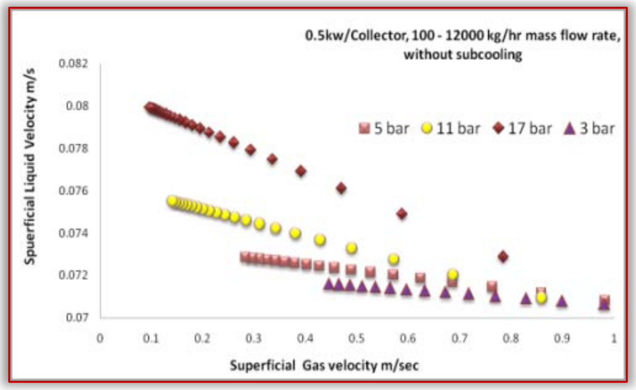
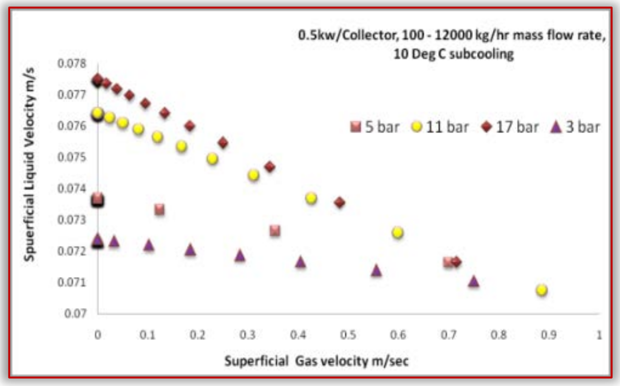


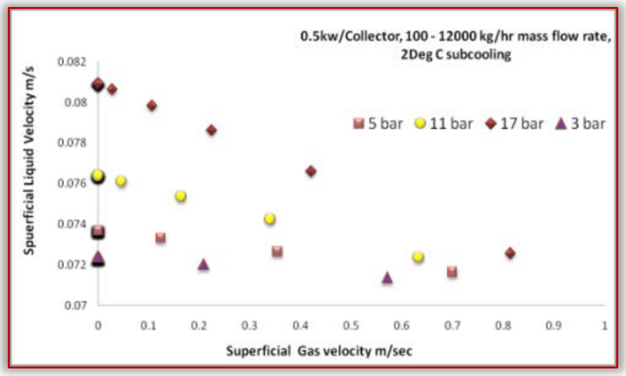
Figure 6. Surface velocities of two pressures 17bar and 3 bar in a DSG flow pattern map by Zarza [15]



a. without subcooling



b. with 10°C subcooling



c. with 20°C subcooling

Figure 7. Flow pattern map with pressure and subcooling as a variable

CONCLUSION

The model has been prepared to analysis the instability in the plant under study. All the results are the confirmation of the actual plant study, theoretical study by Rafael et al. [1-2], Sourav Khanna et al. [3], Murphy et al. [22], Flores et al.[26].

It was observed that if the current system is operating at 14bar or below pressure and mass flow rates below 1200kg/hr the flow pattern is observed as stratified flow which is causing maximum temperature variation around the perimeter of the HCE. . The analysis is in line with the site conditions and results stated by Saad [14]. The plant is operating at the pressure range 0-10 bar and at 780 kg/hr mass flow rate and facing snaking issues.

It was suggested to operate the plant in full capacity at higher pressures more than 12 bar and flow rates more than 1200 kg/hr at high solar insolation hours with less subcooling it to avoid the instabilities.

It has improved the performance of plant and the snaking issue is resolved upto 80%.

As the DSG at low pressure is still facing various challenges as static and dynamic instabilities, the future efforts should be done to build a model for complete power plant to simulate the operating conditions and find the stable operating range for DSG. Also r the non-uniform Heat flux modeling needs to be assessed to make the model more accurate.

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NOMENCLATURE

A	Cross section area, m^2
d	Diameter, mm
e	Thickness, mm
Fr	Froude number
G	Mass flux,
g	Gravity field, m/s^2
h	Convective heat transfer coefficient, $W/(m^2 \cdot K)$
K	Thermal conductivity,
L	Length, m
$\dot{m}$	mass flow rate, kg/hr
Nu	Nusselt number
P	Pressure, bar
Pr	Prandtl's Number
q	Heat flux, W
Re	Reynolds number
We	Weber Number
T	Temperature,°C
x	Steam quality
σ	Absorptance, void fraction
ρ	density, kg/m3
Δ	Difference
a	Absorber tube
cond	Conduction
conv	Convection
f	Fluid
g	Gas
h	Homogeneous
l	Liquid
rad	Radiation
o, out	Out
sat	Saturation
sky	Sky
DSG	Direct steam Generator
HTF	heat transfer fluid
PTC	Parabolic trough collector
HCE	Heat collection Element
DISS	Direct Solar Steam
NIST	National Institute of Standards and Technology

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