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A comprehensive review and evaluation of heat recovery methods from gas turbine exhaust systems

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ABSTRACT

Globally, over 60% of generated electricity is derived from fossil fuels. In the Gulf Cooperation Council (GCC) region, most of electrical power is generated from fossil fuel fired thermal power stations, which are operated either in simple cycle (SC) or combined cycle (CC). The combined cycle is applied in industry to maximize the waste heat recovery. However, 70% of the thermal power plants are in SC configuration in GCC countries, while only 30% are in CC. In the United States, the SC to CC ratio is 54–46%. Considering the large number of SC power plants, exhaust heat recovery upgrades options are of particular interest. This paper aims to investigate and evaluate the potential of incorporating cost-effective new developed heat recovery methods.

In this work, the utilization of extracted heat is categorized into three implementation zones: within the gas turbine flange-to-flange section, auxiliary systems and outside the gas turbine system in the power plant. Moreover, a new methodology has been established to enable qualitative and comparative analyses of the system performance of nine recently developed heat extraction methods according to well identified criteria including safety and risk, complexity of implementation, effectiveness, scale of modifications and the potential market opportunity. The developed methodology is used to advise further system modifications and improvements on the studied inventions for the purpose of enhancing the plant efficiency.

Based on the conducted analysis, no single design out of the investigated in this paper was able to fully achieve the established criteria, a summarized comparison and scoring table was constructed to provide direct comparison between the recent heat recovery patents, among each other, as well as typical waste heat recovery units that are commonly used in the gas turbine industry. It was concluded that each of the presented designs has particular benefits and can be selected for implementation according to the objective, the length of downtime and monetary capital available to the owner of the power station.

Based on the results obtained from the comparative scoring tables, a new design of integrated heat recovery system was proposed. The new system incorporated a circular duct heat exchanger to extract the heat from the exhaust stack and deliver the intermediary heat transfer fluid to a separate fuel gas exchanger. This system showed superiority in improving the thermodynamic cycle efficiency, while mitigating safety risks and avoiding undesired exhaust system pressure drop.

1. Introduction

Over 60% of electricity generation has been sourced worldwide from coal, fuel gas and oil [1]. The majority of the electricity in the Gulf Cooperation Council (GCC) region is being generated from thermal fossil-fuel power plants. These consist primarily of gas turbine (GT), steam turbines (ST) or a combination in the form of a combined cycle, which drive electric generators. Table 1 lists the recent official statistics for power generation mix in the United Arab Emirates (UAE) [2]. It was shown that over 27 GW of power generation is from fossil fuels; and only 60 MW from solar sources. According to the reported statistics, a similar trend was observed in Saudi Arabia, Kuwait and Oman [3–5], where mostly the power is sourced from fossil fuels. The locally generated electricity in Bahrain was fully sourced from three fossil fuel power plants [6] while in Qatar, electricity was derived from four fossil fuel power plants [7].

By considering that turbines have been the essential source of converting the countries’ rich natural fossil fuel sources into electricity, the
A GT consists of several major sections, operating on the Brayton thermodynamic cycle. The compressor is driven by the GT rotor; compressing air to the required pressure and temperature designed for the combustor section downstream; where fuel is added. The combustion products then enter the turbine section, where the released energy is converted to rotation of the rotor. The rotor is coupled with a generator for electric power generation.

There are two main configurations of a thermal GT power plant. In a simple cycle (SC) GT plant, the GT exhaust is released to the ambient for electric power generation. A GT emits heat, which is not used for electricity generation. The exhaust temperature from a 9E GT would typically be range of 700–750 °Centigrade [29]. A typical 9E.04 gas turbine has an efficiency of 64% CC efficiency, and a CC efficiency of 64% [14]. Siemens product information [15] shows the 9000HL is capable of achieving 42.8% in a SC configuration, and 63% in the CC configuration. Prior to the GT exhaust stack of 46% CC efficiency, the world record for the highest efficiency CC plant was held by a GE 7HA-driven GT, at 63.08% [16].

The highest efficiency gas turbines in the market are H-class technologies, such as the GE 9HA or the Siemens 9000HL models. A GE fact sheet [13] shows that 9HA can achieve a SC thermodynamic efficiency of 41%, and a CC efficiency of 64% [14]. Siemens product information [15] shows the 9000HL is capable of achieving 42.8% in a SC configuration, and 63% in the CC configuration. Prior to the GT exhaust stack of 46% CC efficiency, the world record for the highest efficiency CC plant was held by a GE 7HA-driven GT, at 63.08% [16]. The similarity of these achievements displays the close competition by the two biggest global GT manufacturers.

In recent years, there has been a shift away from gas turbines for new renewable energy mix power generation projects. This can be seen in the reduction of new installed generation capacity to the overall energy mix. Various sources [17,18] highlighted that the global orders for GTs gradually increased throughout the 80's, 90's and peaked in the early 2000's. Since then, the number of new installations has reduced or remained flat. Particularly in the last decade, according to the International Renewable Energy Agency [19], the new installed renewable energy capacity has continued to rise, to the detriment of new GT installations as well as other non-renewable energy sources. Since 2015, the number of GTs sold worldwide has steadily reduced. While the number of units sold in 2015 was 472, the number of units sold in 2020 was only 327 [20].

Renewable energy sources are increasingly penetrating the market share. One of the reasons is the implementation of energy policies requiring a reduction of greenhouse gas emissions. Such policies were also published in GCC countries. In the UAE, while in 2012 the power generation was almost fully through natural gas power plants [21], the renewables share increased to 7% by 2020 [22]. The UAE Energy Strategy 2050 calls for reducing this share further to 38%, while increasing the share of clean energy to 44% [23]. Similarly, Saudi Arabia has set a target of generating 9.5 GW of electricity through renewable sources by 2030 [24].

It is commonly known in the power generation industry that the four biggest original equipment manufacturers (OEMs) for GTs are General Electric (GE), Siemens, Mitsubishi Heavy Industries (MHI) and Ansaldo. The aforementioned developments are naturally having an impact on the business outlook of such companies. One way the OEMs can react to the market is to put in place a strategy for larger focus on provision of changes, modifications and upgrades to the existing installed base of gas turbines.

The longest scope option is a conversion of a simple cycle gas turbine to combined cycle. This can take months to implement. Fig. 1 illustrates the differences between SC and CC GT plants; and highlight some of the main structures. In CC configuration, the exhaust gasses of the gas turbine enter the HRSG and provides energy to convert the water coming from the condenser into steam. The steam then drives a steam turbine, which provides the rotational energy to a separate generator.

The exhaust stack is the final downstream section in which the exhaust gasses are passed to the atmosphere. Aside from the lost energy from the exhaust stack, this heat also has negative environmental impact. Chaisson [26] described the long-term effect of the release of such anthropogenic heat to the atmosphere, on the increase in global temperatures. It is worthwhile mentioning that the effect was shown to be lower than the effect of greenhouse gasses [27]. However, it had a noticeable effect due to the contribution to urban heat islands, which can lead to increases in regional temperatures even within decades [28].

Downstream of the GT, there is either an HRSG or a GT exhaust stack. If there is a GT stack, a diverter damper allows the plant to operate in either SC or CC. During SC operation, the damper diverts the exhaust gasses to the GT exhaust stack, while it diverts the exhaust gasses to the HRSG during CC operation. Downstream of the HRSG, the exhaust gasses are released to the atmosphere through the CC exhaust stack. Steam generated in the HRSG is piped to the ST in the ST hall, to drive the steam turbine for more electricity generation.

The exhaust flow temperature from the gas turbine is typically in the range of 500–700 °Centigrade. In a simple cycle gas turbine, such energy is wasted to the atmosphere. In comparison, for a 9E model gas turbine, the HRSG exhaust stack temperature is much cooler in the range of 150–200 °Centigrade [29]. A typical 9E.04 gas turbine has an efficiency of 37% when run in simple cycle mode; while 54.9% in combined cycle.

![Fig. 1. Simple and Combined cycle power plant schematic [25].](image-url)
The downside is that conversion of a power plant from SC to CC is associated with a large capital investment. A comparative study of the SC to CC conversion was illustrated by the following case study. Toyota Tsusho Corporation [31] performed a feasibility study to convert 3 power plants in Iraq from SC to CC. The plants were Amara, Nasiriyah and Najibiya; each having four 125MW GTs [32,33]. On Najibiya, they are identified as 9E units; which makes it plausible to deduce there are a total of twelve 9E GT units on these three sites. The feasibility study estimated the cost of conversion would be between 330 and 475 million USD in total; resulting in 27.5–39.6 million USD per 9E GT unit. The major items considered in the cost consisted of a ST, HRSG, generator, condenser, cooling tower and boiler feed Pump. Additionally, auxiliary facilities, transportation, civil construction and installation were also considered. The cost of the main equipment mentioned above was between 99 and 143 million USD. Analyzing this data allows us to infer the cost of 8.25–11.9 million USD per CC upgrade of a 9E model GT.

Another common method of heat recovery is through Waste Heat Recovery (WHR) units. WHR units represent dense, finned-tube heat exchangers, that are built as a module and transported into an existing SC GT exhaust system. Hence the heat transfer surface structure is similar to HRSGs; but the installation is at a smaller scale. This method of heat extraction has gained popularity, which is emphasized by the fact that the International Organization for Standardization (ISO) is publishing a standard for this particular type of heat exchanger. ISO 21,905, titled “Gas turbine exhaust systems with or without waste heat recovery” [34] is currently in the draft phase. It will be the reference standard that covers non-fired, single phase, tubular waste heat recovery systems housed within the gas turbine exhaust systems, except for the exhaust plenum. It does not cover HRSGs and focuses on modularized tubular heat exchangers. The ISO document can be used as a guideline for manufacturers of such WHR units. It provides a list of typical heat transfer media used as: a water/glycol solution; Hot oil/Thermal oil; Water; or Hydrocarbons, liquid or gaseous. It provides also extensive recommendations to be observed during the scoping, analysis and design of a WHR system into the GT exhaust such as material selection, analysis methods for heat transfer, seismic protection measures, surface treatment, tube fin selection, damper selection, testing, control systems, auxiliaries such as piping and valves and maintainability.

To understand the differences in scope between various GT upgrades and corresponding expected outages, a short familiarization of GT outage interval terminology is provided hereafter. Typically, the shortest outage is a combustion inspection (CI) which, as the name suggests, focuses on the combustion components; with a time-range of approximately a week being common in industry. Next is the hot gas path (HGP) inspection which includes the CI scope, plus downstream sections and ranges approximately two weeks. Finally, there is a major inspection (MI) which comprises the full stationary and rotating components of the flange-to-flange section; with a time-range of approximately a month.

The second-longest scope upgrades comprise compressor and turbine upgrades; using advanced materials to improve robustness and longer usage life. Such upgrades can typically be implemented during an MI. The benefits include increasing the GT maintenance interval and reduction of the overall downtime of the unit. An example of this is the 33 K turbine upgrade Siemens provides for the range of SGT6–5000F gas turbines; enabling an extended operation duration by 33,000 h between maintenance cycles [35]. Similar upgrade options are common among the other major manufacturers.

Combustion source upgrades can be implemented during a CI. These target an increase in the CI interval, reduced NOx emissions, operational and fuel flexibility. One example is the DLN 2.6+ upgrade available to GE’s fleet of F-class GTs [36].

Auxiliary system upgrades can be implemented during short outages or long ones, depending on the scale of the upgrade. There are many upgrade options available as the auxiliaries comprise a long list of systems. As an example, valves such as the ones used for fuel gas control can be converted to upgraded models with reduced maintenance requirements, within two 12-hour working shifts. On the other side of the scale, there are also large and time-consuming upgrades. For example, converting the inlet filtration system from a static system, to a self-cleaning one can take even longer than an MI. There is still a business case as the benefits include lowering the inlet pressure drop and reducing unplanned downtime for filter changes. Control system hardware and software upgrades are also available. Smaller scale software changes such as spanning logic changes to a particular auxiliary system require a short downtime in the range of hours. Complete software model upgrades and hardware upgrades can take several days. Such upgrades are sometimes necessary as hardware components or software security patches reach obsolescence. Digital upgrades, such as advanced monitoring and diagnostic systems can be implemented within a day, if the software platform is already available. The upgrades are highly customizable and can be suited to particular power plant needs.

Another business strategy for major OEMs consists of developing plans to penetrate the competitor’s market space, by offering upgrade or maintenance services for products developed by a different OEM. These are referred to as “other-OEM” products, in the power industry. OEMs may counterbalance by increased research and development into novel programs and products, in order to stay ahead.

With a young and rising population, as well as ambitious industrialization programs in GCC countries such as the UAE 2021–2030 and Saudi 2030 vision, it is expected that the electricity demand in the GCC will continue increasing. A recent study [37] suggested an increase from approximately 850 TWh in 2020 to 1400 TWh in 2030, which is a 65% rise. Most of the gas turbine power plants in this region have been built in the last few decades and still have a long time to go to reach end of life. The outlined OEM strategies, combined with a high ratio of SC to CC gas turbines of approximately 70–30%, creates a large market opportunity in these countries for a lower-cost option of waste heat recovery.

The objective of this paper is to offer a comprehensive review of such waste heat recovery systems for SC gas turbines and describe the most recent developments and inventions in this area of technology. Each of these developments were critically discussed aiming to assess their potential for extracting exhaust energy and suggest further potential areas of improvements to the thermal power cycle in terms of output, efficiency or reliability. Preceding this section, components of gas turbine-based thermal power generation systems were described followed by investigating various types of heat extractions within the gas turbine flange-to-flange section, auxiliary systems and outside the gas turbine system in the power plant. Also, a new methodology was established to enable comparison of heat extraction performance according to several criteria that are important in industry such as safety and effectiveness.

The proposed methodology and relevant analyses are expected to bridge up the gap between scientific research and industrial applications and trends aiming to aid the power plant owners to select the optimum utilization of wasted heat as well as to help academic researchers to gain insight into practical applications of heat recovery systems for gas turbines, and identify possible areas of future research.

2. Description of gas turbine power generation systems and components

The discussion of heat recovery application options, as well as exhaust heat extraction points, is precluded by a description of GT systems and components. In the following pages, these are described and divided into main three sections: the flange to flange section, GT auxiliary systems, and the exhaust system.

2.1. Flange-to-flange section

A GT consists of several major sections, as shown in Fig. 2. The compressor is driven by the GT rotor and compresses filtered air for the combustor section downstream. The compressed air pressure relative to inlet pressure is referred to as pressure ratio. The pressure ratios for
heavy-duty GTs (as opposed to aero-derivatives) in the market ranges from around 10:1 to about 24:1 \[38,39\]. To provide an example for some of the newer and larger models, the pressure ratio of a General Electric (GE) 9HA class GT having a 14-stage compressor, is 21:1; and 24:1 in the case of a Siemens 9000HL class GT \[13–15\].

The compressor discharge air then enters the combustion system, where fuel is added, generating a stable combustion flame. The high temperature and pressure combustion products then enter the hot gas path, that includes a multi-stage turbine downstream, the rotating parts of which are assembled on the same GT rotor. Part of this rotational energy is then transferred, through a coupling, to a generator shaft. The other part is used to drive the compressor at the upstream end; thus, completing the cycle. In summary, the GT units convert the energy stored in fuel into kinetic energy that generates a torque on its rotor, based on the Brayton thermodynamic cycle. The components of the GT described above are described in industry as the “flange-to-flange” section.

### 2.2. GT auxiliary systems

Outside of this section, there are numerous auxiliary systems, which enable unit operation in combination with a control system. The following sections will provide a description of those systems having a fluid flow. Potential benefits for the application of recovered exhaust

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**Fig. 2.** A schematic diagram depicting the major components of a gas turbine engine \[40\].

**Fig. 3.** Simplified P&ID of a GT inlet air and inlet bleed heat system \[45\].
heat, to the flow in each of these systems, will be examined.

2.2.1. Inlet system

The upstream-most section is the inlet system. In comparison, the exhaust system is the downstream-most section of a SC GT. The inlet system orientation is designed to be upstream, in the sense of prevailing wind direction, compared to the exhaust system [41] in order to reduce suction of exhaust gas products back into the inlet system. While the reference states a minimum distance of 7.5 m; that is most likely for a smaller aeroderivative unit. The horizontal as well as vertical distances between the ambient-facing sections of the inlet and exhaust systems vary greatly depending on the size of the unit. Fig. 3 shows a simplified Process and instrumentation diagram (P&ID) for the combined inlet filtration and inlet bleed heat system. Its purpose is to direct filtered air into the GT compressor. The direction is provided by ducting, distributing the air evenly into the front end of the compressor through an inlet plenum. The filter banks may be single or multi-stage. The benefits of multi-stage filtration include decreased chance of air bypass from the filter seals, improved filtration efficiency, and the ability to replace the upstream stage of filters while the unit is in operation. This is typically done in order to reduce GT downtime. One of the filter stages may include a self-cleaning system. This is a system to provide a pulse of compressed air into the downstream side of the filter, in order to push the dust off the upstream side; reducing the amount of contamination embedded in the filter pores and increase filter life.

The inlet system may also comprise power augmentation; which increases the mass flow rate of air into the GT. Increasing the mass flow rate may be achieved by cooling, either evaporative cooling or chilling; or water spraying, which is then referred to as fogging or wet compression. The power augmentation system is installed after the filter banks; in order to mitigate moisture contact with the filters. However, it must be considered that cooling the compressor air can lead to a reduction of exhaust air temperature as well; which may be detrimental to the combined cycle performance. Depending on the combined cycle design and ambient conditions, the higher mass flow rate might not make up for the reduced temperature [42,43]. Real measurements from three power stations confirms this finding [44]. Hence the use of power augmentation systems in inlet systems needs to be considered carefully for a SC GT which involves exhaust heat recovery.

An inlet bleed heating option may also be provided, for added operational flexibility. The system extracts hot air, typically from the compressor discharge, to ensure cleanliness. It is then discharged to a manifold inside the inlet filter house. The manifold has openings to allow the hot air to mix with the colder air coming in from the ambient; thereby raising the overall temperature of the air entering the gas turbine. This system is an obvious potential beneficiary of heat recovered from other parts of the GT cycle; and will be discussed in further details in Section 5.

2.2.2. Cooling and sealing air system

The cooling and sealing air system extracts a small amount of air from various compressor stages; directing them to bearing seals to seal against lubrication oil leakage; or to various stages of the hot gas path for internal cooling. Fig. 4 shows a simplified P&ID. It is obvious that this air extraction has a negative effect on GT efficiency as the compressor has performed work to increase the air pressure from the stages it is being extracted from. It is not surprising that various publications [46,47] agree on this matter. The latter states that up to 0.5 percentage points in efficiency can be lost, per percentage of extracted air for cooling purposes. Hence there is an opportunity, through waste heat recovery, to gain efficiency by reducing cooling air. This opportunity occurs if the heat recovery takes place adjacent to the locations that are typically cooled by cooling air; which are the turbine stages. If heat recovery results in cooling of that area; and if that cooling results in a lower requirement for cooling air, then such an opportunity may present itself.

The cooling and sealing air systems also typically include a bleed valve, which is used to discharge or “bleed” a portion of the GT air.
compressor air during transient conditions, in order to protect the compressor from stall. There is no loss of performance caused by this bleed air; as it is extracted from the compressor during start up and shutdown conditions, where no power generation is taking place.

2.2.3. Bearing lubrication system

The bearing lubrication system typically consisting of a reservoir tank, pumps, filters, cooler and control valves. This is shown in Fig. 5. Best practice calls for the provision redundant pumps, as it is critical for lubrication to be available even in case of the primary alternating current pump failure. Further, a direct current pump, powered by emergency batteries, is typically available in case of a plant blackout. To help the oil drainage from the GT bearings, the lube oil tank is held under vacuum, using a tank vacuum fan. The fan suctions air above the oil level in the tank, to generate a vacuum. The outlet is then routed through a filter prior to release into the ambient, in order to mitigate contamination.

Although the system may provide oil to other auxiliary systems, the primary purpose of the lubrication oil system is to cool and lubricate the GT bearings. Hence there is no opportunity to use recovered heat in this system. The only time the lube oil system is heated is during start up, if the ambient temperature is so low that the increase in viscosity would cause a low flow condition through the piping. However, lube oil systems typically include a small electric heater to overcome this issue; notwithstanding that during start up there is no opportunity to extract waste heat from the GT. The same comment also applies to the hydraulic oil system.

2.2.4. Hydraulic system

Oil is also used for the hydraulic system, which contains similar components to the bearing lubrication system but rated to a higher pressure. Similar auxiliaries as for the lube oil system are typically provided; except for a battery-powered pump. In case of loss of hydraulic oil, the hydraulic valves revert to their fail-safe positions, causing the GT to trip. The high-pressure hydraulic oil is used for the positioning of hydraulic valves; and may also provide lift for the heavier model GT and generator bearings at low-speed conditions. A hydraulic accumulator is typically provided to compensate pressure fluctuation in the system that may be caused by undesired valve position changes or issues such as air presence in the system. Fig. 5 also shows a simplified P&ID of this system.

2.2.5. Fuel system

The gas fuel system includes a treatment system upstream of the GT base; and then mainly a safety stop valve, a safety vent valve, gas control valves and a manifold for a controlled distribution of the fuel to the combustion system. A flow meter is important as it provides feedback to the control system, which compares it to the given command. GTs can be configured to be capable of operating with single, dual or multi-fuels. The liquid fuel system, also contains similar components as mentioned for the gas fuel system. In case the combustion system consists of various cans or evenly distributed flow around the combustion perimeter, a geared flow divider may be used downstream of the pump in order to ensure accurate division of the fuel around this perimeter.

Additionally, some configurations require associated water or air
mixing systems. Air would be used to create an even spray of the liquid fuel. Water can be used to mix with a fuel, resulting in an emulsion. In lieu of that, a separate water injection passage may be used such that the water is mixed further downstream inside the combustion system. Used in this manner, water injection can also be applied to the gas fuel system. Water is mixed further downstream inside the combustion system. Used in lieu of that, a separate water injection passage may be used such that the heating methane from 20 to 200°C led to a heat rate improvement of up to around 5%. Hence, the fuel system is another potential beneficiary for the use of recovered heat. It will be discussed more extensively in Section 5 of this article. The potential of the associated water or air mixing systems as beneficiaries of recovered heat will also be discussed.

Some liquid fuels are highly viscous in nature; and are unable to properly flow through the piping, GT injection and pumping systems; even at the hottest ambient temperatures. They must be heated as required by the operation. These fuels are typically heavier fuel oils that are used as an alternative option, providing a power station with fuel flexibility. John et al. [50] invented a device to heat liquid fuel and injection water simultaneously; using waste energy in the GT exhaust stack. In an example provided, the liquid fuel was heated from about 27°C to about 82°C, to enable unit operation on liquid fuel.

When air is used in the fuel nozzle to create the necessary even fluid particles for efficient liquid fuel combustion, it is referred to as atomizing air [51]. The air is typically pressurized further by an external compressor, after being extracted from the GT compressor itself; and does not require further heating [52]. Alajmi et al. [53] produced a review of novel fuel atomization technologies for gas turbines, in 2018. While air-assisted atomization is one method; there are multiple methods to achieve atomization that do not involve air. In most cases, heating is considered detrimental to the atomization process. The only process where heating is applied is in supercritical fluid assisted atomization. However, this technique is still not applied to gas turbine fuel nozzles. Hence, it can be concluded that heat recovery is neither beneficial nor applicable to fuel nozzle atomization, according to the current available technology.

In the case of water addition into the combustion system; for the purpose of reduced emissions or improved efficiency, it is either done through water or steam injection. In the case of steam injection gas turbines (STIG), the HRSG is used to generate steam which is then injected into the GT combustion system. Thus, by using the exhaust gas heat that is going to the HRSG, some of the GT energy is recovered by generating steam from water prior to injection into the combustion system; thereby increasing the cycle efficiency [54]. Mathioudakis [55] agrees on the advantage of steam injection over water injection; stating that water will consume some heat for evaporation in the combustion system. Therefore, more fuel is consumed in the combustion system in order to keep the combustion outlet temperature constant. This additional fuel consumption reduces the cycle efficiency in comparison to steam injection. This was numerically shown by Cloyd and Harris [56] in their correction factors charts for water injection as well as steam injection performance calculation. As an example, comparing a water/fuel ratio of 1 to a steam/fuel ratio of 1, steam injection provides a 6% improvement to the heat rate calculation, in comparison to water injection.

Additionally, for a CC GT, another advantage is that the supercritical steam created in this manner allows the HRSG to extract more heat [57]. In the case of a SC GT, water is mixed in an HRSG. If another method of exhaust heat recovery is employed to generate steam, a similar improvement in cycle efficiency may be achieved. Hence, water or steam injection can be a valid beneficiary of exhaust heat recovery. By the logic of added fuel requirement for injected water vaporization in the combustor, any heating of water above ambient temperature would provide some level of performance enhancement. This was also stated by John et al. [50].

Purge air is used to provide protection against combustion backflow into the fuel system as well as to cool fuel passages in the combustion system that are not used. Such as when a multi-fuel system is only operating on one fuel at a time. The air used for purging unused passages of a fuel nozzle is typically sourced from GT compressor discharge [58]. The air is provided to the purge air system; which may contain an air compressor, filter and control valves, for processing prior to routing into the fuel nozzles. A cooler is optional, depending on the design temperature of the combustion system components. Fig. 6 depicts the fuel, water injection and purge systems in one simplified P&ID to show the interconnections. However, as the main purpose of the purge air is cooling unused fuel nozzle passages, it would be counterproductive to treat it with heat. Hence, this system cannot be a beneficiary for a heat recovery system.

### 2.2.6. Cooling water system

The cooling water system is used to provide cooled water to the auxiliary system heat exchangers. These are typically used to cool lubrication and hydraulic oils, air systems, as well as some on-base components such as instruments close to the GT hot section or the GT foundation. The heat exchangers may be of various types, such as shell and tube, plate or even printed circuit heat exchanger. A control valve is required to regulate the water flow through the individual heat exchangers. The water outlet from all the heat exchangers is typically directed to a common header, which is then routed through a forced-draft cooler to dissipate the heat to the environment. A common model is the fin-fan cooler; in which water flows through finned tubes that are cooled by fans externally. Fig. 7 shows a simplified P&ID. As the purpose of the system is provision of cooling; it cannot benefit from a exhaust heat recovery system.

### 2.2.7. Compartment, ventilation, and hazardous gas protection system

The GT itself is housed inside a compartment. The function of the compartment comprises personnel protection against heat and noise from the GT during normal operation and against fire or explosion during abnormal conditions. It also contains a ventilation system to provide even cooling for the external surfaces of the GT flange to flange section to avoid distortion of some of the auxiliary equipment housed inside the compartment. Most of the heat generated in the compartment is from the GT flange to flange surface with some smaller amounts stemming from piping, such as purge air. In the example of a 9FA gas turbine [60], the surface temperatures are approximately 267°C while the temperature of the air in the enclosure circulating at 18.4 m³/s, was 47°C. This air circulation was stated to be 80 air changes per hour (ach) for the designed compartment.

The air temperature in the compartment has a linear relationship with the ambient temperature as it is ambient air that is siphoned by the ventilation fans. Ventilation is achieved through either positive pressure or negative pressure fans. Positive pressure fans extract ambient air and pressurize the turbine compartment. A negative pressure fan creates a vacuum inside the compartment. Air enters the compartment through louvers with gravity dampers.

Heaters are also provided to protect components, and the fluids contained within piping, from freezing [61]. The heaters are typically electric. The system cannot benefit from waste heat extraction as the heat is required when the GT is shut down. When the GT is in operation, the compartment is already at high temperature due to thermal radiation, mainly from the GT casing.

A hazardous gas protection system is required for safety reasons and to detect combustible gas in the turbine compartment. The system contains instruments that can detect the percentage of combustible gas lower explosive limits. Typical instruments are based on catalytic bead or infra-red technology. The compartments also contain fire detection and protection systems. The fire detection instruments can be heat sensors or ultraviolet monitors. The protection is typically provided by blanketeting the compartment with carbon dioxide or water mist. The combined compartment ventilation, hazardous gas and fire protection systems are shown in Fig. 8.
The ventilation system in the compartment has a control interconnection with the hazardous gas detection system. The latter’s configuration must take the gas dilution into account, at the level of air exchange achieved by the ventilation system. It is for that reason that the ventilation air flow must be controlled carefully. The heat transfer and gas dilution analysis is performed using computational fluid dynamics (CFD) tools \[60, 61]\. After the analysis, the design simulation is also typically validated at site, by taking detailed temperature, airflow and dilution measurements.

A reference that is commonly used by authors \[60–62]\ for considerations in safe design practices of hazardous gas detection systems, even as late as 2014, is the work conducted by Santon \[63]\ for the UK Health and Safety Executive (HSE) in 1998. While explaining that at the location of the gas leak, a pure concentration of the hazardous gas is inevitable, a suggestion was provided for an acceptable range of dilution. This being a dilution from pure gas to 50\% Lower Explosive Limit (LEL), within a volume of 0.1\% of the net volume of the enclosure from the source of the leak. This explains the concept of “dilution ventilation”, which is a level of ventilation that is able to achieve such a dilution requirement.

Santon further suggested that while an alarm limit of 25\% LEL is typical for hazardous gas detectors; a setting of 5\% is also used by many operators, to be conservative. This is confirmed in the example case of the 9FA gas turbine plant \[60]\ . In GT control systems, there is typically also a second and higher level of detection; which cause a GT to trip. The instruments that are used for detection were described earlier. These do not benefit from a heat source. On the contrary, some of them have sensitive electronics that are damaged by heat. In fact, some of them need to be installed in a cooler area outside the GT compartment for this reason. In such a case, a representative air sample is siphoned from the

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**Fig. 6.** Simplified P\&ID of a GT dual fuel system with air and water injection capability \[51, 59]\.

**Fig. 7.** Simplified P\&ID of a GT cooling water system.
compartment towards these detectors. Hence it is clear that the hazardous gas detection system cannot benefit from recovered exhaust heat.

2.2.8. Turning and starting system

A starting system is necessary to help overcome the initial rotor inertia at zero speed. It is also used to speed up the GT during start, from the speed required to purge out residual unburned fuel from the previous operation, until the GT can sustain its own speed which is referred to as self-sustaining speed. The starting system is also used to crank the GT during offline water wash, which is a system that will be discussed next.

Fig. 8. Simplified P&ID for the turbine compartment; showing the ventilation, fire protection and hazardous gas detection systems [60,61].

Fig. 9. Simplified P&ID for the GT starting system [65].
The starting system is connected to the main GT shaft, as shown in Fig. 9. In a heavy-duty gas turbine, the starting system may contain large motors or hydraulically driven torque systems, connected to the GT shaft via a gearbox and clutch. The clutch enables disengagement after the GT reaches self-sufficient speed. One example is an overrunning and self-disengaging Synchro-self-shifting (SSS) clutch [64]. A separate gearbox may also be provided, to use some of the GT shaft rotational energy to drive rotating equipment such as smaller compressors or pumps used in the previously mentioned auxiliary systems.

While a common configuration for a GT starting system was shown in Fig. 9, there is a wide variety available. Greve, Miller and Shaw [65] discussed various options, such as pneumatic, hydraulic, mechanical starter engines, and electric motors. A load-commutated inverter is also used on heavy duty GTs [66] where the concept is to use the generator as a motor during start-up. While there is a variety of systems, the commonality is that they do not require a source of heat. Hence, there is no application in these systems for recovered exhaust heat.

2.2.9. Water wash system

Industrial gas turbines are usually provided with a compressor and turbine wash system which contains a water tank, together with a detergent dosing system, as well as nozzles directing the water or water-detergent mixture into the compressor or the hot gas path. This is used to clean these areas and recover lost power generation through fouling over time. Fig. 10 shows the water wash system. The compressor requires both online and offline wash options. Hot gas path cleaning options are provided for units that burn fuels that result in higher deposits, such as heavy fuel or crude oil. A drainage system is provided to ensure the GT does not become water-logged during offline water wash (WW). This consists of several valves in the turbine and exhaust sections. These valves can also be used in the case of an unsuccessful GT firing, in order to drain the unburned fuel from the turbine section and avoid the risk of an explosive atmosphere during a subsequent start attempt.

The water is typically heated for both online and offline WW. For the online operation, heating prevents thermal shock to the GT compressor components [67]. For the offline operation, heating is also useful as it will act as a better solvent for contaminants on the compressor blade surfaces. A typical water wash P&ID [67] displays two electric heaters in the water tank. A pair is likely provided for redundancy; as otherwise a single larger heater would have had less connections and therefore less costly to install. In the case of a Frame 6 GT, generating 27 MW, the online WW flow is 10 GPM; while the offline WW flow is 42 GPM. The water is heated up to 82 °C in each case [68]. Other articles [69,70] suggest heating up to 60 °C is sufficient for similarly sized GTs. In either case, electric heaters are sufficient for such applications; as the water does not need to be heated up during flow. The heaters are in the tank; and there is sufficient time for the tank water to be heated prior to WW operation. Hence, recovered exhaust heat is not economical to be routed to the WW system.

Up to this point, the typical auxiliary systems of a SC GT were described. The intention was not to cover all the auxiliary systems but describing system that having fluid flow and thereby potential for application of heat recovery. Each system was analyzed for the potential application of recovered exhaust heat. Prior to examining the exhaust system, it would be beneficial to provide overall summary of the opportunities that were highlighted so far. This summary is shown in Table 2. Out of 15 systems, 5 of them only show a potential to make use of heat recovery in a manner that provides a gain to the power plant operation.

2.3. Exhaust system

The GT exhaust system is the downstream-most section of the GT. Ginter [71] provides a good overview of various components of an exhaust system. Fig. 11 shows a simplified P&ID. As mentioned earlier, the turbine is the downstream-most section of the GT flange to flange envelope. It is typically built in a diverging cone shape to allow the combustion product gasses to expand in the hot gas path.

The exhaust frame is the first connection point after the turbine. It provides a mechanical structure and flow path for the exhaust gasses to move further downstream. Next is the diffuser, which is comprised of a duct connecting GT exhaust frame to the plenum. The diffuser may be inside a protective compartment; or have heavy internal or external insulation, in order to prevent high temperature leakage from the exhaust gasses. The connection of the diffuser to the plenum is through a flexible (or expansion) joint to allow for expansion and contraction during various stages of the GT operation. The plenum itself is a large compartment housed on top of a separate foundation. It is typically manufactured of metallic layers enveloping heat insulation. The exhaust gasses enter the plenum to be directed either into an exhaust stack, or to...
Table 2
Summary of recovered exhaust heat applicability within SC GT auxiliary systems [47,48,50,56].

<table>
<thead>
<tr>
<th>System</th>
<th>Summarized description of benefit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>IBH provides operational flexibility in part load conditions</td>
</tr>
<tr>
<td>Cooling &amp; sealing air</td>
<td>An efficiency gain of up to 0.5% may be achieved, if no cooling air is extracted from the compressor stages [47]. Heat extraction from HGP stages would have a cooling effect; reducing the need for cooling air.</td>
</tr>
<tr>
<td>Bearing lubrication</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>Hydraulic oil</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>Gas fuel</td>
<td>A heat rate improvement around 5% has been shown in an example of heating methane from 25 °C to 200 °C, in a 25 kW unit [48].</td>
</tr>
<tr>
<td>Liquid fuel</td>
<td>Heating to reduce viscosity is a necessity to enable unit operation, for heavier fuel oils [50]</td>
</tr>
<tr>
<td>Fuel-related air and purge</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>Water injection</td>
<td>A heat rate improvement in the range of 6% has been shown, for the same rate of steam injection compared to water [56]. Recovered heat can be used to evaporate the water.</td>
</tr>
<tr>
<td>Cooling water</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>Compartment, ventilation and</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>hazardous gas detection</td>
<td></td>
</tr>
<tr>
<td>Turning and starting</td>
<td>No opportunity identified</td>
</tr>
<tr>
<td>Water wash</td>
<td>No opportunity identified</td>
</tr>
</tbody>
</table>


a heat recovery system.

The exhaust system receives the major focus of this paper. For this reason, it is being introduced in more substantial detail than the other systems. It was also important to introduce the other systems to demonstrate the interconnected systems that enable the gas turbine operation.

In a simple cycle (SC) gas turbine, without heat recovery, the exhaust gasses are directed from the exhaust plenum into an exhaust stack. In a combined cycle (CC) or a GT with heat recovery, the plenum will direct the exhaust gasses to a heat exchanger instead of an exhaust stack. In a unit that can run in either of SC or CC configurations, there is a heavy diverter damper that can move and re-direct the exhaust gasses into the SC exhaust stack or the CC heat exchanger as necessary.

If the exhaust gasses are to go to an exhaust stack, they leave the plenum through a transition duct. This may sit above the plenum or on either side. The transition duct diverges in order to connect into the silencer section. The silencer section contains parallel silencer baffles; and requires a larger outer cross-sectional area in order to have room for the silencer baffles without developing an unacceptably large back-pressure. After the silencer section, there is another converging transition duct, which then connects into the stack cross-section.

The exhaust stack is the final section of the exhaust system. It is a chimney allowing the exhaust gasses to exit to the atmosphere. It is required to be at a certain height, as per local regulations; and is typically the tallest section of a power plant.

A common method for extracting heat from a SC GT exhaust is using an HRSG and thereby re-configuring the plant to a CC unit. The high cost of modification was described earlier may prove to be prohibitive for many power plant operators. Another common method of heat recovery is through WHR units.

WHR units require substantial space and are heavy. A typical WHR unit for boiling water weighs approximately 50 tons and has dimensions of 10 m width, 8 m length and 14 m height [72]. They may be installed between the GT exhaust diffuser and exhaust plenum, in which case they will have a circular inlet for the hot exhaust gasses. An example of such a configuration is shown in Fig. 12 below. As a result, aside from the cost of installation, one would have to pay the following additional costs: Shifting the entire exhaust plenum and exhaust stack to make room for the WHR unit, building additional foundation for the new location of the exhaust plenum; strengthening the foundation underneath the new WHR location, and possibly installing new guide vanes downstream of the WHR to help reduce turbulence caused by the flow over the finned tubes.

Once it is installed, it cannot be bypassed. The pressure drop in the exhaust system remains in place permanently. The only way to bypass it would be to install a bypass damper and a separate exhaust stack. However, this is very expensive to practically implement.

3. Methodology, basis of classification and evaluation of heat recovery methods from GT exhaust systems

The primary objective of this section is to identify a clear methodology which provides classification and evaluation of inventions in the area of exhaust gas heat extraction, which are presented later in Section 5, aside from the typical used heat exchangers, being HRSGs and WHR units. The focus will be on methods for extraction of useful heat from any section of a SC GT exhaust system, comprising the exhaust duct, diffuser, plenum or stack. It is important to note that these sections provide the highest grade heat where the heat was classified according to temperature in the waste heat recovery research area [73]. Further, an analysis of these developments will be presented in terms of ease of implementation, benefits and risks and opportunities for further improvement.

The developments that will be presented have their individual complexities and purposes; and cannot be strictly ranked against each other. However, in order to enable an easier overview, five aspects will be introduced and ranked, according to the evaluation chart shown in Fig. 13. The scoring system mainly comprises three levels: “A”, “B” and “C”, considered relative to each other. In this scoring system, “A” is the best score; “B” the middle; and “C” the lowest score. Other scoring systems such as “high”, “medium” and “low” were not used to avoid any confusion. For example, within the aspect of “pressure drop”, a lower the pressure drop is more advantageous for the exhaust system.

Risk appetite is a very important aspect and therefore was given a separate score ranging from “1” to “5”. The aspect will be named safety and risk. In this case, “1” is the lowest and “5” the highest score. It is important to score the level of risk that exists in case of failure of the added system or its components and compare it to the other aspects in order to determine whether the cost of additional safety measures are justified by the benefits attained.

If a failure can cause a safety risk such as a fire or a human exposure to hazardous gasses, it will be given a score of “1”. If a failure can lead to a prolonged forced outage, such that it would require weeks of repairs to return the unit to operation, it will be given a score of “2”. A forced outage means that the operator does not have an option to continue running the GT safely with the existing damage, and is forced to shut it down. If the damage causes a forced outage that would typically only take a short time, such as days, to repair then it will be given a score of “3”. Damage not causing a forced outage is given a score of “4”. Practically, it means the operator is able to continue running the GT safely with the existing damage. The repair can be planned for a period of planned downtime, such as an offline water wash or compressor borescope inspection. The repair may cause an extension to the planned downtime. If the expected repair time is within 24 h it would be typical for an offline water wash or a borescope inspection, a score of “5” will be given. Such levels of repairs typically do not require an extension to the planned downtime.

The next aspect is the complexity of implementation, considered as Aspect number 1. A configuration is easier to implement in an existing SC GT unit if the modifications are limited to hardware changes on the exhaust system of the GT. Especially if the party performing the modification does not need access to proprietary design documentation, or intellectual property (IP) information of the GT manufacturer such that they are not limited to implementing the modification on a GT of their own make. This configuration will be given a score of “A”. Such modifications typically do not require changes to the GT control logic where a
A stand-alone control system for the heat extraction system can be employed. If the modification is limited to such hardware changes, but requires proprietary design information from the GT OEM, then it will be given a score of “B”. If the modification is not limited to the exhaust system, and also includes operability changes on other GT systems and interactions with them, then it will be given a score of “C”. Such modifications not only require proprietary design information but also need modifications into the GT controls logic. Hence, they are the most difficult to implement; particularly for a service provider lacking access to the GT OEM intellectual property.

The second aspect is effectiveness, which ranks the level of benefits that may be attained from implementing the modification. If the benefit is limited to waste heat recovery that is used for an application outside of the GT, then it will be given a score of “C”. Additionally, if there is an improvement in thermal efficiency, score “B” is given. Modifications that attain the first two benefits, while also improving the GT system operability, such as adding operational flexibility, are given the highest score of “A”.

The third aspect compares the levels of additional pressure drop expected in the exhaust system. In this scoring system, “A” with minimum pressure drop is the best and “C” the lowest score. If existing surfaces are used with no additional obstruction, a score of “A” is attained. Partial flow path obstruction, or partial flow diversion receives a score of “B”. If the full flow is subjected to additional obstruction, then a score of “C” is given.

The fourth aspect deals with the comparative scale of modification required to implement the design. It is closely related to the expected complexity of implementation, affecting the cost and time required. The largest possible modification would comprise of the complete replacement of the exhaust system such as when a completely new exhaust stack is required. This is given a score of “C”. If the modifications comprise significant work on the existing components in the exhaust flow path, such as major modifications to an exhaust diffuser, then score “B” is given. Where no modifications required to the existing components in the exhaust system and external components only need to be added into the flow path with minor penetrations made into the existing components; almost all the manufacturing can be done while the unit is in operation. A short shutdown opportunity of a few days suffices for installation. Such cases will receive a score of “A”.

The fifth and final aspect is the potential market opportunity. If the
modification can be implemented on any gas turbine, as the components are designed for configurations or system components that are considered typical, the market opportunity is the widest. Such cases receive a score of “A”. Some other designs can be implemented on any gas turbine, with certain limitations. These are designs that contain features requiring a certain existing design base or require minor modifications to fit on a different base. Such designs will be given a score of “B”. Some other designs are specially designed for a particular GT model or configuration and hence may not be installed on others without significant modification; and will be given a score of “C”.

Fig. 13 shows an overview of the rankings that were described in detail in the preceding paragraphs. It can be referred to for the purpose evaluation of the various systems in the next section. Three of the aforementioned aspects (i.e. safety and risk, pressure drop and effectiveness) were discussed in a recent conference publication [74] in a shorter context.

In order to compare the new developments against the WHR system described earlier in this section, the scoring will be assigned for WHR first. For the aspect of safety and risk, a failure would be a leak in the heat exchanger. The pressure in the diffuser duct, in the inches of water range is typically much lower than that of the heat transfer fluid, where at least tens of psig are typical. Hence the leak would be from the heat transfer fluid into the exhaust gas flow path. As long as the heat transfer fluid is non-flammable, it would only result in a loss of heat transfer efficiency. The WHR can be isolated and therefore, it would not result in a forced outage. A planned outage would however be extended by several days for repairs. Hence, a score of “4” is applicable.

The complexity aspect depends on the use of the recovered heat. If it will be routed back for use within the GT then it requires system-level and GT controls modifications. As a simpler approach, the heat may be used for an external application such as simply replacing an existing fuel gas performance heater; so that modifications on the exhaust system are limited to hardware. As drawings of the exhaust system would be required, which is intellectual property of the OEM, a score of “B” is applicable for such an example. However, the score may be from “A” to “C” depending on the actual application. If we assume the same usage example for aspect two, then a score of “B” is also applicable here.

For the third aspect, it is clear that the heat transfer surface area will comprise the full cross-sectional area of the exhaust gas flow. Hence a score of “C” is applicable. For the described example, major modifications to the exhaust system are required. However, there is no requirement to replace the entire exhaust system. Hence a score of “B” is applicable to the fourth aspect.

All gas turbines will have an exhaust duct, or exhaust diffuser. Hence, such a modification can be applied to any GT. This means that for the fifth aspect of potential market opportunity, a score of “A” is applicable. The scores are summarized later in Table 7 along with the scores of other systems presented in Section 5 for the purpose of comparison.

4. Review of typical uses of recovered heat from gas turbine exhausts

Prior to discussing the design and construction of exhaust heat recovery systems, it is worthwhile reviewing the existing utilization methods of the recovered heat. Most publications in the field of exhaust heat recovery deal with methods of utilization rather than methods of
extraction. However, in this work more focus is to explore the methods of extraction that used for different heat recovery applications. For instance, Saghafifar et al. [75] provided a techno-economic analysis of an air bottoming cycle, a carbon dioxide-based power cycle and a Kalina bottoming cycle. The purpose of this section is to outline the options for heat utilization beyond the SC GT auxiliary systems. In order to follow the terminology used in Section 2, the utilization will be divided into three categories: (1) Use within the GT flange to flange section; (2) Use

Fig. 13. Flowchart showing an overview of the aspects of evaluation, used for ranking the heat extraction methods against each other.
within one of the GT Auxiliary systems; and (3) Use in the power plant, outside of the GT flange to flange or its auxiliary systems. Each is covered in the following sub-sections; and then summarized in Table 3.

### 4.1. Use within the GT flange to flange section

Referring to Fig. 2, the section of the GT from the front of the compressor casing until the end of the turbine casing, is referred to as the “flange to flange” section in industry. The terminology refers to the main rotating parts of the GT along with the static casing that encompasses them. The upstream connection is to an inlet plenum or ducting. The downstream connection is typically an exhaust diffuser.

#### 4.1.1. Recuperation cycle

A commonly heat recover method used is the gas to gas recuperation cycle. In this configuration, a recuperator is used to pre-heat the GT compressor discharge air before it enters the combustor. As the air is heated, there is less fuel required for heating and thus the efficiency of the cycle is improved. A recuperator is a heat exchanger for which the hot side flow is exhaust gas air, and the cold side is compressor discharge air. They are typically designed as a cross or counterflow metallic matrix through which the exhaust air flows. The cold side flow goes through piping integrated into this matrix as shown in Fig. 14 [76].

The recuperation cycle is utilized for smaller scale GTs up to 25 MW as there is a higher benefit at lower compression ratios. Efficiency improvements between 3 and 18% have been reported in this scale [77]. The reason that at higher compression ratios above 15:1 the heat of compression will exceed that of the exhaust gas temperature, which renders the recuperator ineffective. Modern GTs have compression ratios above this range which would limit the benefit [78].

As an example, the Mercury 50 GT from Solar Turbines, with a compression ratio of 9.9:1 comes with a recuperator as a standard [79]. The recuperator fitted is into the specially designed exhaust system to allow the turbine exhaust to pass over the heat exchanger plates. Air from the compressor discharge is piped into the heat exchanger, with return piping into the combustor. Fig. 15 displays a simplified P&ID of a recuperator system.

#### 4.1.2. Humid air turbine cycle

Another application of the recovered exhaust energy in the GT flange to flange is in the Humid air turbine (HAT) cycle which is shown in Fig. 16 [77]. In this system, air is extracted from the GT compressor at a lower pressure stage and is cooled in an intercooler. The recovered heat is used to saturate air by circulating water, which then flows into the saturator. A similar scenario occurs in the second compression stage of the GT compressor which flows through an aftercooler. The air continues downstream into the saturator. As the humid air leaves the saturator, it is heated by a recuperator and then enters the GT combustor. The GT exhaust heat is used in the recuperator for heating the humid air from the saturator, as well as in an economizer, to pre-heat water that is used for air saturation. An advantage is that water flow can be varied to enable continued use of the cycle at various GT operating loads. A disadvantage is the high consumption of water. Poulliikas [77] further stated the consumption is a third of a steam injection cycle, placing it around 0.5 kg of high purity water per kWh of output. The capital cost of the water treatment system is placed at approximately 5% of the total plant with running costs of approximately 1.5% of the fuel cost.

In 2011 another study of a 600 KW GT was conducted by Nybern and Thern [80]. The study analysed the cycle through simulation to create performance maps. Compared to the simple cycle efficiency of 36.5%, the recuperated cycle efficiency improved to 43% and the HAT cycle produced a further increase to 44%. The economizer added a further 3% and the aftercooler contributed to another 0.5% for a final efficiency of 47.5%. It should be noted that highest jump in the efficiency resulted from using the recuperation rather than the HAT cycle. Another recent simulation study was conducted by Di et al. [81] on a 40 MW aeroderivative GT. It was shown that efficiency increased by less than 1% for a HAT cycle implementation. These studies showed limited improvement produced by the cycle in particular for heavy duty machines.

### 4.2. Use within the GT auxiliary systems

The GT auxiliary systems were described in Section 2 of this review paper along with potential areas of utilization. The auxiliaries also referred to as “accessories”, essentially encompass all the systems that are connected to the GT flange to flange and enable the operation and control of the GT. The control system contains the software logic driving operation during various modes of GT operation such as startup, firing, base load, shutdown and emergency trips. These enable stable and safe operation of the GT flange to flange systems.

#### 4.2.1. Fuel gas heating

One method of using the energy in the exhaust gas to the GT auxiliary equipment is for the purpose of fuel gas heating as shown in Fig. 17. Pre-heating the fuel gas prior to entry into the GT compressor increases the cycle efficiency by reducing the amount of fuel required for the gas in the combustion chamber to reach the firing temperature. Most of the major OEMs provide heat exchangers for fuel gas heating. Typically, these performance heaters heat the fuel gas by utilizing the energy in

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**Table 3**

Summary of the reviewed exhaust heat utilization examples [13,30,77,82,86].

<table>
<thead>
<tr>
<th>Utilization area</th>
<th>Name of utilization area</th>
<th>Short description</th>
<th>Possible detractions from implementation</th>
<th>Increase in efficiency*</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT flange to flange</td>
<td>Gas to gas recuperation</td>
<td>Exhaust energy was utilized to heat compressor discharge temperature prior to combustion</td>
<td>Only applicable to smaller GTs up to 25MW and with a compression ratio below 15:1</td>
<td>Range of 3–18%</td>
</tr>
<tr>
<td></td>
<td>Humid air turbine (HAT)</td>
<td>Exhaust energy recuperated to pre-heat saturated air prior to combustion</td>
<td>High capital and operating costs for water purification and consumption; compared to a limited efficiency improvement compared to recuperation</td>
<td>Range of 1–11%</td>
</tr>
<tr>
<td>GT auxiliary systems</td>
<td>Fuel heating</td>
<td>Fuel gas is heated prior to entering the GT combustor, reducing the fuel consumption.</td>
<td>Would cause additional back-pressure in the exhaust stack due to the presence of the recuperator.</td>
<td>11%</td>
</tr>
<tr>
<td></td>
<td>Pre-heating air into SOFC in a hybrid system with a GT</td>
<td>Compressor discharge air is pre-heated prior to entering a SOFC.</td>
<td>Currently only marketed in the micro-GT scale.</td>
<td>35%</td>
</tr>
<tr>
<td>Power plant, outside of the flange to flange and auxiliaries</td>
<td>ST bottoming cycle</td>
<td>Exhaust gas is used to create steam in an HRSG. The steam is used to drive a ST.</td>
<td>Major capital and operating costs.</td>
<td>20% per [13]</td>
</tr>
<tr>
<td></td>
<td>Stirling bottoming cycle</td>
<td>Exhaust gas heat utilized as Stirling engine heater</td>
<td>Improvement is at smaller scale than for an HRSG. Currently not marketed by major OEMs.</td>
<td>12.2%</td>
</tr>
</tbody>
</table>

*Information per the examples cited in Section 4, unless noted otherwise.
boiler feedwater. According to Gogoi [82], heat recovered from the simple cycle GT exhaust stack may also be utilized using a recuperator. This was demonstrated by showing the potential results through simulation. Gogoi [82] compared the thermodynamic cycle efficiencies for the baseline CC scenario with using only an air recuperator for compressor discharge heating, as well as also using a fuel recuperator. The output of the CC GT without any additions is 17 MW with an efficiency of 46%. By only using the air recuperator, the output increases to 23 MW with an efficiency of 61%. Adding the fuel recuperator, the output increases to 27 MW while improving efficiency a further 11–72%.

4.2.2. Solid oxide fuel cell system (SOFC)

Another method of utilizing the recovered exhaust gas energy for the GT auxiliaries is integration into the hybrid solid oxide fuel cell system. While this having been studied extensively in academia, such hybrid systems are still part of new developments in major industry. Buonomano et al. [83] described various configurations of the GT-SOFC hybrid system and also provided a review of marketability and concluded that this technology is still not mature enough for mass market penetration. The main challenges included economic viability and control systems for such a complex hybrid system. An early 220 kW prototype unit by Siemens Westinghouse was reported by Yi et al. [84] in 2004 which was also referred to by Buonomano et al. [83] as a baseline for various studies and analyses. However, a current search for further development or commercialized versions do not yield any results. Bao et al. [85] provided an extensive review of macroscopic SOFC models in 2017. They stated that the studies are mainly still based on numerical analysis. Hence it can be concluded that commercialization of this technology is still limited.

Out of the Major GT OEMs, Mitsubishi offers a hybrid GT-SOFC system for smaller gas turbines in the hundreds of kW output scale. In a direct integration into the GT system, the SOFC essentially replaces the GT combustion system as shown in Fig. 18. The compressor discharge air as well as gas fuel flow into the SOFC. The SOFC, which in this case will be pressurized and therefore with improved performance, generates electricity and heat. The hot exhaust from the SOFC then continues downstream into the GT turbine. The exhaust air from the turbine goes through a waste heat recovery unit, where the energy is utilized to pre-heat the air from the compressor discharge entering the SOFC. In one such configuration, a 58% cycle efficiency is analysed where the GT produces 104 kW power and the SOFC 359 kW [86].

The Mitsubishi configuration is a hybrid micro-GT-SOFC system that was demonstrated in 2016 and followed up by the company receiving its first order in 2019. In the 2016 demonstration [87] the output of the combined system was 250 kW at an efficiency of 55%. The first order of the system will be used in a cogeneration system, with the same output but a higher efficiency of up to 73%, while also reducing the CO$_2$ emissions by 47% [88]. For comparison, a chart of existing micro-GT outputs and efficiencies [89] showed that the best in class model has an efficiency of 38%. Hence, an improvement to 73% would represent significant improvement of 35 percentage points.

4.3. Use in the power plant, outside of the GT flange to flange or auxiliaries

Use in the power plant but outside the confines of the GT flange to flange or its auxiliary systems, refers to the use either in the balance of plant or a bottoming cycle. The balance of plant refers to systems and components in a power plant that are not directly part of the main drivers for electric generators. For example, cooling towers, fuel tanks, or fuel forwarding systems. A bottoming cycle, in the case of a GT power plant, refers to a thermodynamic cycle that uses waste heat from the gas turbine in order increase the overall thermodynamic efficiency of the plant. The gas turbine cycle is then referred to as the topping cycle.
example, an HRSG with a steam turbine is a bottoming cycle for a gas turbine topping cycle.

4.3.1. Combined cycle power plant with HRSG and steam turbine

The combined cycle with an HRSG and steam turbine is the most common bottoming cycle. It was previously discussed in Section 1.1 that it has a potential for significant plant efficiency improvement. In the cited example the efficiency increased to 54.9% compared to a SC efficiency of 37%. In an example for a same GT model, the cost was up to almost 40 million USD; which is prohibitive for some plant owners.

4.3.2. Stirling cycle

Aside from the ST cycle, there are also many other various bottoming cycles available for increasing the output and efficiency of the overall cycle. It is not the objective of this paper to review all of them. One example is a Stirling bottoming cycle, for which a simplified P&ID is shown in the Fig. 19 below; based on the description in Entezari et al. [90], which is further described as follows. In this arrangement, the GT exhaust gas is used in a Stirling engine’s heater. It is a closed cycle system with a gaseous working fluid. Heating the hot side cylinder causes the piston to move inwards, rotating the connecting shaft. The rotation causes the cold side piston to retract; suctioning the gas towards the cold side cylinder, where it cools. The cooling causes the piston of the cold side to move outwards, pushing the gas towards the hot side cylinder, where the cycle restarts.

It is reported that for a Rolls-Royce RB211 model GT rated for 27.5 MW output and 35.8% efficiency, an additional 9MW were recovered. This leads to an increase in cycle efficiency to 48% [77]. In comparison, using a bottoming cycle consisting of an HRSG with a steam turbine, for this model of GT leads to an additional 14 MW recovery, leading to a combined cycle efficiency of 49% [91]. This can explain why a Stirling bottoming cycle is not commercially offered for the RB211, now renamed SGT-A35 and part of Siemens’ portfolio, nor for other heavy-duty gas turbines by other major OEMs.

Table 3 summarizes the discussion of the exhaust heat utilization examples described in Section 4 and its sub-sections. The purpose of this section was to provide two examples for the utilization of exhaust heat in each of a GT power plant; rather than a review of an exhaustive list of uses.

Fig. 16. The HAT cycle with an intercooler and aftercooler as described by Poullikkas [77]. Solid lines represent air flow; dotted lines are water flow and solid bold lines represent humid air flow.

Fig. 17. Simplified P&ID of fuel gas pre-heating using GT exhaust gas heat [82].
Some of the utilization areas can be combined; while some are mutually exclusive. The uses within GT flange to flange are mutually exclusive. This is because once a heat recovery system is designed for a GT, effort is made to maximize the heat recovery through that system. Therefore, there is no additional heat recovery feasible for another system. For example, if a recuperator is utilized, it is not feasible to install another heat exchanger to recuperate heat for use in a HAT system. The same logic would also hold true for use within the GT auxiliaries.

However, a further advantage of a combined cycle system with an HRSG and steam turbine is that other heat recovery options may still be possible. This is because the mass flow rate and heat transfer power in such a bottoming cycle are so large-scale, that after the heat utilization for the bottoming cycle, there might still be sufficient lower grade heat available for use in another heat recovery system. A common example is fuel gas heating. HRSG feedwater may be utilized in a separate heat exchanger, in order to heat fuel gas. While there is a minor efficiency loss in the bottoming cycle due to the fact that this water needs to be heated again; the improvement in the overall CC efficiency justifies the application. Fuel gas heating options are further discussed in Section 3.2.

It should be noted that aside from their use in power plants, the exhaust heat from GTs can be utilized in neighbouring industries that can benefit from this source of heat. Jouhara et al. [92] emphasized that while almost half of concentrating solar power (CSP) plants utilize molten salt as a heat source for power generation, the large heat exchangers required to convert this heat are at major expense. The medium to high temperature operating range for these power plants lies between 200 and 1000 °C. Modern gas turbines have an exhaust temperature between 600 and 700 °C [15, 40] and therefore the exhaust heat can be utilized to preheat the molten salt, thereby reducing the size of the heat exchanger required for the operation of CSP plants.

The process-relevant electricity supplied to aluminium smelters are also often provided by integrated GT power plants. Brough and Jouhara [93] provide a thorough review of heat requirements, recovery and utilization methods within the aluminium industry. One major drawback of utilizing waste heat from aluminium processing itself was noted to be the effect of corrosive gasses on heat exchangers. Utilization of GT exhaust heat can overcome some of this drawback. Several areas of heat utilization, ranging from low to high temperature ranges, are noted in their review. One specific example included pre-heating combustion air from 30 to 293 °C for an aluminium furnace, recovering 528 kW of energy. Another example was pre-heating scrap by 100 °C, resulting in 21.1% in energy savings. The heat recovery from the GT exhaust stack can be utilized within the above mentioned applications.

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**Fig. 18.** Hybrid GT-SOFC cycle, as described by Chinda and Brault [86].

**Fig. 19.** GT with a Stirling bottoming cycle, as described in Entezari et al. [90].
5. Critical review of recent developments in extraction of heat from gas turbine exhaust system components

The purpose of this section is to review the recent developments for heat extraction from various sections of a SC GT exhaust system. Based on the conducted literature review, it was found that the majority of the developed research within this technical field placed a higher emphasis on the application of the extracted heat, rather than the mechanisms of how the heat is extracted. This review will particularly focus on those inventions developed in the past decade and involved the heat extraction methods from the GT exhaust system.

Within the context of this section, the heat extraction method developed in each patent was described in terms of construction and functionality with the aid of illustrative diagrams. Also, each invention was evaluated and assessed in light of the established aspect criteria described in Section 4. Finally, the opportunities of using additional embodiments for each patent was discussed for the purpose of further performance improvements.

5.1. Integration of heat exchanger downstream of the exhaust diffuser, for HRSG protection and improved operation at varying loads

In this section, the US Patent Application US2018/0135467A1 [94] is described and evaluated. The main purpose of this patent is to control the GT temperature at variable output loads.

5.1.1. System description

The proposed heat exchangers presented in [94] aimed to control the gas turbine temperature at non-base load conditions. Under these conditions, controlling the GT operating temperature can help in improving the GT efficiency as well as protecting the HRSG from high temperature excursions.

In this patent, one heat exchanger was installed inside the exhaust transition duct of a gas turbine and upstream of the HRSG as shown in Fig. 20. It was configured such that a thermal fluid would cycle through the HRSG and the additional heat exchanger. Another option, a heat exchanger was placed in the GT inlet. In addition, both of these additional heat exchangers were connected together, as well as with the HRSG as shown in Fig. 20.

5.1.2. System’s performance evaluation and ranking against the established aspects criteria

Peaking GT power plants cover times of peak demand, or low supply from other sources; such as renewables. In this market, generation flexibility is a key factor for thermal power station operators. They must be able to start and stop frequently; operate at base or reduced load as per the grid demand.

During the start-up cycle of a gas turbine, the exhaust temperature is typically higher. The reason is that the compressor produces less pressure and airflow at lower speed conditions. To counteract this, during start-up, an increase in the compressor inlet guide vane (IGV) angle allows more airflow. Towards the end of the start cycle, the compressor bleed valves will close, which also aids in increasing airflow, and reducing the exhaust temperatures.

IGVs are also used for load control during part-load operation; where load is reduced by closing IGVs. However, there are some repercussions. When the IGVs are at a lower angle, the compressor efficiency is lower. Also, if the angle is too low, it can cause instability to the compressor air flow and the combustion process [95]. To avoid the aforementioned consequences, the air inlet can be heated to reduce the mass flow rate; enabling the IGVs to remain fully open. The common industry method for inlet air heating, is the inlet bleed heat (IBH) system; which bleeds air from the compressor discharge and cycles it back to the inlet of the compressor.

Another benefit of heating inlet air is to increase the GT turndown ratio; the range of GT load at which it can achieve low emissions. Combustion systems with low emissions provided by the major OEMs, are typically called DLN (Dry Low NOx) or DLE (Dry Low Emissions). These operate at various modes; whereby the mode that achieves the highest NOx reduction is usually at higher loads. For example, if a GT is able to operate in that mode at 50% of base load, the turndown ratio is 2.

Reducing the IGV angle at part-load allows the GT to achieve higher firing temperatures to support the emission-compliant combustion mode. IBH allows operation at lower IGV angles through a larger range of part-load, by heating the inlet air and reducing mass flow; therefore, avoiding compressor stall. This benefit is important for plant operators...
in regions where there is a legal emission compliance obligation [95].

The patent highlighted three main embodiments; shown in Fig. 20. In the option of the heat exchanger placed in the exhaust duct; it cools the exhaust gas in order to protect the HRSG, at part-load conditions. The heat exchanger is used to reduce the flue gas temperature by 50–70°C through extracted water or steam from a section of the steam turbine that is cooler than the exhaust gas. There are numerous publications citing the temperature specification requirements of HRSGs and steam turbines [96–104]. The pertinent information is that gas turbine upgrade; a common occurrence in power plants, tend to also increase the exhaust temperature. If the temperature is above the HRSG specification, costly HRSG upgrades are required. This patent allows the operator to take advantage of the higher GT output and efficiency as part of the upgrade, without having to upgrade the HRSG.

A benefit not mentioned in the patent is emission reduction. Generally, the higher the load and firing temperature of a GT, the higher the emissions. For example, in a 7E GT running on natural gas, the amount of NOx emission at half load operation with a firing temperature of approximately 760°C, is below 100 ppmv. On the other hand, during peak load operation with firing temperature close to 1200°C, the NOx emission increases to approximately 200 ppmv [105].

The same source describes a change in emissions as a result of load reduction by further closing of the IGV angle. For example, as the load is reduced from 100% (base load) to 85% by closing IGV angles, the NOx emission reduces by more than 20%. As this invention allows the GT plant owner to have more flexibility in GT operation at part load; the plant will likely have more opportunity to contribute to the grid. These contributions at part load could be at a reduced IGV angle, if only the exhaust diffuser heat exchanger is used; and therefore, at lower emission levels.

In the embodiment shown in Fig. 20, an optional branch directs some of the HRSG fluid to the heat exchanger placed in the inlet system. This option is stated to be capable of increasing the inlet temperature by 30°C. The cooled fluid is then directed back into the exhaust heat exchanger.

The system flow is managed through control valves. A bypass line allows reduced cooling flow to the exhaust heat exchanger; when less heating required during the GT start-up sequence. A drain line allows emptying the heat transfer fluids when there is no need for heat transfer to take place. It is generally not recommended to keep heat transfer fluids inside heat exchangers and piping systems without circulation; as they may cause corrosion, or vaporize and expand, causing leakage.

From the discussion, it is clear that the benefits not only provide waste heat recovery; but also provide system level operability improvements. Hence it scores an “A” on the effectiveness aspect. The score is “C” for the complexity aspect. Because implementation requires changes to several hardware systems, the control logic, and knowledge of OEM level intellectual property. As the system performance is dependant on the GT operational mode; the control system must be integrated with that of the GT. Hence there is no prospect to reduce the complexity level.

The heat exchangers are depicted, in the patent, to be tubular in nature. The main heat transfer surface consists of parallel vertical tubes in multiple rows; which is similar to what is employed in HRSGs or WHR units. Such structures typically cover the entire flow cross-sectional area and would be placed downstream of the exhaust duct and upstream of the heating chamber itself.

The modification will require major changes to the exhaust diffuser duct; as the added heat exchanger tubes typically necessitate an enlargement of the cross-sectional flow area. But there is no need to replace the entire exhaust system. Therefore, a score of “B” will be provided on the comparative scale of modification aspect. Such a modification can be applied to units operating in combined cycle, as shown in Fig. 20. For this reason, a score of “B” is provided on the aspect regarding potential market opportunity.

The safety and risk aspect must consider leakage from both heat exchangers. An exhaust heat exchanger failure would typically result in leakage of the heat transfer fluid into the exhaust gas path. Considering the size of a GT exhaust system, a flow head at least in the range of tens of meters is required; much higher than the flow pressure in the exhaust ducting of a GTs typically only in the range of inches of water. The impact of the leak would be limited, if it can be isolated. A gradual increase in the exhaust temperature towards the HRSG may be observed; but the unit can continue to operate at modes with lower exhaust temperature.

A leakage from the inlet heat exchanger, can lead to an erosion of the compressor blades in the long term. An extensive leakage may cause more immediate damage. Such a leak would typically be observable as an increase in the vibration of the bearing closest to the compressor inlet. The operator may be able to troubleshoot and switch off the heat exchanger flow; and avoid a forced outage. A repair of such a heat exchanger would require an extension to the planned routine outages that typically range 24 h. Hence, a score of “4” is given on the safety and risk aspect.

5.1.3. Opportunities of using additional embodiments for further performance improvements

The US Patent Application US2018/0135467A1 [94] does not describe heat transfer enhancement methods applied to the tubes. Adding fins improves the degree of compactness, in terms of heat transfer surface area per volume. However, it must be balanced against additional exhaust flow pressure drop. WHR units typically utilize finned tubes. With a velocity ranges from 7 to 35 m per second in various sections of the exhaust, the best-case scenario was reported to be a 795 Pa (3.2 inches of water) pressure drop [106]. Another work [107] showed that this kind of pressure drop would cause approximately 0.34% drop in efficiency as well as output for a SC GT.

The only way to overcome the disadvantages of the pressure drop and scale of modification aspects, would be to use a different type of heat exchanger that would not obstruct the exhaust gas flow path. Such heat exchangers are subject of other inventions that will be discussed in upcoming sections of this work.

There is a chance to obtain a score of “5” on safety and risk. In order to return to service within 24 h, one would need to find the leak in the heat exchanger and repair it during this time. Finding the leak can be done within a few hours, with incorporating few simple additions to the system. Placing manual valves at the inlet and outlet flange connections, as well as a test flange downstream of the manual valve at the inlet, allow for the pressure test circuit. The leak can then be found using leak detector fluid, or using a hand-held ultrasonic device. To allow a quick repair, there are two options. The heat exchanger can be specified to the manufacturer to either be weld-repairable, using a weld repair kit, or to specify a modularized system.

5.2. Gas turbine performance heating system, utilizing exhaust plenum heat recovery

Another method of heat extraction is from the exhaust plenum, between the exhaust diffuser and the exhaust stack; as described in US patent application US2007/0271928A1 [108].

5.2.1. System description

Within this invention, the heat recovery device was placed in the corner of the exhaust plenum; for the purpose of fuel heating. Fuel gas tubing enters the triangular heating chamber, as shown in Fig. 21. The tubing has a serpentine shape, with optional fins, to enhance heat transfer. Fig. 21A shows the placement of the heating chamber, within the exhaust system and Fig. 21B provides a more detailed isometric view of the heating chamber itself.

The flow rate of exhaust gas into the heating chamber, was controlled by external actuators, which move the relative position of two
overlapping perforated plates. Full closure was achieved by a position where all the perforations were non-aligned. In one embodiment of the invention, exhaust gasses at 593°C were able to heat the fuel gas up to 288°C; henceforth referred to be the first embodiment.

In another embodiment, henceforth “Alternative 1”, the tubes were located at a higher position, with the area was separated by a baffle running parallel to the vertical axis of the exhaust stack. Adjustable dampers were placed downstream of the silencers; to control exhaust flow across the tubes.

In another configuration, henceforth “Alternative 2”, the heating area was located in a separate parallel duct relative to the existing exhaust stack. Exhaust gasses would re-join the existing stack at a higher position; after passing through flow control dampers. An additional row of silencers may be needed in order to account for the re-entry flow.

5.2.2. System’s performance evaluation and ranking against the established aspects criteria

There are two main categories of fuel gas heating for a GT; perforation heating and dew point heating. A performance heater extracts heat from an available source, and therefore utilizes waste energy. It is well known that heating fuel gas can increase the GT efficiency; as the quantity of fuel required to achieve the firing temperature is reduced. In a typical case where the hot side flow is sourced from the HRSG feedwater, from the intermediate pressure economizer [109], the gas is heated to 185°C (365°F). As combustion temperatures increased over the years, the heating range increased to 315–350°C (600–650°F); achieved by sourcing high pressure feedwater [110,111]. The purpose of dew point heaters, is to eliminate condensates from the gas stream that are harmful for the combustion process. In the example of the electric heater, additional energy is used from the grid towards the electric heating coils. As this system utilizes waste heat, it fits the definition of a performance heater. Therefore, a score of “B” is allocated to the effectiveness aspect.

Each system embodiment requires a varying scale of modification; evaluated as part of the fourth aspect of the discussion. Even the first embodiment, although the simplest, involves major modifications; which means a score of “B” can be given to this aspect. These include:

- Creating openings for piping flange connections and sealing for the sliding plate.
- Adding structural support.
- In order to avoid insulation damage during welding, exhaust insulation needs to be removed and reinstated later.

The main benefits of this invention are heat recovery, as well as lower cost relative to a traditional performance heater. In the first two configurations, there is no intermediate heat transfer fluid between the exhaust flow and the fuel. In a traditional performance heater, additional piping routes boiler feedwater to a separate gas heater. Separate routing requires pipe runs, foundations and heavy insulation around the pipe runs to reduce heat loss.

A potential risk of fire inside the exhaust stack must be considered. In the presence of exhaust temperatures, a fuel gas tube failure carries the risk of combustion. In order to determine the risk, one must compare the composition of the flue gas to the limiting oxygen concentration (LOC).

In gas fuels used for GTs, Methane makes up the most substantial portion of the fuel mixture. The NFPA [112] list the LOC for Methane in the range of 11.1–13.1% in volume. Various publications [113,114] place the percentage of Oxygen in GT exhaust up to 15 percent; pointing to a realistic risk of fire, in case of leakage.

The risk is amplified in certain modes of operation where the fuel gas tubes are still hot, and there is an increased level of oxygen in the exhaust gas stream. Such a mode is possible after GT shutdown; while the unit is still on turning gear. In this mode, there is no fuel fed to the GT combustor; and hence the flow through to the exhaust is ambient air. Such an incident occurred in a power plant in Malaysia [115]. In this plant, gas fuel piping was installed in the exhaust diffuser section of the GT. An unnoticed leak had developed in the piping. During GT shutdown, after flame-out as the unit was slowing down, gradually the percentage of Oxygen in the flow-stream increased to 21% compared to 14% during fired operation. The hot components in the exhaust system, at 570°C, cause methane ignition. The result was a catastrophic failure of the GT exhaust system, which would typically cause months of unplanned outage as well as millions of Dollars of expense. Given the fire risk, a score of “1” is allocated to the safety and risk aspect of the discussion.

Although the patent application took place in 2006, there was no implementation of this invention found in literature, or estimation of the additional pressure drop it creates in the exhaust stack. For an E-class GT in simple cycle mode, just 4 inches of Water in additional exhaust pressure drop can cause the efficiency and output to reduce by 0.42% [107]. Additional pressure drop is certain, as an additional obstruction is placed in the exhaust path. The exhaust gas needs to pass through the orifices, over the tubes and then exit through the orifices again before entering the exhaust air stream. As there is a flow pressure loss through these stages, it will have a lower pressure than the parent exhaust stream. Hence it will require back-pressure build-up at the entrance of the orifices to re-join the parent exhaust stream.

The configurations in alternatives 1 and 2 described above have even higher pressure drop, as there are more obstructions placed in the
exhaust gas path. These are in the form of flow dampers or additional silencer baffles. All embodiments add a partial obstruction of the flow cross-sectional area; or separate a portion of the flow. Therefore, a score of “B” can be allocated to the third aspect, as related to the level of additional pressure drop.

The configurations in these two alternatives are highly complex to implement; as they require significant construction changes to the exhaust stack. It would also require a complete re-construction of the stack, in case the plant owner decides to implement an HRSG for a combined cycle upgrade. The first embodiment, shown in Fig. 21 is the most attractive in terms of the magnitude of modification, cost of implementation and pressure drop in the exhaust stack. The complexity of implementation aspect scores “C”; and requires the following modifications:

- Routing the fuel gas piping into the exhaust stack.
- Additional sealings mechanisms to protect personnel against exhaust gas leaks from the interface between the moving plates and the stack.
- A mechanism between the two plates themselves, to ensure an effective seal in the closed position. An exhaust gas leakage would cause overheat the fuel tubes, leading to deformation and damage.
- Information exchange is required between the gas fuel operating system and the heat exchanger, through the GT control system. Therefore, the modification spans more than one system.

Such a modification can be implemented on any SC GT; as all of them have an exhaust plenum. However, the use would only be limited to gas turbines and fuels that do not run near the LOC risk point. Hence, a score of “C” is provided on the potential market opportunity aspect.

5.2.3. Opportunities of using additional embodiments for further performance improvements

The pressure drop aspect score could be improved by redesigning the heat transfer tubes to a more aerodynamic shape, and eliminating the sliding plates. Another means of controlling the fuel gas temperature, such as a bypass control valve, may be used. A simplified process diagram is shown in Fig. 22. The function of the bypass control valve is to mix cold fuel gas supply with heated fuel gas out of the heat exchanger, in order to control the overall fuel gas temperature. A temperature transmitter would feed the outlet temperature measurement back into a control system, for a decision on valve positioning.

In order to reduce the risk of fire; additional hardware and controls modifications are needed. One option would be to purge the gas from the piping using an inert gas, in the event of a shutdown or trip. Additionally, to install a double block and bleed mechanism to avoid gas fuel leakage into the purged area. Such modifications carry significant expense; as it requires the installation of additional valves, sensors, piping connections, nitrogen tanks, their related accessories and additional control system hardware. Even after the addition of such a protective system, heat exchanger operation would only be limited to gas turbines and fuels that do not run near the LOC risk point. This would severely limit the implementation of this invention.

To overcome this limitation and risk, it would be required to use a non-flammable intermediate heat transfer fluid. In such a case, if either heat exchanger suffers a failure, the heat transfer fluids can be stopped and the unit can continue operation. This is possible if the GT combustion system can operate with both cold and heated fuels; and would improve the safety and risk score to “5”. If the combustion system can only operate with heated fuel, the score would still improve to “3”, to address the damage and return to operation. The market opportunity score also improves to “A”.

5.3. Integration of HRSG components into various parts of the GT exhaust diffuser

This section aims to describe and evaluate the applications of integrated heat exchanger passages within existing sections of a GT exhaust system presented in the US patent 8146,341B2 [116].

5.3.1. System description

The main concept of this invention was to perform some of the heat transfer processes that typically takes place inside an HRSG further upstream; in order to reduce the overall size of the combined cycle. The detailed schematic of the disclosed design is shown in Fig. 23. Heat is
transferred from the GT exhaust air to water in the heat transfer passages, in order to generate steam. Heat exchange passages were placed in the turning vanes, exhaust frame struts, exit guide vanes and associated support structures. The main system components are outlined in Table 4. The aerodynamic shape of the exist guide vanes minimizes the back-pressure to the turbine, besides acting as a support structure for the diffuser outer casing.

Fig. 23 shows an embodiment in which heat transfer passages are integrated into the exhaust frame struts. The water or steam enters through the inlet pipe and extracts heat from the exhaust gas while flowing in the piping structure.

Fig. 23 also shows other locations where the heat transfer piping may be integrated into the system. In one embodiment, they are integrated in the inlet turning vanes. There are a higher number of these vanes compared to the struts; however, they are thinner. Hence, the heat transfer piping also needs to be smaller. The same arrangement is also shown for the exit guide vanes. The size of these vanes is larger than the inlet guide vanes. Also, in connection to the exist guide vanes, there is a support structure shown as item 10. The heat exchange piping is also shown to be integrated into this support structure, for additional heat recovery.

5.3.2. System’s performance evaluation and ranking against the established aspects criteria

It is apparent that the main advantage of the presented heat recovery system in this patent [116] is adding heat transfer functionality to the existing exhaust diffuser. Since the components planned with integrated heat transfer passages already exist in the exhaust diffuser, there is no additional flow area occupied. This implies there is no additional pressure drop penalty; as the outer surfaces of the structures will not be modified. Therefore, a score of “A” can be given to the pressure drop aspect. Also, since the aim is to use the heat for a purpose outside of the GT, a score of “C” is allocated to the effectiveness aspect.

There are some factors that make it difficult to implement the integrated heat exchanger in the field; or add risk to the existing operation. Firstly, it is not practical to upgrade an existing exhaust diffuser to include the modifications of this patent. It would require a complete disassembly of the exhaust diffuser. It would be more practical to replace the entire exhaust diffuser. However, this would require substantial capital expenditure in the range of millions of US Dollars. Therefore, a score of “C” is allocated to the aspect regarding comparative scale of modification.

Regarding the safety and risks aspect, the potential risks of implementing the proposed patent’s design can be viewed from the following perspectives:

a) The inlet and outlet piping connected to the exhaust diffuser would have potential leak points for water or steam onto the diffuser casing. In the case of a smaller GT, the diffuser is closer to the turbine casing. If the leakage is significant and in the form of a spray, it can lead to uneven cooling of the turbine casing; which in turn leads to loss of turbine vane clearances and rubs. This would result in a multiple-week forced outage for repair.

b) The exhaust diffuser inlet turning vanes are thin; requiring tubing to be used for the heat transfer channels, instead of piping. Therefore, pipe flanges cannot be utilized. Instead, mechanical tube connections, such as compression fittings, would be the choice. Experience has shown that, due to higher complexity, they are more prone to leaks. A water leak in this area carries a higher risk due to the closer

Table 4
Reference table describing the components shown in Fig. 23 [116].

<table>
<thead>
<tr>
<th>Item number in the patent embodiment figures</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>GT</td>
</tr>
<tr>
<td>2</td>
<td>Exhaust diffuser</td>
</tr>
<tr>
<td>3</td>
<td>Exit guide vanes</td>
</tr>
<tr>
<td>4</td>
<td>Exhaust frame struts</td>
</tr>
<tr>
<td>5</td>
<td>Diffuser outer casing</td>
</tr>
<tr>
<td>6</td>
<td>Inlet pipe</td>
</tr>
<tr>
<td>7</td>
<td>Outlet pipe</td>
</tr>
<tr>
<td>8</td>
<td>Piping structure</td>
</tr>
<tr>
<td>9</td>
<td>Inlet turning vanes</td>
</tr>
<tr>
<td>10</td>
<td>Support structure</td>
</tr>
</tbody>
</table>
proximity to the turbine casing. Water leakage in the exit guide vane and support structure area can flow to the exhaust plenum; where the inner lining is typically made of overlapping metal plates. A leak past the overlap area and into the mineral insulation underneath the lining, can lead to swelling. That introduces a risk of damage to the exhaust plenum liners and fasteners that hold the liners together. Insulation replacement requires weeks of unplanned outage. If the leak is from exhaust gas into the water, there will be a different issue. The exhaust gas stream carries mainly Carbon Dioxide, Carbon Monoxide, but also Oxides of Sulphur and Nitrogen. These contaminants can cause particle deposits in the water as well as formation of acids; which in turn cause blockages or material damage to the finned tubes carrying the water and steam inside the HRSG; causing further repair expenses. The steam piping exiting the exhaust diffuser towards the HRSG needs to be heavily insulated to avoid heat loss. The farther away the HRSG is, the more material is consumed and added to the capital expenditure.

Another potential risk point exists in the new internal shape of the aerodynamic components used as heat transfer passages. The vanes and struts will be further hollowed out. The thinner material layers are more susceptible to corrosion related damages that initiate cracks. It would take a shorter time for the crack to propagate through the material until fracture. There is also a higher risk of overheating during the periods where the heat recovery system is not in operation and the passages are empty. This risk will be particularly pronounced in case of abnormal operating conditions that cause high exhaust temperature. Since high exhaust temperatures are common, there are temperature sensors and control systems in place to initiate a GT trip. Due to the added risk in this configuration, the trip set-points may need to be more conservative and therefore lead to a higher number of trips. Uniform temperature is critical for those struts that are structural components; as differential temperature leads to exhaust diffuser casing distortion. It follows that equal water distribution to all the struts is of high importance. There are two main issues that can prevent equal water distribution. The first being that various struts will be at different heights according to their location in the circular cross-section. The second being water contamination, which can cause partial or full blockage of some of the passages. Based on the outlined potential risks, it is fair to allocate a score of “2” to the safety and risk aspect of the discussion.

Based on the above discussion, there is a wide potential market for such a configuration. As all GTs have some form of aerodynamic and structural components on the exhaust casing and diffuser, these surface areas are available to be modified for enabling additional heat transfer capability. Therefore, score “A” is allocated to the potential market opportunity aspect.

5.3.3. Opportunities of using additional embodiments for further performance improvements

The proposed design is beneficial as it reduces the HRSG size and therefore the overall CC size; and would result in lowering the construction space costs, especially for plants that are currently in SC mode, and planning to expand to CC but have space limitations.

If the combined cycle system has a diverting damper that allows the GT to run in either SC or CC mode, the heat exchanger developed in this invention could be used for other purposes during SC operation. Many power plants are operated in peaking mode. In times of lower grid demand, the GT may be forced to run in SC mode. In this scenario, there is no further need for steam generation. Instead, the heat exchanger may be used to allow the SC to run at higher efficiency. These functionalities would increase the score of the effectiveness aspect to “B”. For achieving a score of “A”, an option would be to utilize the heated water for inlet bleed heating, in a similar manner described in Section 5.1 where additional inlet system heat exchanger was used. As the water would be diverted away from the HRSG only during lower demand, it may also mean that the unit may have to run at part-load during SC mode. Hence, such an added functionality would allow the plant to gain more operational flexibility. Such uses of the heat exchanger require further investment in terms of hardware and control logic, in order to allow dual usage. An outline of a comprehensive system that allows these functionalities will be further discussed in Section 5.10.

In order to reduce the risks a water leak, detector would need to be installed, which relatively increase the cost. However, the cost of leak detection system would be low compared to the potential cost of repair; and it would allow for the repair to be conducted during a planned shutdown. Hence, the safety and risk score would improve to “4”. Also, the leakage can be resolved by integrating a geared flow distributor that ensures equal flow in all the channels. The issue of blockages or material damage to the finned tubes carrying the water and steam inside the HRSG can be mitigated by water quality control and filtration. This would add to the capital and operating cost.

5.4. Installation of heat recovery module in a gas turbine engine exhaust duct

US Patent 8517,084B2 [72] describes a waste heat recovery unit, installed in a gas turbine exhaust duct. The heat exchanger takes the form of parallel circular tubes, shown in Fig. 24. The company suggested the use of finned tubes [117], which is a good choice to increase the heat transfer surface area.

5.4.1. System description

The system components shown in Fig. 24 are listed in Table 5. The exhaust gasses pass over the heat exchanger tubing; with a diverter damper enabling flow control between the heat exchanger and bypass areas. The size of channel openings is controlled by the overlap between the stationary and rotating parts. A higher overlap leads to more exhaust gas being directed to the heat exchanger. A linkage allows the shaft to be rotated, in order to control the valve openings, through movement of the arm Table 6.

5.4.2. System’s performance evaluation and ranking against the established aspects criteria

The invention is designed to be a lightweight construct that can be directly bolted on to an existing exhaust duct. It would not require a shift in the entire exhaust plenum and stack, as in a standard WHR unit. Therefore, a score of “A” is allocated to the scale of modification aspect.

In order to benefit from the heat recovery in this system, then at least a partial diversion of the flow to the heat exchanger is necessary; considering the description of the flow channels in Section 5.4.1. This provides a score of “B” towards the pressure drop aspect.

Considering that larger heavy-duty gas turbines tend to have a conical shaped exhaust diffuser or duct, for control over exhaust gas expansion [71], and noting that this design needs to be connected to a cylindrical exhaust duct, it would mainly be suited to smaller aero-derivative units; resulting in a score of “C” for the market opportunity aspect.

Over time, contamination will build up over the contacting surfaces of the overlapping components, especially in case of using high ash content fuels; leading to increased friction in this area. Metal components in the exhaust system may deform over time due to conditions such as high exhaust temperatures. To prevent this from affecting the overlapping movement, some clearance needs to be factored in. This leads that some of the exhaust gasses meant for the heat exchanger, will leak into the bypass ducting.

On the other hand, the patent claimed that the seal is good enough to allow full bypass while conducting maintenance on the heat exchanger. Safety considerations would require the leak towards the heat exchanger to be close to zero. This means there is no clearance to reduce the risk to the overlapping movement. However, it does not cause forced outage.
Repairs to the damper may be conducted during a planned outage. The repairs would require more than 24 h, including cooldown time to enable access. This means a score of “4′′ can be allocated to the safety and risk aspect.

As there is no description of the use of the heat that is gained within the GT, it is likely used for an outside application; leading to a score of “C” to the effectiveness aspect. It also indicates the modification to an existing GT would be of a hardware-only nature; resulting in a score of “A” for the complexity aspect.

5.4.3. Opportunities of using additional embodiments for further performance improvements

Since the patent does not describe any lubrication system, the shaft bearings are likely double-shielded or sealed bearings with permanent grease lubrication [118]. These would typically require frequent replacement, at the range of temperatures seen in the GT exhaust. The end of life can be extended by selecting a long-life grease, managing the operating temperature and cleanliness [119]. While the patent does not mention a driving force for the arm movement, it is likely to be a hydraulic actuation system. Installing this system outside the hot area will extend its life. In both cases, extending life leads to avoidance of unplanned downtime.

5.5. Compressor clearance control system using GT exhaust heat transfer

The purpose of the configuration proposed by US Patent 8172,521B2 [120] is to heat the GT compressor casing using heat extracted from the turbine. The benefit is to control over the clearance between the compressor rotor blades and the compressor casing.

5.5.1. System description

In this patent, the compressor inlet guide vanes (IGVs) narrow during GT low load operation to reduce the air flow. The increase of pressure drop across the IGVs leads to lower temperature entering the compressor. The casing of the compressor is more massive than the rotor blades and have a slower thermal response. The rotor blades contract and expand quicker than the casing. When the GT load reduces, the

Table 5
Reference table describing the component shown in Fig. 24 [72].

<table>
<thead>
<tr>
<th>Item number in the patent embodiment figures</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cylindrical channel</td>
</tr>
<tr>
<td>2</td>
<td>Parallel circular tubes - heat exchanger housing</td>
</tr>
<tr>
<td>3</td>
<td>Cylindrical channel for heat exchanger housing</td>
</tr>
<tr>
<td>4</td>
<td>Exhaust gas flow area</td>
</tr>
<tr>
<td>5</td>
<td>Diverter damper</td>
</tr>
<tr>
<td>6</td>
<td>Bypass duct</td>
</tr>
<tr>
<td>7</td>
<td>Stationary part of diverter damper</td>
</tr>
<tr>
<td>8</td>
<td>Rotating part of diverter damper</td>
</tr>
<tr>
<td>9 and 10</td>
<td>Opening channels</td>
</tr>
<tr>
<td>11</td>
<td>Shaft bearing</td>
</tr>
<tr>
<td>12</td>
<td>Structural component</td>
</tr>
<tr>
<td>13</td>
<td>Linkage</td>
</tr>
<tr>
<td>14</td>
<td>Shaft</td>
</tr>
<tr>
<td>15</td>
<td>Arm</td>
</tr>
<tr>
<td>16</td>
<td>Ribs</td>
</tr>
<tr>
<td>17</td>
<td>Seal</td>
</tr>
</tbody>
</table>

Table 6
Reference table describing the components shown in Fig. 27 [129].

<table>
<thead>
<tr>
<th>Item number in the patent embodiment figures</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Inner surface</td>
</tr>
<tr>
<td>2</td>
<td>Outer surface</td>
</tr>
<tr>
<td>3</td>
<td>Inlet piping</td>
</tr>
<tr>
<td>4</td>
<td>Outlet piping</td>
</tr>
<tr>
<td>5</td>
<td>Dimples</td>
</tr>
<tr>
<td>6</td>
<td>Exhaust stack inner diameter</td>
</tr>
<tr>
<td>7</td>
<td>Exhaust stack outer diameter</td>
</tr>
<tr>
<td>8</td>
<td>Heat exchanger inner diameter</td>
</tr>
<tr>
<td>9</td>
<td>Heat exchanger outer diameter</td>
</tr>
<tr>
<td>10</td>
<td>Circumference of heat exchanger half section</td>
</tr>
<tr>
<td>11</td>
<td>Flow pattern around dimples</td>
</tr>
</tbody>
</table>
clearances will increase and vice-versa. Increased clearances cause loss of performance and compressor stall; while reduced clearance can cause a rotor blade rub against the casing.

Fig. 25 depicts the invention schematically. Some of turbine air is extracted and routed to a heat exchanger integrated into the compressor casing. The GT control system selects the control valve position and thereby controlling the level of heating.

5.5.2. System’s performance evaluation and ranking against the established aspects criteria

Since turbine air is extracted through a port rather than using an additional heat transfer cross-section, a score of “A” can be given on the differential pressure aspect of the ranking criteria. This modification can be applied to any turbine and therefore score of “A” is given on the market opportunity aspect.

The modification requires OEM level knowledge of the GT design. Dimensional drawings, and clearances during various modes of operation must be known as well as the capability to make multi-system changes to the GT control logic. Hence a score of “C” is given on the complexity aspect. The modification to the exhaust system only requires the addition of an extraction port only giving a score of “A” on the scale of modification aspect.

During part load operation, the updated system can enable the IGVs to close further leads more turndown. Alternatively, it can be used instead of or in collaboration with an inlet bleed heat system during cold weather operation. Greater control over the clearances, without an external energy input, enables improved GT efficiency. Since the patent claimed the heat used is waste heat, then it is expected that the extraction is from after the last stage of the turbine which is not utilized for GT rotation. Hence, a score of “A” can be given to the effectiveness aspect.

In case of a heat exchanger leak, the direction depends on whether the pressure is higher at the turbine extraction, or at the location of the leak on the compressor casing. An exhaust gas leak into the compressor might not be detectable. However, over time the compressor contamination would increase, resulting in performance loss. It would manifest itself through the requirement of more frequent offline water washes, which require unplanned downtime. Contaminants, such as acids formed from sulphur reacting with moisture, can cause compressor corrosion. On the other hand, if the cold side pressure is higher, the compressor air would be lost. This leads to a performance loss that is not recoverable even by offline water washing. These risks do not lead to a forced outage but reduce intervals between planned outages. Hence it is akin to an extension of the planned downtime resulting in a score of “4” is given on the safety and risk aspect.

5.5.3. Opportunities of using additional embodiments for further performance improvements

Polytetrafluoroethylene (PTFE), a common valve seat material is only suitable for temperatures up to 260°C [121]. Therefore, the control valve needs to be of a metal-to-metal contact type. Therefore, the seat leakage classification cannot be better than Class IV. These permit a leakage up to 0.01% of the rated valve capacity [122]. This represents energy loss when the system is not in use which needs to be factored in to the overall performance gains. An additional shut-off valve can reduce this energy loss. Data analytics, based on measurements of the GT compressor discharge air temperature, could help to detect an exhaust air leakage into the compressor. This can improve the safety and risk aspect.

The casing heat exchanger maintenance also needs to be considered. GT compressor air cleanliness depends on the inlet system filtration. The requirement for increased GT compressor cleanliness is driven by several factors, such as reducing compressor degradation and increasing performance as well as prevention of blocked turbine blade cooling passages, that are cooled by compressor air. Nowadays, it is common to have filters in the range between F9 and HEPA [59]. The ratings are according to the terminology in European standard EN779:2002 [123], which is traditionally used in industry. Note that as of 2018, filters equivalent to F9 are covered under a newer ISO 16,890 [124]; while HEPA filters are covered under European standard EN 1822 [125]. On the other hand, turbine exhaust gasses contain contaminants as a result of the combustion process. The smaller the casing heat transfer passages, the higher the risk of fouling. To mitigate this risk, the air must be filtered. The challenge with filtration would be the high temperature. The extraction temperature leaves metal [126] or ceramics [127] as the only options for filter media. To avoid an unplanned shutdown, the filter housing needs to be installed in an accessible area outside the GT compartment. Further, there would have to be a parallel filtration system, allow filter replacement without operation interruption. This would increase the system cost but allow the retention of the score of “4” for the safety and risk aspect.

5.6. Fuel heating using exhaust gas extraction from the GT exhaust stack

The US patent 8015,793B2 [128] described a method of heat extraction from a simple cycle GT exhaust stack, without the use of an intermediary heat transfer fluid.

5.6.1. System description

Fig. 26 shows extraction through a duct, from which 4–7% of the exhaust gas is directed to a fuel gas heat exchanger, to later be ducted back into the exhaust stack. The purpose of gas heating is to improve the overall efficiency of the GT thermodynamic cycle. Isolation dampers and valves, at the suction and discharge connections enable isolation, during periods of non-operation; combine with an air purge system. A gas fuel leak sensor is needed in the exhaust stream; due to higher pressure of the fuel gas compared to the exhaust gas. The amount of unburned fuel in the exhaust gas stream is approximated to be in the range of 0.01 to 1 percent. Leak sensors enable the controller to raise an alarm above this threshold indicating a leak of fuel gasses inside the heat exchanger.

5.6.2. System’s performance evaluation and ranking against the established aspects criteria

As the system effectively replaces a separate fuel gas performance heater, the modification is limited to hardware elements such as exhaust
stacks that have common designs. A stand-alone control system eliminates the need for GT control system modification. Hence a score of “A” can be given on the complexity aspect. A score of “B” is given on the effectiveness aspect in line with the discussion in Section 5.2.

In case of system failure, such as heat exchanger leaks, the system can be switched off and purged. The only effect would be the reduced efficiency of the GT, assuming that a separate dew point heater exists in the plant. Hence a score of “5′′ can be applied to the safety and risk aspect. As there is no intermediate heat transfer fluid, another possible risk of fire applies as described earlier in Section 5.2. However, it is inferred that the isolation dampers, valves, blowers and leak sensors are included to mitigate this safety risk.

Suction blowers shown in Fig. 26 are used to drive the flow from the extraction duct and back into the exhaust stack. The blowers are driven by Variable Frequency Drive (VFD) motors that enable control over the volumetric flow rate of the exhaust gasses through the heat exchanger. The parallel blower configuration is typically designed for redundancy. However, it also allows both blowers to be put into operation, should there be a requirement due to added pressure drop, induced by heat exchanger fouling.

In this system, it is expected to have a localized exhaust gas pressure drop, at the location where the duct connects back into the exhaust stack. The reason that the pressure in the duct is higher than the stack, with a safety margin, is to allow positive flow. The flow passages in the heat exchanger will foul considerably over time, due to the contamination in the exhaust gas. This design allows a score of “B” for the pressure drop aspect.

The fuel heating system can be built as a retrofit on existing simple cycle gas turbines, without a requirement to shift the exhaust stack. There will be a requirement for additional foundation to support the heating system and ducting. The heat exchanger cannot have very small passages on the exhaust gas side, as a fouling factor needs to be considered. Therefore, it should not be of a compact heat exchanger type. A conventional shell & tube heat exchanger is recommended to be used with the exhaust gasses on the shell side for easier cleaning. The tubes will need to be of a costly high-grade stainless steel to protect against corrosion from the humidity, nitrogen and sulphur products in the exhaust gas stream. The modification portion strictly on the exhaust duct is minimal; and a score of “A” is given on the scale of modification aspect.

The main disadvantage is the expected incurred cost which would be in the range of millions of US Dollars due to the large size of ducting, heat exchanger and numerous hardware. For example, the leak detection system needs to be accurate enough to detect fuel gasses (primarily methane) in a low percentage range. Typical detectors used in the industry include catalytic bead-based chemical sensors or infra-red based optical sensors. The former model requires frequent calibration, every several months. This would require an unplanned downtime as the fuel gas heating system is critical to operation. More expensive infra-red sensors can typically be calibrated without shutdown; and hence will not cause unplanned downtime.

The system can be implemented on any GT as they all have exhaust stacks. However, operation of the heat exchanger would only be limited to gas turbines and fuels that do not run near the LOC risk point. This would limit the implementation of this invention. In the current embodiment, a score of “C” is applicable on the marketability aspect.

5.6.3. Opportunities of using additional embodiments for further performance improvements

A potential opportunity for improvement of the configuration would be to extract the exhaust gasses at a higher pressure location, further upstream such as the exhaust diffuser. The exhaust gasses are not doing any additional work for the GT at this point. Furthermore, the suction blowers create a vacuum pressure at the extraction point which effectively lower the back-pressure on the turbine and therefore improve the GT cycle efficiency. An intermediary heat transfer fluid can be used to improve the score of the marketability aspect to “A”.

5.7. Heat extraction from the GT exhaust stack, through a circular duct carrying a heat transfer fluid

The invention under US Patent 7874,156B2 [129] proposed a circular metallic duct containing a heat transfer fluid as heat extraction method from the exhaust stack.
5.7.1. System description

Reference to Fig. 27A, the purpose of the duct is to encompass a GT exhaust stack and transfer the exhaust heat to the heat transfer fluid. The inlet and outlet piping are shown in Fig. 27C. There are several installation options provided. The first option to install the circular duct on the outer side of the exhaust stack, as shown in Fig. 27A. The second option would be on the inner circumference. Another option would be inside the exhaust stack, with a gap between the heat exchanger and exhaust ducts. Finally, to replace a section of the exhaust stack with the circular duct, as shown in Fig. 27B.

5.7.2. System’s performance evaluation and ranking against the established aspects criteria

There is no heating application referred to in the patent, within the GT. Hence, it is likely to be for an external application. This enables a score of “C” on the effectiveness aspect.

Installing on the outer circumference of the exhaust stack, minimizes the required downtime. If site safety procedures allow, and if there is no modification required to the stack itself, the heat exchanger installation may also take place during GT operation. Additional support structures may be required.

Fig. 27. Circular duct assembly as described in [129]. A: the embodiment of the circular duct installed on the outer surface of the exhaust stack. B: the embodiment of the circular duct installed as part of the exhaust stack. C: the circular duct heat exchanger.
For the embodiment per Fig. 27A of the patent, it is explained that the inner surface of the heat transfer duct is connected to the outer surface of the stack through a heat transfer mastic. Such a description typically refers to a chemical substance with a high conductive heat transfer coefficient. One example is a thermal conducting cement. The type required for the temperatures of an exhaust stack are hardening-type cements which have a range of temperature up to 677 °C [130]. One model has a thermal conductivity of 170 W/m²°C [131]. This is even higher than carbon steel, which has a conductivity of around 32 W/m²°C at such elevated temperatures [132].

The associated benefit is to reduce the installation time of the heat transfer circular duct. In this scenario, the duct can be fastened around the exhaust stack using mechanical fasteners. The cement adheres to each surface and remains in place, filling any air gaps between the exhaust stack and the circular duct and improving thermal conductivity. It also avoids the requirement to make modifications to the structure and insulation. On the other hand, the thermal cement adds another layer of thermal resistance, which increases the overall thermal coefficient.

It is worthy to note that heat transfer is limited with this option. The inner radius of an exhaust stack is made of layers of steel sheeting and thermal insulation. To remove this limitation, the outer steel layer and insulation needs to be removed and to install the heat exchanger in contact with the inner wall.

In the embodiment in Fig. 27B, the inner surface of the heat exchanger is already flush with the rest of the exhaust stack inner surface. As additional pressure drop in the stack is avoided, a score of “A” can be given for the pressure drop aspect. However, for both safety and sound-attenuation reasons, a layer of insulation and outer metal lining must be installed on the outer perimeter, in both cases. As significant modifications to the exhaust stack are necessary; a score of “B” is obtained for the scale of modifications aspect. The complexity is limited to hardware changes to the exhaust stack. Hence a score of “A” can be applied to the complexity aspect. The marketability aspect score is “B”; because all SC gas turbines have an exhaust stack.

A leak may result in the heat transfer fluid leaking into the exhaust stack flow path, or into the insulation. If the heat transfer fluid is water, there is no significant impact. If the leak is into the insulation, it can damage it by causing it to expand and eventually break down. It would extend a planned outage while not causing a forced outage. There will be at least several days required to erect scaffolding at the exhaust stack and perform repairs. A score of “4” is therefore applied to the safety and risk aspect.

5.7.3. Opportunities of using additional embodiments for further performance improvements

The modification can only be applied to circular exhaust stacks, due to its shape. The marketability aspect score can improve to “A” if a rectangular option is also provided. This should not be a complicated endeavor as the connections as well as the heat transfer passages can remain the same.

Regular external inspections of the exhaust stack, using an infra-red camera, can be added to the site maintenance plan. This will help to detect early damages to the structure and leaks, and to plan corrective action ahead of time. Fig. 27B shows an example where one circular heat exchanger duct is installed in the stack. There is room for more of these ducts to be installed in series which will provide a level of redundancy until a longer planned outage is reached. The maintenance plan, combined with redundancy, would enable a score of “5” for the safety and risk aspect.

The aim of the dimples in Fig. 27 are to induce turbulent flow leading to increase the heat transfer surface area. However, it must be noted that the outer surface of the circular duct is also made of metal and exposed to the ambient. There would be a significant amount of heat loss to the environment unless the outer surface is insulated. Mineral wool is an option, as it has a thermal conductivity in the range of 0.16 W/m²°C, at elevated temperatures of around 500 °C, which would be expected for this type of application [133]. The insulation and fastening will add to the cost of the heat exchanger installation. Aside from the application of dimples, another option to improve the overall heat transfer coefficient would be to utilize a nanofluid, instead of water. Awaits et al. [134] provided an extensive review of nanofluid classifications, applications and improvement of heat transfer coefficients. One of the reviewed articles [135] analyzed the utilization of an Al₂O₃-water based nanofluid, as the coolant in a supercritical water reactor where the working temperature is approximately 600 °C which is at the range of the GT exhaust temperature.

5.8. Heat extraction from the GT exhaust diffuser using a circular tubing network

This design published under US Patent 9845,689B2 [136] aims to control pressure drop inside the exhaust diffuser with a capability for exhaust heat extraction.

5.8.1. System description

In this invention, a circular structure is installed just downstream of the exhaust frame strut. As shown in Fig. 28, it comprises a series of tubes to serve three purposes. The first is to direct more flow towards the outer circumference of the structure, which is the inner circumference of the exhaust diffuser. This reduces flow separation of the exhaust gasses on the inner walls of the exhaust diffuser and gain a better control over the pressure drop along the flow direction. The second purpose is to provide additional structural stiffening to the exhaust diffuser. The third purpose is of primary interest to this review paper as the tubular structure allows flow of a heat transfer fluid, to extract heat from the exhaust flow.

5.8.2. System’s performance evaluation and ranking against the established aspects criteria

There is no mention of an improvement in GT performance. Hence a score of “C” is applied to the effectiveness aspect. The tubular heat exchanger can be installed inside the diffuser during a planned outage, reducing downtime. The only work required would be to drill the heat transfer fluid connections into the diffuser; and to fasten the heat exchanger on the inner circumference. A score of “A” is therefore applied to both the scale of modification and complexity aspects.

As the diffuser is typically installed in a compartment insulated for personnel protection, an external leak would not typically lead to a
safety risk. However, if a mechanical fastener is damaged, it can cause vibration that is higher than the allowed limit on the bearing closest to the exhaust. This would require a forced outage. The outage would last at least several days, in order access the area, find the damaged component and make repairs. The quickest repair may entail removing the heat exchanger and isolating the connections. Hence, a score of “3” is applied to the safety and risk aspect.

The amount of heat extraction from the single row of tubes has not been quantified. There may need to be multiple rows to extract a meaningful quantity. As previously mentioned, for an E-class GT in SC mode, only 4 inches of water in additional exhaust pressure drop can cause the efficiency to reduce by 0.42% and the output to also reduce by 0.42% [107]. The tubes would be exposed to a similar type of flow as an HRSG would. Therefore, a similar finned tubular construction can be utilized. As the entire flow path is covered, a score of “C” is applied to the pressure drop aspect. It can be applied to any GT, as exhaust diffusers are commonly applied to all gas turbines. A score of “A” can be applied to the marketability aspect.

5.8.3. Opportunities of using additional embodiments for further performance improvements

There are some steps that can be taken to improve the safety and risk aspect. First, an analysis to ensure the added structure inside the diffuser is not prone to unacceptable levels of vibration at various GT operating ranges. Secondly, applying adequate flashing and insulation added to the penetrations in the exhaust diffuser, to avoid a risk of leaks from these points. Thirdly, having a prepared repair plan and spare parts available to reduce any unplanned downtime.

5.9. Heat recovery enhancement using a common exhaust stack for multiple GTs

This heat recovery system was proposed by US Patent Application 20130327052A1 [137]. The aim of this system was to improve heat transfer as well as exhaust gas dew point control.

5.9.1. System description

Improved heat transfer is achieved by routing the exhaust gas flow of several gas turbines into a common exhaust stack by increasing turbulence as shown in Fig. 29. This also leads to lower condensate formation because some of the constituents of the exhaust flow stream, can form acids when they react with the moisture in the flow stream.

The temperature across the heat exchanger is monitored by sensors such to stop the flow in case of dropping below the dew point. Fig. 30 shows the heat exchanger placement in a parallel flow-path to the common exhaust stack, with a separate flow damper. The position command is based on input from the temperature sensors.

Fig. 29. Overall exhaust routing as described in the patent [137].

Fig. 30. Detailed view of the heat exchanger and flow control system as described in the patent [137].

5.9.2. System’s performance evaluation and ranking against the established aspects criteria

Higher flow rates in the common exhaust leads to an improvement in the convective heat transfer coefficient of the hot side flow. Also, condensation control increases the life of the metallic components in the stack. However, there is no benefit to the GT operating efficiency. Hence a score of “C” is given to the effectiveness aspect.

The patent did not describe changes to the back-pressure on the individual GTs. There is no additional surface area added as an obstruction to the flow path. Hence a score of “A” can be applied to the pressure drop aspect.

One drawback is the scale of modification which is in the same order of magnitude as construction of the entire power station. Therefore, a “C” score was applied. While it may be implemented during the initial construction of a new power plant, however, it is unlikely to be feasible as an upgrade. It is technically possible to implement on any GT. Hence a score of “A” can be applied to the marketability aspect.

The other drawback is reduction of operating flexibility for the power plant. In a typical plant, the operator has more control over the operation of the individual GT. In such a common exhaust configuration, there is one large heat exchanger parallel to the stack. It needs to be a heat exchanger larger than an HRSG of a single GT, to utilize the heat in the common exhaust effectively. The operator would need to keep all the GTs in operation in to keep the flow in the common exhaust steady and not to deviate from the design specifications. Aside from hardware changes to the exhaust system, modification to the controls of all the GTs in the power station is also required. Hence a score of “C” is given to the complexity aspect.

The above-mentioned arrangement has other operating limitations. For example, if there is less demand from the grid, the operator might need to run several GTs in part load or full speed no load (FSNL) instead of shutting one or more GTs down. This adds to the total fired hours of the GTs and may affect the maintenance intervals and add to the operating costs. This means a score of “4” is applied to the safety and risk aspect.
5.9.3. Opportunities of using additional embodiments for further performance improvements

In order to improve the safety and risk aspect, there would need to be a segmentation of the heat exchanger in order to allow a fraction of its capacity to be used at times. One option is to use several smaller heat exchangers in parallel. If the extracted heat is then branched out to several applications, it would improve the effectiveness aspect. However, it needs to be balanced by the design complexity.

Up to this end, many of recent methods for exhaust gas heat extraction from simple cycle GT systems were investigated. A summarized results sheet which shows a comparison between the extraction methods with respect to the established evaluation criteria is outlined in Table 7. Table 7 also provides the score across all the categories for a typical WHR unit according to ISO 21,905. As Table 7 outlines a comparison between many exhaust heat extraction methods that are common in the market, a SC plant operator or owner can benefit from it while reviewing various options to improve their plant capability by extracting heat from the exhaust system. For example, if a plant owner

Table 7
Comparison between heat extraction methods 5.1–5.9 and typical WHR rankings across the safety and risk, as well as the next five comparative aspects.

<table>
<thead>
<tr>
<th>Application reference number</th>
<th>Title of application</th>
<th>Summary of application</th>
<th>Safety and risk</th>
<th>Complexity of implementation</th>
<th>Level of benefits</th>
<th>Comparison of additional pressure drop</th>
<th>Comparative scale of modification</th>
<th>Potential market opportunity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Score of 5.1</td>
<td>Integration of heat exchanger downstream of the exhaust diffuser, for HRSG protection and part improved part-load operation</td>
<td>Finned-tube heat exchanger rows added downstream of the turbine exhaust duct, for the purpose of cooling the exhaust air while simultaneously heating the GT inlet air, thereby allowing operating flexibility.</td>
<td>4</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td>Score of 5.2</td>
<td>Gas turbine performance heating system, using heat recovered from the exhaust stack</td>
<td>Tubular heat exchanger installed in a portion of the exhaust plenum, with features allowing isolation, for the purpose of fuel performance heating.</td>
<td>1</td>
<td>C</td>
<td>B</td>
<td>B</td>
<td>B</td>
<td>C</td>
</tr>
<tr>
<td>Score of 5.3</td>
<td>Integration of HRSG components into various parts of the GT exhaust diffuser</td>
<td>Existing exhaust system surface areas modified to add a functionality allowing heat extraction; for the purpose of integrating parts of an HRSG into the GT exhaust.</td>
<td>2</td>
<td>C</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>A</td>
</tr>
<tr>
<td>Score of 5.4</td>
<td>Heat recovery module suitable for addition to a gas turbine engine exhaust duct</td>
<td>A tubular heat exchanger, added as a module to an existing exhaust duct, with full isolation, partial or full heat extraction features.</td>
<td>4</td>
<td>A</td>
<td>C</td>
<td>B</td>
<td>A</td>
<td>C</td>
</tr>
<tr>
<td>Score of 5.5</td>
<td>Compressor clearance control system using GT exhaust heat transfer</td>
<td>Heat extraction through a port in the turbine exhaust, for the purpose of GT compressor clearance control at various modes of operation.</td>
<td>4</td>
<td>C</td>
<td>A</td>
<td>A</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td>Score of 5.6</td>
<td>Fuel heating via exhaust gas extraction from the GT exhaust stack</td>
<td>A portion of the exhaust stack flow is ducted to a separate heat exchanger module, for the purpose of fuel heating.</td>
<td>5</td>
<td>A</td>
<td>B</td>
<td>B</td>
<td>A</td>
<td>C</td>
</tr>
<tr>
<td>Score of 5.7</td>
<td>Heat extraction from the GT exhaust stack, through a circular duct carrying a heat transfer fluid</td>
<td>Partial replacement of a section of the exhaust stack, and replacement with a circular duct heat exchanger, clamped-on to the location.</td>
<td>4</td>
<td>A</td>
<td>C</td>
<td>A</td>
<td>B</td>
<td>B</td>
</tr>
<tr>
<td>Score of 5.8</td>
<td>Heat extraction from the GT exhaust diffuser, through a circular tubing network</td>
<td>Tubular heat exchanger module, in circular format, added into an existing exhaust diffuser for the purpose of heat extraction as well as exhaust flow balancing.</td>
<td>3</td>
<td>A</td>
<td>C</td>
<td>C</td>
<td>A</td>
<td>A</td>
</tr>
<tr>
<td>Score of 5.9</td>
<td>Common flow GT exhaust stacks, for improvement of heat recovery</td>
<td>Modification of the exhaust stack of a plurality of GTs, in order to route them to a single combined common exhaust stack, for the purpose of improvement of the convective heat transfer coefficient of the exhaust flow to a heat exchanger in the new common stack.</td>
<td>4</td>
<td>C</td>
<td>C</td>
<td>A</td>
<td>C</td>
<td>A</td>
</tr>
</tbody>
</table>
needs an option with a higher safety and risk aspect score, the finned tube heat exchanger in the exhaust duct (described in 5.1) and the innovations described in Sections 5.4–5.7, as well as the common exhaust stack of Section 5.9 fare equal or better to the typical WHR system.

There are many systems shown in Table 7 that have score higher than the WHR system across all aspects such as inventions reported in Sections 5.5–5.7. In invention 5.5, a portion of the exhaust gas is extracted from the turbine duct and is solely used for compressor clearance control. Invention 5.6 is used for fuel gas heating while 5.7 represents a circular heat exchanger duct in the exhaust stack for the purpose of heat extraction. Hence, the intent for using the recovered heat is important to make the correct choice. In this example, it can be concluded that if the intent is fuel gas heating, system 5.6 has higher potential than a WHR system.

6. Methodology for integrating new heat recovery methods in GT’s exhaust system

The main aim of this section is to develop a methodology which can be used to design a novel integrated heat recovery systems based on the several applications and methods of heat extraction from the GT exhaust system described in Sections 4 and 5. The proposed system was evaluated according to the established criteria in Section 3. The methodology used in this section can be followed by power plant operators, or academics interested in this field, to devise a system with maximized potential for their needs.

The first step is to outline all possible configurations of heat extraction methods and discussions applied in earlier sections. Table 8 provided an overview summary of the various methods for heat extraction and relevant applications presented in Section 5. Table 9 showed all possible applications have been discussed so far with appropriate link to the relevant subsection. The next step is to link the most suitable heat extraction method among the nine heat extraction methods described in Table 8 and match it to one or more of the nine applications described in Table 9. A power plant operator, or academic researcher, might have limitations on the method or application based on the configuration of their plant or research case.

Referring to Table 7 and the discussion presented in Section 5.7, different options were suggested to improve the Safety and risk aspect score to “5” and to improve the Potential market opportunity aspect to “A”. If these options were implemented, the heat extraction from the GT exhaust stack, through a circular duct carrying a heat transfer fluid will have one of the most positive overall scores in Table 7. Accordingly, the circular duct heat exchanger of Section 5.7 shall be used as the primary heat extraction method.

Referring to Table 9, the applications 1 through 4 have several detractors as summarized earlier in Table 3. Application 6 which cools exhaust air for HRSG protection also has a limited scope as it only has the potential to be applied on some CC units and not applicable for SC GT plants. In addition, the GT compressor clearance control is one of the most complex applications, as discussed in Section 5.5, and is therefore not the most practical application for marketability.

Therefore, fuel gas heating, and inlet air heating applications showed many advantages over other applications outlined in Table 9. Recall that inlet air heating implementation, as described in Section 5.1 is complex. The hardware modification spans locations inside the GT auxiliary systems, specifically inside the inlet filter house. Plant owners are typically reluctant for large-scale modifications in the inlet system flow path. The reason is the difficulty to control a foreign material exclusion (FME) policy for the amount of scaffolding, tooling and movement over long periods. Even a small part left behind could lead to a potential foreign object damage (FOD) in the compressor. In comparison, the fuel gas heating hardware modification is outside the GT auxiliary systems. Also, the complexity is less as it can operate with a stand-alone control system, as described in Section 5.6 in further detail.

In conclusion, fuel gas heating as application (i.e., option 5 in

<table>
<thead>
<tr>
<th>Invention reference number</th>
<th>Title of invention</th>
<th>Heat extraction method</th>
<th>Application of extracted heat</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.1</td>
<td>Integration of heat exchanger from the exhaust stack</td>
<td>Finned-tube heat exchanger rows added downstream of the turbine exhaust duct.</td>
<td>1. Cooling the exhaust air for HRSG protection</td>
</tr>
<tr>
<td>5.2</td>
<td>Gas turbine performance heating system, using heat recovered from the exhaust stack</td>
<td>Tubular heat exchanger installed in a portion of the exhaust plenum, with features allowing isolation.</td>
<td>Integrating parts of an HRSG into the GT exhaust; to reduce the HRSG size</td>
</tr>
<tr>
<td>5.3</td>
<td>Integration of HRSG components into various parts of the GT exhaust diffuser</td>
<td>Existing exhaust system surface areas, in particular aerodynamic vanes, modified to allowing heat extraction functionality.</td>
<td>Not provided</td>
</tr>
<tr>
<td>5.4</td>
<td>Heat recovery module suitable for addition to a gas turbine engine exhaust duct</td>
<td>A tubular heat exchanger, added as a module to an existing exhaust duct, with full isolation, partial or full heat extraction features.</td>
<td>Not provided</td>
</tr>
<tr>
<td>5.5</td>
<td>Compressor clearance control system using GT exhaust heat transfer</td>
<td>Heat extraction through a port in the turbine exhaust</td>
<td>GT compressor clearance control at various modes of operation</td>
</tr>
<tr>
<td>5.6</td>
<td>Fuel heating via exhaust gas extraction from the GT exhaust stack</td>
<td>A portion of the exhaust stack flow is extracted and ducted to a separate heat exchanger module, through a blower system.</td>
<td>Fuel gas heating.</td>
</tr>
<tr>
<td>5.7</td>
<td>Heat extraction from the GT exhaust stack, through a circular duct carrying a heat transfer fluid</td>
<td>Partial replacement of a section of the exhaust stack, and replacement with a circular duct heat exchanger, clamped-on to the location.</td>
<td>Not provided</td>
</tr>
<tr>
<td>5.8</td>
<td>Heat extraction from the GT exhaust diffuser, through a circular tubing network</td>
<td>Tubular heat exchange module, in circular format, added into an existing exhaust diffuser.</td>
<td>1. Exhaust air flow direction control</td>
</tr>
<tr>
<td>5.9</td>
<td>Common flow GT exhaust stacks, for improvement of heat recovery</td>
<td>Modification of the exhaust stack of a plurality of GTs, in order to route them to a single combined common exhaust stack.</td>
<td>1. Improvement of the convective heat transfer coefficient of the exhaust flow to a heat exchanger in the new common stack</td>
</tr>
</tbody>
</table>

Table 9) has the highest potential. Accordingly, the circular duct heat extraction method was integrated with fuel gas application. The proposed new integrated P&ID diagram and system components are shown in Fig. 31 and Table 10, respectively.
The fuel gas arrives from the fuel gas (FG) supply and treatment skid to the FG heat exchanger, which represents the performance heater, according to the definition in Section 5.2. Manual isolation valves are provided in upstream and downstream of the heat exchanger, for emergency or maintenance situations. A pressure safety valve (PSV) is installed downstream of the heat exchanger for protection in case of uncontrolled heating of the fuel gas in the heater. Temperature, pressure, and flow transmitters were provided, in communication with the GT control system. A scrubber and filtration skid help avoid contamination of the GT fuel nozzles; in case of corrosion or leakage inside the FG heater. Prior to the GT combuster, another skid contains emergency stop as well as pressure control valves.

A temperature control valve (TCV) was installed at the water connection upstream of the FG heater; in the form of a 3-way bypass valve. The TT-01 temperature reading was provided to the GT control system which compares it to the required setpoint for the GT operational mode, and to provide a command to the TCV as needed.

On the water side downstream of the FG heater, a vent is provided to avoid compressible fluids from entering the pumps and cause cavitation. Air may enter the flow stream after maintenance during which water is drained. A temperature control valve (TCV) was installed at the water connection upstream of the FG heater; in the form of a 3-way bypass valve. The TT-01 temperature reading was provided to the GT control system which compares it to the required setpoint for the GT operational mode, and to provide a command to the TCV as needed.

Table 9
List of the extracted heat applications discussed within the context of the paper.

<table>
<thead>
<tr>
<th>Application number</th>
<th>Application of extracted heat</th>
<th>Refs. section</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Gas to gas recuperation. Exhaust energy utilized to heat compressor discharge temperature prior to combustion</td>
<td>Section 4.1.1</td>
</tr>
<tr>
<td>2</td>
<td>Humid air turbine (HAT). Exhaust energy recuperated to pre-heat saturated air prior to combustion</td>
<td>Section 4.1.2</td>
</tr>
<tr>
<td>3</td>
<td>Pre-heating air into SOFC in a hybrid system with a GT: Compressor discharge air is pre-heated prior to entering a SOFC</td>
<td>Section 4.2.2</td>
</tr>
<tr>
<td>4</td>
<td>Stirling bottoming cycle: Exhaust gas heat utilized as Stirling engine heater</td>
<td>Section 4.3.2</td>
</tr>
<tr>
<td>5</td>
<td>Fuel gas heating</td>
<td>Sections 4.2.1, 5.2 and 5.6</td>
</tr>
<tr>
<td>6</td>
<td>Cooling the exhaust air for HRSG protection</td>
<td>Section 5.1</td>
</tr>
<tr>
<td>7</td>
<td>Heating the GT inlet air, thereby allowing operating flexibility on part load</td>
<td>Section 5.1</td>
</tr>
<tr>
<td>8</td>
<td>HRSG for ST bottoming cycle</td>
<td>Sections 4.3.1 and 5.3</td>
</tr>
<tr>
<td>9</td>
<td>GT compressor clearance control at various modes of operation</td>
<td>Section 5.5</td>
</tr>
</tbody>
</table>

Table 10
Reference table for the P&ID, describing the component names.

<table>
<thead>
<tr>
<th>P&amp;ID item number</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>dP-01</td>
<td>Differential pressure transmitter for the lead filter</td>
</tr>
<tr>
<td>dP-02</td>
<td>Differential pressure transmitter for the lag filter</td>
</tr>
<tr>
<td>FT-01</td>
<td>Flow transmitter for heated fuel gas</td>
</tr>
<tr>
<td>PT-01</td>
<td>Pressure transmitter for heated gas downstream of the FG heater</td>
</tr>
<tr>
<td>PT-02</td>
<td>Inlet pressure transmitter for lead water pump</td>
</tr>
<tr>
<td>PT-03</td>
<td>Inlet pressure transmitter for lag water pump</td>
</tr>
<tr>
<td>PT-04</td>
<td>Outlet pressure transmitter for lead water pump</td>
</tr>
<tr>
<td>PT-05</td>
<td>Outlet pressure transmitter for lag water pump</td>
</tr>
<tr>
<td>PT-06</td>
<td>Pressure transmitter for heated water downstream of the exhaust stack heat exchanger</td>
</tr>
<tr>
<td>TT-01</td>
<td>Temperature transmitter for heated gas downstream of the FG heater</td>
</tr>
<tr>
<td>TT-02</td>
<td>Pressure transmitter for water downstream of the water pumps</td>
</tr>
<tr>
<td>TT-03</td>
<td>Pressure transmitter for water downstream of the exhaust stack heat exchanger</td>
</tr>
<tr>
<td>Vent-01</td>
<td>Water vent downstream of FG heater</td>
</tr>
<tr>
<td>Vent-02</td>
<td>Water vent downstream of the exhaust stack heat exchanger</td>
</tr>
</tbody>
</table>

Fig. 31. Process and instrumentation diagram for the integrated concept of Section 5.11.
drained from the system. Gas may enter in case of a FG heat exchanger leak. After that the water enters the pumping station which consists of two parallel pumps operating in lead-lag configuration, for redundancy. Check valves downstream of the pumps were installed to prevent flow into the lag pump. The pressure transmitters upstream of each of the pumps monitor the pressure to ensure it does not drop below the minimum required suction head; according to the pump design. Such a situation shall induce an alarm in the GT control system and automatically shut down the pump, to prevent cavitation damage. Each pump leg has a pressure control valve; positioned according to feedback based on the pressure transmitter.

The outlet of both pumps is connected to a common header, prior to the exhaust stack heat exchanger. Two sets of stack heat exchanger can be provided, as shown in the figure, for redundancy. After the water is heated here, it exits to a connection line where another temperature as well as pressure transmitter are present. The feedback from the temperature transmitters upstream and downstream of this heater are provided to the GT control system which can control the position of the 3-way bypass control valve. Further downstream, a stop valve is installed, for times at which no water heating is required or during maintenance. A drain connection is needed upstream of the stop valve, to allow water drainage. Also a pressure safety valve is installed, in case of uncontrolled heating or water phase change in the heat exchanger. Finally, a set of redundant filters were provided prior to the water connection into the heat exchanger, to protect the heat exchanger passages against fouling. A selector valve was provided to allow selection of the filter to be in service, while isolating the other one thus, allowing filter replacement without interrupting the operation.

7. Conclusions

In this paper, various power plant systems were reviewed, and potential heat recovery applications were presented. The heat recovery systems were classified according to the area of implementation within the plant such as flange to flange section, GT auxiliary systems, and the exhaust system. Considering that many of the thermal power plants still operate in a simple cycle mode (e.g. 70% of the thermal power plants in GCC and almost 54% in the United States), the major objective of this paper was to develop a methodology to evaluate the traditional and recently developed heat extraction methods for simple cycle gas turbines. The methodology was developed based on many aspects that are important for thermal performance and practical implementation such as safety and risk in case of failure, complexity of implementation, effectiveness, level of additional pressure drop in the exhaust, the comparative scale of modifications and the potential market opportunity. The methodology established in this paper can be followed by power plant operators, or academics interested in this field, to devise a system with maximized potential for their needs.

Nine inventions were reviewed critically and compared to the traditional heat recovery methods that dominate the market. Detailed evaluation was conducted for each invention based on the proposed methodology along with devising the methods to improve the performance where possible.

Based on the critical review of the various heat extraction methods, a new integrated heat recovery system was proposed with combined benefits and high potential for practical implementation. The proposed integrated heat recovery system incorporated the circular duct heat exchanger for fuel gas heating application.

The system showed higher potential for practical implementation in a SC GT system, compared to the previously reviewed systems with significant heat recovery capabilities which would be sufficient to meet the fuel gas heating requirements. The proposed integrated heat recovery system provided a step-by-step worked example of following the established methodology and relevant scoring system in matching a heat extraction method with the most suitable application according to the defined criteria.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

References


