SOLAR HYBRID GAS TURBINE SYSTEMS

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Abstract: The provision of a sustainable energy supply is one of the most important issues facing humanity at the current time, and solar thermal power has established itself as one of the more viable sources of renewable energy. The dispatchable nature of this technology makes it ideally suited to forming the backbone of a future low-carbon electricity system. The combination of high solar shares with high conversion efficiencies is one of the major advantages of solar gas turbine systems compared to other solar-fossil hybrid power plants. Solar gas-turbine power plants are a promising new alternative, allowing increased conversion efficiencies and a significant reduction in water consumption. Hybrid operation is a further attractive feature of solar gas-turbine technology, facilitating control and ensuring the power plant is available to meet demand whenever it occurs.

Keyword: solar thermal power, hybrid gas-turbine

I. INTRODUCTION

SOLAR power is a leading contender for the future provision of clean, sustainable and indigenous energy, due largely to the fact that more solar energy falls on the Earth's deserts in 6 hours than the entire population of the planet consumes in a year. Harnessing only a fraction of this available energy would allow the whole world to meet its electricity demand from a clean and sustainable source. Over the last decades, two dominant solar power technologies have emerged on the international market: solar photovoltaics and solar thermal power. Solar photovoltaic systems convert the Sun's energy directly into electricity, whereas solar thermal power plants use this energy to create a high-temperature heat source, which can then be used to drive conventional power plant equipment. The new generation of thin-film photovoltaic panels are currently the solar power technology that provides the lowest cost of electricity. However, due to the effects of rapidly changing weather conditions and the lack of a large-scale, low-cost technique for electricity storage, their output is both uncontrollable and highly variable. On the other hand, the more expensive solar thermal power plants can take advantage of the thermal conversion step to integrate hybridisation and thermal energy storage, and thereby supply reliable and controllable power on demand to consumers. By decoupling the electrical output from the instantaneous solar input, solar thermal power plants can continue to generate electricity during times when the Sun is not shining, such as during inclement weather, or at night. Capable of being deployed on a utility sized multi-megawatt scale, they can benefit from economy of scale effects, which are expected to lead to significant cost reductions through mass production in the coming years.

In coming years, the solar thermal power industry will need to focus on exploiting the unique opportunities offered by the combination of hybridisation and thermal energy storage. As costs fall and the penetration of renewable energies grows, the dispatchability offered by solar thermal power plants will make them ideally suited to forming the cornerstone of a future energy grid based on renewable energy sources.

II. CHALLENGES FOR SOLAR THERMAL POWER

The relatively high cost of solar thermal power plants remains a key barrier to greater deployment. The high cost of solar thermal power compared to other renewable technologies can be at least partially explained by a lack of innovation, and a failure to exploit the most promising market niches.

A number of different approaches can be taken to reduce the cost of electricity from solar thermal power plants:

- A first focus can be placed on reducing the cost of the power plant components. Fortunately, a large number of solar components are still in the early stages of their learning curve and costs are dropping rapidly. This is especially true of the solar field components, which currently represent up to 50% of the total power plant cost.
- A second focus can be placed on optimising existing designs, reducing parasitic electricity consumption as well as improving the operational strategy, all of which serve to increase the annual electrical output of the power plant.
- A final focus can be placed on new power plant concepts. This can involve moving to more efficient thermodynamic cycles (generally requiring higher temperatures), new receiver designs and improved collector field layouts. This can lead to significant reductions in the cost of electricity, as long as the increase in power output compensates for any increase in the investment cost.

Recent developments in solar tower power plants have begun to address these issues. Increases in cycle efficiency and more effective operation strategies have led to some reductions in production costs but, in truth, a step change in technology is needed to boost productivity and drive down electricity costs.

A typical breakdown of water consumption in steam-cycle based solar thermal power plants is shown in Table 1.1, along with the values for parabolic dish systems, which employ water-free Stirling engine technology. Water consumption in steam-cycle solar thermal power plants includes:

• Condenser cooling, with especially large volumes for evaporative or oncethrough cooling.

• Make-up for water lost from the cycle during steam drum blowdown.

• Collector field mirror washing to ensure high reflectivity and thus maintain a high efficiency of the solar field.

Power Plant Type	Condenser [ltr/MWh$_e$]		Make-Up Water [ltr/MWh$_e$]	Mirror Washing [ltr/MWh$_e$]
	Wet	Dry		
Parabolic Trough	3000	0	40	125
Solar Tower	2800	0	35	150
Linear Fresnel	3600	0	50	200
Parabolic Dish	-	-	-	100

Table 1.1: Water consumption in solar thermal power plants

III. SOLAR GAS - TURBINE TECHNOLOGY

The use of gas-turbines instead of steam-turbines in solar thermal power plants offers a way to address all these issues: increasing efficiency, reducing water consumption and maximising the flexibility and dispatchability of the power plant through hybridisation. The development of pressurised air receivers for solar tower systems allows the integration of solar heat at high temperatures directly into the gas-turbine circuit, potentially increasing the conversion efficiency of the solar energy. Gasturbines in combined-cycle configuration are currently the technology that offers the highest conversion efficiency for a thermal power generation system, which should allow the cost of electricity from solar gas-turbine systems to be lower than that of other solar power concepts. Gas-turbines are also a largely water free technology. There is no condenser cooling, or make-up water for the cycle, only compressor and mirror washing remain, the water consumption of which is an order of magnitude lower. In combined-cycle configuration, the higher energy delivery temperature allows the use of dry-cooling technology for the bottoming-cycle with only a minor efficiency penalty [67]. By reducing water conflicts, new, high-insolation, regions are opened up for solar thermal power plant deployment, leading to increased capacity. Greater deployment of solar thermal power plants will allow the technology to ride out the learning curve and reduce the associated costs. The integration of solar heat directly into the compressed air circuit of the gasturbine also simplifies hybridisation. Solar preheating of the compressor air allows fuel consumption to be dramatically reduced. At the same time, fuel flow in the combustion chamber can be controlled extremely rapidly, allowing the combustor to compensate for rapid variations in the solar heat input and thus maintain stable operation of the gas-turbine. Hybrid operation is an attractive feature of solar gas-turbine technology, facilitating control and ensuring the availability of the power plant to meet demand whenever it occurs. The increased dispatchability of the hybrid power plant also presents the advantage of eliminating the need for the allocation of additional spinning reserve that accompanies the connection of conventional, non-dispatchable, renewable energy resources to the grid.

IV. HYBRID SOLAR GAS-TURBINES

A. Gas-Turbines for Power Production

Developed in its modern form by the NorwegianÆgidius Elling in the early 1900s, the gas-turbine is a rotary engine designed to convert heat into mechanical work. Working on a close approximation of the theoretical Brayton cycle, the gas-turbine unit comprises three main components: a compressor, a turbine-expander and a heat source, which can be either internal, such as a combustor, or external, by means of a high-temperature heat exchanger. To date, almost all utility-scale industry gas-turbines have been internally fired.

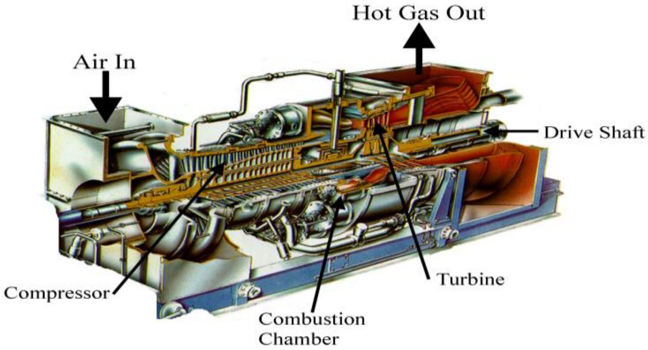

Figure 1: A typical single-shaft axial gas-turbine. Image Source: General Electric Power Systems

A diagram of a typical single-shaft gas-turbine unit is shown in Figure 1. In the combustion chamber, fuel is burnt to provide energy to the high-pressure gas delivered by the compressor, raising its temperature. The hot gas is then expanded through the turbine, providing the power to drive the compressor as well as additional power that can be recovered at the shaft. Gas-turbines have significantly higher power densities than other internal combustion engines, such as reciprocating piston engines. However, achieving high efficiencies with a gas-turbine engine is complicated by the fact that the compressor consumes a large fraction of the power produced. In order to boost the overall conversion efficiency it is necessary to either increase the efficiency of the compressor, or increase the output of the expander. As modern compressors already achieve polytropic efficiencies in the region of 0.91, recent focus has been placed on increasing the turbine entry temperature and thus the power output of the expander. The evolution of the typical temperature values at the inlet of the turbine are shown in Figure 2.

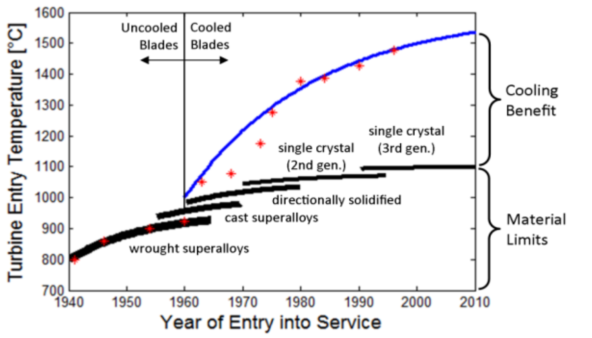

Figure 2: Historical evolution of gas-turbine turbine entry temperatures.

B. Solar Gas -Turbine Hybridisation

One of the most promising sources of low-carbon heat with which to reduce emissions from gas-turbine units is concentrated solar energy. As described, solar concentration systems can deliver heat at the high temperatures needed by gas-turbine systems without any of the carbon emissions associated with fossil fuel combustion.

C. Hybridisation Schemes

A number of different hybridisation schemes can be imagined for the integration of solar heat into a gas-turbine cycle, depending upon the level at which the heat is available and the heat transfer medium employed. However, developments in the field of high-temperature pressurised air receivers mean that it is now possible to integrate solar energy directly into the gas-turbine circuit, supplying heat to the air leaving the compressor without the need for intermediary heat exchangers. As such, two principal hybridisation schemes can be imagined for use with hybrid gas-turbine systems, namely serial and parallel hybridisation. In a serial hybridisation scheme, shown in Figure 3, the entire main airflow from the compressor is routed through the solar heat source and heated to the desired temperature. This pre-heated air is then sent to the combustion chamber, where the higher air inlet temperature results in reduced fuel consumption to reach the desired turbine entry temperature.

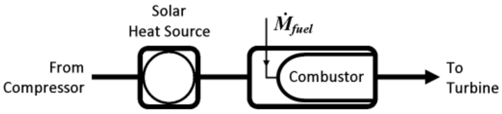

Figure 3: Serial hybridisation scheme for a gas-turbine.

In a parallel hybridisation scheme, shown in Figure 4, the airflow from the compressor is split, with part being sent to the solar heat source, and part being sent directly to the combustion chamber. The hot gases from both heat sources are then mixed together before being sent to the turbine.

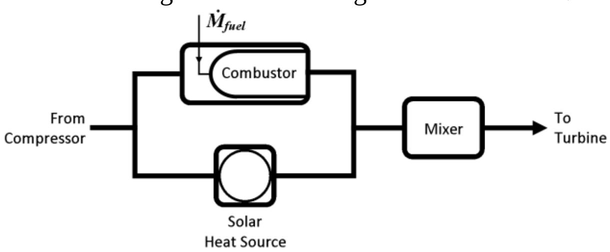

Figure 4: Parallel hybridisation scheme for a gas-turbine.

Parallel hybridisations systems simplify the operation of the gas-turbine, as it is easier to isolate the combustion and solar sub-systems and thus operate the unit using only one heat source; this is particularly important during start-up of the gas-turbine when fuel-only operation may be desired. It is also simpler to control operation of the solar sub-system, as the mass flow can be controlled to maintain nominal operating temperatures. However, parallel hybridisation is a poor choice thermodynamically. Despite recent improvements, material limits currently result in a maximum sustained outlet temperature from the solar heat source of around 1200°C, if the receiver and gas-turbine are closely integrated [6], or 950°C, if the hot gas from the receiver needs to be piped to the gas-turbine. This is significantly below the outlet temperature from the combustion chamber, which can be above 1400°C for contemporary F-class gas-turbines. With these temperature limitations, post-combustion mixing of the gas-streams will reduce the turbine entry temperature below the maximum level achievable in the combustion chamber, penalising the conversion efficiency of the gasturbine. Furthermore, the greater the degree of solar integration in a parallel system, the lower the final temperature of the gases delivered to the turbine. As such, a serial hybridisation scheme has been selected for all the solar gasturbine power plants considered in this work. In a serial scheme, both the solar receiver and the combustion chamber can be operated up to their maximum temperatures, with the final inlet temperature to the turbine equal to the combustor outlet temperature. Additionally, in a serial scheme, the final temperature delivered to the turbine is independent of the degree of solar integration, allowing greater amounts of solar heat to be integrated into the power generation cycle without adversely affect gas-turbine performance.

D. Hybrid Solar Gas - Turbine Power Plants

The layout of a typical simple-cycle hybrid solar gas turbine with a serial hybridisation scheme is shown in Figure 5; the hybridised gas-turbine cycle can also be represented in a temperature-entropy diagram, shown in Figure 6. The layout shown in Figure 5 is representative of a utility-scale hybrid solar power plant, in which the large size of the power block requires the gas-turbine to remain at the base of the tower. It is thus necessary to pipe the hot gases up and down the central tower, imposing additional temperature restrictions. As the Sun's energy is to be harnessed at high temperatures, high concentration ratios are required in order to limit radiation losses and maintain an acceptable efficiency at the receiver. Coupled with relatively large size of the utility-scale solar gas-turbine power plants studied in this work, high temperature operation will require the use of central receiver (i.e. heliostat field) solar collectors.

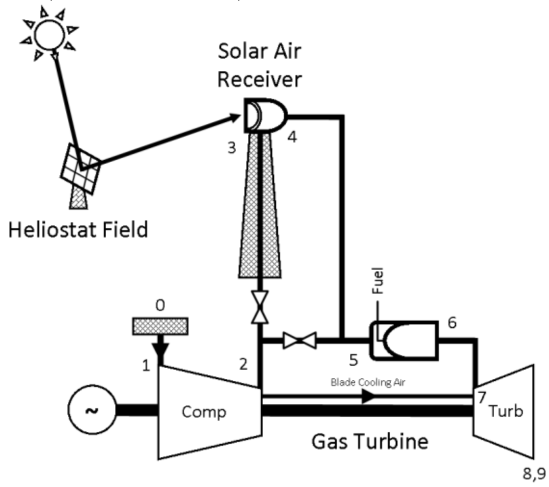

Figure 5: Layout of a simple-cycle hybrid solar gas-turbine.

Starting from the ambient conditions (state 0) the air is drawn through a series of filters to remove dirt, sand and other solid objects, resulting in a drop in pressure (state 1); the air is then compressed to a higher pressure (state 2).

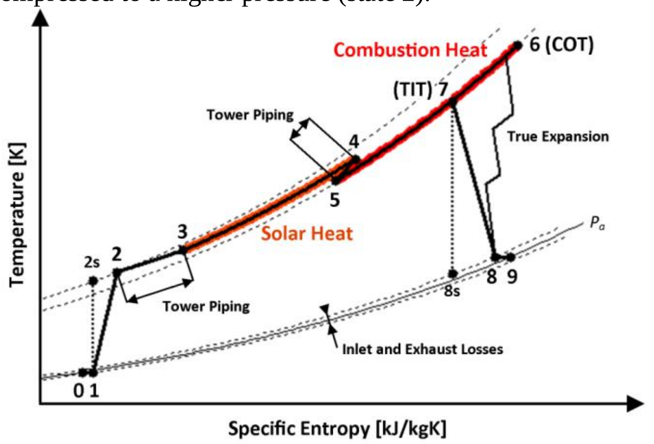

Figure 6: Temperature-entropy diagram of the hybrid solar gas-turbine.

Losses in the tower piping have been exaggerated for emphasis. Depending on the temperature at the outlet of the combustion chamber, a certain fraction of the high-pressure compressor air is withdrawn from the main flow and used for cooling of the turbine blades; the higher the temperature of the combustor gases, the greater the fraction of air that is required for cooling. Some additional air is also withdrawn for purging and sealing of the hot-gas path, preventing high-temperature gases from reaching unwanted areas. The remaining compressor air flow is sent to the solar sub-system, where it first passes up the piping in the central tower (state 3). A concentric piping arrangement has been assumed in this work, meaning that the compressor air is heated by the hotter air descending from the tower. At the top of the tower, concentrated solar energy is used to heat the air in a solar receiver (state 4) and this hot air then descends back down the tower (state 5), losing some heat in the process to the cold air from the compressor. The pre-heated air from the solar receiver is then sent to the combustion chamber where a certain mass flow of fuel is injected and burnt in order to raise the temperature of the gas mixture to the desired firing temperature (also known as the combustor outlet temperature, COT, state 6). Solar preheating of the compressor air allows fuel usage to be dramatically reduced. Additionally, fuel-flow to the combustion chamber can easily and rapidly be adjusted to compensate for fluctuations in the solar heat input, allowing a stable operating point to be maintained.

E. Operation of Hybrid Solar Gas - Turbines

The variable nature of the solar flux means that the solar heat input to the gasturbine is not constant, but varies with current meteorological conditions. In order to maintain a constant temperature at the entrance of the turbine, the fuel-flow to the combustion chamber is continuously controlled. An example of the operation of a hybrid solar gas-turbine is shown in Figure 7.

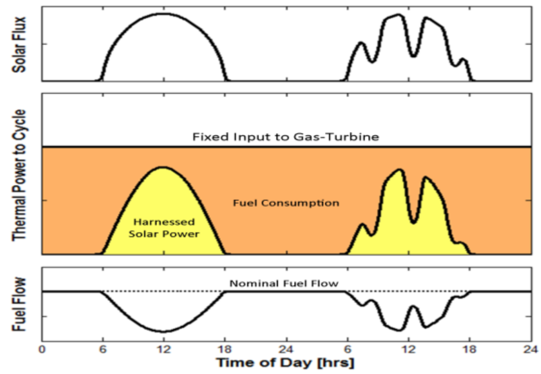

Figure 7: Variation in operation of hybrid solar gas-turbine.

The response time of the combustion chamber mass flow controller is significantly faster than that of the solar receiver, which always has a certain thermal inertia. Typical response times for solar air receivers are on the order of 5 to 10 minutes, whereas fuel-flow to the combustion chamber can be controlled on the order of a second or less [96]. Perfect control of the combustion chamber can therefore be assumed. The relative distribution of the heat input to the gas-turbine cycle depends upon the available solar flux. During daytime, heat from the solar sub-system can be harnessed by the gas-turbine, partly (or completely) replacing the heat input from fuel combustion and fuel flow to the combustion chamber decreases below the nominal value. Despite the drop in fuel flow, the combination of solar and fuel heat input provides the required nominal heat input to the gas-turbine, maintaining nominal electricity production. The overall fuel-electric conversion efficiency varies throughout the day, with values greater than 100% possible when a large fraction of the heat is supplied by concentrated solar energy. At night-time, operation of the power plant continues in pure fossil-fuel mode. As such it is important to maintain a high overall conversion efficiency for the power plant. If the power block efficiency is low, high carbon emissions during cloud passages and night-time operation can outweigh any savings achieved during solar operation.

V. ENVIRONMENTAL ASPECTS

In regions where incentive measures are in place to reward the production of carbon-free electricity, it is important to be able to determine the share of solar electricity generated by a hybrid power system. In a correctly designed electricity market, hybrid solar gas-turbine power plants should only receive incentive payments for the electricity that is actually produced from solar energy.

A. Solar Share of Electricity Production

The degree of solar integration into a hybrid solar power system can be measured in terms of the solar share f_{sol}, which is defined in Equation (1) as the ratio of solar heat input Q_{sol} to total heat input Q_{tot} for the cycle.

$$f_{sol} = \frac{\dot{Q}_{sol}^{+}}{\dot{Q}_{tot}^{+}} \qquad (1)$$

The nominal solar share of a hybrid solar gas-turbine is directly related to certain key cycle temperatures, in particular the nominal receiver temperature T_4 and the combustor outlet temperature T_6. As the receiver temperature approaches the combustor outlet temperature, the solar share increases, as the amount of heat to be supplied by combustion drops. At the limiting case, where $T_4 = T_6$, the nominal solar share equals 100% and no fuel needs to be burnt. Neglecting losses in the tower piping, the nominal solar share can be estimated using Equation (2), where T_2 is the compressor discharge temperature and T_3 the receiver inlet temperature.

$$f_{sol,num} \simeq \frac{T_4 - T_3}{T_6 - T_3} \tag{2}$$

As mentioned, the combustor outlet temperature of a contemporary industrial gas-turbine is in the region of 1400°C, while tower-mounted solar receivers are currently limited to temperatures below 950°C. As a result, the maximum nominal solar share that can be achieved by a typical industrial gasturbine is in the region of 50%. Higher nominal solar shares could be obtained by using gas-turbines with lower firing temperatures, or by finding a way to integrate the gas-turbine at the top of the receiver tower. There is an inherent trade-off between machine efficiency, which increases with increasing combustor outlet temperature, and the achievable solar share, which decreases with increasing combustor temperature. Designing hybrid solar power systems purely for high solar shares is therefore not always the optimal solution; the need for high conversion efficiencies needs to be weighed against increasing the degree of solar integration. As the solar heat input is highly variable, the hybrid solar gas-turbine power plant will not always be able to operate under nominal conditions. The nominal solar share cannot therefore be used to determine the true distribution between solar- and fuel-generated electricity. In order to take into account the variation in operation throughout the year, an annualised value for the solar share must be used instead. The annual solar share will always be lower than the nominal solar share, as the solar receiver will operate at least part of the time at less than nominal power.

B. Extending Solar Operation

In order to reduce the overall level of carbon emissions, a key parameter for the power plant is the annual fuel-electric conversion efficiency, which depends not only on the power block conversion efficiency and the nominal degree of solar integration but, more importantly, on the duration of solar operation. For an idealised cycle, the annual value of the fuel-electric efficiency can be estimated using Equation (3), based on the conversion efficiency η_{cycle} and the nominal solar share $f_{sol,nom}$ of the power generation cycle, as well as on the total number of operating hours h_{op} and the equivalent number of full-load solar operating hours $h_{sol,eq}$.

$$\eta_{fuel} = \frac{\eta_{cycle}}{1 - f_{sol,ann}} = \frac{\eta_{cycle}}{1 - f_{sol,ann} \cdot \frac{h_{sol,eq}}{h_{op}}} \tag{3}$$

Once the conversion efficiency and nominal solar share of the hybrid solar gas-turbine power plant have been fixed, the annual fuel electric depends solely on the fraction of the total operating hours during which solar heat is supplied to the system. By increasing the time during which the solar collector provides nominal heat to system, the overall fuel electric efficiency can be increased. The duration of solar operation is intrinsically linked to the size of the solar collector field. Larger fields can collect larger amounts of energy, and thus provide nominal output with lower levels of solar radiation input. Over sizing the solar collector field is thus one means of extending the duration of nominal operation, albeit at a relatively high cost. The size of the solar collector field can be expressed in terms of the solar multiple SM, defined using Equation (4) as the ratio of the nominal thermal power delivered by the field Q_{field} to the nominal power demanded by the receiver Q_{rec}. The nominal output from the heliostat field is typically defined considering a direct normal irradiation of 850 W/m2 at solar noon on the Equinox (21st of March or 22nd September).

$$SM = \frac{\dot{Q}^{+}_{field,nom}}{\dot{Q}^{+}_{rec,nom}} \tag{4}$$

With a nominally sized solar field (SM = 1), the thermal power delivered to the receiver reaches the nominal value only at midday on the design day. At other times, when the solar irradiation is weaker, the receiver outlet temperature is less than the design value. Annually, the equivalent number of full-load solar operating hours is thus relatively low. When the solar collector field is oversized (SM > 1) the thermal power delivered by the field reaches the nominal value required by the receiver earlier in the day, allowing the receiver to be operated at nominal conditions for a longer duration. In this way the equivalent number of full-load solar operating hours can be increased, and the annual solar share of the hybrid solar gas-turbine rises. However, in the absence of a storage unit, thermal power above the nominal demand cannot be harnessed (as the temperature of the receiver is limited) and as such this excess energy needs to be spilled from the system by defocusing a certain fraction of the collector field.

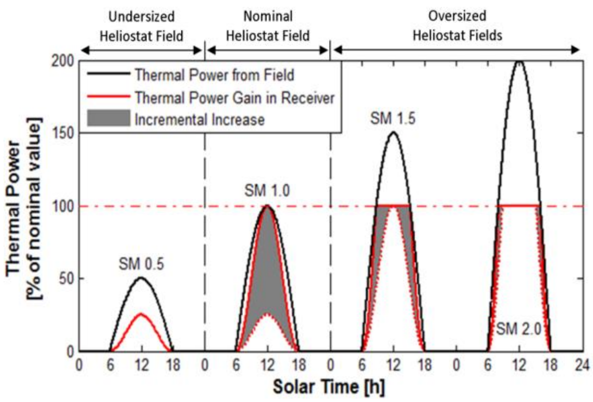

Figure 3.11: Receiver thermal power as a function of the solar multiple.

Increasing the solar multiple allows the nominal operating duration of the receiver to be extended; however, this increase suffers from rapidly diminishing returns. As can be seen in Figure 3.11, moving from SM 1.0 to SM 1.5 results in a much greater incremental increase than moving from SM 1.5 to SM 2.0, and this trend continues to higher solar multiples. The cost of the solar field is roughly proportional to the solar multiple and, as such, the marginal cost of increasing the duration of nominal receiver operation rises exponentially as the solar multiple is increased above a value of SM 1.0.

VI. CONCLUSION

The results of the system layout indicate a potential field of application under modified funding schemes, allowing for co-firing. Further R&D is required to develop the system to a marketable status. A key limitation of the studies performed in this article is that the hybrid solar gas-turbine power plants are considered in isolation. For calculation of the electrical output, the power plants are operated at a fixed load for a fixed period each day. In a liberalised electricity market, this will not be the case, and each power plant would have to submit production bids alongside other producers. This will result in a significantly altered load profile for the power plant, and the specific characteristic of a given electricity market may favour different hybrid power plant designs to the ‘optimal’ configurations presented here.

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