Hybrid Solar Heat Generation Modelling and Cases

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Abstract

Renewable thermal energy systems (RTES) harness renewable energy sources to provide services for space heating and cooling, district heating, domestic hot water, and industrial process heat (IPH). The use of low-pressure steam generated by the combustion of fossil fuels is common today to provide process heat for industrial facilities. Solar IPH (SIPH) technologies could economically replace the steam or heat needs at many industrial sites by providing high-temperature pressurized hot water, a heat transfer fluid (HTF) such as synthetic-oil, or direct steam (Kurup and Turchi, 2015). RTES could be hybridized with technology options or combined with the existing heat supply (e.g. fuels), to give options for targeted IPH application and the reduction of fuel consumption. This work has tested hybrid system modelling approaches. Initial results show when a natural gas (NG) burner that feeds an IPH application of 300°C, has both air and NG streams pre-heated with a solar field/RTES exit temperature of 180°C (via an HTF), a 13% NG offset is possible. NG offsets reach up to 26%, when the RTES exit temperatures are at 300°C for a given annual capacity factor of 24%. This can be even higher with addition of thermal energy storage (TES).

Keywords: Solar Industrial Process Heat (SIPH), Hybrid System Modeling, System Advisor Model (SAM), RTES

1. Introduction

In 2017 the International Energy Agency (IEA) estimated that 32% of total global energy is consumed by industry. Of that, it was found that 74% of the total industrial energy is used for industrial process heat (IPH). It should be noted that 90% of the IPH provided comes from fuels such as coal, NG and oil (Philibert, 2017). Varying fuel prices, along with calls to divest from fossil fuels, creates a strong incentive to switch to renewable heat solutions in industry, but currently renewably sourced heat, including biomass, represents only 10% of the global heat demand (IEA, 2018). At the time of writing, only one country (Denmark) had a 100% renewable heat target, compared to 57 countries with 100% renewable electricity targets (REN21, 2020). Denmark is also the only country to have a 100% final energy target to be met by renewables (REN21, 2020). Buildings and industrial thermal energy applications require different temperature ranges, quantities, and rates of thermal energy (Schoeneberger et al., 2020). Hybrid solutions and thermal energy storage (TES) will be important for the dispatch of heat, at optimal times needed by the demand side of the buildings and industrial applications. Denmark is a prime example where hybrid renewable thermal energy system (RTES) solutions are being deployed and are cost competitive with the current regional NG costs in Denmark. A hybrid RTES in Tårs, Denmark, that combines flat plate collectors (FPCs) and concentrating solar power (CSP) parabolic trough collectors (PTCs) coupled in series, with water storage has been operational since 2015 (Perers et al., 2016). This hybrid RTES connects to the existing gas fired district heating system and can meet approximately 30% of the town's annual district heating needs (Aalborg CSP, 2015; Putz and Epp, 2019). It is worth noting the hybrid RTES for the town of 840 households, had an estimated levelized cost of heat (LCOH) of approximately €30/megawatt hour thermal (MWh_{th}) or USD $\$33/MWh_{th}$ [as of 29th Nov. 2019 (X-Rates, 2020)], compared to $€62/MWh_{th}$ ($\$68/MWh_{th}$ as of 29th Nov. 2019) for the average heat cost from the existing gas boilers (Putz and Epp, 2019). As can be seen, the LCOH of a hybrid RTES system even in latitudes such as Denmark, can be competitive with the costs of operations from NG, given sufficient solar thermal yield and high enough gas prices.

The National Renewable Energy Laboratory (NREL) System Advisor Model (SAM) is a well-established tool for modeling solar heat systems (Wagner and Gilman, 2011). Solar water heating (SWH) with glazed and evacuated tube FPCs, Linear Fresnel Collectors (LFCs), and PTCs can already be modeled to evaluate the thermal yield. SAM's CSP models for IPH can use PTC and LFC technologies, that can either deliver heat to an HTF, or directly via direct steam generation (DSG). The HTF for the CSP process heat module can be molten salt, synthetic oil, or pressurized steam. To note, no SIPH plants where molten salt as the HTF have been found to date, and as such this is not considered viable yet for industry. The SWH and CSP annual thermal outputs can be post-processed with existing financial models from the SAM to calculate financial metrics such as the LCOH and net present

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value (NPV), though for the highlighted cases studies, detailed economic evaluation is outside of the current scope of this paper. Currently, the public version of SAM (2020.2.29) can do single system modelling very well but it is not yet capable of hybrid RTES modelling at different temperatures or combining technologies such as FPCs and CSP. For this paper and work, SAM 2020.2.29 has been used for the CSP process heat modules (NREL, 2020a).

2. Methodology

We have developed an initial framework and a variety of approaches to hybrid system modeling for RTES at different temperatures or combinations of technologies. This framework is detailed in a paper expected to be published through SolarPACES 2020, while this paper focuses on some of the main results from the modelling scenarios. Initial models to test key hybrid systems are designed based on commercially available solar heat technologies such as FPCs, LFCs and PTCs, which are suitable for integration with TES and conventional NG burners on industrial sites. As highlighted in Morocco, modelling studies have been undertaken looking at the use of LFCs which act as a pre-heat to an air stream entering an industrial NG burner (Laadel et al., 2018), where the modelling was done using the EBSILON Professional software. In this context, we used SAM, MATLAB, and the IPSEpro thermal process modeling software. The results from the three scenarios highlighted in this paper are:

- FPCs and PTCs with TES (using a synthetic oil)
- Constant and variable temperature PTCs with an HTF and NG burner
- DSG LFCs and Phase Change Material (PCM) storage

The first case scenario, using the separate SAM modules, is designed to pre-heat the HTF (Therminol-VP1) through the FPC system, and then send it to a PTC solar field coupled to the TES with an exit temperature of 300°C, which is more in line with medium temperature IPH applications. The IPH application could use the heated synthetic oil to heat a process or generate steam for a desired application.

The second case scenario, modelled in IPSEpro, uses a constant and varied solar field/RTES exit temperature (via an HTF) to provide heat to an air and/or fuel streams, that supply an existing NG burner system. This is expected to be suitable for hybridization of existing industrial systems that use NG burners today.

The third case scenario uses an array of DSG LFCs coupled with TES, which uses a PCM to improve the system's flexibility and capacity factor. The annual thermal output from SAM of individual heat generation models can be post-processed and combined with existing dispatch models for IPH and TES. PCMs store energy in the latent heat of the phase change and can thus achieve relatively high energy densities (Sharan et al., 2019).

3. Results

3.1. FPCs and PTCs with TES

The first scenario uses process heat modules that exist separately within SAM, i.e. the SWH, and the parabolic trough heat modules. The hybrid system is also combined with a TES. For this paper, the heated fluid exiting the PTCs can be used to either heat a process or generate steam for a desired application (Figure 1). The key innovation in this hybrid model is then to combine the modules for the hybrid RTES energy generation model. Note, the financials are not included currently in this hybrid model. This scenario is designed to pre-heat the HTF through an FPC system to an intermediate temperature, and then send it to a PTC system to reach the higher process heat temperature. This hybrid RTES is an existing and commercially operating system (e.g. the 6.8MW_{th} site in Tårs, Denmark), where the FPCs heat unpressurized water to approximately 75 °C, and the heated water is sent to the PTCs, where the exit temperature of the water is 98 °C (Aalborg CSP, 2015; Putz and Epp, 2019). This paper's scenario though allows for a high temperature HTF instead of an unpressurized water, and the concept is more suited for IPH application.

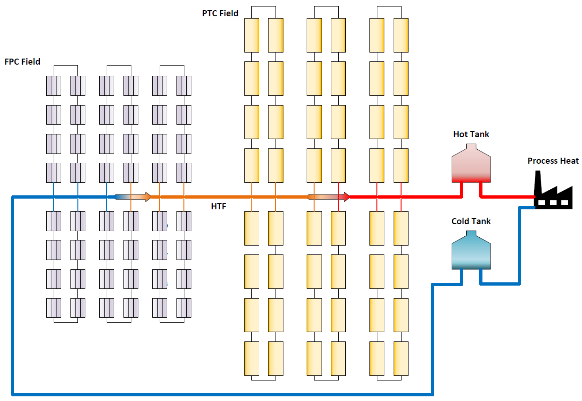

Figure 1 Schematic for Hybrid FPC and PTC model with TES for process heat application. (The relative size of the FPC and PTC system varies by IPH application and the land availability)

The hybrid FPC and PTC plant is sized according to the desired process heating power, temperature, hours of thermal storage, mass flow constraints and the nominal temperature into and out of the FPC field. The sizing procedure is similar to the sizing of a regular PTC-only plant: the heating power dictates the total size of the field, the process heat temperature dictates the number of PTCs in series, and the mass flow constraints of the PTCs dictate the number of subfields. However, with the hybrid plant, the PTC field is sized using a higher inlet temperature, resulting in fewer PTCs in series.

For this analysis, the FPC field is and sized according to the design mass flow, the relatively constant process heat, cold outlet temperature, and the target intermediate FPC outlet/PTC inlet design temperature. Note, this is an example FPC collector which can operate with the needed pressure from the available and pre-built SWH collectors for modelling which are available in SAM 20.2.29 (NREL, 2020a). A single HTF (Therminol VP-1) flows from the FPC field directly to the PTC field. The design mass flow is dictated by the design plant power and temperature, and in turn determines the number of FPCs in parallel. The temperature rise from the cold inlet to the intermediate temperature determines the number of FPCs in series. This sizing is performed at a constant standard ambient temperature and irradiance; however, since the FPCs are stationary and experience a range of cosine losses throughout the day and seasons, the intermediate temperature is always changing.

The PTC field is made up of troughs with 6m aperture width, and 80mm receivers. Again, this is an available selectable collector in the SAM parabolic trough process heat module, others are also available (NREL, 2020a). This variable intermediate temperature requires that the plant controller be more sophisticated than that for a regular PTC or regular FPC plant. The controller is similar to that for a PTC-only plant where the mass flow through the entire system regulates the outlet temperature. The PTCs are also still used to provide a high-temperature limit control via defocusing or pointing away from the sun. However, model convergence for this hybrid plant requires more algorithmic logic as the PTCs cannot as easily predict their inlet temperature iteration to iteration.

For this simulation, the default or base case TES sizing was for 8hrs of storage, to cover ramp-downs in the evening and ramp-ups in the morning. Both the FPCs and the PTCs are sized such that $5MW_{th}$ is delivered at the process heat sink or PTC to process heat exchanger (HX) via the Therminol VP-1. This is with a solar multiple of approximately 1.9. Table 1 shows the key assumptions and solar field sizes for this scenario. As can be seen, the FPCs have a solar field area of 918 m^2 and the PTCs a single loop aperture area of 2,624 m^2 with 8 actual loops. The system design is a test case to prove the operation of the hybrid model and it is not optimized to scale the solar field area for PTC. The ongoing work is focusing on optimization and dispatch modeling to scale down the PTC solar field area.

Table 1 Key assumptions and solar field sizes for FPCs and PTCs with TES scenario

Parameter	FPCs	PTCs
Collector Type	Heliodyne Gobi 400	SkyFuel SkyTrough (80mm)
HTF	Therminol VP-1	Therminol VP-1
Solar Field (m^2)	918	20,992
Design Inlet Temperature (°C)	27	150
Design Outlet Temperature (°C)	150	300

The SAM hybrid model produces hourly simulations for a specific location based on the annual typical meteorological year (TMY) file, which includes the hourly ambient temperature and the direct normal irradiation (DNI). In this scenario, the location of the simulated site is Lancaster, California, which has an annual average DNI of 7.93 kWh/m^2/day (NREL, 2020a). Note, the SAM tool can input TMY or weather files for simulations across the world, and for the United States the National Solar Radiation Database (NSRDB) has been used (NREL, 2020b).

The FPCs can potentially work with Therminol VP-1 which allows to use the same HTF in the PTC field. To give a sense of the overall impact of the hybrid RTES, annual simulations at hourly timesteps were performed. Figure 2 shows the annual profile of the heat sink inlet temperature, PTC inlet temperature, and the FPC exit temperature over a 24-hour period. It is important to highlight, that while there are minor differences between the PTC outlet and the process load HX inlet (e.g. due to thermal losses in the piping from the PTC solar field and the HX inlet), over the year it can be considered effectively the same. The variable PTC inlet temperature (i.e. the fluid temperature from the FPCs) is shown in orange in the top plot in Figure 2, in addition to the cold tank temperature (blue), hot tank temperature (green) and the heat sink inlet temperature (maroon). On average the output temperature from the FPCs during the peak time of the in the day is around 210 °C. This significant temperature increase of the FPC inlet allows the PTCs to then raise the HTF temperature to 285 °C (Figure 2). The heat sink inlet varies during most of the day, especially dropping significantly during the nighttime. Addition of TES increases the thermal output after daylight hours and provides constant inlet temperature of 27 °C to the FPC field.

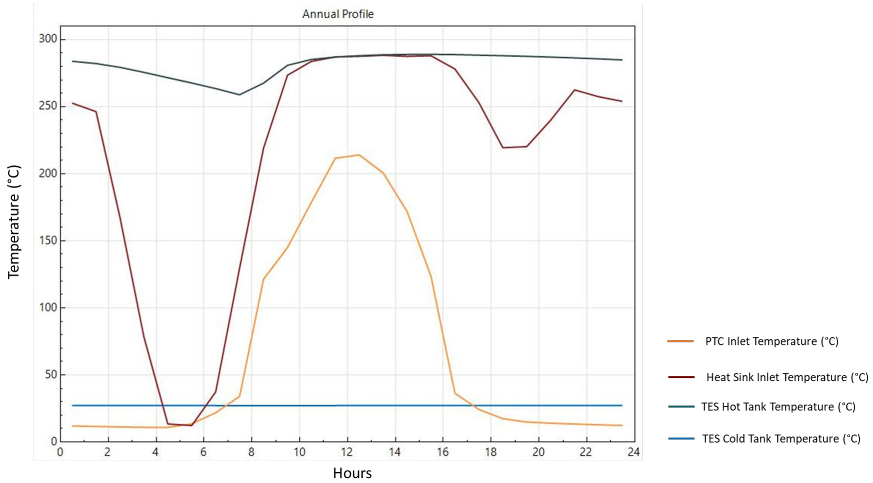

Figure 2 Annual average daily temperature profile of hybrid FPC + PTC + TES system with 8 hours of storage. Orange curve: variable PTC inlet temperature; maroon curve: heat sink inlet temperature or temperature delivered from the hybrid RTES; green curve: hot tank temperature; and blue curve cold tank temperature or inlet temperature to FPC system)

To highlight the difference between operating the hybrid RTES with 8hrs of TES and without storage, Figure 3 shows the impact of the storage on the temperature of the heated fluid delivered to the process load HX. The blue line represents the direct PTC output with no storage, and the red line shows the difference in daily generation for the 8hr storage situation, which is the base case for this scenario. As seen, the TES increases the energy delivered to the HX.

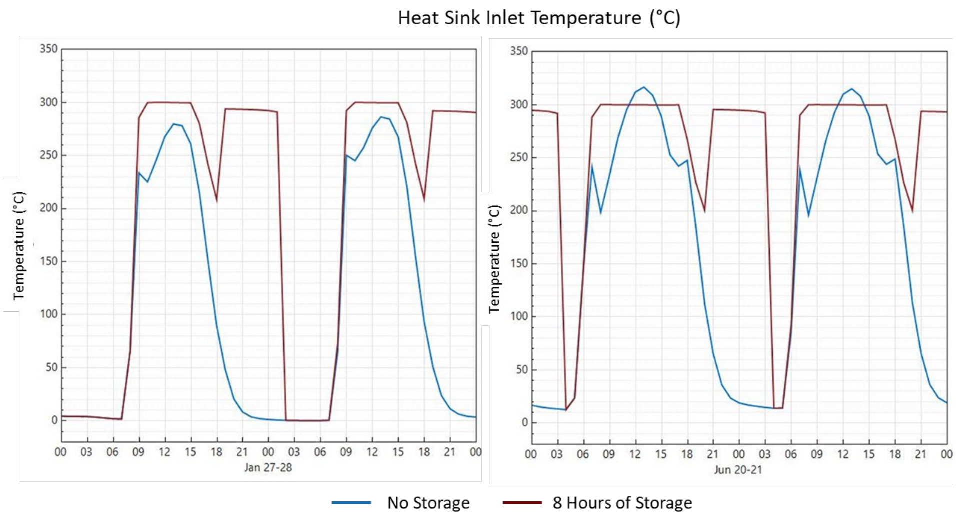

Figure 3 Temperature delivered to process heat form the hybrid FPC-PTC-TES system for typical days in January and June (blue curve represents TES off case, red curve represents 8 hours of TES case)

Figure 3 left shows the hourly simulation results of the process HX inlet temperature for 2 days in January (20th and 21st) as representative winter days, and Figure 3 right shows 2 days in June (20th and 21st) as representative summer days based on the TMY file. As seen, the key effects of storage increase the temperature delivered to the process HX to nearly the same temperatures as during the day when the PTCs are operating. The storage (8hrs) for winter and summer have similar effects, and on average extend the number of hours of delivering nearly 300 °C heat by approximately 6-8hrs. Without storage it is clear that the PTCs send heated fluid at much more variable temperature, as the heated fluid temperature is coupled to the extent of solar radiation available in the day. Delivered heat is suitable for a range of industrial processes such as; precipitation of primary metals, distillation of plastics and drying of chemicals which require around 300 °C process heat (Schoeneberger et al., 2020)

A parametric analysis for this scenario has been performed, where the numbers of hours of storage were varied. Table 2 shows the variation of the capacity factor, and the resulting impact on the annual net generation by changing the hours of storage in comparison to the base case of 8hrs.

Table 2 Parametric analysis showing the change in system annual capacity factor and annual net energy generation for various TES hours (*base case for thermal storage hours)

Thermal Storage (hours)	Capacity Factor (%)	Annual Net Energy Generation (MWh)	Change in net generation from 8hrs case (%)	Thermal Energy Stored in Charging State (MWth)
0	33 %	14,262	-52 %	0
2	46 %	20,287	-32 %	3,381
4	54 %	23,542	-21 %	6,815
8*	68 %	29,770	0 %	13,346
12	80 %	35,255	18 %	19,087

As seen in Table 2, increasing storage has a significant impact on the net annual generation of the combined solar fields, and that at 8hrs of storage the hybrid RTES can operate with a capacity factor of approximately 68% in Lancaster, California. When the storage is increased to 12hrs, the net generation increases by 18%, similarly if the storage is decreased to 4hrs, the net generation decreases by 21%, both compared to the 8hrs base case.

3.2. Constant and variable temperature solar field PTCs with an HTF and NG burner

As highlighted, scenario two in this paper is the use of PTCs with a liquid HTF providing heat input into the air and NG streams that feed a constant load NG combustor that provides heat for an IPH application at 1,000 °C or 300 °C. The HTF can be used to pre-heat the NG stream, the airstream entering the NG burner, or both. This scenario is a potential near term representation of what industrial sites could utilize to hybridize their current existing system with a renewable thermal input and as such reduce fuel consumption. To develop this hybrid RTES model, the first stage was to simulate a constant solar field outlet temperature, and then develop it further where a variable temperature PTC solar field is used instead of a constant exit temperature.

The first test case of this scenario, the heated HTF (Therminol VP-1, a commonly used synthetic oil in CSP electricity generation plants) is set at a constant 250 °C exit temperature from the solar field/RTES (e.g. with PTCs) and the air feed flow rate is set at 50 kg/s. Effectively the RTES exit temperature is constant through the year. As mentioned for the high temperature IPH application, the outlet of the NG burner that feeds the IPH application is 1,000 °C. IPSEpro software is used to calculate heat balances, enthalpies, and simulate processes for the HX and NG burner. Figure 4 shows the base case NG burner without solar heating (A) and constant temperature solar field heating of the NG and air streams (B) prior to input into the NG combustor for 1,000 °C process heat application.

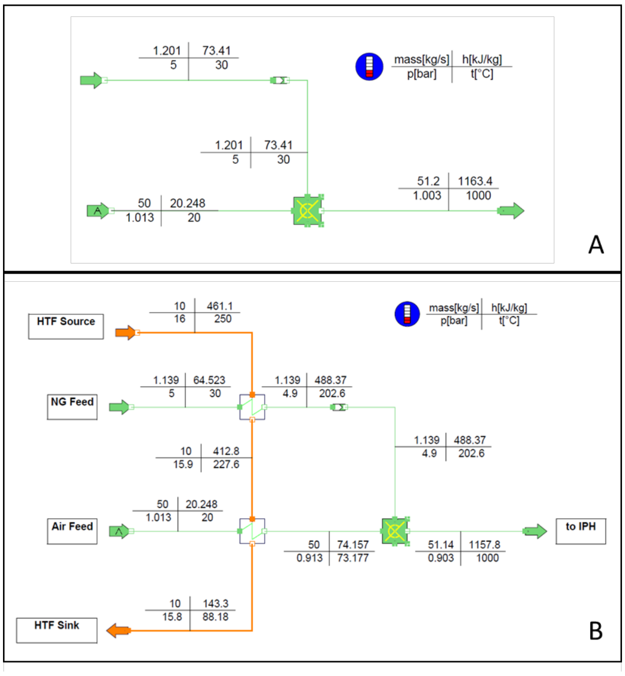

Figure 4 Constant temperature solar field and NG combustor for 1,000 °C process heat application, the air feed flow rate is set at 50 kg/s. (A: base case NG burner without solar heating, B: solar heating for both NG and air streams)

The constant temperature HTF, can heat the NG and air stream separately, or as in Figure 4 both streams. Table 3 shows the impact of the heat input into the NG stream, the air stream and then both. The biggest single impact is when the heated fluid from the solar field heats the air stream prior to entering the NG combustor, where due to the increased enthalpy of the air, slightly less gas is required (5% less) to reach the 1,000 °C air temperature needed for the IPH application. Note, due to the 1.2 kg/s of NG and 50 kg/s of air for the base case, most of the energy from the HTF is used for the NG heating. When the NG and air streams entering the NG burner are both heated to 203 °C and 72 °C respectively through two separate HXs, it is possible to reduce NG consumption by approximately 5% compared to the base case where no renewable heat is added (Table 3). This analysis has not done optimization of the delta temperature rise across the air and NG HXs highlighted in Figure 4.

Table 3. Summary of results from constant solar field and NG hybrid system models for 1000 °C process heat application with 50 kg/s air inlet mass flow to the NG burner.

Heating from solar field	No heating	NG only	Air only	NG and Air
NG burner air feed temperature (°C)	20	20	78	72
NG burner gas feed temperature (°C)	30	202	30	203
NG mass flow (kg/s)	1.201	1.199	1.145	1.139
Change in NG mass flow (%)	**0.0 %**	**- 0.1 %**	**- 4.7 %**	**- 5.1 %**

In second test case (medium IPH temperature), the exit temperature of the solar field/RTES is set as 180 °C, the air feed flow rate is set at 50 kg/s and the outlet of the NG burner that feeds the IPH application is now 300 °C. Note, due to the reduction in the IPH temperature (i.e. 300 °C instead of 1,000 °C), 0.3.01 kg/s of NG is needed compared to 1.2 kg/s. When the NG and air streams entering the NG burner are heated to 161 °C and 58 °C respectively through two separate HXs, it is possible to reduce NG consumption by approximately 13% compared to the base case where no renewable heat is added (Table 4).

Table 4 Summary of results from constant solar field and NG hybrid system models for 300 °C process heat application with 50 kg/s air inlet mass flow to the NG burner

Heating from solar field	No Heating	NG only	Air only	NG and Air
NG burner air feed temperature (°C)	20	20	59	58
NG burner gas feed temperature (°C)	30	166	30	161
NG mass flow (kg/s)	0.301	0.299	0.274	0.262
Change in NG mass flow (%)	**0.0 %**	**- 0.3 %**	**- 9.0 %**	**- 13.0 %**

In third case, a fully operational PTC system has been modelled with a variable HTF temperature. The PTC system is designed to operate with a 24% capacity factor (approximately 2,076 hours of annual full load operation) and providing a maximum outlet temperature between 180 °C and 300 °C. Figure 5 shows the process flow diagram of the variable temperature PTC solar field and NG burner model where both the NG and air streams are heated.

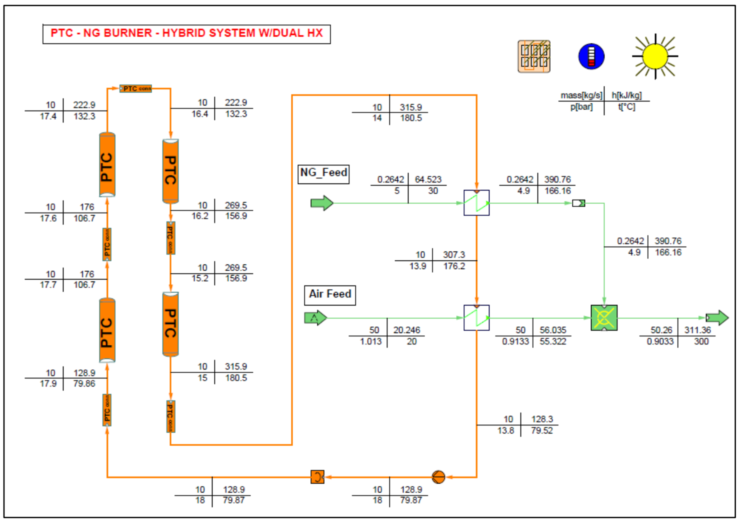

Figure 5 Variable temperature PTC and NG combustor for 300 °C process heat application, and air feed flow rate is set at 50 kg/s

Table 5 shows the progression of having the PTC outlet temperature increase from 180 °C to 300 °C, when the capacity factor is 24%. The reduction in NG mass flow can be as high as 26% when the HTF reaches 300 °C and offsets the NG consumption of the burner (Table 5). To note, this result is valid for a capacity factor of about 24%, and the model does not include TES and so the reduction of NG occurs within the sunlight hours of the day (8-9 hours). As seen, when the PTC solar field exit temperature is closely aligned to the process temperature (i.e. 300 °C) compared to 1000 °C, a greater amount of NG can be offset (-26% compared to -14%).

Table 5 Change in NG mass flow with respect to variable temperature HTF from PTC system for 300 °C process heat application with 50 kg/s air inlet mass flow to the NG burner

Heating from solar field	No Heating	180 °C	200 °C	250 °C	300 °C
NG mass flow (kg/s)	0.301	0.262	0.256	0.242	0.223
Change in NG mass flow (%)	0.0 %	-13 %	- 15 %	- 19 %	- 26 %

3.3. DSG LFCs with PCM storage

The third scenario uses DSG LFCs coupled with TES to improve the system's flexibility and capacity factor. PCMs are chosen as the storage medium as they have relatively high energy densities. PCMs store energy in the latent heat of the phase change are well suited for integration with systems that use steam as the working fluid since both media go through a phase change allowing the temperature profiles to be matched which improves the effectiveness of heat transfer (Sharan et al., 2019).

The DSG LFC system is designed for 1 MW_{th} capacity with a solar multiple of 1.2 and target steam quality of 0.75. Single loop aperture area is calculated as 3,081 m^2 which generates actual thermal output leaving in steam up to 2.11 MW_{th} . The PCM TES size adjusted to maintain the thermal output correspondingly.

The DSG LFC system is modelled with SAM which evaluates the annual performance of the solar system. The annual performance of the DSG SAM module has been compared and validated to an operating DSG LFC solar field (Kurup et al., 2017). The annual thermal yield from the DSG LFC solar field is 3,470 MWh_{th} in Tucson, Arizona, without storage. When the solar multiple is increased to 2, and a TES of 6hrs is added, the thermal yield can increase by 60%, and thereby increasing the capacity factor.

The steam properties such as temperature, mass flow rate and steam quality are calculated for each hour of the year depending on the available solar resource. Figure 6 shows the annual average daily temperature profile and field average outlet steam quality of the DGS LFC system. As seen in Figure 6, during the day the DSG LFC produces on average approximately a stable 270 °C steam output (red curve) during daylight, at a quality of approximately 60%.

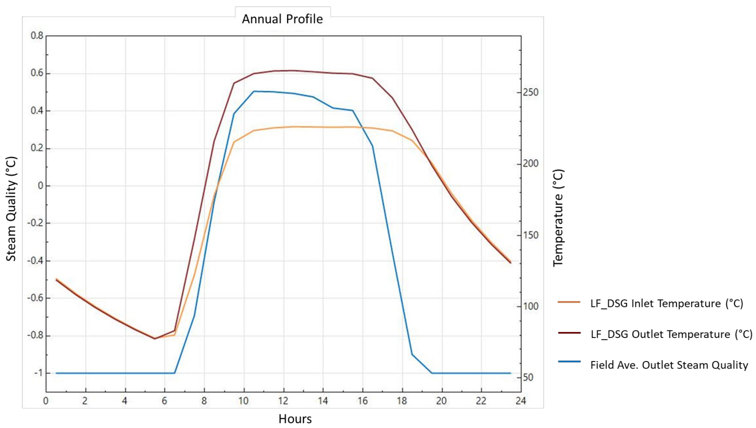

Figure 6 Annual average daily temperature profile and field average outlet steam quality of LF-DGS system. Orange curve: LF-DSG inlet temperature, maroon curve: LF-DSG outlet temperature, and blue curve field average outlet steam quality)

Figure 7 shows the hourly steam mass flow from the DSG LFC solar field for a typical week in January and July (blue and orange respectively). As seen, the steam output in winter compared to summer days is approximately 50% less (i.e. mass flow of 0.8 kg/s compared to 1.6 kg/s). Figure 7 also shows how the steam output from the solar field is impacted by transients (e.g., weather events and cloud passage), where the steam output e.g., on day 9 in winter drops significantly. Similarly, the weather event and reduction of steam output drops to nearly 0 and increases again once the clouds have passed on day 10 in July.

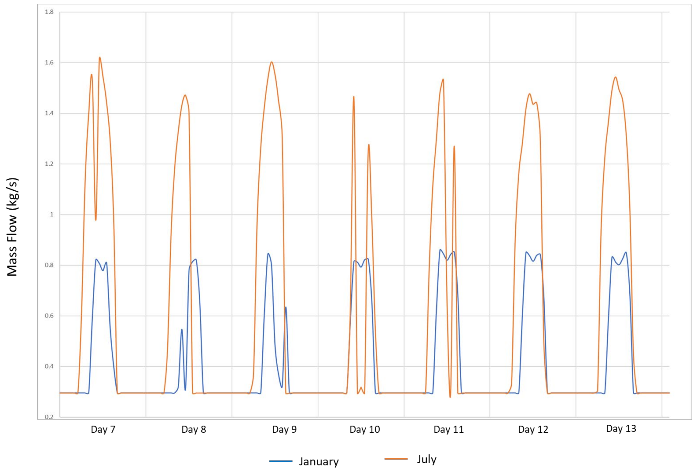

Figure 7 Hourly steam mass flow from DSG-LF solar field for a typical week in January and July

This thermal output is then used to determine the charging and discharging behavior of the TES system. The PCM-TES is modelled in MATLAB following prior NREL work (Sharan et al., 2019), albeit with several adaptations to capture the varying heat transfer coefficients of condensing and evaporating steam. Steam travels through steel pipes which are surrounded by the PCM, as illustrated in Figure 8. Numerous tubes are bundled together into large 'tanks' to store the required quantity of energy.

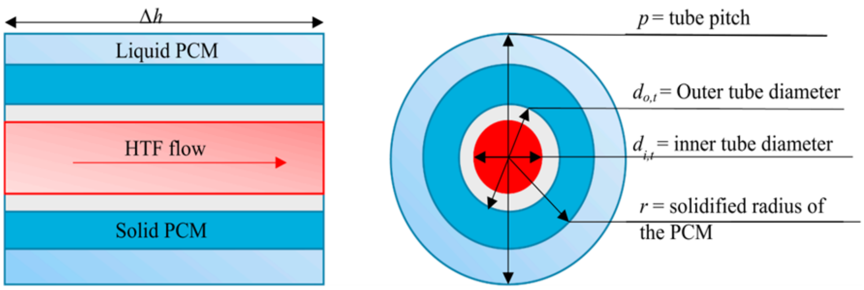

Figure 8 Schematic diagram of a shell-and-tube PCM thermal storage system (Sharan et al., 2019)

The system is operated assuming that the load requires a constant power input which is delivered by steam. The DSG system generates steam, and if the produced power exceeds the load's requirement then the excess steam passes through the TES thereby storing energy. When the solar resource reduces below the power requirements, the storage is discharged. In this example, it is assumed that the load has a power requirement of 1 MW_{th} at a steam temperature of 240 °C. The DSG LFC must generate steam at a higher temperature than this, so that the steam produced by the PCM meets the 240 °C requirement. In this example, the DSG LFC produces steam at 5 bar, corresponding to a temperature of 270 °C. Sodium formate is chosen as the PCM and this material has a phase change temperature of 258 °C (see Table 6). The discharging steam is generated at a pressure of 35 bar, which corresponds to a discharging temperature of 243 °C. Thus, the melted PCM is sufficiently hot to generate the required temperature of steam.

Table 6: Properties of the PCM sodium formate

Parameter	Unit	Property
Melting point	°C	258
Density	kg/m^3	1,920
Heat capacity	kJ/kgK	1.216
Latent heat	kJ/kg	245

Figure 9 and Figure 10 show the effect of the solar field and thermal storage size for several days of operation during the summer and winter. Results are shown for systems with a solar multiple of 2 and 4 with storage durations of 6 and 12 hours. The figures show the power produced at each hour by the DSG LFC, as well as the power that charges (negative) or discharges the storage (positive). The power that is then sent to the load is also shown. For a solar multiple of 2 with 6 hours of storage, solar energy is rarely stored during the winter. During the summer energy is stored, but the storage is too small so that later in the afternoon solar energy cannot be stored and too much power is sent to the load (and may be curtailed). Increasing the storage duration to 12 hours means that solar energy is not dumped and that the load can be supplied with power for most of the night.

Increasing the solar multiple to 4 enables the system to provide power to load for a large proportion of the time during the winter. During the summer large quantities of power are curtailed with only six hours of storage. With a storage duration of 12 hours, power can be delivered continuously in the summer. Interestingly, the storage is not fully discharged during the first evening. As a result, the storage cannot store all the excess energy during the second day and some of this energy is dumped (or sent to the load). However, even this very large system is unable to provide power continuously in the winter and an auxiliary power supply is required.

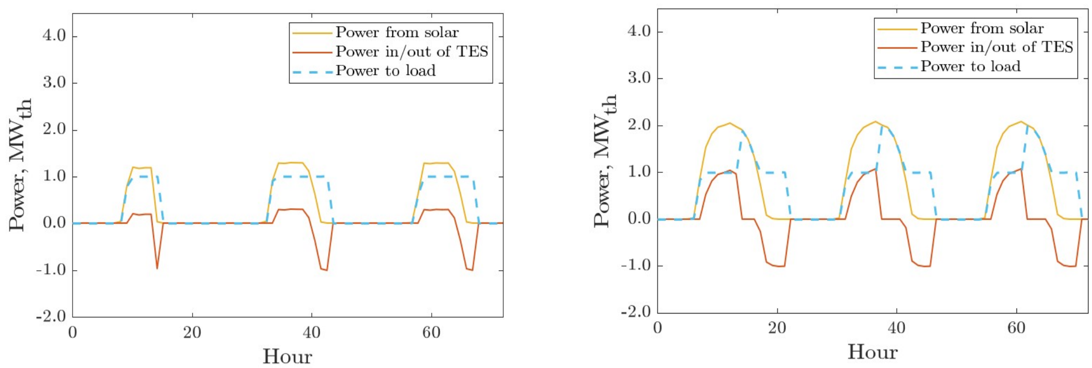

Figure 9 Thermal powers in DSG-TES system. Solar multiple = 2, thermal storage duration = 6 hours. (left) February (right) June

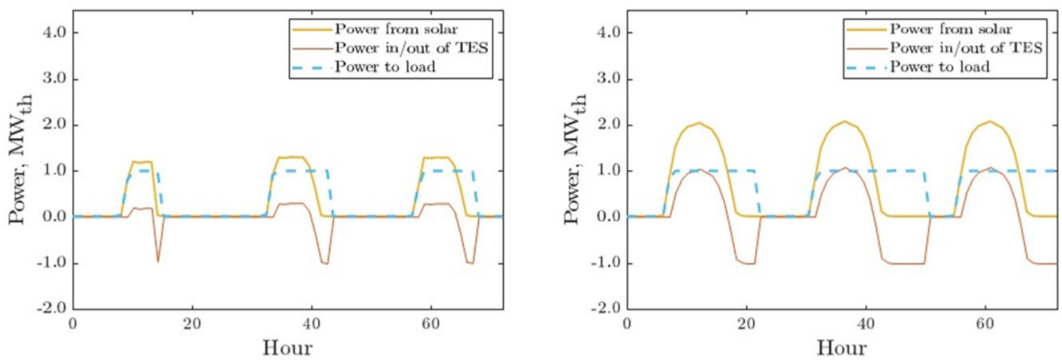

Figure 10 Thermal powers in DSG-TES system. Solar multiple = 2, thermal storage duration = 12 hours. (left) February (right) June

4. Discussions and Future Work

This paper has created and modified existing energy generation tools such as the SAM, to develop 3 main hybrid RTES models for the delivery of renewable heat to IPH applications. To begin to generate results, code modifications and hybrid RTES models have been developed and require further testing and refinement.

4.1. FPCs and PTCs with TES

The benefit of this hybrid system for district heating and potentially industrial applications, is that it can potentially reduce the plant cost relative to a PTC-only plant having the same output specifications if, for example, the same HTF can be utilized in the FPCs and the PTCs. Also, the FPCs are more efficient at lower temperatures (e.g. less than 100 °C) and the PTCs are more efficient above this and the PTCs allow for an overheat protection if needed. The lower temperature heating can be achieved by cheaper FPCs, while the higher temperature heating outside the range of the FPCs is accomplished by the PTCs.

Future work for the FPC-PTC-TES hybrid case scenario and modelling efforts include:

- Use of different HTFs like mineral oil or unpressurized water (i.e. similar to the Tårs plant)
- If the collector characteristics can be modelled, use of a smaller PTC (e.g. aperture width of 2m), that is designed for SIPH application instead of a 6m aperture trough designed for CSP electricity generation
- Connecting the FPC field to the PTC field by a HX for better pressure, TES dispatch and temperature control. Also optimizing the field sizing between the FPCs and PTCs.
- Additional SAM user interface for FPC module selection and modifications
- Adding more capital cost estimates for both FPC and PTC system and improving the SAM financial models to reflect a real hybrid case project economics
- Potentially releasing the hybrid add-on to the SAM public version e.g. with validation

This hybrid RTES model has shown that the utilization of Therminol VP-1 through the FPC and PTC solar fields as a single working fluid is feasible. This is not currently done in industry but was undertaken to create a working hybrid model. For further model developments, we aim to also create a version of this hybrid RTES where the solar fields are separated such that unpressurized water/glycol runs in the FPC field, and with a HX coupling to the PTC field which operates Therminol-VP1. This would then allow the use of unpressurized fluids, smaller volumes of Therminol VP-1 and TES tanks, and therefore potential cost savings for the system integration. This needs to be investigated.

It is expected that while this could be an effective solution, the economics are unclear and require further work to determine the benefit. Further work is needed to compare a pressurized system (if possible) compared to one with Therminol VP-1. With improvements and continued development of the FPC-PTC-TES model, the aim will be to validate the thermal yields, operation modes, and TES dispatch with real operating plants such as the Tårs plant, with weather files for Denmark and for the U.S.

4.2. Constant and variable temperature solar field PTCs with an HTF and NG burner

The hypothetical scenario was to test hybrid modelling and potential impacts. As such a real IPH application will have further refinement e.g. specific temperatures for the process. The results of the constant solar field PTCs with HTF and NG burner showed only a 5% offset due to the temperature difference between the process temperature (1,000 °C) and the solar field outlet temperature (250 °C), for a 50 kg/s air flow to the NG combustor. As mentioned, this is for the situation where the NG flow into the combustor without renewable heat is 1.2 kg/s compared to 50 kg/s of air, as such most of the temperature rise was seen in the NG stream rather than the air stream. Optimization of the delta temperature rise across the air stream HX and NG HXs will need to be investigated to highlight the best NG offset. For the 1000 °C case of scenario two, this basically shows that with today's technology, solar fields can offset a limited amount of NG consumption for an energy intense process application. The optimization in the future will show whether the NG offset can be increased, for example by increasing the heat delivered to the air stream.

For the medium temperature process heat case (300 °C) and the solar field outlet temperature of 180 °C, with a 50 kg/s airflow into the combustor, the NG offset could increase up to 13%. However, when the PTCs exit temperature is raised to 300 °C, the offset of NG consumption is expected to be 26%, for a given annual capacity factor of 24%. This system, without TES, can only provide heat during daylight hours. When TES is added e.g. 4-8hrs, the capacity factor can be increased to approximately 50% based on the DNI conditions. This would provide is a significant financial saving. This is a hypothetical study to show the functionally of a hybrid RTES integrated to the NG burner, which requires optimization based on heat demand and air flow to the process. The economic analysis would also highlight aspects such as NPV and payback if for example the costs of the hybrid RTES are identified, to then determine the LCOH as found in other analysis (Kurup and Turchi, 2019).

SIPH could be used to meet high temperature applications (i.e. 1,000 °C) in the longer term, where CSP towers are utilized. For example, CSP towers with particles that are directly heated are already showing promising test results at 965 °C (DLR, 2018). Similarly, air towers are reporting greater than 1,000 °C with on-sun tests (Heliogen, 2020). Therefore, future SAM modules with the CSP Tower could be adjusted for SIPH where the powerblock is removed.

As a future work, economic analysis is expected to be conducted to optimize the RTES in a cost-effective hybrid scenario. Other alternative scenarios including TES addition to the PTC system, co-operation of PTC and NG burner to provide heat to process in two separate heat streams, and waste heat recovery with a recuperator after NG burner will also be analyzed to compare effectiveness of use of hybridization and systems costs. We are also planning to add an optimized dispatch model to the variable temperature solar field PTCs with an HTF and NG burner using hourly DNI data as an input and providing hourly thermal output to maximizing the NG offset.

4.3. DSG-LFCs with PCM storage

A solar thermal system that is designed to produce the required power at the peak solar insolation is defined as having a solar multiple of 1. Since this modelled system requires a power of 1 MW_{th} is delivered continuously, a solar field with a solar multiple of 1 would produce less than 1 MW_{th} for most of the year. Thus, the solar field must be oversized compared to this design point. For example, a system with a solar multiple of 2 would be twice the size of the nominal design (Figure 11). For applications which require power continuously the solar multiple is typically quite large between 3 and 4. This means that during peak daytime hours, most of the solar energy that is generated is stored in the TES. For example, the solar field may generate 4 MW_{th} of which 3 MW_{th} passes through the thermal storage. However, during the night, the storage only discharges at a rate of 1 MW_{th} – i.e. the mass flow rate of steam during charge is approximately three times higher than during discharge (the actual mass flow rates are slightly different to this since the heat transfer coefficient differs during condensation and boiling of steam). This is for future work.

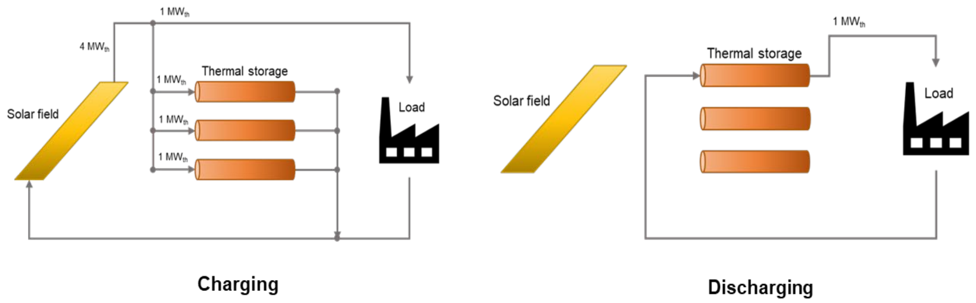

Figure 11: Schematic of a DSG-TES system with multiple PCM storage tanks. (left) Charging. All storage tanks are charged simultaneously, and the load is also powered by the solar resource. (right) Discharging. Solar energy is not available, and the storage tanks discharge one at a time to provide power to the load.

Key design variables for these systems are the sizing of the solar field and the thermal storage. The solar multiple should be large enough that the load is supplied with the required power throughout the night, even on low solar days. The reduced solar availability in the winter requires extremely large and uneconomical solar fields and an auxiliary power supply is necessary. The thermal storage should be large to store all the excess energy from the solar field otherwise the excess energy will be dumped. The storage system should also be designed to charge and discharge at the required power ratings. The sizing of the solar field, thermal storage, and auxiliary power supply is an interconnected problem, which requires the economics of the application to be considered to find the optimal configuration.

Ideally, the storage system should be charged and discharged at a similar rate. Charging and discharging at very different rates reduces the storage efficiency. For example, Figure 12 illustrates the performance of a tank that is designed to be charged at 3 MW_{th}. As can be seen, the steam enters the tank with a quality of 0.6 and is fully condensed by the exit of the storage. The tank is then discharged at a third of the rate (1 MW_{th}), as illustrated in Figure 12. Due to the lower mass flow rate, the saturated water at the inlet boils about halfway along the pipe. Thus, most of the heat transfer occurs in the inlet section of the pipe and the PCM solidifies here first, whilst still being liquid in the rest of the pipe. As time progresses, the solid-liquid front in the PCM gradually moves axially along the pipe. The discontinuities and uneven shapes can be explained by significant changes in the overall heat transfer coefficient, which varies substantially as steam goes from saturated water to saturated steam and goes through nucleation-dominated and convection-dominated boiling processes.

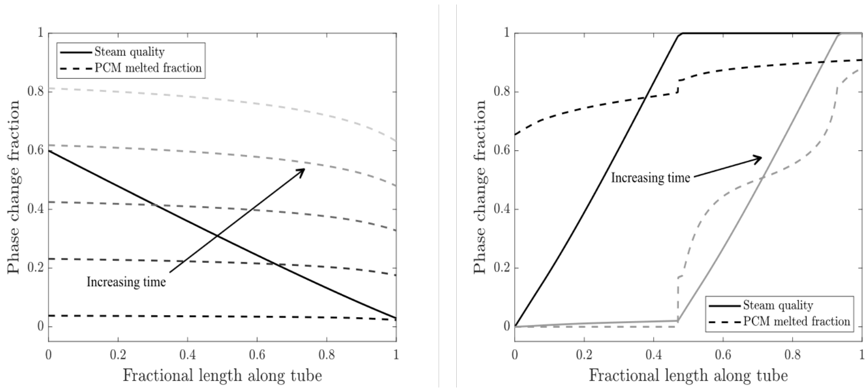

Figure 12 Phase change fraction along the length of the PCM-TES during charge (left) and discharge (right).

It is preferable for the PCM to solidify and melt uniformly along the length of the pipe. In order to achieve this, the thermal storage is discretized into several tanks. The system is designed so that the charging and discharging rates of each tank are approximately equal. For the example above, the system charges at 3 MW_{th} with all tanks being charged simultaneously – i.e. each tank is charged at 1 MW_{th}. During discharge, the tanks are then discharged one at a time at a rate of 1 MW_{th}. This ensures that the PCM melts and solidifies almost uniformly along the length of the tank, and thereby improves the TES efficiency under the wide range of scenarios required by realistic load cycles.

The choice of PCM allows the TES to closely match the process steam temperature. Sodium formate has a melting temperature of 258 °C, latent heat of 245 kJ/kg, heat capacity of 1.2 kJ/kg-K and a unit cost of $0.4/kg (Sharan et al., 2019). Other available PCMs that could be suitable for a similar industrial application are sodium nitrate ($0.8/kg), sodium nitrate-potassium nitrate ($0.8/kg) and lithium nitrate ($8.4/kg). Lithium nitrate can be an alternative to sodium formate, which has a similar melting point of 253 °C, latent heat of 200 kJ/kg and heat capacity of 1.5 kJ/kg-K. However, the cost difference makes it very unlikely to be used. Based on both process temperature and cost of material, sodium formate is an attractive option for the process temperature of 240 °C at 35 bars.

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