Humidified gas turbines—a review of proposed and implemented cycles

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Abstract

Gas turbines with air–water mixtures as the working fluid promise high electrical efficiencies and high specific power outputs to specific investment costs below that of combined cycles. Different humidified gas turbine cycles have been proposed, for example direct water-injected cycles, steam-injected cycles and evaporative cycles with humidification towers. However, only a few of these cycles have been implemented and even fewer are available commercially. This paper comprehensively reviews the literature on research and development on humidified gas turbines and identifies the cycles with the largest potential for the future. In addition, the remaining development work required for implementing the various humidified gas turbine cycles is discussed. This paper can also be used as a reference source that summarizes the research and development activities on humidified gas turbines in the last three decades.

1. Introduction

The world energy demand has increased steadily and will continue to increase in the future: the International Energy Agency (IEA) predicts an increase of 1.7% per year from 2000 to 2030. This increase corresponds to two thirds of the current primary energy demand, which was 9179 Mtoe in 2000, and in 2030, fossil fuels will still account for the largest part of the energy demand. In...
addition, the IEA predicts that the demand for electricity will grow by 2.4% per year and that most of the new power generating capacity will be natural gas-fired combined cycles [1]. Therefore, it is important to find improved technologies for power generation with high electrical efficiencies and specific power outputs (kJ/kg air), low emissions of pollutants and low investment, operating and maintenance costs for a sustainable use of the available fuels. Advanced power cycles based on gas turbines can meet these requirements, since gas turbines have relatively high efficiencies, low specific investment costs (USD/kW_e), high power-to-weight ratios and low emissions. The power markets have been deregulated in several countries and distributed generation and independent power producers have become more competitive. These changes require flexible power plants with high efficiencies for small-to-medium power outputs. As a result of this, it was estimated that more than half of the orders for new fossil-fueled power plants in the last part of the 1990s were based on gas turbines [2], since non-expensive and clean natural gas was available, and the demand for gas turbines continues to increase [3].

To increase the gas turbine efficiency and specific power output, the research and development work has focused on:

- Increased firing temperature
- Increased pressure ratio
- Improved component design, cooling and combustion technologies, and advanced materials
- Technology transfer from aircraft gas turbines to stationary gas turbines for power generation and conversion of aircraft gas turbines to power generation applications
- System integration (e.g., combined cycles, intercooling, recuperation, reheat, chemical recuperation)

Combinations of the above options are often applied for the improvement of gas turbine performance. Moreover, the efficiency and power output of a gas turbine can be augmented by humidifying the working fluid. Many different cycles with water or steam injection have been suggested, although only a few have been commercialized. The driving forces for gas turbine humidification have been the potential of high electrical efficiency and specific power output and reduced specific investment cost, decreased formation of nitrogen oxides (NO_x) in the combustor, reduced power output degradation caused by high ambient temperatures or low ambient pressures (i.e., at high elevations) and improved part-load performance compared with combined cycles.

This paper is a review of proposed and implemented humidified gas turbines with the purposes of identifying systems with a potential for high efficiency and specific power output and low costs, highlighting the research and development in the area, and addressing the technical issues to be solved. A large number of studies on different humidified gas turbine cycles are cited in this paper. It is not possible to cite every paper in the area, thus the most interesting papers, for example describing some new cycle configuration or application, have been selected. The paper begins with a description of the basic principle of humidified gas turbines and a brief history of the early work in the area. Then, proposed and implemented humidified gas turbines found in literature are described. Humidified gas turbines have some features in common that separate them from dry gas turbines; therefore, studies on water recovery and quality, thermophysical properties of air–water mixtures, combustion with humid air, normalizing humidified gas turbine performance data and the potential for less expensive carbon dioxide recovery than for dry cycles are discussed separately.
The paper ends with a discussion and evaluation of the suggested and realized humidified gas turbines. The thermodynamic potential, costs, applications and critical questions for the implementation of different gas turbine cycles with air–water mixtures as the working fluid are also considered.

2. Introduction to humidified gas turbines

2.1. Principle of humidified gas turbines

The basic idea of gas turbine humidification is that the injected water or steam increases the mass flow rate through the turbine. This augments the specific power output, since the compressor work remains constant (i.e. if water is injected after the compressor) and much less work is required to increase the pressure of a liquid than a gas. When energy in the gas turbine exhaust is recovered by preheating water or generating steam for injection or preheating the combustion air in a recuperator, the cycle efficiency is raised. In addition, for a recuperated gas turbine, water added before the combustor reduces the temperature of the compressed air at the inlet to the recuperator, thus increasing the energy recovery rate from the exhaust gas. Moreover, in a dry gas turbine, the combustion air mass flow rate is smaller than the exhaust gas mass flow rate, since a large amount of air is used for combustor and expander cooling, and there is a difference in specific heat capacity between the air and the exhaust gas; these factors reduce the effectiveness of the recuperator heat recovery. In contrast, humidification increases the combustion air mass flow rate and heat capacity, thus increasing the performance of the recuperator. When water or steam is injected in the combustion chamber, a part of the combustor cooling air is replaced and the amount of work for compression is decreased. For the same injection mass flow rate, water cools the combustor more than steam, since the energy for water evaporation is provided by the combustion gases. An additional advantage of humidifying the gas turbine working fluid is a reduced formation of NO\textsubscript{x} in the combustion process.

In this paper, the humidified gas turbines found in literature have been divided into three categories:

- Gas turbines with injection of water that evaporates completely. This category includes systems with water injection at the compressor inlet for power augmentation on hot days, water injection in the compressor for intercooling and water injection after the compressor combined with recuperation. Few systems of the two types mentioned last have been implemented, while water or steam injection in the combustion chamber is an established method to reduce the NO\textsubscript{x} formation.
- Gas turbines with injection of steam. In this category, there are several commercial systems.
- Gas turbine cycles with injection of water in a humidification tower, with a recirculation water loop. Generally, these cycles are called evaporative gas turbines (EvGT) or humid air turbines (HAT). To the authors’ knowledge, a pilot plant in Sweden is the only evaporative cycle in operation in the world today.

Fig. 1 schematically shows the systems in the first category. The power output of a gas turbine decreases with 0.5–0.9% when the ambient temperature increases with 1 °C [4], thus the power output of a simple cycle gas turbine is strongly dependant on the ambient temperature. The compressor volumetric flow rate
is constant at constant shaft speed; hence, high ambient temperatures reduce the air density and the compressor mass flow rate. Inlet air cooling reduces the compressor inlet temperature, thus increasing the air density and the compressor mass flow rate. In water-injected systems, the water cools the air and adds to the working fluid mass flow rate. Fig. 1 also shows a wet compression system, where water is injected inside the compressor, and an evaporative or spray intercooling system, where water is injected between the low-pressure and high-pressure compressors. Both these systems reduce the compression work by intercooling the air. Many researchers have investigated evaporatively aftercooled and recuperated gas turbines, schematically shown in Fig. 1, although the authors of this paper have not been able to find any publications on such systems in operation. If the cycle is surface intercooled, the water heated in the intercooler is often injected. The water is most often assumed to be injected into a chamber, to slow down the air and allow the water appropriate time to evaporate. However, Rogers and Liese [5] investigated direct water injection with a pressure-swirl nozzle into a high-pressure fast-moving air stream without a saturation chamber. Hot water flash atomizes when sprayed, resulting in smaller droplet diameters, higher water evaporation rates and a closer approach to saturation than cold water. Direct water injection in a gas turbine cannot completely saturate the air; on the other hand, the pressure drop resulting from the water injection is insignificant and the cost and size of the equipment is small.

In steam-injected gas turbines, schematically shown in Fig. 2, steam generated from the exhaust gas energy is injected into the gas turbine before, in or after the combustor or between turbine stages. The steam must be superheated to prevent water droplets in the combustor and corrosion, and at a higher pressure than the compressor discharge pressure.

One typical configuration of the evaporative gas turbine cycle is schematically shown in Fig. 2. In this cycle, energy is recovered by humid air in the recuperator and water in the economizer, intercooler (if included) and aftercooler. In the evaporative cycle, the boiler of a steam-injected cycle and the evaporation chamber of a water-injected cycle are replaced by a humidification tower, which is a key component in the cycle. In the tower, which in most studies is a column with a packing that increases the air–water contact area, air and water are brought into direct counter-current contact. In the simultaneous mass and heat transfer process, some of the water evaporates and, in most cases, saturated heated humid air leaves at the top of the tower. The water evaporates at a temperature determined by the partial
pressure of water in the air and not at the boiling temperature corresponding to the total pressure in the system. Therefore, the evaporation occurs at a lower temperature than is possible in a conventional boiler and low-temperature heat can evaporate water. In contrast, the pinch point limits the heat recovery in the boiler in the steam-injected cycle and energy at low temperature levels cannot be used to evaporate water. In the evaporative cycle, the air meets water of a higher temperature as it ascends in the tower. The humidity ratio in the air increases upwards in the tower, thus raising the evaporation temperature. Hence, the evaporation process is non-isothermal, the air and water temperature profiles are closely matched and the exergy destruction in the humidification process is lower than in the boiler in a steam-injected cycle, where the isothermal boiling process results in a poor matching of the exhaust gas and water temperature profiles.

2.2. Early history of humidified gas turbines

Water or steam injection was used in the first gas turbines constructed in the beginning of the 20th century for cooling the combustion chamber, since the available materials could not withstand high temperatures. At the same time, the cooling air need was reduced and the power output increased. In one of the first gas turbines with a positive net power output, constructed by Elling in 1903, the compressor was cooled by water injection and the combustion gases were cooled by a water jacket; the water that evaporated in the jacket was injected in the gas turbine [6]. Armengoud and Lémâle constructed a gas turbine in 1905–1906 in which steam generated from combustion chamber cooling was injected to achieve a positive net power output [7]. However, the first commercial gas turbine, manufactured by Brown Boveri, that started operation in Neuchâtel, Switzerland, in 1940 was not humidified [8].

Following the early gas turbines, studies on gas turbines with water injection in the combustion chamber continued [9–18]. Water injection inside the compressor for reduced compressor work was further investigated as well [19], also combined with recuperation [20,21]. Water injection before or after the compressor in aircraft jet engines has been used for thrust augmentation at take-off [8]. Recuperated gas turbines with water injection after the compressor (i.e. evaporative aftercooling)
have been studied at least since the 1930s [22–27], and more studies on such systems followed in the 1980s.

Steam-injected gas turbines have been investigated since the beginning of the 20th century [13,23,28–35], and some steam-injected gas turbines were installed: Rice [36] mentioned gas turbines with steam addition to the combustor in the 1950s and with steam injection in the compressor discharge air in the 1960s. The interest in steam injection for both power augmentation and NO\textsubscript{x} control increased in the 1970s: Ediss [37] reported on work on steam-injected combustion chambers by Rolls-Royce and Westinghouse, General Electric (GE) patented a steam-injected gas turbine [38], steam was injected into a GE MS 3000 gas turbine in 1969 for increased power output [39] and Golod et al. [40] reported on experimental investigations on steam injection in a two-shaft gas turbine. After the first patents for the Cheng cycle [41], several commercial steam-injected gas turbines were developed. Steam-injected gas turbines became more common in the middle of the 1980s and today there are several hundreds in operation. The development of steam-injected gas turbines was reviewed by Tuzson [42], and Cheng and Nelson [43] presented an overview of the development of the Cheng cycle.

The study of gas turbine cycles including humidification towers probably started with the patent by Martinka [44] for a recuperated gas turbine where water heated in the intercoolers was injected in a packed tower for humidification of the compressed air. Several similar gas turbine cycles were patented in the following years [45–47]. The interest in this cycle increased in the 1980s, as shown by, for example, the work by Mori et al. [48] and the introduction of the humid air turbine (HAT) cycle [49].

It should be noticed that the early efforts with water or steam injection in gas turbines were focused mainly on technical issues. However, efficiency and power output improvements by adding water or steam have become more important for the gas turbine research and development in the last two decades.

3. Water-injected gas turbines with complete evaporation

3.1. Water injection for inlet air cooling

The interest in inlet air cooling systems for gas turbines has increased in recent years due to the increasing need for power to a low specific investment cost, especially during the summer when the ambient temperature is high. The available inlet air cooling systems can be classified into the following groups:

- Evaporative media coolers. Water is distributed over pads of fibers through which the air passes to be humidified.
  - Spray inlet coolers or fogging systems. Water is injected into the air through nozzles and creates a fog of small water droplets, 5–20 \(\mu\text{m}\) in diameter. The systems are divided into two groups:
    - Saturated systems, where the air is saturated before the compressor.
    - Overspray systems, where more water than is needed for saturation is injected. Water droplets enter the compressor, evaporate and cool the air.

- Mechanical vapor compression or absorption chillers, where a heat exchanger cools the inlet air. Chillers can increase the gas turbine power output by 15–20\% and the efficiency by 1–2\% (i.e. if gas turbine exhaust gas energy is recovered). A chiller can cool the inlet air regardless of the ambient conditions; however, the specific investment cost is higher than for media and spray coolers.
The investment cost is low for media coolers and spray systems; however, the water consumption results in an extra operational cost. Spray systems require demineralized water to avoid depositions in the compressor, while media coolers can use untreated municipal water [50]. The evaporative media cooler is a common inlet air cooling system, although spray systems have gained in popularity since the beginning of the 1990s. Evaporation in a media cooler requires a driving force; therefore, the performance of media coolers is limited by the ambient temperature and relative humidity. A media cooler can increase the relative humidity of the inlet air to about 90%, thus increasing the power output by 5–10% and the efficiency by 1.5–2.5% and decreasing the NO\textsubscript{x} emissions by 10% for a conventional diffusion combustor [50]. Saturated spray systems provide similar performance improvements as media coolers, while overspray systems can increase the power output by 10–20%, the efficiency by 1.5–3% and decrease the NO\textsubscript{x} emissions by 20–40% for a conventional diffusion combustor [50]. The possible increase of power output and efficiency resulting from a water-injected system depends on the ambient air temperature and relative humidity; however, overspray systems are not as sensitive to high ambient humidity as media coolers and saturated spray systems. In overspray systems, the mass flow rate of excess water corresponds to approximately 0.5–2% of the compressor inlet air mass flow rate [4]. Injection of hot water results in an aerosol, where the water droplet diameter is between 2 and 3 \textmu m, with a low risk for compressor blade erosion [51]. Hot water provides smaller droplets than cold water, since the water flashes upon leaving the nozzle. Compared with inlet air cooling systems with cold water, an overspray hot water system can operate at lower ambient temperatures and is not limited to the same extent by high ambient humidity levels, while the investment cost is higher. Water injection at the compressor inlet for air cooling or on-line compressor washing affects the air temperature and specific heat capacity; hence, the compressor runs off-design with a possible risk of stalling [52]. However, spray inlet air cooling systems have been installed and operated successfully on approximately 600 gas turbine units [4]. Models of the effect of overspray on the compression work, and references to the work of others, were presented by HärteI and Pfeiffer [53] and White and Meacock [54].

3.2. Water injection in the compressor—wet compression and spray intercooling

Wang et al. [55] experimentally investigated the effects of water injection between the stages in a small compressor. Ingistov [56] tested a system for water injection between compressor stages. The system was first introduced for cleaning the compressor blades from depositions, although with increased water injection rates the power output could be improved. Zheng et al. [57] presented a thermodynamic model for wet compression that could be applied for wet inlet cooling as well. The above-mentioned inlet air cooling system with hot water [51] is to be further developed into the TopHat cycle [58], where a large amount of water preheated by the gas turbine exhaust gas is injected in the compressor, thus reducing the compressor outlet temperature and enabling efficient recuperation of the gas turbine. A comparison showed that the TopHat cycle had higher efficiency and specific power output than, for example, a combined cycle, a Cheng cycle and a HAT cycle, and the highest TopHat efficiency was 57.4% \footnote{When not otherwise stated, the efficiencies cited in this paper are based on the lower heating value of the fuel.} (PR = 12, TIT = 1200 °C). For a large gas turbine, the authors of the study calculated an efficiency of 58.2% for the TopHat cycle, while a combined cycle had an efficiency of 57.6%.
advanced TopHat cycle, an efficiency of over 60% was expected. The investment cost for a large power plant (350 MW<sub>e</sub>) was reported to be 20% lower for a TopHat cycle than for a combined cycle.

Spray intercooling was investigated in the AGTJ-100A gas turbine pilot plant [59]. The testing of this reheat twin-spool gas turbine started in Japan in 1984. In order to increase the power output and efficiency and to decrease the generation of NO<sub>x</sub>, evaporative intercooling was chosen [60]. After the intercooler, the relative humidity was 90%. In 1998, GE introduced the Sprint spray intercooling system for the LM6000 gas turbine, where water is injected at the low-pressure and high-pressure compressor inlets. The Sprint system increases the power output by more than 8% at ISO conditions and by more than 32% at 32 °C ambient temperature and at high ambient temperatures, the gas turbine efficiency is improved as well [61].

3.3. Recuperated water-injected gas turbines

The basic evaporatively aftercooled and recuperated gas turbine is shown in Fig. 1. This system can be modified by injecting water at various points in the cycle. The central focus is on the effective recovery of the exhaust energy of the gas turbine.

The basic water-injected cycle has been studied by, for example, Najjar and Zaamou [62], who found that this cycle had 57% higher power output and 13% higher efficiency compared with a recuperated gas turbine without water injection. Camporeale and Fortunato [63] studied water-injected cycles based on a two-shaft industrial gas turbine. With a water injection rate corresponding to 13% of the inlet air mass flow rate, the water-injected cycle had 30% higher power output compared with the simple cycle gas turbine; the efficiency was 44.8% for the water-injected cycle (PR = 16.7, TIT = 1250 °C) and 32.8% for the simple cycle gas turbine. For part-load and high ambient temperatures, the performance of the water-injected cycle was superior to the simple cycle regarding power output and efficiency. For an aeroderivative gas turbine, Camporeale and Fortunato [64] found a power output and an efficiency of 23.8 MW<sub>e</sub> and 47.1% for the water-injected recuperated cycle (PR = 16.3, TIT = 1250 °C) and 31.7 MW<sub>e</sub> and 41.6% for the steam-injected cycle (PR = 21.4, TIT = 1250 °C). Camporeale and Fortunato [65] further studied water-injected aeroderivative gas turbines with consideration of the expander design.

Basic cycles modified with intercooling have been studied as well. Manfrida et al. [66] analyzed the recuperated water-injected cycle and steam-injected gas turbines; both cycle types were investigated with and without surface intercoolers. The water-injected cycles had lower efficiencies than the steam-injected cycles due to a lower amount of heat recovery from the exhaust gas. The authors suggested combined cycles with injection of bottoming cycle steam in the gas turbine for improved heat recovery. At base load, this system is operated without any steam injection for optimal efficiency, while at peak load conditions, steam is injected in the gas turbine for increased power output. The water-injected cycles had the highest efficiencies for low pressure ratios and high water injection rates. The water injection rates corresponded to up to 20% of the inlet air mass flow rate, thus the air was oversaturated in the aftercooler and the remaining water evaporated in the recuperator. Frutschi and Plancherel [67] predicted an efficiency of 43–45% for a water-injected intercooled gas turbine. Chiesa et al. [68] calculated an efficiency of 55.1% (PR = 33, TIT = 1500 °C), or 52.9% (PR = 21, TIT = 1250 °C), for an aeroderivative surface intercooled water-injected gas turbine. Evaporative intercooling resulted in slightly lower efficiencies. In the aftercooler, more water than was required for saturation was injected and the remaining droplets evaporated in the recuperator. Traverso and Massardo [69] investigated
evaporatively intercooled water-injected cycles with power outputs of about 50 MW$_e$. The efficiency of the water-injected cycle was lower than for a steam-injected or evaporative gas turbine and a combined cycle, since the water-injected cycle was limited by the high irreversibility of the air–water mixing process and the water addition was limited by air saturation. Rolls-Royce has studied a recuperated cycle based on the aeroderivative gas turbine Trent with water injection for intercooling and aftercooling. An efficiency of over 50% was expected, which is lower than for a combined cycle and higher than for a simple cycle (42%) [70]. Compared with a combined cycle, the water-injected cycle should be more flexible and have a 30% lower specific investment cost. A water-injected cycle is one alternative for enhancing the Trent performance, since Rolls-Royce deems that the technical risk of increasing the pressure ratio, currently 35, and the firing temperature is high. However, it is expected that the water-injected cycle might have control problems and the durability of the recuperator is a risk.

Another possible modification of a gas turbine is water injection in the recuperator. Horlock [71] investigated open and closed water-injected gas turbines, with intercooling and water injection in the recuperator for the open cycle. The optimum pressure ratio for high efficiency was rather low (i.e. between 8 and 10). Horlock [72] analyzed the performance of a recuperator with water injection before or in the recuperator. El-Masri [73] investigated a recuperated two-shaft gas turbine with water injection in the intercooler, aftercooler and recuperator. This cycle was 2.75% points and 5% points, respectively, more efficient than a non-intercooled steam-injected gas turbine and a non-water-injected intercooled recuperated cycle. Without water injection in the recuperator, the efficiency was reduced by about 2% points. The author claimed that the evaporative aftercooler was easy to design and impurities in the water should pass through the engine without problems. The water-injected recuperator could be fouled if high-quality water is not used; therefore, the water should be injected as a fine mist that evaporates without impinging on the recuperator walls. Another possible modification of the basic water-injected cycle is evaporation and recuperation in several steps. Annerwall and Svedberg [74] investigated surface intercooled reheated gas turbines where evaporative aftercooling was followed by a recuperator, followed by water injection and another recuperator. The thermal efficiency obtained for this system was 52.4%, while a combined cycle had an efficiency of 49.4% and an intercooled, recuperated and reheated steam-injected cycle had an efficiency of 49.0%. All cycles were based on a simple cycle gas turbine with a power output of 21.3 MW$_e$ and a thermal efficiency of 32.5% (PR = 13.5). Szargut [75] studied a water-injected gas turbine for cogeneration of power and district heating. This cycle had two-step intercooling: in the first step, water for injection was heated and in the second step, external cooling water was used. After the compressor, water was injected for aftercooling, the air was recuperated, more water was injected, the air was further recuperated before injection of water heated by the exhaust gas and further recuperation. Multi-stage injection enabled addition of more water than single-stage injection, without water droplets evaporating in the recuperator. The high concentration of water in the exhaust gas raised the dew point, which increased the district heating production. Szczygiel [76] also investigated cycles with multi-stage injection [75] and single-stage injection, with or without an external aftercooler and extraction of expander cooling air from different locations in the cycle.

More extensive modifications of the basic design, where the compressed air stream is split for improved heat recovery, have also been suggested. Mori et al. [48] proposed two cycles designed for a low air–water mixing exergy destruction: one with direct water injection and one with a humidification tower, which is described in Section 5.3.1. In the direct water-injected cycle, the intercooling process was divided into two parts and the second intercooler was water-cooled. Following the surface aftercooler, the air stream was split into three streams and water heated in the second intercooler was
injected into each stream. The resulting humid air streams were passed in parallel through the first intercooler, the aftercooler and a recuperator, then the air streams were mixed and passed through a second recuperator. Water evaporated in the first recuperator; thus, this was not a standard heat exchanger and the cycle with a humidification tower was considered more technically feasible. A cycle similar to the water-injected cycle by Mori et al. [48] was patented by Nakamura et al. [77], although with an important difference: the air stream was split directly after the compressor. One part of the stream was passed through the heat recovery system as in the cycle by Mori et al. [48], while the other part bypassed the heat recovery system and was mixed with the humid air before a third and last recuperator. When 28% of the intake air was passed through the humidification system, the efficiency was 50.4% and for 53% part-flow, the efficiency was 51.2% (TIT = 1000 °C). Nakamura [78] patented a simpler water-injected part-flow cycle. In this cycle, one of the streams resulting from the split after the compressor was evaporatively aftercooled and recuperated before mixing with the dry air stream and further recuperation. An evaporatively intercooled configuration where all of the compressed air was aftercooled was also patented. Nakamura [79] presented a recuperated gas turbine where the air was intercooled in two steps: first in a heat exchanger and then by injection of water heated in the intercooler. Water heated in the intercooler was also injected in the compressed air for aftercooling and in the fuel. Bram and de Ruyck [80] used exergy and pinch analysis to devise a humidified cycle similar to the HAT cycle, although without a humidification tower. The layout of this cycle closely resembled the water-injected cycle by Mori et al. [48]; the main difference was that the energy from the second intercooler was not recovered into the cycle. This cycle had about the same net efficiency as an evaporative cycle with a humidification tower (54%).

Other cycle configurations have been investigated as well. A combined cycle with water injection was presented by Aronis and Leithner [81]. After the gas turbine compressor, water was injected and evaporated by low-temperature energy (below 170 °C) in a heat exchanger. The low-temperature energy could be provided by biomass combustion, solar, geothermal or industrial waste heat. Since the combustion temperature was lowered to the allowed turbine inlet temperature by the water and not by air, the gas turbine fuel could be combusted with an amount of air close to the stoichiometric air amount and the compression work was reduced. Since the water content in the exhaust gas was high, condensation occurred at temperatures suitable for district heating. Possible cycle modifications included aftercooling the compressed air before the evaporator by superheating the humid air or superheating the humid air with an external heat source. Another innovative recuperated water-injected gas turbine for cogeneration of power and hot or cold water was presented by Dodo et al. [82]. In this system, an absorption refrigerator recovered the exhaust gas energy. Depending on the operational conditions, cold water generated by the chiller could be injected at the compressor inlet or hot water generated by the chiller could be injected for aftercooling.

Gas turbines with injection of both water and steam have also been proposed, for example by El-Masri [83], who presented a cycle where the compressed air was aftercooled by water injection; the humid air was heated by exhaust gas in a recuperator and then mixed with saturated steam, generated from the exhaust gas energy, before further recuperation. The cycle could be intercooled by water injection to further increase the efficiency. Dual-pressure combined cycles had slightly higher efficiencies than the suggested cycle, which in turn had higher efficiency than steam-injected cycles. The amount of exhaust gas energy used for recuperation and steam generation could be varied, which enabled flexible cogeneration of power and steam. Bolland and Stadaas [84] investigated a modified version of the intercooled cycle presented by El-Masri [83] for power generation only for current state-of-the-art gas
turbines. In this cycle version, the steam was superheated before injection in the combustor instead of mixing saturated steam with humid air before the final recuperator. For the investigated gas turbines, a steam-injected cycle had 8–10% points higher efficiency than a simple cycle and the suggested cycle had 3–4% points higher efficiency than a steam-injected cycle. For large industrial gas turbines, a triple-pressure combined cycle had the highest efficiency (52–55%), followed by the suggested cycle (49%) and a steam-injected cycle (45–46%). For medium-size industrial and aeroderivative gas turbines, the difference between a combined cycle and the suggested cycle was small and for small industrial gas turbines (4–6 MWₑ), the efficiency of the suggested cycle was equal to or higher than the efficiency of a single-pressure combined cycle.

One demonstration plant where water was to be injected directly into a gas turbine after the compressor has been found in literature: a proposed demonstration plant for combined generation of power and district heating at the Vrije Universiteit in Belgium. In this gas turbine, the compressed air was heated in a metallic heat exchanger by gasified biomass that had been combusted with the gas turbine exhaust gas in an external combustor. The heat exchanger could heat the air to 800 °C, so natural gas topping combustion could be added to increase the turbine inlet temperature to 1000 °C. Water, preheated by the exhaust gas, could be injected before the external heat exchanger, for augmented power output and efficiency and flexible power-to-heat ratios. Calculations showed that without water injection and topping combustion, the power output would be 300 kW and the heat output 1200 kW (thermal efficiency 13%, total efficiency 65%), while injection of 8.5% by weight of water would increase the power output to 565 kWe (efficiency 21.5%). With water injection and topping combustion, the power output would be 800 kWe (efficiency 26%) [85]. However, the authors of this review paper have not been able to find any publications about operation of this plant with water injection. Bechtel and Parsons [86] patented a similar externally fired water-injected gas turbine, although without topping combustion. A so-called Vapor, Air and Steam in conjunction with a work Turbine (VAST) cycle has also been proposed [87]. This cycle is a variation of steam injection gas turbine. However, the inventor claimed that the cycle overcomes the saturation limits of the HAT and evaporative cycles resulting in higher specific power and lower costs, and also overcomes the conventional limit on water injection caused by flame stability.

3.4. Water or steam injection in the combustor for NOₓ control

Water or steam injection in the combustor to decrease the formation of NOₓ was suggested in the beginning of the 1960s [88] and was, at least up to the mid 1990s, the most common NOₓ control method [8]. For gas turbines, thermal NOₓ is the predominant source of NOₓ emissions. The formation of thermal NOₓ is determined by the temperature and residence time; hence, adding a diluent, for example water or steam, to the combustion zone reduces the NOₓ generation by lowering the temperature [89]. An additional advantage of water or steam injection is augmented power output; however, if the exhaust gas energy is not recovered by preheating the water or generating the steam for injection, the efficiency is decreased. Steam injection reduces the efficiency less than water injection, since the energy to evaporate the water is taken from the fuel; furthermore, this means that for the same NOₓ reduction effect, a smaller amount of water is needed compared with steam [89]. Usually, water corresponding to about 50% of the fuel mass flow rate is injected and steam corresponding to 100–200% of the fuel mass flow rate is injected [8]. Dry low-NOₓ burners were introduced in the beginning of the 1990s. In these burners,
the combustion is lean, thus reducing the temperature and the NO\textsubscript{x} formation. NO\textsubscript{x} levels of 25 ppmvd (parts per million by volume, dry, 15% oxygen) can be achieved by both wet [89] and dry systems [8]. Mathioudakis [90] proposed an analysis method to assess the effects on the power output and efficiency of water injection in the gas turbine combustor.

### 4. Steam-injected gas turbines

Steam-injected gas turbines are mostly used for small-to-medium-size cogeneration of power and steam in industry (e.g. food industries, breweries, chemical and paper industries) and community facilities (e.g. hospitals, universities, district heating), especially in applications where the steam load or the electricity price varies. In a cogeneration system, the steam demand often determines the operation. A system consisting of a simple cycle gas turbine and an HRSG (heat recovery steam generator) is efficient for constant heat demand, whereas the system is inefficient at reduced heat demands, since the steam production or the gas turbine power output must be reduced; however, in a steam-injected cycle, the steam not needed for other purposes can be injected in the gas turbine for increased power generation. A duct burner can be installed before the HRSG to increase the steam production.

This section begins with a description of the available commercial steam-injected gas turbines and then the systems proposed in literature are presented.

#### 4.1. Commercial steam-injected gas turbines

The steam-injected gas turbines available in the Gas Turbine World Handbook [91] have been summarized in Table 1.

**4.1.1. The Cheng cycle**

The first commercial Cheng cycle, the 501-KH5, started operation in 1985 at a university in California and generated power and steam for space heating and cooling; surplus power could be exported to the grid. The 501-KH5 was based on the single-shaft aeroderivative Rolls-Royce Allison 501-KB gas turbine. This gas turbine has a large compressor surge margin, which enables injection of all the steam generated from the exhaust gas without compressor stalling. The Kawasaki M1A-13 gas turbine has also been converted to a Cheng version, the M1A-13CC, and the first plant started operation in Japan in 1988 [96]. Since then, over a hundred Cheng cycles have been installed [43]. In addition, there are retrofit Cheng steam injection systems for GE industrial gas turbines, suitable for gas turbines used for peaking power generation or cogeneration [43]. In 2001, Cheng retrofit systems were installed on three GE Frame 6B gas turbines [97].

In addition to the 501-KH5 and the M1A-13CC, Cheng cycles have been designed, although not implemented, for several other gas turbines. Nelson et al. [98] proposed a single-shaft medium-size (43 MW\textsubscript{e}) Cheng gas turbine with an efficiency of 50%. All of the generated steam could be injected: superheated steam was injected in the combustor and saturated steam was used for turbine blade cooling and to reduce the NO\textsubscript{x} formation by premixing of steam with the fuel. A Cheng version of the steam-cooled GE 7H gas turbine was calculated to have an efficiency of 60.5% and a power output of 600 MW\textsubscript{e} [43]. A Cheng version of the GE LM2500 gas turbine with an
Table 1
Available steam-injected gas turbines [91]

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Model</th>
<th>$P$ (MW&lt;sub&gt;e&lt;/sub&gt;)</th>
<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT (°C)</th>
<th>$m_{steam}$ (kg/s)&lt;sup&gt;a&lt;/sup&gt;</th>
<th>$m_{air}$ (kg/s)&lt;sup&gt;b&lt;/sup&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cheng cycles</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Rolls-Royce&lt;sup&gt;c&lt;/sup&gt;</td>
<td>501-KH5</td>
<td>6.4&lt;sup&gt;d&lt;/sup&gt;</td>
<td>39.9</td>
<td>10.2</td>
<td>N/A</td>
<td>2.7&lt;sup&gt;e&lt;/sup&gt;</td>
<td>18.4&lt;sup&gt;d&lt;/sup&gt;</td>
</tr>
<tr>
<td>Kawasaki heavy industries</td>
<td>M1A-13CC</td>
<td>2.3&lt;sup&gt;f&lt;/sup&gt;</td>
<td>31.9</td>
<td>8.9</td>
<td>N/A</td>
<td>1.4&lt;sup&gt;g&lt;/sup&gt;</td>
<td>N/A</td>
</tr>
<tr>
<td><strong>STIG cycles</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ishikawajima-harima heavy industries</td>
<td>STIG-LM1600</td>
<td>16.9</td>
<td>39.7</td>
<td>25.1</td>
<td>735</td>
<td>2.5 (1.5 &lt;sup&gt;g,h&lt;/sup&gt; + 1.2 &lt;sup&gt;b&lt;/sup&gt;)</td>
<td>52.6</td>
</tr>
<tr>
<td>GE aero energy products&lt;sup&gt;j&lt;/sup&gt;</td>
<td>LM2500 STIG</td>
<td>26.5/27.8</td>
<td>39.4/40.6</td>
<td>20.0</td>
<td>N/A</td>
<td>2.3 &lt;sup&gt;e,h&lt;/sup&gt; + 4.0 &lt;sup&gt;i&lt;/sup&gt;</td>
<td>75.8</td>
</tr>
<tr>
<td>Ishikawajima-harima heavy industries</td>
<td>STIG-IM5000</td>
<td>50.1/51.2</td>
<td>42.9/43.9</td>
<td>30.6</td>
<td>788</td>
<td>10.4 + 9.1&lt;sup&gt;k&lt;/sup&gt;</td>
<td>156.0</td>
</tr>
<tr>
<td><strong>Aquarius</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mashproekt</td>
<td>Aquarius-16</td>
<td>15.5</td>
<td>41.7</td>
<td>20.0</td>
<td>N/A</td>
<td>5.6</td>
<td>44.8</td>
</tr>
<tr>
<td>Mashproekt</td>
<td>Aquarius-25</td>
<td>25.0</td>
<td>42.0</td>
<td>17.9</td>
<td>N/A</td>
<td>8.1</td>
<td>72.7</td>
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<tr>
<td>Mashproekt</td>
<td>Aquarius-40</td>
<td>40.7</td>
<td>42.8</td>
<td>19.8</td>
<td>N/A</td>
<td>12.9</td>
<td>93.5</td>
</tr>
<tr>
<td><strong>Miscellaneous</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ishikawajima-harima heavy industries</td>
<td>IM400 IHI-FLECS</td>
<td>6.2</td>
<td>35.7</td>
<td>12.4</td>
<td>1010</td>
<td>N/A</td>
<td>18.2</td>
</tr>
<tr>
<td>Kawasaki heavy industries</td>
<td>M7A-01ST</td>
<td>6.6</td>
<td>32.9</td>
<td>12.7</td>
<td>N/A</td>
<td>1.7&lt;sup&gt;l&lt;/sup&gt;</td>
<td>22.2</td>
</tr>
</tbody>
</table>

All data from the Gas Turbine World Handbook [91], if not otherwise stated.

<sup>a</sup> Injected steam mass flow rate.

<sup>b</sup> Gas turbine exhaust mass flow rate.

<sup>c</sup> Centrax Gas Turbine (6.4 MW<sub>e</sub>, 36.7%, PR = 12.3) and Hitachi Zosen (6.8 MW<sub>e</sub>, 42.2%, PR = 10.2) sell this system as well.

<sup>d</sup> Without steam injection, the power output is 3.8 MW<sub>e</sub> and the exhaust gas flow rate is 15.7 kg/s [92].

<sup>e</sup> Ref. [93].

<sup>f</sup> Up to 8.5 tons/h of process steam can be generated with a supplemental burner. The power output is 1.3 MW<sub>e</sub> without steam injection [94].

<sup>g</sup> Ref. [95] (60 Hz).

<sup>h</sup> Steam injection at fuel nozzle for NO<sub>x</sub> abatement [95].

<sup>i</sup> Steam injection at compressor discharge for power augmentation [95].

<sup>j</sup> Ishikawajima-harima heavy industries (50/60 Hz, 26.7/28.0 MW<sub>e</sub>, 39.5/40.8%, PR = 19.7/20.0, TIT = 783/793 °C, 6.3 kg steam/s) and MTU Motoren- und Turbinen-Union Friedrichshafen GmbH (27.6 MW<sub>e</sub>, 40.4%, PR = 20.2, TIT = 807 °C) also sell this system.

<sup>k</sup> High-pressure steam + low-pressure steam.

<sup>l</sup> Ref. [96].
efficiency of 44.5% has been designed [99]. The LM2500 STIG had an efficiency of 41.0% and the higher efficiency of the Cheng cycle was due to several factors: the HRSG in the STIG cycle had a fixed pressure, while the Cheng HRSG pressure was self-regulated, the steam injection rate was determined manually in the STIG, while it was automated in the Cheng cycle and the technology for steam injection differed between the two cycles: in the Cheng cycle, steam was injected as in the cycle described by Nelson et al. [98]. These design differences enabled a steam injection rate of 9.1 kg/s in the LM2500 Cheng, while the LM25000 STIG steam rate was 6.3 kg/s. The authors reported that the simulated power output and efficiency of the Cheng cycle were relatively unaffected by high ambient temperatures, while the impact was larger on the STIG cycle.

Operational experiences with Cheng cycles are provided by Kellerer and Spangenberg [100], who described a 501-KH5 Cheng cycle for cogeneration of power and district heating that commenced operation in 1996 at a German university. Strasser [94] described an M1A-13CC Cheng cycle for cogeneration of power and steam installed in a German factory. Penning and de Lange [101] discussed two 501-KH5 Cheng cycles for cogeneration installed in a cardboard factory with fluctuating steam demand and constant power demand in operation since 1993. Macchi and Poggio [102] described the first 501-KH5 Cheng cycle with water recovery, which was located at an Italian car-manufacturing factory. The Cheng cycle, that started commercial operation in 1993, supplied all the power, steam and hot water required by the factory and the surplus power was sold to the grid. A comparison between a Cheng cycle and a simple cycle gas turbine with a supplementary-fired HRSG for the power and heat demand curves of the plant showed that the Cheng cycle had a lower electricity production cost.

4.1.2. The STIG cycle

GE offers steam injection systems (STIG) for power augmentation of their aeroderivative gas turbines. The first STIG system started operation in 1985 at a paper mill in California, where a LM5000 gas turbine was retrofitted with partial steam injection of high-pressure steam in the combustor and the high-pressure compressor discharge. Steam injection increased the power output from 29.9 MW<sub>e</sub> to 41.9 MW<sub>e</sub> and the efficiency increased from 36.0 to 41.8% and the NO<sub>x</sub> levels were below 25 ppmvd (parts per million by volume, dry, 15% oxygen) [103]. The LM1600, LM2500 and LM5000 gas turbines have been converted to STIG versions. These systems have higher steam injection rates than the first STIG system, since low-pressure steam is injected before the gas generator<sup>2</sup> low-pressure turbine in addition to the high-pressure steam injection [104]. Peltier and Swanekamp [105] reported on operating experiences with a LM5000 STIG cogeneration plant. The net power output was sold to the grid and the steam was used by a food products company. The plant was in operation only during the weekday summer months and then the plant was started and stopped each day. The reliability and availability of the plant was similar to other aeroderivative gas turbines for base-load cogeneration. Noymer and Wilson [106] presented a model for evaluating steam injection in an aeroderivative gas turbine with a two-spool gas generator and a free power turbine. Steam injection before the high-pressure, low-pressure and

<sup>2</sup> The gas generator consists of the combustor and the turbines that drive the compressors, while the net power output of the gas turbine is generated by a separate power turbine.
power turbines were investigated, and the highest increase of power and efficiency was found for steam injection into the high-pressure turbine.

4.1.3. The Aquarius cycle

The Ukrainian company Mashproekt offers the Aquarius system, a steam-injected gas turbine with water recovery. The system is developed for power generation and mechanical drive. The water in the exhaust gas is recovered in a direct contact condenser, where water is sprayed over a grid. The water used in the condenser is cooled in an air-cooled heat exchanger and the recovered water is cleaned and re-used in the cycle. The water recovery system increases the plant price by 15–20% compared with conventional steam-injected gas turbines [107]. An Aquarius-25 test rig, in operation since 1995, has accumulated 12,000 h of operation [108]. Steam was injected in the combustor for NO\(_x\) suppression and after the combustor for power augmentation. For cooling water temperatures below 30 °C, the plant was self-supporting with water. The water quality was satisfactorily and no fouling of the gas turbine could be detected. An Aquarius-16 mechanical drive unit has been factory tested for 1600 h between 1998 and 2000. Gas turbines with partial and full steam injection without water recovery are also available [109]. An Aquarius-16 was scheduled for installation in 2001 at a natural gas compression station [109]; however, there are no operating experiences from this plant available in literature.

4.1.4. The IHI-FLECS

Ishikawajima-Harima Heavy Industries (IHI) manufactures a steam-injected cogeneration system based on the Allison 501-KH5 gas turbine called IM400 IHI-FLECS (FLexible Electric Cogeneration System). In comparison to the Cheng cycle based on the same gas turbine, this system includes an ejector that mixes saturated steam with a part of the compressed air and injects the mixture in the combustor. In the steam-air mixture, the steam is superheated by 40 °C due to the reduced partial pressure and there is no risk of water droplets in the combustor. By excluding the superheater, more steam can be generated, the system can respond faster to changes in steam demand and the steam temperature is lower than the compressor outlet temperature, thus decreasing the turbine cooling air temperature. There is also a configuration called TRI-FLECS that produces compressed air in addition to power and steam. Eleven IM400 IHI-FLECS and three IM400 TRI-FLECS plants are in commercial operation and three more IM400 IHI-FLECS plants were under construction in 1999 [110].

4.2. Proposed steam-injected gas turbines

4.2.1. Steam-injected gas turbines for power generation

Modified steam-injected gas turbines cycles including intercoolers, recuperators, reheat or topping steam turbines (where the steam is expanded before injection in the gas turbine) have been investigated in many studies.

4.2.1.1. Steam-injected gas turbines with intercooling. One possible modification of the basic steam-injected gas turbine is intercooling, which was studied by Larson and Williams [111] for the LM5000 STIG for use in small central power plants. The efficiency of this cycle could be 47–48% (HHV, about
52–53% LHV\(^3\)) and the power output 110 MW\(_e\), as estimated by GE. The intercooled STIG could compete with combined cycles in this size range regarding investment cost and cost of electricity; in the late 1980s, the best combined cycle generated about 250 MW\(_e\) at 40% efficiency (HHV, about 44% LHV). Macchi et al. \([112]\) investigated intercooled steam-injected aeroderivative gas turbines. A triple-pressure HRSG was considered: high-pressure steam was injected in the combustor and the intermediate-pressure and low-pressure steam was injected into the gas turbine. The investigated cycle included a topping steam turbine that expanded the steam before injection in the combustor when the highest HRSG pressure was higher than the injection pressure. A surface intercooled steam-injected gas turbine had an efficiency slightly above 51% (PR = 36, TIT = 1250 °C) or slightly above 53% (PR = 45, TIT = 1500 °C). At high pressure ratios, the optimum high-pressure steam evaporation pressure was below the pressure required for steam injection and the topping steam turbine was not beneficial. At the optimum pressure ratio for the low-pressure compressor, there were no efficiency differences between cycles with surface and evaporative intercooling. If reheat was included, the efficiency of an intercooled steam-injected gas turbine could increase by 3% points and the specific power output could be doubled. According to Macchi et al. \([112]\), in the end of the 1980s, GE investigated an intercooled steam-injected LM8000 gas turbine, for which an efficiency of 52% (PR = 34, TIT = 1371 °C) was predicted; however, this cycle was not commercialized.

4.2.1.2. Steam-injected gas turbines with reheat. Gas turbine reheat is another possible modification of steam-injected gas turbines. Fischer et al. \([113]\) studied steam injection for a gas turbine with sequential combustion and a simple cycle efficiency of 38.5% and power output of 268 MW\(_e\) (PR = 30, TIT\(_{hp\ exp}\) = 1235 °C). The turbine cooling air was cooled by heating the steam for injection and by indirect cooling by water in surface heat exchangers or by injection of water. Partial steam injection (i.e. only a part of the steam that can be generated in the HRSG is injected into the gas turbine) could be used for existing gas turbines without redesign to increase the power output by more than 30% and the gross thermal efficiency by about 14%. Cycles with full steam injection had net efficiencies of 47.3% when the turbine cooling air was indirectly cooled and efficiencies of 50.2% when the turbine cooling air was cooled by water injection. When steam was injected and the compressor size was kept constant, the turbine size increased: the blade height of the last turbine stage in the low-pressure turbine increased by about 33%. This requires extensive redesign, thus the authors recommended keeping the turbine size constant and using a smaller compressor.

4.2.1.3. Steam-injected gas turbines with intercooling, recuperation and reheat. Intercooling can also be combined with recuperation and reheat. Annerwall and Svedberg \([74]\) investigated steam-injected gas turbines with these modifications. A basic steam-injected gas turbine had an efficiency of 44.3%. Application of only reheat, only intercooling or reheat and intercooling reduced the efficiency, while application of recuperation or recuperation and reheat increased the efficiency. Maximum efficiency (49.0%) was reached for intercooling, reheat and recuperation combined. However, both a recuperated water-injected cycle and a combined cycle had higher efficiencies. Additional information about this study can be found in Section 3.3. De Paepe and Dick \([115]\) found that a basic steam-injected gas turbine

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\(^3\) Efficiencies based on the higher heating value (HHV) of the fuel have been converted to LHV efficiencies by multiplying the HHV efficiency with the factors 1.11 for natural gas \([91]\), 1.06 for liquid fuels \([91]\) and 1.04 for coal \([114]\).
had an efficiency of 49% (PR = 30, TIT = 1250 °C), with steam blade cooling, the efficiency increased to 52% (TIT = 1400 °C) or 51% (TIT = 1250 °C). Reheat, intercooling or intercooling and recuperation of the steam-injected cycle increased the efficiency slightly, although not enough to justify the more complex cycle; however, the specific power output increased significantly. A combined cycle had an efficiency of 58.5% and an intercooled recuperated gas turbine had an efficiency of 48%. Component data corresponding to large axial gas turbines were used in the simulations.

4.2.1.4. Steam-injected gas turbines with a topping steam turbine. Systems with a topping steam turbine have also been proposed. In such systems, the steam is generated at a high pressure and expanded in a steam turbine before injection in the gas turbine. Fraize and Kinney [116] suggested a steam-injected gas turbine with a non-condensing bottoming cycle for utility power production. In this system, steam was generated from the exhaust gas energy, expanded in a steam turbine, reheated by exhaust gas and injected in the gas turbine. This system had a thermal efficiency of about 56% (PR = 20, TIT = 1427 °C), a combined cycle had an efficiency of 57% and a simple steam-injected cycle had an efficiency of 52%. The turbine cooling flow was 5% of the compressor mass flow rate. Brown and Cohn [117] found that a simple steam-injected gas turbine had a net efficiency of 40.5% (HHV, about 43% LHV) and a power output of 127 MW, a steam-injected cycle with a non-condensing bottoming cycle had an efficiency of 40.9% (HHV, about 43% LHV) and a power output of 152 MW and a combined cycle had an efficiency of 42.5% (HHV, about 45% LHV) and a power output of 476 MW (four gas turbine units). A recuperated steam-injected gas turbine had an efficiency 1% point higher than the steam-injected cycle with a topping steam turbine. However, the specific work was reduced and the cost of the recuperator was therefore not justified. The study was based on distillate fuel-fired heavy-duty industrial gas turbines (TIT = 1204 °C). The specific investment cost was higher for the combined cycle than for the steam-injected cycles, while the cost of electricity was the lowest for the combined cycle, slightly higher for the steam-injected cycle and highest for the simple cycle. Manfrida and Bosio [118] found higher efficiencies for combined cycles than for steam-injected gas turbines. The authors suggested a steam-injected combined cycle to increase the efficiency so that the steam-injected cycle could compete with the combined cycle for utility power generation. This cycle had a dual-pressure HRSG, the high-pressure steam was first expanded in a steam turbine, then it was mixed with low-pressure steam from the HRSG and one part of this stream was injected in the gas turbine. The other part was expanded in a condensing low-pressure steam turbine after which it was fully condensed and re-used in the cycle. Frutschi and Plancherel [67] compared combined cycles to gas turbines with water or steam injection. An efficiency of almost 45% for a simple steam-injected gas turbine was possible. A recuperator increased the efficiency, although the specific power output was reduced. The combined steam-injected cycle and the turbo steam-injected cycle could reach efficiencies of 50%. The turbo steam-injected cycle was suggested by Foster-Pegg [119], who called the system the Turbo-STIG cycle. Both these cycles had dual-pressure HRSGs. In the combined steam-injected cycle, the high-pressure steam was expanded in a backpressure turbine before injection in the gas turbine and the low-pressure steam was injected into the combustion chamber. In the Turbo-STIG, the low-pressure steam was added to the combustor. The high-pressure steam was used to drive a steam backpressure turbine, which in turn drove a compressor that compressed the air further before the gas turbine combustor. The expanded steam was injected in the gas turbine. The authors concluded that the combined cycle was the best available alternative for power generation and cogeneration, since it was a proven technology. Foster-Pegg [119] stated that the advantage of the Turbo-STIG was that the additional air compressor increased the pressure ratio of
the gas turbine and all the steam generated from an unfired HRSG could be injected in the gas turbine without a decrease of the surge margin. Compared with other gas turbines with full steam injection, the Turbo-STIG thus required less extensive design changes. Foster-Pegg calculated the performance of the Turbo-STIG for five commercial single-shaft gas turbines; the efficiencies ranged from 40.7 to 46.7%, an increase of 38–46% compared with the simple cycle gas turbines, and the power outputs increased by 72–123% compared with the simple cycles. Foster-Pegg claimed that Turbo-STIG cycles should have similar electrical efficiencies as unfired combined cycles, but higher power outputs. A combined cycle that is supplementary fired to achieve the same power output as a Turbo-STIG cycle would have a lower efficiency than the Turbo-STIG cycle.

4.2.1.5. Steam-injected gas turbines with a topping steam turbine and reheat. A topping steam turbine can be combined with sequential combustion. Hofstädter et al. [120] investigated steam-injected gas turbines with sequential combustion. The cycles included a backpressure steam turbine through which steam was expanded before injection in the gas turbine to increase the cycle efficiency. After the turbine, some of the steam was heated by cooling the turbine cooling air before injection. With a supercritical boiler pressure, this system had a thermal efficiency of 55.9% (PR = 40, TIT = 1300 °C, 1394 kJ/kg). A simple steam-injected cycle had an efficiency of 54.5% (PR = 40, TIT = 1300 °C, 1147 kJ/kg). The steam exiting the backpressure turbine could also be split in two parts: one part was reheated in the HRSG and the other part was heated by the high-pressure cooling air before injection; this cycle had an efficiency of 56.8% (PR = 40, TIT = 1300 °C, 1096 kJ/kg). A reference combined cycle had an efficiency of 58.3% (700 kJ/kg). The authors concluded that the proposed cycles had a potential for peak- and medium-load power generation due to the high specific power output and low investment cost compared with a combined cycle, while the combined cycle was more feasible for base-load due to its higher efficiency. Rice [36] analyzed different steam-injected cycles for varying steam injection rates, turbine inlet temperatures and pressure ratios, to determine the power output for a certain amount of steam injection compared with using the steam in a condensing steam turbine. For steam-injected gas turbines with a topping steam turbine and steam-injected reheat gas turbines, the steam injection rate for a certain power output was lower than for simple steam-injected gas turbines. For steam-injected reheat gas turbines, a topping steam turbine halved the steam rate for a certain power output compared with a condensing steam turbine. Steam-injected intercooled reheat gas turbines were also discussed. The efficiency of the power generation by steam injection when the gas turbine exhaust gas energy was used to generate the steam for injection was considered as well [121]. The efficiency for power generation by injecting steam in a gas turbine was higher than for expanding the steam in a condensing steam turbine. For reheat steam-injected gas turbines, a topping steam turbine increased the efficiency. Rice [122] analyzed the same steam-injected gas turbine cycles with injection of all the steam generated in the HRSG. Full steam injection increased the power output and efficiency. The largest gains of steam injection were found for gas turbines with high pressure ratios and turbine inlet temperatures and the reheat gas turbine had the highest potential for full steam injection.

4.2.1.6. Other alternative steam-injected gas turbines. There are other possible modifications of steam-injected gas turbines in addition to those discussed above. For example, Ågren et al. [123] presented a steam-injected gas turbine where a steam injector was used to inject the steam in the combustor, thereby raising the gas pressure and reducing the irreversibility resulting from injecting high-pressure steam into gas with a lower pressure. It was possible to increase the power output by 1.5% and the efficiency by
0.6% points compared with a steam-injected gas turbine without an injector. Ubertini and Lunghi [124] investigated a steam-injected gas turbine combined with a molten carbonate fuel cell (MCFC) for medium-to-large-scale power plants. The waste heat (650 °C) from the fuel cell stack, that had a net power output of 14.8 MW\(_e\) and an efficiency of 59%, was indirectly transferred to the gas turbine since the fuel cell operated at atmospheric pressure. For an unfired gas turbine cycle, the total electrical efficiency was 67% (PR = 13) without steam injection and 69% (PR = 10) with steam injection. With a combustor included in the steam-injected gas turbine cycle, the total electrical efficiency was 66% (PR = 10, TIT = 1200 °C). The gas turbine increased the power output of the system by 2 MW\(_e\) for the unfired cycle and by 6 MW\(_e\) for the fired cycle.

### 4.2.1.7. Gas turbines with injection of saturated steam

Gas turbines with injection of saturated steam have also been suggested. Wei [125] presented a version of the aeroderivative gas turbine Trent with injection of saturated steam in the combustor. Excluding the superheater increased the steam mass flow rate and the power output, but decreased the efficiency compared with a cycle with injection of superheated steam. The steam-injected Trent had an efficiency of over 50%, which was slightly lower (3–4% points) than for a combined cycle. The specific investment cost was 30% lower than for a combined cycle, while the operational flexibility was the same as for a simple cycle. Mechanical Technology, Inc., developed a saturated steam-injected version of the Allison 501-KB for industrial cogeneration, and one such gas turbine was installed at a General Motor plant in 1985. Without steam injection, the power output was 3 MW\(_e\); with steam injection of 2.27 kg/s the power output was raised to 4 MW\(_e\) and the efficiency increased by 20%. Saturated steam (18.25 bars) is often used in cogeneration systems; therefore, this steam quality was used for injection although this penalized the efficiency compared with injection of superheated steam.

### 4.2.2. Steam-injected gas turbines for cogeneration of power and steam and/or hot water

#### 4.2.2.1. Steam-injected gas turbines for cogeneration of power and district heating and district cooling

Frutschi and Wettstein [126] compared steam-injected gas turbines with combined cycles for large-scale cogeneration of power and district heating, both cycle alternatives had dual-pressure boilers. The district heating temperatures were 70/120 °C and for maximum cogeneration, all the generated steam was used for heat production and no steam was injected in the gas turbine. For cogeneration applications, the authors concluded that steam-injected gas turbines only had an advantage compared with simple cycle gas turbines with an HRSG and not compared with combined cycles, since the combined cycle had a higher power output and electrical efficiency than the steam-injected cycle. Krause et al. [127] investigated the economics of a 501-KH5 Cheng cycle for small-scale cogeneration of power and district heating. The steam-injected gas turbine was flexible and could follow the thermal demand curve; hence, a Cheng cycle could be cost effective for this application. The district heating water was preheated by the exhaust gas and further heated by saturated steam. The remaining steam was superheated and injected into the gas turbine. The simulation model was compared with performance data from a Cheng cycle at a German university.

The heat recovery for district heating generation can be improved by flue gas condensation. This was investigated by Annerwall [128], who studied small-scale cogeneration of power and district heating. By cooling the exhaust gas from the steam-injected gas turbine, indirectly or by water injection, to 50 °C, 75% of the injected steam could be recovered at full-load and maximum steam injection and...
the recovered heat could be used in a low-temperature (50–60 °C) district heating network. The recovered energy can be upgraded by a heat pump to be used for district heating, as investigated by Lundberg [129], who considered mechanical vapor compression and absorption heat pumps for this purpose. The net total efficiency of the mid-size steam-injected gas turbine with or without intercooling was 80–82% (HHV), while a combined cycle had a total efficiency of 74%. However, the combined cycle had a higher electrical efficiency than the non-intercooled steam-injected cycle.

Steam-injected gas turbines have been considered for cogeneration of power and district heating and cooling, for example for office buildings and hotels [130]. The cogeneration system included a vapor compression refrigerator and an absorption chiller. Steam for injection, space heating and driving the absorption chiller was generated in the supplementary-fired HRSG and auxiliary boilers. Compared with a cogeneration system with simple cycle gas turbines, the steam-injected gas turbine had a higher efficiency and was more flexible, thus reducing the operational cost. Pak and Suzuki [131] considered modifications for increased electrical efficiency of small-scale (about 10 MWₑ) gas turbine-based cogeneration systems for power and district heating and cooling. Steam injection resulted in a system with a high electrical efficiency for low heat demands, a very low heat output for maximum steam injection and a high flexibility.

4.2.2.2. Steam-injected gas turbines for cogeneration of power and process steam. Steam-injected cycles have been considered for cogeneration of power and process steam, for example by Baken and van den Haspel [132], who presented a design and operation optimization method. For the investigated case (i.e. the Netherlands in the late 1980s), steam-injected gas turbines should be operated as peak shaving units, since the electrical and total efficiencies decrease when steam must be generated by supplemental firing to cover the demands for injection and process steam. Bartolini et al. [133] presented a model, validated with data from an existing 501-KH5 Cheng cogeneration unit, for the off-design performance of steam-injected gas turbines.

4.2.2.3. Modified steam-injected gas turbines for cogeneration. Steam-injected gas turbines with various cycle modifications can be used for cogeneration. Furuhata et al. [134] proposed a steam-injected reheat gas turbine for small-scale cogeneration. Hot water (75 °C) was produced in a condensing heat exchanger. Since all the available steam was injected in the gas turbine, combustion instabilities in the low-pressure combustor due to low levels of oxygen were possible. Experiments showed that a temperature above 1100 K and an oxygen concentration above 6% (molar percentage) were needed to achieve stable combustion. At a power output of 1 MWₑ, the electrical efficiency was 40.2% (HHV, 44.6% LHV) and the total efficiency 97.5% (LHV); at 7 MWₑ the efficiency was 44.3% (HHV, 49.2% LHV) and the total efficiency 96.3% (PR = 20, TIT = 1200 °C). Hisazumi et al. [135] presented a similar small-scale system with reheat, steam injection and intercooling that generated power and steam (process steam or for driving an absorption chiller). This cycle had 5% points (HHV) higher efficiency than a 501-KH5 Cheng cycle. Chodkiewicz et al. [136] investigated intercooled steam-injected gas turbines where the turbine expansion continued below the ambient pressure, the water in the exhaust gas was recovered by condensation and hot water was generated. The remaining non-condensable gases were compressed to ambient pressure before the stack. The cycle could include a solid fuel-fired boiler for steam generation. In this cycle, the HRSG preheated the feedwater before the boiler and the generated steam was expanded in a topping steam turbine before injection in the gas turbine. The electrical efficiency was about 47% (TIT = 1300 °C, PR = 20) for a system without a boiler, while a cycle
similar to the Cheng cycle had an efficiency of 43.5% (TIT = 1300 °C, PR = 30). Korakianitis et al. [137] investigated GE LM5000 combined cycles for cogeneration of power and hot water for district heating with supplemental firing and steam injection. Flue gas condensation recovered the injected water and preheated the steam cycle feed water and the district heating water. The electrical efficiency of the plant was about 50%. Steam injection increased the electrical efficiency and power output, while supplemental firing increased the power output and decreased the electrical and total efficiencies. Bartlett et al. [138] studied mid-size cogeneration of power and district heating from waste and natural gas in so-called hybrid cycles. In the steam-injected hybrid cycle, steam was generated from a waste incinerator and the gas turbine exhaust gas, superheated by the gas turbine exhaust gas and expanded in a topping steam turbine. The steam was re-superheated and injected in the gas turbine. Saturated steam was used for blade and rotor cooling and a flue gas condenser recovered the injected steam and generated district heating. The hybrid steam-injected cycle had an electrical efficiency of 43.5%, while a hybrid combined cycle had an efficiency of 40.2%. The hybrid steam-injected cycle had a higher total efficiency than a hybrid combined cycle, a waste-fired steam cycle and a natural gas-fired combined cycle. The steam-injected cycle was the most viable for cogeneration of power and district heating due to its high electrical and total efficiencies and low specific investment cost.

4.2.3. Solid fuel-fired steam-injected gas turbines

Externally fired and syngas-fired steam-injected cycles have been proposed. Larson and Williams [139] discussed biomass-fired steam-injected gas turbines for cogeneration. In a direct-fired system, the combustion gases were used directly in the gas turbine after cleaning; however, extensive development work remained for this system. In indirectly fired systems, the turbine working fluid was indirectly heated in an atmospheric biomass combustor. The gas turbine could also be fired with gasified biomass, and such systems were deemed the most technically and economically promising of the three systems discussed in the paper. Stahl and Perugi [140] patented an externally fired gas turbine where the gas turbine exhaust gas was used to combust solid fuel in a combustor that externally heated the working fluid in the gas turbine. The combustor exhaust gas was used to generate steam for injection in the gas turbine before the external heat exchanger. Injection of water before the external heat exchanger was also suggested.

5. Evaporative gas turbines

In the 1980s, the interest in gas turbine cycles with humidification towers increased and a research program on the HAT, the evaporative cycle patented by Fluor Daniel [49], was commenced in the USA. In the beginning of the 1990s, another research program on EvGT was initiated in Sweden, the EvGT (evaporative gas turbine) project. There are many publications in the area of EvGT; therefore, the authors have divided the cited papers into sections. The HAT and EvGT projects are described first, then follows other publications in the area, and in each section, the publications have been divided into those on natural gas-fired and solid fuel-fired cycles.

5.1. The humid air turbine (HAT) research program

The HAT research program involved the Electric Power Research Institute (EPRI), consultant companies, gas turbine manufacturers and utility companies. The program investigated different HAT
cycle configurations integrated with coal gasification (IGHAT) or fueled with natural gas and an aeroderivative gas turbine for humid air operation.

5.1.1. Natural gas-fired HAT cycles

The HAT cycle that was patented by Rao [49] has the same configuration as shown in Fig. 2, except that the compression is intercooled and the water outlet stream from the tower may be cooled by external cooling water. Versions with second intercooling and aftercooling steps with external cooling were also patented. Humidification of the gas turbine working fluid results in an increased volumetric flow rate in the expander compared with the compressor, and a flow mismatch results when an existing gas turbine is used in a HAT cycle. Rao [141] patented several versions of the HAT cycle where some of the humid air was expanded in a separate turbine to avoid this problem. In some versions, an external waste heat source was used to heat additional water for the humidification tower. However, in the subsequent studies on the HAT cycle, the cycle layout shown in Fig. 2 was used.

EPRI [142] presented a report on HAT and combined cycles IGHAT or fired with natural gas. Two gas turbine manufacturers, GE and Asea Brown Boveri (ABB), supplied data for the gas turbines used in the study. For natural gas, the GE HAT cycle had an efficiency of 53.5% while the triple-pressure reheat combined cycle efficiency was 49.5% and the ABB HAT cycle efficiency was 57.4% while the combined cycle efficiency was 53.4%. The gas turbine firing temperature was 1260 °C in all cases (PR = 23–24). The cycles had power outputs between 200 MW_e and 270 MW_e. Compared with the combined cycles, the HAT cycles had higher part-load efficiencies and were less sensitive to high ambient temperatures. The water consumption was lower for the HAT cycles when the combined cycles used wet cooling towers; however, the water quality requirements were higher for the HAT cycle than for a wet cooling tower according to the authors of the report. The specific investment cost for a HAT cycle was higher (11–26%) than for a combined cycle; however, the higher efficiency resulted in slightly lower costs of electricity for the HAT cycles.

Previous work had shown that the optimal HAT would have a high pressure ratio (35 ± 5) and the turbine outlet mass flow rate would be 20–30% greater than the compressor inlet flow. Therefore, an aeroderivative gas turbine, based on the Pratt and Whitney PW 4000 aircraft engine, was investigated for operation in the HAT cycle, since aeroderivative gas turbines have high pressure ratios and can comply with higher flow rate variations than industrial gas turbines [143]. EPRI [144] presented a report on an IGHAT based on the proposed gas turbine FT 4000 HAT. The turbine was designed for fueling with syngas and was modified for natural gas operation as well. With natural gas, the HAT efficiency was 55.5% and the power output was 157 MW_e (PR = 37), while for a triple-pressure reheat combined cycle, the efficiency was 52.6% and the power output 201 MW_e. However, the HAT specific investment cost was higher than for the combined cycle (3%), resulting in an only slightly lower cost of electricity for the HAT. It was estimated that the development cost for the FT 4000 HAT would be above 200 million USD (1992 USD). Day and Rao [145] presented results for a redesigned natural gas-fired FT 4000 HAT that had a net power output of 200 MW_e and a net efficiency of 55.4% (PR = 40, TIT = 1320 °C), while an advanced combined cycle had an output of 201 MW_e and an efficiency of 52.6% for an ambient temperature of 21.5 °C. The HAT cycle had superior part-load performance compared with the combined cycle: the heat rate was 5% lower at full-load and 20% lower at 50% load for the HAT cycle.

HAT cycles combined with fuel cells have also been investigated. Rao and Samuelsen [146] presented a HAT integrated with a solid oxide fuel cell (SOFC) with a thermal efficiency of 69.1% (PR = 15) for small-scale distributed power generation. A combination of a HAT cycle and two fuel cells...
had an efficiency of 76.0% (PR = 15). A system without a humidification tower (SOFC-GT) had an efficiency of 66.2% (PR = 9). In these systems, the air from the intercooled compressor was used in the fuel cell after recuperation with the gas turbine exhaust gas, the fuel cell exhaust gas was expanded in the high-pressure turbine and additional fuel was burned in a reheat combustor before the low-pressure expander. Both the specific investment cost and the cost of electricity were 4% lower for the single SOFC-HAT than for the SOFC-GT, while the dual SOFC-HAT was more expensive. Rao et al. [147] presented integrated systems with power outputs of about 300 MWₑ with humid air gas turbines and SOFCs, of the same type as the systems by Rao and Samuelsen [146] but without the reheat combustor. These could reach efficiencies of 75% based on natural gas.

5.1.1.1. Cascaded humidified advanced turbine (CHAT). The CHAT (cascaded humidified advanced turbine) was proposed to solve the problem of flow mismatch between the compressor and turbine in the HAT. The CHAT could be based on standard gas turbine components in order to reduce the cycle development cost. Compared with a combined cycle, the CHAT may have higher part-load efficiency, lower specific investment cost, faster startup, improved part-load performance and retains its power output at high ambient temperatures. This intercooled, humidified, recuperated and reheated cycle has two shafts: one power-generation shaft with a low-pressure compressor and expander and one power-balanced shaft with intermediate-pressure and high-pressure compressors and expanders. Compared with a simple cycle gas turbine, the work input to the low-pressure compressor is reduced, since the high-pressure expander pressure ratio is lower than the total compressor pressure ratio on the power-balanced shaft. Hence, the intermediate-pressure and high-pressure compressors supply most of the required compression work. Several large-scale CHAT designs, presented in Table 2, based on existing heavy-duty gas turbines integrated with a high-pressure shaft with industrial compressors and expander have been proposed.

For the CHAT design with an efficiency of 55.5% in Table 2, the specific investment cost was estimated to be 10–15% lower than for a combined cycle at ISO conditions and 20–25% lower at 35 °C ambient temperature [148]. Nakhamkin and Gulen [150] presented a transient analysis of the first CHAT cycle design. The plant’s dynamic response to startup from cold conditions, sudden load changes and shutdowns were studied. It was found that the CHAT cycle was comparable to a simple cycle gas turbine in operation flexibility at startup. Further studies of the CHAT have shown higher power outputs and electrical efficiencies than combined cycles based on commercially available gas turbines for mid-size power generation (30–150 MWₑ). For a GE Frame 6FA core engine, a combined cycle had a power

Table 2
Data for some of the proposed CHAT designs based on heavy-duty gas turbines

<table>
<thead>
<tr>
<th>P (MWₑ)</th>
<th>ηₑ (%)</th>
<th>pₑₘₐₓ (bar)</th>
<th>TITₑₚₑₙₑxp (°C)</th>
<th>TITₑₜₑₚₑₙₑxp (°C)</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>288</td>
<td>51.3</td>
<td>53</td>
<td>1350</td>
<td>760</td>
<td>Ref. [148] (The first CHAT cycle presented.)</td>
</tr>
<tr>
<td>317</td>
<td>54.7</td>
<td>79</td>
<td>1394</td>
<td>871</td>
<td>[149]</td>
</tr>
<tr>
<td>316</td>
<td>55.5</td>
<td>68</td>
<td>1394</td>
<td>871</td>
<td>[148]</td>
</tr>
<tr>
<td>354</td>
<td>59.2</td>
<td>67</td>
<td>1394</td>
<td>1149</td>
<td>[148]</td>
</tr>
<tr>
<td>N/A</td>
<td>63</td>
<td>N/A</td>
<td>1482</td>
<td>1149</td>
<td>[148]</td>
</tr>
</tbody>
</table>
output of 107.4 MWₑ and an efficiency of 53.0%, while a CHAT cycle had a power output of 143.5 MWₑ and an efficiency of 54.9% \( (p_{\text{max}} = 68 \text{ bar}, \text{TIT}_{\text{lp exp}} = 1374 ^{\circ} \text{C}, \text{TIT}_{\text{hp exp}} = 871 ^{\circ} \text{C}) \) [151].

The CHAT cycle is suited for distributed generation because of fast startup, high efficiency at full- and part-load, low power and efficiency degradation due to ambient temperature and site elevation, and competitive cost. A CHAT demonstration plant based on the Rolls-Royce Allison 501-KB7 gas turbine, with a power output of 12.1 MWₑ and a net efficiency of 46.4% \( (p_{\text{max}} = 60 \text{ bar}, \text{TIT}_{\text{lp exp}} = 1102 ^{\circ} \text{C}, \text{TIT}_{\text{hp exp}} = 816 ^{\circ} \text{C}) \), has been proposed for distributed generation [152]. Gaul [153] proposed an 11 MWₑ CHAT design, also based on the 501-KB7, for distributed generation. In this CHAT cycle, the second shaft was not power-balanced to avoid costly modifications to the gas turbine compressor.

5.1.1.2. Compressed air storage with humidification (CASH). In the recuperated and reheated CASH (compressed air storage with humidification) cycle, compressed air produced at off-peak periods for a low cost is stored in, for example, an underground cavern, for power generation during periods with high electricity prices. There exist two non-humidified compressed air energy storage (CAES) plants: a 290 MWₑ plant in Germany and an intercooled, recuperated and reheated 110 MWₑ plant in the USA [154]. In the CASH cycle, the stored air is extracted and humidified in a tower by hot water heated in the compressor intercooler and aftercooler (the water is stored in a tank) and by exhaust gas. Power generated by the turbines drives the compressors via an electric motor. When the power demand is high, none of the power generated by the turbines is used for compression and all the generated power can be delivered to the grid, resulting in a plant with a low specific cost. An additional advantage of the CASH cycle is that the flow mismatch of the HAT is avoided [155]. An economic evaluation of a 530 MWₑ CASH plant showed that at an ambient temperature of 32 °C, when cycling power is most needed, the plant specific investment cost was 405 USD/kWₑ, compared with 403 USD/kWₑ for a simple cycle and 489 USD/kWₑ for a combined cycle. At ISO conditions, the difference in specific investment cost between the CASH system and a combined cycle was smaller. The CASH plant had lower overall costs than the simple cycle for operation of over 400 h/year and lower overall costs than a combined cycle for operation of less than 4000 h/year [154].

5.1.1.3. Combustion turbine humidified air injection (CTHAI). Nakhamkin et al. [156] proposed the CTHAI (combustion turbine humidified air injection) system, which is to be used for augmenting the power output of gas turbines at peak demand periods. Air is compressed by a motor-driven compressor and humidified in a humidification tower by water heated by the gas turbine exhaust gas. The humid air is heated by the gas turbine exhaust gas in a recuperator and injected before the gas turbine combustor. Average industrial-quality water can be used. Injection of humid air could increase the power output by 20% at 35 °C ambient temperature and reduce the heat rate by 8–15%, to a turnkey cost of 180–220 USD/kWₑ [157]. The system could also be used for retrofit of combined cycles to increase the power output. Nakhamkin et al. [158] summarized the development of the CTHAI technology and suggested the use of different types of boilers for generating steam to humidify the air instead of a humidification tower.

5.1.2. Solid fuel-fired HAT cycles

5.1.2.1. Integrated gasification HAT cycles. EPRI [142] presented a report on HAT cycles fueled with gasified coal or natural gas. The IGHAT was predicted to have about 2–3% (1% point) higher efficiency, about 20% lower specific investment cost, decreased operation and fixed maintenance costs and a 15%
lower cost of electricity compared with a combined cycle integrated with coal gasification (IGCC). The cycle electrical efficiencies were slightly above 40% and the power outputs were in the range of 460–600 MWₑ. The IGHAT investment cost was lower since it could recover low-temperature energy in the form of hot water, used in the humidification tower, from an inexpensive quench-cooled gasifier, while the IGCC required expensive syngas coolers for generation of steam. The variable operation cost was higher for the IGHAT since it required demineralized makeup water. The IGHAT had an improved part-load performance compared with the IGCC, since the IGHAT had a twin-shaft gas turbine and was regenerative. The IGHAT was also less sensitive to high ambient temperatures than the IGCC. The study was based on gas turbine data supplied by GE and ABB. EPRI [144] presented a report on a 410 MWₑ IGHAT based on two FT 4000 HAT gas turbines. The design calculations showed that the IGHAT had approximately the same efficiency (about 42.5%) as a 502 MWₑ IGCC based on two industrial gas turbines, although the specific investment cost was 11% lower and the cost of electricity was 8% lower. The same gasification technology as in the 1991 EPRI report was used. A CHAT cycle integrated with coal gasification (IGCHAT) has also been proposed. Compared with an IGCC, the exclusion of the steam bottoming cycle reduced the investment cost in the order of 100 USD/kWₑ. In addition, the IGCHAT could use low-temperature energy; hence, it could use a low-cost quench gasifier that further reduced the investment cost [159].

For intermediate-load cycling power generation, a CASH cycle with integrated coal gasification (IGCASH) has been designed. This system was estimated to save 20–40% in investment cost per peak kilowatt of capacity compared with an IGCC. The advantage of an IGCASH system was a smaller and less expensive gasifier for a given power output [155]. In addition, it was estimated that the IGCASH would save 15–20% in investment cost per peak kilowatt of capacity compared with a standard cycling pulverized-coal plant with flue gas scrubbing [143]. The IGCASH could be integrated with a natural gas-fired CASH cycle in a compressed air storage with humidification integrated with natural gas (CASHING) plant. The natural gas-fired system includes a turbine generator, saturator, recuperator and a heat exchanger for heating water, but not an additional compressor system or cavern. The CASHING plant could be used for peak- and intermediate-load operation [155]. The CASHING system should have lower investment cost and higher flexibility than an IGCASH plant [143].

5.1.2.2. Closed and externally coal-fired HAT cycles. Rao et al. [160] presented a closed intercooled, aftercooled and recuperated HAT cycle. External cooling water cooled the working fluid (helium) in a precooler before water separation and the recovered water was re-used in the cycle. The heat source for the cycle could be nuclear, solar or solid fuels. The thermal efficiency of the closed HAT cycle was 6% (2.3% points) higher than for a dry recuperated intercooled closed cycle and the specific power output was 37% higher.

Robson and Seery [161] presented the HIPPS (high performance power system) system, which was a combined cycle integrated with a coal-fired high-temperature advanced furnace (HITAF). The furnace preheated the compressor discharge air before a natural gas-fired topping combustor. Steam for the bottoming cycle was generated in the furnace and from the gas turbine exhaust gas. With over two-thirds of the energy input in coal, the efficiency was 47% (HHV, about 50% LHV). A HIPPS system with an intercooled FT 4000 gas turbine (i.e. without humidification) could reach an efficiency of 53% (HHV, about 57% LHV) when coal accounted for 55% of the fuel energy input. If the FT 4000 HAT gas turbine
was used in a HAT/HIPPS system (without a steam turbine), the efficiency could be 54.3% (HHV, about 58%) and the power output 352 MW\textsubscript{e} for a system where 53% of the heat input was from coal.

5.2. The EvGT project and related work

A research program on EvGT was initiated in Sweden in the beginning of the 1990s. The program involves utility companies, gas turbine manufacturers, research organizations and universities. The objectives of this program are to demonstrate the evaporative gas turbine technology in a pilot plant, investigate the humidification process and the water circuit chemistry, and to propose future plant designs. An evaporative gas turbine pilot plant has been constructed and different small-to-mid-scale cycle configurations and applications have been investigated. A commercial demonstration plant is under consideration [162]. Ågren [163], Rosén [164], Lindquist [165] and Bartlett [166] presented some of the work within the project. All the studies cited in this section were not directly financed by the EvGT project; nevertheless, they are connected to this project.

5.2.1. Natural gas-fired EvGT cycles

In one study, a mid-size (70–80 MW\textsubscript{e}) intercooled evaporative cycle was investigated for generation of power only or power and district heating [167]. For power generation only, the evaporative cycles had electrical efficiencies on the same level as combined cycles with dual-pressure HRSGs, about 55%. The specific investment cost was about 20% lower and the cost of electricity was 5% lower for the evaporative cycle compared with the combined cycle. The cycles were based on the same gas turbine (PR\textsubscript{Z}20, TIT\textsubscript{Z}1427\textdegree C). The evaporative cycle had a potential of 2% points higher efficiency if the cooling air was evaporatively cooled, the pressure ratio was increased to about 40 and the intercooling pressure split was optimized. Lindquist [165] performed parameter studies of evaporative cycles and concluded that the maximum electrical efficiency potential ranged from 45% for systems with power outputs of 1 MW\textsubscript{e} up to almost 60% for systems with power outputs of 100 MW\textsubscript{e}. An intercooled evaporative cycle had a 1% point higher efficiency (about 54.1%) than a dual-pressure reheat combined cycle for a turbine entry temperature of 1200\textdegree C, 0.5% points higher efficiency (about 56.1%) for a turbine entry temperature of 1400\textdegree C and about the same efficiency (about 57.5%) for a turbine entry temperature of 1600\textdegree C, while the specific power output was higher for the evaporative cycle in all cases. Rosén et al. [168] studied small-scale steam-injected and evaporative cycles based on two industrial gas turbines with simple cycle power outputs below 3 MW\textsubscript{e}. The thermal efficiency was increased by almost 20% with steam injection and by 42–56% with an evaporative cycle. The power outputs were increased by 41–52% with steam injection and by 42–90% with an evaporative cycle.

Cogeneration of power and district heating with evaporative cycles has also been investigated. In a report by Nilsson [167], evaporative cycles for cogeneration had electrical efficiencies on the same level as combined cycles with dual-pressure HRSGs, 49–50%, whereas the evaporative cycles had higher total efficiencies (94% compared with 89%). The specific investment cost was about 30% lower and the electricity and heat production costs were lower for the evaporative cycles compared with the combined cycles. Rydstrand et al. [169] studied generation of power and district heating with three humidified cycles: a conventional steam-injected cycle, a part-flow evaporative cycle with steam injection and a saturated steam-injected cycle without a steam superheater. Part-flow evaporative cycles will be discussed in Section 5.2.3. The cycles were based on the ALSTOM GTX100 gas turbine (PR\textsubscript{Z}20). Flue gas condensation produced the most of the district heating.
To increase the district heating production, an inlet air humidification tower, where low-level heat in the exhaust gas was used to heat water that humidified the inlet air, was added in some cases. All the humidified cycles had electrical efficiencies (up to 50%) similar to a combined cycle and higher total efficiencies. The humidified cycles had much lower specific investment costs, 40% lower per kilowatt of electricity and 60% lower per kilowatt of heat, and were economically feasible at lower costs of electricity compared with the combined cycle. The influence of the district heating water temperature on the performance of an evaporative cycle for power and district heating production, modeled on the evaporative cycle pilot plant in Sweden, was investigated by von Heiroth et al. [170]. Lower water temperatures increased the condensation in the flue gas condenser, resulting in higher heat production and total efficiency. Kaakinen [171] investigated three different evaporative cycles for production of power and district heating \((\text{PR} = 36, \ \text{TIT} = 1427 ^\circ \text{C})\). The most complex evaporative cycle was intercooled, aftercooled and recuperated and supplementary firing between the condensing district heating heat exchangers increased the district heating production (electrical efficiency 55.5%). Two simplified configurations with lower electrical efficiencies were also investigated: one version without an economizer for heating water for the humidification tower and one version without economizer and aftercooler. The total efficiencies were approximately 85%.

Another investigated application for the evaporative cycle is industrial cogeneration of power (1–50 MW\(_e\)) and steam and/or hot water [172]. For generation of steam, a steam boiler was included in the evaporative cycle. The best economic performance of the evaporative cycle was found for applications with low demands of electricity and demand of low-temperature energy, for example hospitals and airports. In addition, industries like the food and pulp and paper industries were interesting for the evaporative cycle. If electricity and hot water (50–60 \(^\circ\)C) are generated, the evaporative cycle could have an electrical efficiency of 45% and a total efficiency of over 90%. High humidification rates (i.e. full-flow humidification) raise the power output and a flue gas condenser can generate low-temperature heat. The pressure ratio should be below 15 for high electrical efficiencies. If a high total efficiency is sought, the pressure ratio should be above 15 and the recuperator and intercooler excluded. In addition, Nilsson et al. [173] discussed evaporative cycles for industrial cogeneration (a few to about 80 MW\(_e\)). If there is a demand for low temperature energy, it is favorable with high humidification rates (i.e. full-flow humidification), while for power production only or cogeneration of process steam, lower humidification rates can be used (i.e. part-flow humidification combined with a boiler for steam generation). The steam-injected gas turbine was deemed to be the strongest competitor to the evaporative cycle, since it is commercialized and can be used in the same applications. The evaporative cycle is flexible and the split between power and heat production can be chosen depending on the current needs.

Replacing the dry cooling air with humid air or steam can increase the efficiency of an evaporative cycle. Jordal and Torisson [174] investigated open-loop vane cooling with humid air in an evaporative cycle based on a modern industrial medium-size gas turbine. For an intercooled cycle, cooling with humid air instead of dry air from the compressor increased the thermal efficiency by 0.2% points, while for a non-intercooled cycle, the increase was 0.5% points. The efficiency increase was due to the low temperature of the humid air taken from the humidification tower outlet, the higher specific heat compared with dry air and the reduced compression work for the cooling medium, since a part of the coolant was compressed in the liquid state instead of in the gas turbine compressor.
5.2.2. Solid-fuel fired EvGT cycles

5.2.2.1. Externally fired EvGT cycles. Eidensten et al. [175] investigated small-scale externally biomass-fired evaporative cycles for power generation. The study was based on the same two gas turbines that were used in the study by Rosén et al. [168] for natural gas-fueled evaporative cycles described in Section 5.2.1. In the externally fired gas turbine, an atmospheric combustor and a high temperature heat exchanger, where the compressed air was heated by the combustion gases before an optional natural gas topping combustor, replaced the combustor of the open cycle gas turbine. The evaporative cycle was not aftercooled, while intercooling was used in some versions. The humid air from the humidification tower was heated by the exhaust gas streams from the gas turbine and the combustor before the high temperature heat exchanger. The thermal efficiency was about 32% for the smaller system and 41–43% for the larger system.

Externally biomass-fired evaporative cycles for cogeneration of power and district heating were investigated by Yan et al. [176]. The investigated system was the same as in Eidensten et al. [175], except that aftercooling and intercooling were included. Flue gas condensation recovered the water vapor in the exhaust gas streams and generated district heating. Yan et al. [177] investigated the same system, without the flue gas condenser, for power generation only. Intercooling could improve the efficiency compared with systems without intercooling. Aftercooling lowered the humidification tower water outlet temperature, thus enabling recovery of exhaust gas energy to lower temperatures. Electrical efficiencies slightly over 45% could be reached. Yan [178] and Yan and Eidensten [179] presented a review of externally solid fuel-fired gas turbines for power generation, including combined cycles and evaporative gas turbine cycles. The evaporative cycle had the largest potential for small systems. The high-temperature heat exchanger, metallic or ceramic, requires further development. To reach high electrical efficiencies with a metallic heat exchanger, topping combustion with natural gas or, for example, gasified biomass [180] is required.

Maunsbach et al. [181] investigated integration of combined cycles, steam-injected gas turbines and evaporative gas turbine cycles in the pulp and paper industry. The evaporative cycle that was externally fired with bark had a high electrical efficiency, although the overall effect on the pulp and paper mill system was small since the fraction of the total fuel amount used in the evaporative system was low. The steam-injected cycle and the combined cycle were integrated with gasification of black liquor or biomass.

Ahlroth et al. [182] investigated small-scale biomass-fueled closed evaporative cycles with part-flow humidification. Flue gas condensation and low-temperature heat recovery from the cycle were used to generate district heating. Compared with dry cycles, the humidified cycles in the study had 2–3% points higher electrical efficiencies. The electrical efficiencies for the evaporative cycles were in the range of 30–34%.

Biomass, refuse or waste heat can replace part of the natural gas input to an evaporative cycle, and evaporative cycles for base-load power generation using such energy sources were studied by Simonsson et al. [162]. The external energy can heat the water outlet stream from the humidification tower, generate steam for injection in the cycle and superheat the humid air after the tower. The use of external energy was limited by the maximum water content of the combustion air for stable combustion, which was approximately 35% by mass. Cases with costs of electricity for the additional electricity below the cost of electricity for the basic evaporative cycle were: preheating the evaporative cycle water circuit with heat from a waste incineration plant and superheating the humid air with flue gas from a steel
plant. Steam injection from a biofuel-fired steam boiler resulted in a cost of electricity just below the cost for the base case and steam injection with steam from a waste incineration plant should provide a very low cost of electricity.

5.2.2.2. EvGT cycles integrated with gasification. Steinwall [183] investigated evaporative cycles integrated with biomass drying, by steam or flue gases, and gasification, atmospheric or pressurized, for generation of power (45–52 MWₑ) and district heating. The electrical efficiencies for the different alternatives were 40–45% (50% moisture in fuel initially) and the total efficiencies were 90–92%. The highest electrical efficiency was for a system with pressurized gasification and steam drying.

Olsson [184] studied combined and evaporative cycles for cogeneration of power and district heating from gasified biomass. The drying and gasification of the biomass was integrated in the cycle. For a combined cycle with pressurized gasification, the electrical efficiency approached 45% and the evaporative cycle efficiency was close to this value. With near atmospheric gasification, the electrical efficiency was 4–5% points lower. The total efficiencies were 78–94% for the IGCC cycles, while the total efficiencies were lower for the evaporative cycles unless for low district heating temperatures.

Cavani [185] investigated medium-scale evaporative cycles fired with gasified biomass for generation of power (23–48 MWₑ) and district heating. The biomass was dried and gasified at atmospheric pressure. To avoid unstable combustion of the low heating value syngas, only a fraction of the compressed air was humidified and the remaining dry air was used for combustion in the primary zone, while the humidified air was used for diluting and cooling. In some cycle versions, a separate turbo expander was included for expansion of some of the humid air to reduce the flow mismatch between the compressor and turbine and decrease the need for extensive gas turbine design changes; however, the electrical efficiencies for these systems (34–38%) were lower than for a system without an additional expander (40%).

5.2.3. EvGT cycles with part-flow humidification

In most of the evaporative cycles studied, the whole compressed air flow is sent to the humidification system; however, cycles with part-flow humidification have been suggested. Within the EvGT project, part-flow evaporative cycles have been investigated and Westermark [186] proposed a cycle where the part-flow was between 10 and 30% of the compressor inlet air mass flow rate. Previously, Nakamura et al. [77,187], had suggested evaporative and water-injected cycles with higher percentages of part-flow. In part-flow cycles, only a fraction of the air is passed through the humidification system; the remaining air bypasses the tower and is mixed with the humidified air before the recuperator or the combustor. Part-flow humidification reduces the heat exchanger area and tower volume, which decreases the investment cost compared with a cycle with full-flow humidification. In addition, it has been shown that it is unnecessary to cool the whole air flow in the aftercooler and afterward heat it up again in order to achieve an efficient heat recovery; hence, compared with full-flow cycles, the electrical efficiency is retained or increased for part-flow cycles.

Different cycle configurations with part flow humidification have been suggested, for example including innovative tower designs. Aagren et al. presented part-flow evaporative cycles based on an intercooled industrial gas turbine (PR = 19, TIT = 1400 °C) [188] and a non-intercooled aeroderivative gas turbine (PR = 35, TIT = 1267 °C) [189] with a packed bed humidification tower divided in two sections. A two-section tower can provide a lower water outlet temperature than a one-section tower, thus increasing the heat recovery, and a two-section tower raised the electrical efficiency for all cases. Part-flow humidification reduced the efficiency slightly for the industrial gas turbine cycle with
a one-section tower, although both the part- and full-flow cycle with a two-section tower had the same efficiency (55.3%). Removing the intercooler reduced the efficiency by 2–3% points. For the aeroderivative cycle, part-flow increased the efficiency to 49.7% for a cycle with a one-section tower and to 50.9% for a cycle with a two-section tower. The power outputs were about 70 MW\(_e\) for the industrial gas turbine cycles and about 100 MW\(_e\) for the aeroderivative gas turbine cycles.

Part-flow evaporative gas turbine cycles where the humidification tower is combined with steam generation have also been suggested. Ågren and Westermark investigated evaporative cycles with part-flow humidification for a non-intercooled aeroderivative gas turbine [190] and an intercooled industrial gas turbine [191]. In the cycles, high-temperature energy was used to generate steam that was mixed with the working fluid and the humidification tower recovered low-temperature energy below the boiling point corresponding to the total pressure in the system. For the aeroderivative case (PR = 35, TIT = 1336 °C), the efficiency had a maximum (52.9%, 98 MW\(_e\)) for a part-flow corresponding to 12% of the compressor intake mass flow rate. For the industrial gas turbine (PR = 20, TIT = 1358 °C), full-flow humidification resulted in the highest electrical efficiency (52.6%, 78 MW\(_e\)); however, the efficiency was relatively constant down to a part-flow of 50%. Jonsson and Yan [192] compared these evaporative cycles with combined cycles with Kalina cycles as the bottoming cycle. The Kalina cycle is a further development of the steam Rankine cycle with an ammonia-water mixture as the working fluid. For the industrial gas turbine, the evaporative gas turbine had a higher efficiency than the combined cycles with single-pressure Kalina bottoming cycles, while a reheat Kalina cycle provided a higher efficiency. For the aeroderivative gas turbine, the combined cycles with Kalina bottoming cycles had higher efficiencies than the evaporative cycle.

Bartlett [166] found that steam-injected part-flow evaporative cycles with high pressure ratios were better suited for non-intercooled gas turbines than full-flow cycles, since the part-flow cycles required less gas–gas heat exchange and had higher electrical efficiencies (up to almost 54%, PR = 35, TIT = 1500 °C) and specific power outputs. For intercooled gas turbines, the full-flow evaporative cycles had the highest efficiency (55%, PR = 40, TIT = 1500 °C), although the part-flow cycles had similar efficiencies for high pressure ratios (PR = 40, TIT = 1500 °C). The part-flow cycles heated the humid air in a humid air superheater that had lower effectiveness and cost than the recuperator in the full-flow cycles. An economic comparison showed that part-flow evaporative and steam-injected cycles (that were steam-cooled) had the largest potential. The full-flow cycles required intercooling and a high-performance recuperator with a lower cost than today’s recuperators in order to be competitive. The non-intercooled humidified gas turbine cycles had lower specific investment costs (15–35%) and lower costs of electricity (3–12%) than combined cycles and intercooling should reduce the cost of electricity further.

Jonsson and Yan [193] made a comparative economic analysis of non-intercooled steam-injected part-flow evaporative, full-flow evaporative and steam-injected gas turbine cycles based on existing gas turbines for small-to-mid-size power generation. For the GTX100 (ALSTOM, PR = 20, TIT = 1358 °C) and Trent (Rolls-Royce, PR = 35, TIT = 1336 °C) cases, a steam-injected part-flow evaporative cycle had the lowest cost of electricity. The Trent cases had the lowest costs of electricity, mostly because they had the highest electrical efficiencies (52.1% for the part-flow cycle), followed by the GTX100 and the Cyclone (ALSTOM, PR = 14, TIT = 1279 °C) cases. However, for all cycles studied, the costs of electricity varied by only 1–3% for a specific gas turbine. The steam-injected cycles had the lowest specific investment cost, followed by the part-flow evaporative cycles, while the full-flow evaporative cycles had the highest specific investment cost. For the GTX100 and the Trent, the part-flow case with
the lowest costs of electricity was compared with literature data for combined cycles with dual-pressure HRSGs based on the same gas turbines. The costs of electricity for the evaporative and combined cycles were in the same range, while the investment cost was 16–25% lower and the specific investment cost was 12–15% lower for the evaporative cycles.

5.2.4. The evaporative gas turbine pilot plant

There is only one evaporative gas turbine in operation: a pilot plant in Lund, Sweden, that started operation as a simple cycle in 1997. The plant is based on a Volvo VT600 gas turbine with a simple cycle power output of 600 kW and an efficiency of 22%. In the final plant configuration, the compressed air is aftercooled, humidified in a packed bed tower, by water heated in the aftercooler and an exhaust gas economizer, and recuperated. As an evaporative cycle without an aftercooler, the efficiency was approximately 35%, while addition of a plate aftercooler increased the efficiency by 0.4% [194]. The plant is not optimized for electrical efficiency or low emissions; its only purpose is to demonstrate the technology. The flow mismatch between the compressor and turbine is met by bleeding off some of the compressed air. The water content of the tower outlet air was 0.14 kg water/kg air at 75% load, which is the maximum load available in the plant [195]. Lindquist et al. [196] described the operation of the plant with startup strategies and water circuit control. Full power output can be reached almost as fast for the evaporative cycle as for the simple cycle. The pilot plant can be self-sufficient with water by cooling the exhaust gas to 35 °C in the flue gas condenser and the water treatment system can achieve a sufficient feedwater quality [197]. Static [198] and dynamic models [199] for the pilot plant have been presented and validated with measured data. An evaporative cycle contains large volumes due to, for example, the humidifier and recuperator; hence, the cycle has slower dynamic responses during power transitions than a simple cycle, although this should not result in any major operational problems. Dynamic simulation of the pilot plant showed that the compressor surge margin decreased during transition from dry to evaporative operation, thus this transition should occur at low power levels.

5.3. Other work on evaporative gas turbines

5.3.1. Natural gas-fired evaporative gas turbines

In this section, all evaporative cycles are assumed to include intercooling, aftercooling and recuperation. If these components are excluded or other cycle modifications are made in some studies, this is mentioned.

The evaporative cycle has been compared with other humidified cycles, for example by Chiesa et al. [68], who compared evaporative cycles with recuperated water-injected cycles based on an intercooled aeroderivative gas turbine. An evaporative cycle had an efficiency of 57% (PR = 48, TIT = 1500 °C), while a water-injected cycle had an efficiency of 55.1% (PR = 33, TIT = 1500 °C). In both cycles, the fuel was preheated by water and exhaust gas. Exergy analysis showed that the exergy destruction due to air–water mixing was smaller in the humidification tower of the evaporative cycle than in the mixing process of the water-injected cycle. In the water-injected cycle, aftercooling and air–water mixing were performed in one step that had a large exergy destruction, since the evaporation temperature was much lower than the temperatures of the air and water that were mixed. In the evaporative cycle, the exergy destruction was smaller, since the aftercooling and humidification were separated. In addition, the evaporative cycle had a higher efficiency since a larger part of the intercooler and low-temperature exhaust gas energy could be recovered. A smaller part of this heat was needed in the water-injected cycle.
to heat the optimum water flow. The mixing of steam and air in an intercooled steam-injected cycle had a higher exergy destruction than in the water-injected and evaporative cycles; hence, the efficiency of the steam-injected cycle cannot be as high as the efficiencies for the other two cycles. The efficiency of the evaporative cycle was 1.5% points higher than for an intercooled combined cycle based on the same gas turbine (PR = 46, TIT = 1500 °C). Chiesa and Lozza [200] continued the investigations on intercooled aeroderivative gas turbines. A simple cycle had an efficiency of 45.7% and a power output of 125 MW_e (all cycles PR = 46, TIT ≈ 1500 °C), a triple-pressure reheat combined cycle had an efficiency of 57.1% and a power output of 153 MW_e and an evaporative cycle had an efficiency of 57.7% and a power output of 287 MW_e.

Traverso and Massardo [69] compared steam-injected gas turbines, recuperated gas turbines with water injection for intercooling and aftercooling, and evaporative cycles with power outputs of about 50 MW_e. The water-injected cycle is discussed in Section 3.3. The authors proposed a simplified recuperated evaporative cycle, where water injection replaced the surface heat exchanger for intercooling, for small-to-medium-size systems. A combined cycle with a dual-pressure HRSG had the highest efficiency (52.9%, PR = 18, TIT = 1400 °C (the same for all cases)), although the specific power output was higher for the humidified cycles. The evaporative cycle had the highest efficiency of the wet cycles studied (52.0%, PR = 30). The simplified evaporative cycle had an efficiency of 50.6% (PR = 30), the water-injected cycle had an efficiency of 47.7% (PR = 30) and the steam-injected cycle had an efficiency of 50.7% (PR = 30). The evaporative and water-injected cycles had similar costs of electricity, while the combined cycle was penalized by a high investment cost due to the steam bottoming cycle.

Jin and Ishida [201] used graphical exergy analysis to compare different humidified gas turbine cycles. An evaporative cycle had an efficiency of 56.4% (TIT = 1200 °C, PR = 20), while a steam-injected gas turbine had an efficiency of 48.4% (TIT = 1200 °C, PR = 20). In the steam-injected cycle, the temperature profiles were mismatched in the medium-temperature section of the heat recovery system due to the isothermal boiling. In the evaporative cycle, the temperature profiles were more closely matched in the medium-temperature section, since the high-temperature exhaust gas was used to superheat the humid air. Another beneficial feature of the evaporative cycle was that water was evaporated by energy at a lower temperature level than in the steam-injected cycle. In addition, the exergy destruction for air–water mixing was smaller in the evaporative cycle.

Higdon et al. [202] compared an evaporative cycle, where a part of the tower outlet water was sent to an external cooling loop to increase the heat recovery, with an intercooled steam-injected gas turbine. The evaporative cycle efficiency was 54.8% (PR = 15, TIT = 1149 °C) compared with the intercooled steam-injected cycle that had an efficiency of 47.9% (PR = 30, TIT = 1149 °C). The pressure ratio for maximum electrical efficiency for the evaporative cycle was rather low, although the efficiency varied only little with the pressure ratio.

There are also investigations on the optimal layout of the heat recovery system in the evaporative cycle where different cycle parameters are varied. Lazzaretto and Segato [203] presented an optimization of the evaporative cycle structure, using pinch technology and exergy analysis to optimize the heat recovery system (i.e. the number and position of intercoolers, aftercoolers and economizers). A recuperator was included in all cycle configurations and the fuel was preheated by water and exhaust gas. The highest cycle efficiencies were obtained for values of the intermediate pressure ratio (i.e. at the intercooler) and the intercooling final temperature that minimized the compression work. Aftercooling only increased the cycle efficiency when the heat from the aftercooler could be used to heat a part of
the water recycled from the tower. In the range of pressure ratios investigated (10–30), the pressure ratio had little impact on the cycle efficiency. For a pressure ratio of 20, different heat exchanger networks that provided the same cycle efficiency were designed (54.6%, TIT = 1250 °C). The network with the smallest number of heat exchangers had one aftercooler, two economizers and two intercoolers. Xiao et al. [204] proposed an evaporative cycle for maximum specific power output without a recuperator (thermal efficiency 43.1%, 1300 kJ/kg, PR = 16, TIT = 1260 °C). Two recuperated configurations with different water circuits for high thermal efficiency were also presented. In these, the air leaving the tower was not saturated (thermal efficiency 60.3%, PR < 10, TIT = 1260 °C). Xiao et al. [205] continued the studies and found that the optimum pressure ratio for a high efficiency (thermal efficiency over 60%) of an evaporative cycle was low (PR < 10, TIT = 1260 °C), while higher pressure ratios resulted in higher specific power outputs. Exergy analysis showed that heating the water to different temperatures in the intercooler, aftercooler and economizer and adding it to the humidification tower at different heights could increase the cycle efficiency. Krause and Tsatsaronis [206] made an exergoeconomic evaluation of an evaporative cycle. The authors found that intercooling and a high turbine inlet temperature resulted in a high efficiency. The exergetic efficiency of the heat utilization within the cycle was low since heat transfer, mixing and saturation in several steps downgraded the heat. Jin et al. [207] investigated an evaporative cycle by considering the temperature difference at the bottom of the humidification tower, between the inlet air and outlet water streams. When the temperature difference at the bottom of the tower was increased, the cycle efficiency decreased and the specific power output increased. The case when the water outlet temperature is lower than the inlet air dry bulb temperature was discussed, since such a situation had occurred at the authors’ humidification tower test rig.

One alternative for increased cycle efficiency is to use external cooling water to reduce the humidification tower water outlet temperature for increased heat recovery. This was suggested already by Rao [49]. The water could also be cooled by ambient air, as proposed in a study by Stecco et al. [208]. In this study, the whole outlet water flow could be cooled, when the humidity ratio after the tower was high (this ratio was varied in the optimization process), or only the part that was used in the intercooler and aftercooler, which was mixed with the makeup water. The cycle with partial cooling of the tower outlet water had the highest efficiency and specific power output. An efficiency of 58% (PR = 4.5, TIT = 1000 °C) was possible for a tower outlet humidity ratio of 40%, although the specific power output was low for this set of parameters. Stecco et al. [209] optimized this configuration and found efficiencies in the range of 54–56% (PR = 9, TIT = 1000 °C) with higher specific power outputs than the configuration optimized for efficiency. Gallo et al. [210] made exergy and thermoeconomic analyses of evaporative cycles where external cooling water cooled the part of the tower outlet water to be used for intercooling and aftercooling. Considering only the impact of the fuel and water costs on the cost of electricity (i.e. the investment, operation and maintenance costs were not included in the analysis), it was found that the cost of electricity was almost the same for pressure ratios between 12 and 20. Gallo et al. [211] performed an exergy analysis of the same evaporative cycle configuration. Water evaporation in the humidification tower had a smaller exergy destruction than the evaporation of water in a boiler. The external cooler increased the cycle exergetic efficiency from 46.0 to 48.2%. Gallo [212] compared the evaporative cycle with an external cooler with the simple cycle gas turbine, the recuperated gas turbine, the recuperated intercooled gas turbine, the steam-injected gas turbine and the combined cycle. For pressure ratios of 12 and 30 (TIT = 1300 °C), the evaporative cycle had the highest efficiency (54.8 and 54.7%, respectively). For a low pressure ratio, the steam-injected cycle had the highest specific power output, followed by the evaporative cycle, while for a high pressure ratio, the evaporative cycle
had the highest specific power output. The consumption of demineralized water was approximately the same for the evaporative cycle and the steam-injected gas turbine, while a combined cycle had a much lower consumption of demineralized water.

Many different cycle modifications to increase the efficiency have been suggested, for example part-flow humidification. Mori et al. [48] proposed two humidified gas turbine designs: one with direct water injection, described in Section 3.3, and one with a humidification tower. The thermal efficiency of the evaporative cycle was 52.8% (PR = 12, TIT = 1100 °C). Nakamura et al. [187] patented the cycle presented by Mori et al. [48], although modified with part-flow humidification. After the compressor, the air stream was split. One part of the air passed through the optional aftercooler, the humidification tower and a recuperator, then it was mixed with the air that had bypassed the tower before a second recuperator. The tower could be divided into two parts to increase the heat recovery: in the top part, water heated in the intercooler, aftercooler and high-temperature part of the economizer was injected, while in the lower part, water heated in the low-temperature part of the economizer was added. In this configuration, a compressor was included after the aftercooler to counteract the pressure drop in the tower and recuperator. In yet another cycle variation, part of the intercooling and the aftercooling was performed inside two humidification towers. Ambient air was compressed, passed through a tower for indirect intercooling, further intercooled in an indirect water-cooled heat exchanger, further compressed, indirectly aftercooled in a tower, and then humidified in the two towers. The fuel (natural gas) was also humidified in a tower. The inventors suggested that 60–100% of the air should pass the tower in the different cycle versions. In all the cycles presented above, the fuel was preheated by exhaust gas. Sayama and Nakamura [213] patented a similar gas turbine cycle, although with full-flow humidification and no aftercooling, where untreated water, for example industrial water, could be used. In a first humidification tower, industrial water heated in the intercooler was used. The water that was not evaporated was heated in an economizer and re-used in the tower. The air went from this tower to a second tower where pure water heated in the intercooler was used to clean the air. The water not evaporated was heated in the economizer and re-used in the tower.

Another modification for increased efficiency is a combination of an evaporative cycle and an organic Rankine cycle (ORC) [214]. The ORC used heat rejected from an external cooler and the water recovery system. The external cooler cooled the water recirculated from the tower that was to be used for intercooling and aftercooling. The ORC raised the cycle power output by 1.6–2.2% and the efficiency by 0.5–1% points, depending on the working fluid.

An evaporative cycle can be combined with inlet air cooling to increase the efficiency. Song and Ro [215] suggested an evaporative cycle where the evaporation of the liquefied natural gas (LNG) used as the cycle fuel was employed to cool the compressor inlet air. Compared with the simple cycle gas turbine, the evaporative cycle increased the power output by 48% and the efficiency by 16%, while an evaporative cycle integrated with evaporation of LNG increased the power output by 54% and efficiency by 17% (PR = 10, TIT = 1000 °C). Water injection can also be used for inlet air cooling, as in the non-intercooled evaporative cycle proposed by Kiguchi and Hatamiya [216] (employees of Hitachi). More water than required for saturation was injected and some droplets evaporated in the compressor. The water vapor in the exhaust gas was recovered by a direct-contact condensation system. For a 40 MW_e power output, the thermal efficiency was 54.3% (PR = 15, TIT = 1400 °C). Hitachi has patented several gas turbine cycles where the working fluid is humidified, for example a system where water, recovered from the exhaust gas, was injected into the compressed air before recuperation [217], an evaporative cycle with an improved humidification tower with a smaller water flow in the lower part of the tower.
[218], an evaporative cycle without intercooling and aftercooling to reduce the investment cost [219] and a cycle where the air was humidified first in a tower and then by water injection before the recuperator [220].

An evaporative cycle could be integrated with a fuel cell to raise the efficiency. Li [221] investigated integration of an evaporative cycle and a SOFC. With methane as the fuel, the cycle efficiency was 71.4% and the power output 35.7 MWₑ (TIT = 1203 °C). The SOFC received humid air from the recuperator. After the fuel cell, all of the fuel was not spent and a combustor burnt this fuel before the gas turbine. An evaporative cycle without a SOFC could have a net efficiency of 58% (PR = 23, TIT = 1250 °C) for an optimized configuration using modern technology with a power output above 200 MWₑ.

Ishida and Ji [222] suggested an evaporative cycle integrated with a two-stage absorption heat transformer for increased efficiency. A heat transformer is a variant of the absorption heat pump that upgrades medium-temperature heat to a higher temperature. A water–lithium bromide mixture heat transformer, driven by heat in the exhaust gas, was used to heat water for the humidification tower. The match of the temperature profiles in the medium and low-temperature range was close, since the heat transformer improved the matching in the low-temperature range. By incorporating the heat transformer, the cycle thermal efficiency was increased from 54.2 to 56.1% (PR = 20, TIT = 1200 °C) and the specific power output was increased by 7%.

The evaporative cycle could be integrated with chemical looping combustion, as suggested by Ishida and Jin [223]. In chemical looping combustion, a metal oxide is circulated between an oxidation reactor, where it takes the oxygen from the combustion air, and a reduction reactor, where the oxygen is delivered to the fuel. In this way, the combustion system generates two separate streams: a stream of carbon dioxide (CO₂) and water from the reduction reactor and a stream of mainly nitrogen from the oxidation reactor. Gas turbines can generate power from both high-pressure exhaust streams. After separation of the water, CO₂ of a high concentration is obtained and can be sequestered. If chemical looping combustion is integrated with an evaporative cycle, the water condensed from the exhaust gas streams could be used as makeup water for the evaporation process. A graphical exergy method was used to analyze the system. The evaporative cycle electrical efficiency was 55.1% with water recovery, while a steam-injected gas turbine with a conventional combustor had an efficiency of 48.4%.

Cooling the gas turbine with humid air to increase the efficiency has also been considered, for example by Gallo et al. [224], who considered the turbine blade cooling. For a cycle with an external cooler for the humidification tower outlet water, cooling the expander with humid air from the tower outlet resulted in the highest cycle efficiency (54.1%, PR = 12, TIT = 1200 °C). Cooling air from the high-pressure compressor exit or the aftercooler exit resulted in a lower efficiency (53.4%) and cooling air from the recuperator exit provided the lowest efficiency (52.2%).

Comparisons of the evaporative cycle to other advanced gas turbine cycles, for example chemically recuperated or semi-closed gas turbines, have also been performed. Carapellucci et al. [225] compared evaporative cycles and chemically recuperated gas turbines based on heavy duty industrial and aeroderivative gas turbines. In chemical recuperation, the fuel is reformed endothermically in the presence of water (i.e. steam reforming) or without (i.e. simple decomposition). The energy needed is taken from the gas turbine exhaust gas. The evaporative cycle and gas turbines with chemical recovery had performances comparable to combined cycles. The authors concluded that the evaporative cycle was more suitable for heavy-duty gas turbines, while chemical recovery was more suitable for aeroderivative gas turbines. Corti et al. [226] compared the evaporative cycle with semi-closed gas turbines. The semi-closed gas turbine was a combined cycle where most of the exhaust gas was recirculated to
the compressor inlet after the bottoming cycle HRSG. Some ambient air was also compressed to supply oxygen for the combustion. After the HRSG, the exhaust was cooled to condense the water, which was injected after the compressor. The advantages of the cycle were that some of the nitrogen in the exhaust gas was replaced with water and CO₂, thus reducing the cost for CO₂ removal, and the exhaust gas recirculation and water injection decreased the NOₓ emissions. Both the semi-closed and the evaporative cycle had high efficiencies, while the investment cost was higher for the evaporative cycle.

The turbinomachinery for an evaporative cycle has been considered by Desideri and di Maria [227], who discussed the conversion of a 2 MWₑ single-shaft GE/Nuovo Pignone PGT2 gas turbine (25%, PR = 12.7) to evaporative operation. Calculations were made for the gas turbine in a recuperated cycle (28.5%, 2.0 MWₑ), an intercooled recuperated cycle (36.1%, 2.7 MWₑ) and an evaporative cycle with an external cooler (for the tower outlet water used for intercooling and aftercooling) and flue gas condensation (42.7%, 4.6 MWₑ). Combining the compressor of the PGT2 with the expander from the larger gas turbine PGT5 could avoid the flow mismatch in the evaporative cycle. An economic analysis showed that the specific investment cost was 1% higher for the recuperated cycle and 3% higher for the evaporative cycle than for the simple cycle (1059 USD/kWₑ). Since the efficiency was the highest for the evaporative cycle, it should have the lowest cost of electricity.

The possibility to modulate the evaporative cycle power output at constant speed by changing the water evaporation rate was discussed by di Maria and Desideri [228]. In order to change the amount of evaporated water, a part-flow evaporative cycle was adopted, where the air stream after the aftercooler was split and only one stream passed the humidification tower. The other stream bypassed the tower and was mixed with the humid air before the recuperator. When the percentage of bypass air was increased, the power output was reduced while the efficiency was relatively constant, which implied an advantageous part-load performance. An economic analysis, based on a medium-size industrial gas turbine (ALSTOM GTX100), showed that the specific investment cost was lower for a combined cycle (62 MWₑ, 54.4%, 500 USD/kWₑ) than for the evaporative cycle (75 MWₑ, 50.0%, 545 USD/kWₑ).

The evaporative cycle has an advantage compared with the simple cycle and combined cycle at high ambient temperatures, as shown by Kim et al. [229]. When the ambient temperature increased, the turbine work of the evaporative cycle was raised since more water could be added in the humidifier due to the higher humidifier air inlet temperature. At the same time, the compression work increased; hence, in total, the net specific power output was almost the same. For both a simple cycle gas turbine and a combined cycle with a single-pressure HRSG, high ambient temperatures resulted in greater reductions in specific power output and efficiency than for the evaporative cycle.

The evaporative cycle has been considered for micro gas turbines. Parente et al. [230] found that a theoretical evaporative cycle could increase the electrical efficiency of a recuperated, non-intercooled micro gas turbine by nearly 5% points and the specific power output by close to 70%. For two existing recuperated micro gas turbines (dry power outputs 100 and 500 kWₑ, respectively), the calculated efficiency increase was 3% points and the specific power output increase was 30%. The smaller increases for the existing gas turbines were due to various technical aspects of humidifying the gas turbine working fluid, as were discussed by the authors. A thermoeconomic analysis [231] showed that evaporative micro gas turbine cycles for power generation or combined heat and power generation had lower specific investment costs than conventional micro gas turbines. The evaporative micro gas turbine cycle also offered more flexible operation for heat and power generation than a dry micro gas turbine.

In most studies, it is assumed that the evaporative cycle will be used for stand-alone power generation, although application of evaporative cycles in energy intensive industries has also been discussed.
Hansen and Nielsen [232] stated that the pinch point of the evaporative cycle is usually at a low temperature; hence, additional heat in the range 100–200 °C can be utilized with a high marginal efficiency. Energy intensive industries (e.g. refineries, cement production, steelworks) typically have a surplus of heat from 250 to 300 °C and down. A case study where pinch analysis was used to integrate an evaporative cycle into a refinery was presented.

5.3.2. Solid fuel-fired evaporative gas turbines

5.3.2.1. Integrated gasification evaporative gas turbines. Ong’iro et al. [233] investigated IGCC and intercooled IGHAT cycles for coal. The IGHAT had a higher thermal efficiency than the IGCC, since more energy was recovered from the gasification process by generating hot water for the HAT cycle than by generating steam for the combined cycle. The IGHAT performance was less sensitive to high ambient temperatures than the IGCC, since the water absorbed in the tower balanced changes in the airflow rate through the compressor. The IGCC had a non-reheat single-pressure or dual-pressure steam bottoming cycle.

Chiesa and Lozza [200] investigated intercooled aeroderivative gas turbines in combined or evaporative cycles integrated with coal gasification with power outputs of 130–8140 MW. In the IGHAT, water quench cooling of the syngas generated hot water for use in the humidification tower. In the IGCC, high-pressure steam generated from syngas cooling or intermediate-pressure steam generated in a quench cooling system was used in the steam bottoming cycle. The efficiency of the IGHAT (44.5%, in all cases: PR = 36, TIT ≈1500 °C) was lower than for the IGCC with a syngas cooler (47.4%) or quench cooling (44.9%).

Li [221] investigated an integrated gasification intercooled evaporative cycle with open-loop steam blade cooling with a net efficiency of 50.6%. With a hot gas cleaning system, 0.6% points higher efficiency was possible. Combining an open-loop steam-cooled evaporative cycle with a pressurized fluidized bed combustion (PFBC) plant was also suggested. In this system, coal is combusted and the combustion gases are cleaned while hot before expansion in the turbine. The use of fluidized bed combustion limited the turbine inlet temperature to 800 °C; however, topping combustion with syngas increased the turbine inlet temperature and a net efficiency of 50.8% was reached.

Smith et al. [234] discussed HAT or CHAT cycles integrated with gasification and a cryogenic air separation unit, which generates oxygen and nitrogen. The power cycles could be integrated with the air separation unit by extracting air from the gas turbine cycle to the air separation unit. The oxygen was used in the gasification process and the nitrogen could be used for expander cooling or for injection in the turbine, combustor or humidification tower.

Ländlv [235] patented an evaporative cycle integrated with gasification of a solid fuel. In this system, the integration between the gasification and power generation systems was different to previously suggested IGHAT cycles, where the energy was recovered by heating water for the humidification tower. In the patented system, energy was transferred to the evaporative cycle by quench cooling the syngas until saturation and then cooling it further to generate intermediate pressure steam, that was injected after the humidification tower.

5.3.2.2. Externally fired evaporative gas turbines. Externally fired evaporative cycles have been studied both in the HAT project, see Section 5.1.2.2, and in the EvGT project, see Section 5.2.2.1. In addition, Jin et al. [236] investigated an externally coal-fired intercooled evaporative cycle. After recuperation
with the gas turbine exhaust gas, the humid air was heated in the external coal combuster before the turbine. The water in the exhaust gas was condensed and re-used in the cycle and the heat from the condenser was used to heat water for the tower. The net efficiency was 46.9% (TIT = 850 °C). An externally fired combined cycle had a net efficiency of 40.9%. As a comparison, a supercritical coal-fired steam cycle had an efficiency of 42% and an ultra-supercritical steam cycle had an efficiency of 46%; however, these systems were more complex than the externally fired evaporative cycle.

Huang and Naumowicz [237] investigated an externally fired intercooled evaporative cycle. With an increased water injection ratio (at a given pressure ratio), the thermal efficiency increased while the second-law efficiency decreased, since the second law efficiency included the exergy in the exhaust gas in the stack. The optimum thermal efficiency was achieved at a low pressure ratio (i.e. PR = 8–14) (TIT = 1260 °C).

5.4. The humidification tower

The humidification tower is a key component in the evaporative cycle that has not been employed previously in power cycles. Most of the design methods for humidification towers that have been developed are adapted from chemical engineering methods for cooling tower design. However, since the pressure, temperature and humidity levels differ between a humidification tower and a cooling tower, care must be exercised when using simplified methods for cooling tower design. Modeling of humidification towers and different methods for humidification of air were discussed by Dalili [238]. Accurate modeling of the simultaneous mass and heat transfer in the humidification tower is required for reliable cycle design and cost estimation. Aramayo-Prudencio and Young [239,240], Enick et al. [241], Gallo et al. [242], Jin et al. [243], Parente et al. [244], Wang et al. [245], Agren [163] and Lindquist et al. [246] proposed models for packed bed humidification towers, which have been used in most evaporative cycle simulations, and Dalili and Westermark [247] performed an experimental investigation of the packed bed humidifier in the Swedish pilot plant. Zhao et al. [248] presented a humidification tower model developed from a test rig tower without a packing and Wang et al. [249] proposed a gas turbine cycle where the air was humidified in a spray dryer, which is a column without any surface enlarging elements where water is injected at the top and flows counter-currently with the air stream. Gillan and Maisotsenko [250] suggested a tube-and-shell heat exchanger that combined the aftercooler, economizer, recuperator and humidification processes into one component.

An alternative to the packed bed humidifier is the tubular humidifier. Dalili and Westermark [251, 252] presented experimental results from a tubular humidifier pilot plant, which consists of a tube with rods on the outer surface. Inside the tube, ascending air is brought into counter-current contact with a water film falling downward, while the exhaust gas flows downward on the outside of the tube. Thus, the separate economizer required for a packed bed humidifier is avoided with a tubular humidifier. A tubular humidifier is favorable for small-size gas turbines due to its effective heat and mass transfer characteristics and compactness.

6. Water recovery and water quality

Water consumption and quality have been much discussed for humidified gas turbines, since the cost for water increases the operational cost for the plant and water and its contaminants may cause erosion
and corrosion in the cycle. The cost for a water recovery system that condenses the water and treats it before re-use must be compared with the cost of buying makeup water of a sufficient quality. The water can be condensed in an indirect surface heat exchanger that uses water (e.g. from the environment or cooled by ambient air) or ambient air to cool the exhaust gas or in a direct-contact condenser where water (e.g. cooled by ambient air) is sprayed into the exhaust gas. Industrial applications of flue gas condensation for cogeneration may provide experiences for water and heat recovery for humidified gas turbines. There are some examples of water recovery in humidified gas turbines: the steam-injected gas turbine Aquarius recovers water with a direct-contact condenser [107], in an Italian car-manufacturing factory, there is a Cheng cycle with water recovery by an indirect condenser that generates hot water for the factory [102] and the Swedish evaporative gas turbine pilot plant is self-supporting with water due to an indirect condenser [197]. In all cases, the recovered water is cleaned from substances that may cause corrosion in the water circuit or gas turbine before re-use in the cycle. Different alternatives for water recovery in steam-injected cycles have been investigated by De Paepe and Dick [253] (indirect water- or air-cooled condensers and direct condensers), De Paepe et al. [254] (water- or air-cooled indirect condensers), Bettagli and Facchini [255] (direct-contact condenser), Nguyen and den Otter [256] (indirect water- or air-cooled condensers) and Blanco and Ambs [257] (direct and indirect water-cooled systems). Water recovery in evaporative cycles has been investigated by Cataldi [258] (direct and indirect air- or water-cooled condensers) and Desideri and di Maria [259] (indirect water-cooled condenser). The effect of water recovery on the thermodynamic performance of the evaporative cycle was investigated by Poels and Gallo [260]. The indirect water-cooled condenser introduced an additional pressure drop in the exhaust gas, which decreased the specific power output by 1.2–2.0% and the efficiency by 0.3–0.5% points. Experiments on water recovery from humid exhaust gas from gas turbines have been conducted. De Paepe and Dick [261] constructed a test rig consisting of a finned tube condenser where they accomplished full water recovery. A direct-contact condenser was also tested and achieved full recovery [253]. Zheng et al. [262] described experiments on a condensing glass tube (to avoid corrosion) heat exchanger installed after the HRSG of an experimental steam-injected gas turbine. The condensed water required only a little treatment before re-use. Cerri and Arsuffi [263] suggested an alternative method to make a steam-injected gas turbine self-sufficient with water. In this system, energy in the exhaust gas drove a demineralizing unit that distilled water from seawater for injection.

Gas turbines are sensitive to contaminants that may enter the cycle with the inlet air, the fuel and, in the case of humidified cycles, the water. For direct water-injected and steam-injected gas turbines, demineralized water is required. Water of lower quality contains contaminants, for example salts that can cause corrosion in the high-temperature parts of the gas turbine. For evaporative cycles, water of a lower quality could be used since only a part of the water is evaporated into the air in the humidification tower and non-volatile contaminants remain in the liquid phase. Bartlett and Westermark [264] presented a model for the flow of alkali salts in an evaporative cycle. Alkali metals and alkaline earth metals may cause corrosion in the turbine and in the water circuit. Substances can be transferred between the air and water streams in the humidification tower and in a direct-contact flue gas condenser. If there are contaminants in the water, only volatile compounds and compounds in entrained droplets can be transferred to the air, therefore a droplet separator should be included after the humidification tower. Bartlett and Westermark [265] presented results from experimental investigations in the evaporative gas turbine pilot plant regarding the beneficial effect of different inlet air filters and flue gas condensation on the air and water quality. For evaporative cycles with water recovery, efficient inlet filters reduce
the need for water treatment, since a filter greatly decreases the amount of salt particles entering the cycle. If an air filter is not used, the water in the humidification tower captures most of the particles in the air stream by acting as a scrubber.

7. Humid air as a gas turbine working fluid

7.1. Thermophysical properties of air–water mixtures

For humidified gas turbines, accurate thermophysical properties (i.e. thermodynamic properties and transport properties) for air–water mixtures are required for reliable simulation of the cycle performance, design of the cycle components and cost estimating. The air–water system deviates from ideal behavior, especially at high pressures, low temperatures and high humidity levels. Ji and Yan [266] reviewed the existing theoretical thermodynamic property models and experimental data of air–water systems. Dalili et al. [267] reviewed the available thermodynamic property models for air–water mixtures and considered the impact by different property models on the simulation of humidified gas turbine cycles. The models found were limited in temperature, pressure and/or humidity range. The best available model was the Hyland and Wexler real gas model [268], despite a limited temperature range (valid up to 200 °C). A comparison between ideal and real gas data showed that the real model predicted a higher saturation humidity at a specific temperature, since water is more volatile in the real model. The cycle efficiency seemed to be only slightly affected by different thermodynamic models. Ji and Yan [269] proposed a real gas model, based on a modified Redlich–Kwong equation of state, for the thermodynamic properties (i.e. humidity, enthalpy and entropy) of air–water mixtures as well as nitrogen–water mixtures [270]. The new model was compared with other real gas models: up to 50 bars and 400 K, the models agreed with each other, while the deviations were larger at higher pressures. Ji and Yan [271] validated the new model by comparing saturated properties calculated with the model to experimental data available in literature for oxygen–water, nitrogen–water and air–water systems and to other thermodynamic property models. Experimental data for the air–water system are scarce. The comparisons showed that the new model is valid for saturated properties up to 300 °C and 200 atm. Yan et al. [272] calculated some components in an evaporative cycle with ideal, ideal mixture and real models, among them the Ji-Yan model, for the air–water mixture thermodynamic properties. It was found that the humid air superheater outlet temperature varied up to 18 °C for the different models. Comparisons of the superheated properties calculated with the Ji-Yan model, the ideal model and the ideal mixture model showed that the Ji-Yan model can be used up to 1500 °C and 200 bars.

7.2. Combustion with humid air

One possible problem in humidified gas turbines is the combustion, since the water may cause combustion instability and reduced combustion efficiency resulting in increased emissions of carbon monoxide and unburned hydrocarbons. Day et al. [273] described experiments and calculations performed to determine the effect of combustion air water content on emissions, stability limits, operation and ignition. The moisture in the air reduced the NO\textsubscript{x} emissions, while the carbon monoxide emissions were not significantly increased. Bhargava et al. [274] investigated humid air premixed flames experimentally. The moisture levels in the air were 0–15% by mass. When moisture is added,
the equilibrium temperature is reduced and this decreases the NO\textsubscript{x} formation. At a constant equilibrium temperature, the presence of steam reduces the NO\textsubscript{x} production by lowering the concentration of oxygen atoms and by increasing the concentration of OH atoms, which showed that the NO\textsubscript{x} reduction is not only an effect of temperature reduction by water or steam injection. Hermann et al. [275] investigated premixed combustion of natural gas with humid air by experiments and calculations. In the experiments, the water content in the air was varied between 0 and 33% by weight. Low emissions of carbon monoxide were measured, except at very high humidification levels. Chen et al. [276] investigated the effect of humid air on combustion of liquid fuels in gas turbines. If 10% water (by mass) was added to the air, a 90% reduction of the NO\textsubscript{x} was possible compared with a case with dry temperature at the same flame temperature. The comparison was made for the same flame temperature, and the NO\textsubscript{x} reduction was due to the same mechanisms as described by Bhargava et al [274]. Bianco et al. [277] made simulations of premixed flame combustion chambers for EvGT. A lean burn combustor fed with a homogenous mixture of methane and humid air (0–5% water–air mass ratio) was simulated. The moisture reduced both NO\textsubscript{x} and carbon monoxide at a constant equivalence ratio. The lean blowout limit was shifted toward higher equivalence ratio. The water increases the specific heat capacity of the mixture, reducing the flame temperature and thus slowing the chemical reactions forming NO\textsubscript{x}. Belokon et al. [278] experimentally investigated diffusion flame combustors fired with methane and humid air with up to 20% water by mass. The experimental data was compared with data from steam injection in an LM5000 combustor. The experiments were used to create a model for prediction of combustion efficiency and NO\textsubscript{x} emissions for humid air cycles. In the evaporative cycle pilot plant, the NO\textsubscript{x} emissions were below 10 ppm and the levels of unburned hydrocarbons and carbon monoxide were very low [195].

7.3. Normalizing gas turbine data for humidified gas turbines

Normalizing gas turbine data to ISO conditions from different ambient conditions for an evaporative gas turbine with conventional techniques is difficult. In the evaporative cycle pilot plant, the performance test had to be performed on a standard day to avoid this problem [195]. However, experimental data from the pilot plant and an artificial neural network system were used to generate an empirical model of the cycle. This model was successfully used for data normalization and performance prediction [279]. Mathioudakis [280] discussed how the data from acceptance testing of gas turbines could be corrected when water injection is used.

7.4. Carbon dioxide recovery from humidified gas turbines

Humidified gas turbines have a high concentration of water in the exhaust gas. If the water is condensed, the remaining exhaust gas flow rate is decreased and the CO\textsubscript{2} concentration increased. This enables less expensive post-combustion CO\textsubscript{2} removal than for dry gas turbines. Rao and Day [281] studied CO\textsubscript{2} removal for the proposed FT 4000 HAT gas turbine and an advanced (triple-pressure, reheat, closed circuit steam cooling) combined cycle. The exhaust gas from the power plant was first cooled in a direct-contact condenser that condensed the water before the CO\textsubscript{2} recovery unit, an amine scrubber driven by steam. The HAT cycle had a high water concentration in the exhaust gas, resulting in a low flow rate with a high CO\textsubscript{2} concentration to the recovery unit. Thus, the operating and investment costs of the CO\textsubscript{2} recovery unit was lower for the HAT cycle compared with the combined cycle. The HAT cycle with CO\textsubscript{2} recovery had a specific investment
cost saving of 50 USD/kWₑ (6%) compared with the combined cycle. With CO₂ removal, the net efficiencies and power outputs were 55.0% (> 60% without CO₂ recovery) and 215 MWₑ for the HAT cycle and 54.2% and 361 MWₑ for the combined cycle. The authors estimated that optimizing the compressor pressure split could increase the efficiency of the HAT without CO₂ removal by 2% points. Rao et al. [147] proposed a system integrating a HAT with a SOFC and CO₂ removal. The fuel cell used pure oxygen instead of air, thus the exhaust gas consisted of CO₂ and water. The water was condensed in a direct-contact condenser; a part of the CO₂ was purged from the system while the rest was recycled to the HAT cycle to moderate the temperature in the SOFC. The cycle efficiency was over 60%. Carcasci et al. [282] presented a semi-closed evaporative cycle. In this cycle, some of the exhaust gas was recirculated and only the stoichiometric amount of air was used for combustion, which resulted in reduced compressor work and an exhaust gas with a high CO₂ concentration that facilitated the removal of CO₂ with absorption methods. An efficiency of over 50% was possible with CO₂ separation and delivery at ambient pressure and temperature. After the recuperator, the exhaust gas passed CO₂ absorption and water recovery units. Some of the exhaust gas was mixed with ambient air and sent back to the compressor. Corti et al. [283] calculated an efficiency of 49% including CO₂ removal by absorption for an intercooled recuperated semi-closed gas turbine with water injection for aftercooling. Desideri and Corbelli [284] studied a Cheng cycle, based on an existing Italian Cheng cycle operational since 1991 for cogeneration of power and district heating, with the inclusion of an absorption scrubber for CO₂ removal. The CO₂ emissions could be reduced with only slight decreases in efficiency and power output.

8. Discussion

In this chapter, various aspects of humidified gas turbines will be discussed. The thermodynamic performances of the humidified gas turbines suggested in literature will be compared to data for existing gas turbine cycles, the costs and potential applications of humidified gas turbines will be addressed, and the development status and critical questions for humidified gas turbines will be considered.

8.1. Thermodynamic performance

Performance data for some of the natural gas-fired humidified gas turbines for power generation only proposed in literature have been compared with data for commercial gas turbine cycles in Figs. 3 and 4. To assemble as complete a picture of the performance of the theoretical cycles as possible, most feasible cycles in the studies cited in this paper have been included in Figs. 3 and 4. In order for a cycle to be feasible, the electrical efficiency and the approximate power output must be given in the paper, and the authors of this paper have preferred to include only cycles based on relatively new gas turbine technology. However, the included cycles have been simulated using different assumptions, for example for gas turbine cooling, pressure drops in the heat exchangers and the maximum possible amount of water or steam that can be added to the cycle without operational problems. Therefore, the resulting trend lines contain some uncertainty. For example, the trend line for the water-injected gas turbines in Fig. 4 is much affected by just two points, numbers 8 and 9. However, the authors deemed it wiser to include these points, since it is not possible to judge which studies are the most reliable regarding the accuracy of the assumptions and calculation methods. Details and sources for the literature data are given in Tables 3–5, and in Table 6, some additional data for humidified gas turbines, not shown in
Fig. 3. Literature data for natural gas-fired water-injected gas turbines (black rhombs Nos. 1–9, Table 3), steam-injected gas turbines (black triangles Nos. 10–33, Table 4) and evaporative gas turbine cycles (black squares Nos. 34–59, Table 5) compared with existing simple cycle gas turbines, steam-injected gas turbines and combined cycles [91].

Fig. 4. Trend lines (black) for literature data for humidified gas turbine cycles compared with trend lines (gray) for existing simple cycle gas turbines, steam-injected gas turbines and combined cycles [91].
the figures, are given. The abbreviations used in the tables are explained in the nomenclature list in the appendix. The literature data are compared with commercial simple cycle gas turbines, steam-injected gas turbines and combined cycles with power outputs below 500 MW, for which data have been taken from the Gas Turbine World Handbook [91]. The simple cycle gas turbines in the Gas Turbine World Handbook have power outputs between 0.2 and 334 MW, and efficiencies between 16 and 42%, the steam-injected gas turbines have power outputs between 2 and 51 MW, and efficiencies between 32 and 44% and the combined cycles have power outputs between 7 and 972 MW, and efficiencies between 40 and 60%. The method for cooling the condenser of the steam cycle in the combined cycles is not given in the Gas Turbine World Handbook. The efficiency of a combined cycle depends on this method: cooling water from, for example, the sea or a lake has a lower temperature than cooling water from a cooling tower, resulting in a lower condenser pressure (i.e. a higher electrical efficiency). In Fig. 3, 59 theoretical humidified gas turbines are compared with logarithmic trend lines calculated by Excel for the commercial gas turbines. The individual data points for the commercial gas turbines are included as well, since the trend lines only show the mean value. In Fig. 4, logarithmic trend lines for the literature data are compared with trend lines for the commercial gas turbine cycles.

Performance data for different cycles cited in this study should only be compared qualitatively and not quantitatively, since the sources for the data in some cases lack complete information about the calculation methods (e.g. shaft or net power output, lower or higher heating value for the fuel), the cycle

### Table 3

<table>
<thead>
<tr>
<th>No.</th>
<th>$P$ (MW)</th>
<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT (°C)</th>
<th>Cycle features</th>
<th>Comments</th>
<th>Source</th>
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<tr>
<td>1</td>
<td>17</td>
<td>44.8</td>
<td>16.7</td>
<td>1250</td>
<td>WI</td>
<td>Ind. GT (Cyclone); SC: 12. 4 MW, 32.8%</td>
<td>[63]</td>
</tr>
<tr>
<td>2</td>
<td>23.8</td>
<td>47.1</td>
<td>16.3</td>
<td>1250</td>
<td>WI</td>
<td>Aero. GT; SC: 19.6 MW, 32.7%</td>
<td>[64]</td>
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<tr>
<td>3</td>
<td>28.9</td>
<td>44.2</td>
<td>16.3</td>
<td>1307</td>
<td>WI</td>
<td>Aero. GT (GE LM2500); SC: 24.4 MW, 35.0%</td>
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<tr>
<td>4</td>
<td>34.1</td>
<td>52.4</td>
<td>13.5</td>
<td>1062</td>
<td>WI IC RH</td>
<td>Thermal eff., ABB GT10; SC: 21.3 MW, 32.5%; CC: 32.4 MW, 49.4%</td>
<td>[74]</td>
</tr>
<tr>
<td>5</td>
<td>50</td>
<td>47.7</td>
<td>30</td>
<td>1400</td>
<td>WI EIC</td>
<td>Approximate P; SC: 50 MW, 39.3%; PR = 40, TIT = 1400; CC: 50 MW, 52.9%, PR = 18, TIT = 1400</td>
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<tr>
<td>6</td>
<td>60</td>
<td>50.4</td>
<td>22</td>
<td>1377</td>
<td>WI EIC</td>
<td>Gross eff.; SI IC REC: 60 MW, 45.0%; PR = 22, TIT = 1377</td>
<td>[73]</td>
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<tr>
<td>7</td>
<td>80</td>
<td>50.0</td>
<td>35</td>
<td>N/A</td>
<td>WI EIC</td>
<td>Rolls-Royce Trent; P = 50-80 MW, SC: 51.2 MW, 41.6%; CC: 66.0 MW, 54.3%</td>
<td>[70]</td>
</tr>
<tr>
<td>8</td>
<td>160</td>
<td>55.1</td>
<td>33</td>
<td>1500</td>
<td>WI IC</td>
<td>Aero. GT; SC: 40 MW, 39.7%; PR = 30, TIT = 1250</td>
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<td>9</td>
<td>272</td>
<td>58.2</td>
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<td>N/A</td>
<td>WI TopHAT</td>
<td>Large GT; SC: 160 MW, 39.4%; CC: 234 MW, 57.6%</td>
<td>[58]</td>
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</table>

The data are included in Fig. 3 as black triangles and in Fig. 4 as a black trend line.

*a* TIT or a similar temperature before the turbine.
<table>
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<tr>
<th>No.</th>
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<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT ($^\circ$C)\textsuperscript{a}</th>
<th>Cycle features</th>
<th>Comments</th>
<th>Source</th>
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<td>10</td>
<td>6.5</td>
<td>37.8</td>
<td>10.0</td>
<td>957</td>
<td>Turbo-STIG</td>
<td>Solar Centaur; SC: 3.8 MW\textsubscript{e}, 27.1%, PR = 10.2, same TIT</td>
<td>[119]</td>
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<tr>
<td>11</td>
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<td>42.3</td>
<td>10.1</td>
<td>982</td>
<td>Turbo-STIG</td>
<td>Allison 501; SC: 3.3 MW\textsubscript{e}, 29.4%, PR = 9.7, same TIT</td>
<td>[119]</td>
</tr>
<tr>
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<td>12</td>
<td>40.7</td>
<td>9.5</td>
<td>970</td>
<td>Turbo-STIG</td>
<td>Sulzer Type 3; SC: 6.3 MW\textsubscript{e}, 29.6%, same PR and TIT</td>
<td>[119]</td>
</tr>
<tr>
<td>13</td>
<td>27.5</td>
<td>44.5</td>
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<td>N/A</td>
<td>SI</td>
<td>LM2500; STIG: 28.1 MW\textsubscript{e}, 41.0%</td>
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</tr>
<tr>
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<td>41.6</td>
<td>21.4</td>
<td>1250</td>
<td>SI (Cheng)</td>
<td>Aero. GT; SC: 19.6 MW\textsubscript{e}, 32.7%</td>
<td>[64]</td>
</tr>
<tr>
<td>15</td>
<td>32.7</td>
<td>49.0</td>
<td>13.5</td>
<td>1062</td>
<td>SI IC REC</td>
<td>Thermal eff.; ABB GT10; SC: 21.3 MW\textsubscript{e}, 32.5%; CC: 32.4 MW\textsubscript{e}, 49.4%</td>
<td>[74]</td>
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<tr>
<td>16</td>
<td>38.5</td>
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<td>SI IC REC</td>
<td>Thermal eff.; ABB GT10; SC: 21.3 MW\textsubscript{e}, 32.5%; CC: 32.4 MW\textsubscript{e}, 49.4%</td>
<td>[74]</td>
</tr>
<tr>
<td>17</td>
<td>43</td>
<td>50.0</td>
<td>24.4</td>
<td>1250</td>
<td>SI (Cheng)</td>
<td>SC: 18.8 MW\textsubscript{e}, 35.2%, PR = 18, same TIT</td>
<td>[98]</td>
</tr>
<tr>
<td>18</td>
<td>45.1</td>
<td>52.7</td>
<td>40</td>
<td>1500</td>
<td>SI (Cheng)</td>
<td>Approximate P; SC: 50 MW\textsubscript{e}, 39.3%, PR = 40, same TIT; CC: 50 MW\textsubscript{e}, 52.9%, PR = 18, same TIT</td>
<td>[69]</td>
</tr>
<tr>
<td>19</td>
<td>50</td>
<td>50.7</td>
<td>30</td>
<td>1400</td>
<td>SI EIC SCO</td>
<td>Approximate P; SC: 50 MW\textsubscript{e}, 39.3%, PR = 40, same TIT; CC: 50 MW\textsubscript{e}, 52.9%, PR = 18, same TIT</td>
<td>[115]</td>
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<tr>
<td>20</td>
<td>50</td>
<td>52.0</td>
<td>32</td>
<td>1400</td>
<td>SI SCO</td>
<td>Assumed mid-size P; SC: 49.0%, PR = 30, TIT = 1250; CC: 58.3%, PR = 30, TIT = 1250, 3-p HRSG; SI SCO: 51.0%, TIT = 1250</td>
<td>[69]</td>
</tr>
<tr>
<td>21</td>
<td>60</td>
<td>47.6</td>
<td>20</td>
<td>1377</td>
<td>SI</td>
<td>Gross eff.; SI IC REC: 60 MW\textsubscript{e}, 45.0%, PR = 22, same TIT</td>
<td>[73]</td>
</tr>
<tr>
<td>22</td>
<td>80</td>
<td>50.0</td>
<td>N/A</td>
<td>N/A</td>
<td>SAT</td>
<td>Rolls-Royce Trent; P = 50-80 MW\textsubscript{e}</td>
<td>[125]</td>
</tr>
<tr>
<td>23</td>
<td>83.3</td>
<td>45.3</td>
<td>12.2</td>
<td>1187</td>
<td>Turbo-STIG</td>
<td>Canadian Westinghouse 251 B 10; SC: 37.4 MW\textsubscript{e}, 31.0%, PR = 13.8, same TIT</td>
<td>[119]</td>
</tr>
<tr>
<td>24</td>
<td>89</td>
<td>46.7</td>
<td>15.5</td>
<td>1085</td>
<td>Turbo-STIG</td>
<td>ABB Type 8; SC: 47.7 MW\textsubscript{e}, 32.8%, same PR and TIT</td>
<td>[119]</td>
</tr>
<tr>
<td>25</td>
<td>100</td>
<td>51.2</td>
<td>36</td>
<td>1250</td>
<td>SI IC</td>
<td>Assumed P</td>
<td>[112]</td>
</tr>
<tr>
<td>26</td>
<td>100</td>
<td>53.2</td>
<td>45</td>
<td>1500</td>
<td>SI IC</td>
<td>Assumed P</td>
<td>[112]</td>
</tr>
<tr>
<td>27</td>
<td>110</td>
<td>53.0</td>
<td>N/A</td>
<td>N/A</td>
<td>SI IC</td>
<td>GE LM5000</td>
<td>[111]</td>
</tr>
<tr>
<td>28</td>
<td>160</td>
<td>56.8</td>
<td>40</td>
<td>1300</td>
<td>SI RH ST</td>
<td>Assumed P; steam reheat</td>
<td>[120]</td>
</tr>
<tr>
<td>29</td>
<td>200</td>
<td>55.9</td>
<td>40</td>
<td>1300</td>
<td>SI RH ST</td>
<td>Assumed P; SI: 54.5%, same PR and TIT</td>
<td>[120]</td>
</tr>
<tr>
<td>30</td>
<td>220</td>
<td>53.2</td>
<td>45</td>
<td>1500</td>
<td>SI IC</td>
<td>Aero. GT; SC: 40 MW\textsubscript{e}, 39.7%, PR = 30, TIT = 1250</td>
<td>[68]</td>
</tr>
</tbody>
</table>

(continued on next page)
parameters differ (e.g. turbine inlet temperature, pressure ratio) and the technology assumptions differ (e.g. how much water that may be injected without operational problems). However, the trend lines in Fig. 4 confirm the conclusions from exergy analyses performed in several studies: the evaporative cycle has the potential for the highest efficiency of the different humidified gas turbine cycles, followed by the water-injected gas turbine and then the steam-injected gas turbine. The low exergy destruction for evaporation and air–water mixing in the humidification tower of the evaporative cycle results in a higher cycle efficiency for this cycle than for the water-injected cycle, where the exergy destruction for evaporation and mixing is larger due to the larger temperature difference in the evaporation chamber. The steam-injected gas turbine has a lower efficiency than the other humidified cycles since the exergy destruction for steam-air mixing is relatively large and the HRSG pinch point limits the heat recovery from the exhaust gas. One feature of humidified gas turbines not shown in Figs. 3 and 4 is that the specific power output is larger than for simple cycle gas turbines and combined cycles due to the humidification.

In some of the cited studies, the turbine inlet temperatures and pressure ratios (as shown in Tables 3–6) are higher than the current state-of-the-art values. Today, gas turbines with firing temperatures of 1430 °C (i.e. the GE MS7001H and MS9001H) and pressure ratios of 35 (i.e. the Rolls-Royce Trent) are available [91]. The most efficient gas turbine cycle available today is a combined cycle based on a GE H technology gas turbine. In this cycle, advanced gas turbine materials and a triple-pressure steam cycle with reheat of the steam by gas turbine cooling will result in a net efficiency of 60% and a power output of 480 MW (50 Hz) or 400 MW (60 Hz) [285]. The construction of the first system of this type (50 Hz) was completed in 2002 and the plant is currently undergoing testing before commercial operation starts [286]. Some humidified gas turbines that outperform this combined cycle are shown in Fig. 4. However, the largest potential for humidified gas turbines is for smaller power outputs (below 100 MW), where the efficiency of the combined cycles are decreased relatively more than for the humidified cycles, as shown in Fig. 4. If steam is available, in steam-injected gas turbines or in EvGT including steam boilers, steam cooling of the gas turbine can improve the efficiency. Very high efficiencies could be achieved for systems combining humidified gas turbines and fuel cells, as shown in Table 6.
<table>
<thead>
<tr>
<th>No.</th>
<th>$P$ (MW_e)</th>
<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT (°C)$^a$</th>
<th>Cycle features</th>
<th>Comments</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>34</td>
<td>4.6</td>
<td>42.7</td>
<td>12.7</td>
<td>N/A</td>
<td>EG</td>
<td>SC: 25.0%; SC REC: 28.5%; SC IC REC: 36.1%</td>
<td>[227]</td>
</tr>
<tr>
<td>35</td>
<td>12.1</td>
<td>46.4</td>
<td>N/A</td>
<td>1102</td>
<td>CHAT</td>
<td>Allison 501-KB7; TIT$<em>{lp}$ exp = 1102, TIT$</em>{hp}$ exp = 816, $p_{max}$ ≈ 60 bar</td>
<td>[152]</td>
</tr>
<tr>
<td>36</td>
<td>37</td>
<td>45.1</td>
<td>–</td>
<td>N/A</td>
<td>CHAT</td>
<td>Rolls-Royce Avon; SC: 28.2%</td>
<td>[151]</td>
</tr>
<tr>
<td>37</td>
<td>40</td>
<td>54.3</td>
<td>15</td>
<td>1400</td>
<td>EG</td>
<td>Water inj. for inlet air cooling</td>
<td>[126]</td>
</tr>
<tr>
<td>38</td>
<td>42.6</td>
<td>55.1</td>
<td>40</td>
<td>1500</td>
<td>EG</td>
<td>Full-flow humidification</td>
<td>[166]</td>
</tr>
<tr>
<td>39</td>
<td>46.4</td>
<td>55.1</td>
<td>40</td>
<td>1500</td>
<td>EG</td>
<td>Part-flow humidification</td>
<td>[166]</td>
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<tr>
<td>40</td>
<td>50</td>
<td>50.6</td>
<td>30</td>
<td>1400</td>
<td>EG EIC</td>
<td>Approximate P; SC: 50 MW_e, 39.3%; PR = 40, same TIT; CC: 50 MW_e, 52.9%, PR = 18, same TIT</td>
<td>[69]</td>
</tr>
<tr>
<td>41</td>
<td>50</td>
<td>52.0</td>
<td>30</td>
<td>1400</td>
<td>EG</td>
<td>Same as above</td>
<td>[69]</td>
</tr>
<tr>
<td>42</td>
<td>73.8</td>
<td>55.3</td>
<td>20</td>
<td>1400</td>
<td>EG</td>
<td>Ind. GT; 2-section hum. tower</td>
<td>[188]</td>
</tr>
<tr>
<td>43</td>
<td>74</td>
<td>55.2</td>
<td>20</td>
<td>1427</td>
<td>EG</td>
<td>Full-flow hum.; CC: 62 MW_e, 54.5%</td>
<td>[167]</td>
</tr>
<tr>
<td>44</td>
<td>78.3</td>
<td>52.6</td>
<td>20</td>
<td>1358</td>
<td>EG</td>
<td>ALSTOM GTX100; full-flow hum.</td>
<td>[191]</td>
</tr>
<tr>
<td>45</td>
<td>82.6</td>
<td>56.0</td>
<td>29</td>
<td>1400</td>
<td>EG</td>
<td>SC: 41.2 MW_e, 38.2%; CC: 59.9 MW_e, 55.6%, 2-p HRSG reheat</td>
<td>[165]</td>
</tr>
<tr>
<td>46</td>
<td>94.3</td>
<td>52.3</td>
<td>–</td>
<td>N/A</td>
<td>CHAT</td>
<td>GE Frame 6B; CC: 48.7%; SC: 32.0%</td>
<td>[151]</td>
</tr>
<tr>
<td>47</td>
<td>97.9</td>
<td>52.9</td>
<td>35</td>
<td>1336</td>
<td>EG non-IC</td>
<td>Rolls-Royce Trent; part-flow hum.</td>
<td>[189]</td>
</tr>
<tr>
<td>48</td>
<td>102.5</td>
<td>50.9</td>
<td>35</td>
<td>1267</td>
<td>EG non-IC</td>
<td>Aero. GT; part-flow hum.; 2-section tower; CC: 51.6%; SC: 41.5%</td>
<td>[189]</td>
</tr>
<tr>
<td>49</td>
<td>143.5</td>
<td>54.9</td>
<td>–</td>
<td>1374</td>
<td>CHAT</td>
<td>GE Frame 6FA: $p_{max}$ = 68 bar, TIT$<em>{lp}$ exp = 1374, TIT$</em>{hp}$ exp = 871; CC: 53.0%; SC: 34.2%</td>
<td>[151]</td>
</tr>
<tr>
<td>50</td>
<td>157.4</td>
<td>55.5</td>
<td>37</td>
<td>N/A</td>
<td>EG (HAT)</td>
<td>FT 4000 HAT</td>
<td>[144]</td>
</tr>
<tr>
<td>51</td>
<td>202.2</td>
<td>53.5</td>
<td>23</td>
<td>1260</td>
<td>EG (HAT)</td>
<td>GE; CC: 205.7 MW_e, 49.5%; TIT = 1260</td>
<td>[142]</td>
</tr>
<tr>
<td>52</td>
<td>207.5</td>
<td>57.4</td>
<td>24</td>
<td>1260</td>
<td>EG (HAT)</td>
<td>ABB; CC: 265.9 MW_e, 53.4%, TIT = 1260</td>
<td>[142]</td>
</tr>
<tr>
<td>53</td>
<td>209.1</td>
<td>55.7</td>
<td>40.6</td>
<td>1319</td>
<td>EG (HAT)</td>
<td>FT 4000 HAT</td>
<td>[145]</td>
</tr>
<tr>
<td>54</td>
<td>220.0</td>
<td>62.0</td>
<td>N/A</td>
<td>N/A</td>
<td>EG (HAT)</td>
<td>FT 4000 HAT, optimized compressor pressure split</td>
<td>[281]</td>
</tr>
<tr>
<td>55</td>
<td>221.1</td>
<td>60.0</td>
<td>N/A</td>
<td>N/A</td>
<td>EG (HAT)</td>
<td>Aero. GT; CC: 153 MW_e, 57.1%, 3-p HRSG reheat IC; SC IC: 125 MW_e, 46.5%</td>
<td>[281]</td>
</tr>
<tr>
<td>56</td>
<td>287</td>
<td>57.7</td>
<td>46</td>
<td>1498</td>
<td>EG</td>
<td>FT 4000 HAT</td>
<td>[200]</td>
</tr>
</tbody>
</table>

(continued on next page)
Table 5 (continued)

<table>
<thead>
<tr>
<th>No.</th>
<th>$P$ (MW$_e$)</th>
<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT ($^\circ$C)$^a$</th>
<th>Cycle features</th>
<th>Comments</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>57</td>
<td>310</td>
<td>57.0</td>
<td>48</td>
<td>1500</td>
<td>EG</td>
<td>Aero. GT; SC: 40 MW$_e$, 39.7%, PR = 30, TIT = 1250; CC IC: 170 MW$_e$, 55.5%, PR = 46, TIT = 1500</td>
<td>[68]</td>
</tr>
<tr>
<td>58</td>
<td>354</td>
<td>59.2</td>
<td>–</td>
<td>1394</td>
<td>CHAT</td>
<td>$p_{max} = 67$ bar, TIT$<em>{lp \ exp} = 1394$, TIT$</em>{hp \ exp} = 1149$</td>
<td>[148]</td>
</tr>
<tr>
<td>59</td>
<td>400</td>
<td>63.0</td>
<td>–</td>
<td>1482</td>
<td>CHAT</td>
<td>assumed P, TIT$<em>{lp \ exp} = 1482$, TIT$</em>{hp \ exp} = 1149$</td>
<td>[148]</td>
</tr>
</tbody>
</table>

The data are included in Fig. 3 as black squares and in Fig. 4 as a black trend line.

Table 6

<table>
<thead>
<tr>
<th>$P$ (MW$_e$)</th>
<th>$\eta_{el}$ (%)</th>
<th>PR</th>
<th>TIT ($^\circ$C)$^a$</th>
<th>Cycle features</th>
<th>Comments</th>
<th>Source</th>
</tr>
</thead>
</table>

Water-injected gas turbine cycles

| N/A | 45.3 | 20  | 1277              | WI EIC EREC   | Utility GT, gross eff. | [73] |
| N/A | 49.0 | N/A | N/A               | WI EIC SI     | Large ind. GT, SI: 45–46%; CC: 52–55% | [84] |
| N/A | 49.1 | 9.5 | 1277              | WI SAT        | Shaft eff., SI: 47.9%, PR = 30, TIT = 1377 | [83] |
| N/A | 50.5 | 22  | 1427              | WI EIC EREC   | Aero. GT, gross eff. | [73] |
| N/A | 51.4 | 30  | 1377              | WI EIC SAT    | Shaft eff., SI EIC: 48.9%, same PR and TIT | [83] |

| N/A | 58.1 | 6   | 1200              | WI EREC       | Thermal eff. | [71] |
| N/A | 56.1 | 6   | 1200              | WI IC EREC    | Thermal eff. | [71] |

Steam-injected gas turbine cycles

| N/A | 42.5 | 19  | 1277              | SI            | Utility GT, gross eff. | [73] |
| N/A | 47.8 | 20  | 1327              | SI            | Aero. GT, gross eff. | [73] |
| 17.2| 69.0 | 10  | 590               | SI MCFC       | Topping combustor | [124] |
| 20.8| 66.0 | 15  | 1200              | SI MCFC       | Topping combustor | [124] |

Evaporative gas turbine cycles

| N/A | 54.0 | N/A | 1200              | WI IC EIC EREC |             | [80] |
| 3–5 | 69.1 | 15  | N/A               | HAT + SOFC    | Two fuel cells | [146] |
| 3–5 | 76.0 | 15  | N/A               | HAT + SOFC    | Two fuel cells | [146] |
| 300 | 75.0 | 20  | 1200              | HAT + SOFC    |                | [147] |

$^a$ TIT or a similar temperature before the turbine.
Although the use of natural gas will probably increase in the future, solid fuels are less expensive and more abundant, and humidified gas turbine cycles could use these fuels efficiently. Solid fuels can be utilized in gas turbines, either by burning the fuel and using the cleaned combustion gases directly in the gas turbine or by gasifying the fuel and firing the cleaned gases in the gas turbine combustor. Some coal-fueled gas turbine power plants have been constructed with PFBC or gasification processes [287]. Other alternatives for solid or low-quality fuels are externally fired or closed cycle gas turbines, where the fuel energy is transferred indirectly to the gas turbine through a heat exchanger that replaces the combustion chamber of an open cycle gas turbine. A few closed cycle gas turbines have been in operation [8]. For reduced natural gas consumption, humidified cycles can integrate external heat sources (e.g. refuse) effectively. In addition, humidified cycles, especially evaporative cycles, fueled with gasified coal or biomass have a potential for higher efficiency and lower investment costs and costs of electricity than combined cycles. Externally fired humidified cycles show efficient performance as well.

8.2. Costs

The specific investment cost for a natural gas-fired humidified cycle, disregarding the development cost, should be lower than for a combined cycle, since the cost for the steam turbine in the combined cycle is avoided and the specific power output is high. The difference in specific investment cost should be larger for small power outputs, since the steam turbine accounts for a larger part of the combined cycle investment cost when the power output is reduced. In evaporative gas turbine cycles, part-flow humidification is one possible alternative for reduced specific investment cost, since the heat exchanger area and humidification tower volume are decreased with part-flow, while the electrical efficiencies of part-flow cycles are on the same level or higher than for full-flow cycles. The cost of electricity is mainly determined by the electrical efficiency; hence, the evaporative cycle should have a cost of electricity on the same level or below the combined cycle.

Most economic studies on the evaporative gas turbine confirms the assumption of a lower specific investment cost for the evaporative cycle compared with a combined cycle, for example the studies by Nilsson [167], Bartlett [166] and Jonsson and Yan [193]. However, in the two reports by EPRI on the HAT cycle [142,144], the specific investment costs for the HAT cycles were higher than for the combined cycles. The specific power output of the gas turbine has the largest impact on the specific investment cost: if a higher power output can be generated by a certain gas turbine (i.e. a higher specific power output can be achieved), the specific investment cost is reduced. Thus, the design assumptions for handling the extra working fluid mass flow rate resulting from the humidification are essential for the specific investment cost. As a not very realistic example: if it is assumed that the compressor size is constant for a certain gas turbine used in an evaporative cycle and a combined cycle, the extra mass flow of working fluid in the expander of the evaporative cycle increases the specific power output of the evaporative cycle and thus decreases the specific investment cost (if the cost for the gas turbine is assumed to be the same in both cycles). Since the exact design assumptions for the HAT cycle gas turbines are not discussed in the EPRI reports, it is impossible to evaluate the reasons for the higher specific investment costs for the HAT cycles in comparison with the combined cycles.
8.3. Applications

The potential applications for humidified gas turbines are many, for example, base-load power generation and cogeneration of power and steam or district heating. Humidified gas turbines have improved performance compared with dry gas turbines at part-load and at high ambient temperatures, since the water addition rate can be changed to compensate for changes in load and ambient conditions. The market for small-to-mid-size combined cycles is growing [288] due to power market deregulation and the introduction of independent power producers. Deregulated markets require small, flexible power plants with high efficiencies that can generate electricity at a low cost, thus humidified gas turbines, featuring high efficiencies, specific power outputs and flexibility, should be able to compete with the combined cycle on these markets. For cogeneration of power and steam or district heating, humidified gas turbines are more flexible than a combined cycle and can generate power and district heating to a low cost.

8.4. Development status and critical questions

Before the humidified gas turbines can compete with the conventional technology, they must be fully developed and tested. Technologies for water injection inside the compressor for intercooling (wet compression) are on the experimental stage, while similar on-line compressor washing systems are commercially available. There is also the GE Sprint system, where water is injected between the low-pressure and high-pressure compressors in a LM6000 gas turbine for intercooling. Spray intercooling seems to be a less expensive alternative for intercooling of multi-shaft gas turbines than surface intercooling, since it requires smaller design changes. Recuperated gas turbines with water injection for aftercooling are not commercially available and successful operation of such a system has not been reported in literature. Steam-injected gas turbines are an established technology most often used for small-to-medium-size cogeneration for power and steam in applications where the heat demand or electricity prices vary. However, steam-injected gas turbines with the high efficiencies cited in some theoretical studies are still to be realized. The operation of the evaporative gas turbine cycle has been validated in a pilot plant, although the technology still needs to be demonstrated in a larger scale to evaluate the full potential of the technology.

For successful operation of humidified gas turbines, some critical questions first need to be answered. One possible problem is that when steam or water is injected into a gas turbine, the compressor back pressure (i.e. the pressure ratio) is increased. The injection rate must be restricted to keep the pressure ratio below the surge line, which is the limit of stable operation for the compressor. In the surge region, some compressor blades are stalled, which means that the flow between the blades is reduced and flow separation may occur. Some working fluid may rotate in the opposite direction to the rotor, which results in vibrations that may damage the compressor [8].

Another problem is the volumetric flow rate mismatch in humidified gas turbines. When the gas turbine working fluid is humidified, the expander volumetric flow rate is increased in relation to the flow rate in the compressor. A gas turbine, for example with a compressor from a smaller gas turbine and a turbine from a larger gas turbine, could be designed for accommodating the increased flow; however, the cost would be substantial, as indicated by the studies on the FT 4000 HAT gas turbine [144]. Nilsson et al. [173] suggested using a free power turbine and/or multiple spools, although this is most suitable for medium- and large-size gas turbines, while for single-shaft gas turbines, a compromise for
the compressor and turbine design can be made or a gear used between them. In the commercially available steam-injected cycles, the flow mismatch problem has been solved by using partial steam addition or by using gas turbines with large surge margins; however, there are only a few gas turbines that can operate with full steam injection. For the evaporative cycle, the CHAT concept has been suggested to enable using standard gas turbine components in a humidified cycle. The CASH and CASHING cycles, which were designed for cycling power generation, solve the flow mismatch problem as well. Rao [141] suggested a HAT cycle where some of the humid air is expanded in a separate turbine to avoid mismatch of the flow rates in the compressor and turbine if an existing gas turbine is used in the cycle. Many authors suggest that aeroderivative gas turbines should be better suited for operation in humidified cycles than industrial gas turbines. Aeroderivative gas turbines are designed to handle flow rates in excess of the design value and their pressure ratios are higher than for industrial gas turbines; many studies have found high pressure ratios beneficial for the efficiency of humidified gas turbines. In addition, the multi-shaft design of aeroderivative gas turbines facilitates the integration of an intercooler. Another driving force for using aeroderivative gas turbines in humidified cycles is that aeroderivative gas turbines are not as suitable as industrial gas turbines for combined cycle applications, since the high pressure ratio results in a low exhaust gas temperature. Therefore, other cycle layouts than combined cycles are required for high electrical efficiencies for power cycles based on aeroderivative gas turbines.

Another critical question is the development of recuperators and intercoolers, which are required for humidified gas turbines to reach the highest efficiencies cited in this paper. Recuperators and intercoolers are not standard components in power plants today. Gas turbine recuperators are relatively scarce and have mostly been used for propulsive gas turbines, for which a low specific fuel consumption (kg/kWh) is important [289]. Furthermore, recuperators are required for competitive electrical efficiencies for micro gas turbines and primary surface recuperators [290] and plate-fin recuperators [291] have been developed for this application. Saidi et al. [292] reviewed different types of intercoolers. One example of an intercooled recuperated gas turbine is the WR-21, which will be used for warship propulsion with reduced fuel consumption [293]. The evaporative gas turbine cycle includes a component not previously used in power plants: the humidification tower; however, this component has extensive use in the chemical process industry. The knowledge transfer between mechanical and chemical engineers will be greatly helpful for the development and design of such components.

Yet another critical question is gas turbine operation with humidification. For direct water injection and steam injection, demineralized water is required, while water of a lower quality may be used in the evaporative cycle. For all humidified cycles, the water consumption increases the operational cost. The water can be recovered from the exhaust gas and re-used in the cycle after treatment; however, this increases the investment and operational costs. It has been shown in several installations that the water can be recovered from the exhaust gas and treated to achieve a sufficient quality for re-use in the cycle. If the water should be recovered or not depends on the cost of makeup water of a suitable quality. For the water-injected cycle, there are different recommendations for how much water that should be injected: some authors recommend that the air should be saturated or undersaturated at the inlet to the recuperator so as to not damage the recuperator by evaporation of remaining water droplets, while other authors propose systems where oversaturated air enters the recuperator or systems where additional water is injected inside the recuperator. If the air cannot be oversaturated at the recuperator inlet, this limits the water injection rate. One possible solution is to inject water and recuperate in several steps. Operating a gas turbine with humid air could also cause combustion instabilities and lower the combustion efficiency. However, the evaporative gas turbine pilot plant and many steam-injected gas turbines have
been operated without any significant problems. An advantage of using humid air as a gas turbine working fluid is that the water suppresses the generation of NO\textsubscript{x} in the combustion process, without the use of an expensive low-NO\textsubscript{x} burner.

8.5. Other non-technical issues

The research and development on humidified gas turbines in various cycle versions have been investigated by different groups of distinguished institutions supported by different resources in Europe, US and other regions. However, except for the Cheng and STIG cycles, the commercialization of other advanced humidified gas turbine configurations, which might provide a better performance, has not been fully implemented by industry. This is partly due to some technical problems that need to be solved as discussed before. However, there are other issues which might be ‘non-technical’ barriers which obstacle the further commercialization of the new technologies. For example, the market competition among the gas turbine manufacturers is tougher today than before. This may make the industry more conservative to adapt new technology, which may involve the risk and additional costs of the new market development. Industry would like to take the benefits of the technology advance if others can prove the technology in a commercial scale. But they are very reluctant to be the first to build such a pilot plant which has a substantially different configuration compared with their existing plants. The transfer of knowledge and collaboration between academia and industry could be helpful to solve the problems. A joint effort for the first commercialization of the advanced humidified gas turbine may involve all stakeholders such as government, industry and academia. Experiences from the Cheng and STIG cycles are of importance for the future commercialization of other humidified gas turbines.

9. Conclusions

In this review paper, a large number of papers discussing different humidified gas turbines have been summarized. The main thermodynamic conclusions of the review are:

- Humidified gas turbines have a thermodynamic potential of electrical efficiencies similar to, or higher than, combined cycle efficiencies.
- The evaporative gas turbine including a humidification tower has the highest efficiency of the humidified gas turbine cycles.
- Humidified gas turbines have higher specific power outputs than dry gas turbine cycles.

The conclusions regarding the costs of humidified gas turbines are:

- The specific investment cost for a mature humidified gas turbine should be lower than for a combined cycle, since the cost for the steam turbine in the bottoming cycle is avoided. The difference in specific investment cost is larger for small power outputs.
- Evaporative cycles with part-flow humidification promise lower specific investment costs than evaporative cycle with full-flow humidification, with the same or higher electrical efficiency.
Regarding the remaining development work and applications for humidified gas turbines, the following conclusions can be drawn:

- Only steam-injected gas turbines are available commercially today, although not with the high efficiencies cited in many theoretical studies. The evaporative gas turbine has been demonstrated in a pilot plant, while the water-injected cycle seems to be the least experimentally investigated humidified gas turbine cycle.
- Humidification of the gas turbine working fluid results in a volumetric flow rate mismatch between the compressor and expander. Gas turbines that can accommodate the increased flow rate are still to be developed.
- For humidified gas turbines to achieve the highest efficiencies cited in this review paper, intercooling and recuperation are required. Further development work remains before these heat exchangers are standard components in gas turbine power cycles.
- Aeroderivative gas turbines seem to be more suited for operation in humidified cycles than industrial gas turbines.
- It has been shown in several power plants that gas turbines can operate with a humidified working fluid and that the water can be recovered from the exhaust gas and re-used in the cycle.
- Cogeneration of power and district heating seems to be a promising application for humidified gas turbines. It has been estimated that the specific investment costs and costs of electricity and heat should be significantly lower than for combined cycles.
- Power production integrated with removal of carbon dioxide is feasible for humidified gas turbines, since the CO₂ removal system could be smaller than for a combined cycle and therefore less expensive. The cycle efficiency including CO₂ removal might also be higher than for combined cycles. In addition, humidification reduces the generation of NO in the combustion process.

Acknowledgements

Financial support from the Royal Institute of Technology and the Swedish Energy Agency (Statens energimyndighet) is gratefully acknowledged.

Appendix A

Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>ABB</td>
<td>Asea Brown Boveri</td>
</tr>
<tr>
<td>aero.</td>
<td>aeroderivative</td>
</tr>
<tr>
<td>CAES</td>
<td>compressed air energy storage</td>
</tr>
<tr>
<td>CASH</td>
<td>compressed air storage with humidification</td>
</tr>
<tr>
<td>CASHING</td>
<td>compressed air storage with humidification integrated with natural gas</td>
</tr>
<tr>
<td>CC</td>
<td>combined cycle</td>
</tr>
<tr>
<td>CHAT</td>
<td>cascaded humidified advanced turbine</td>
</tr>
</tbody>
</table>
CO₂ carbon dioxide
CTHAI combustion turbine humidified air injection
eff. efficiency
EG evaporative gas turbine (with intercooling, aftercooling, recuperation)
EIC evaporative/spray intercooler
EPRI Electric Power Research Institute
EREc water injection in recuperator or evaporation of previously injected water in the recuperator
EvGT evaporative gas turbine
GE General Electric
GT gas turbine
HAT humid air gas turbine
HHV higher heating value (MJ/kg)
HRSG heat recovery steam generator
IC surface intercooler
IEA International Energy Agency
IGCC integrated gasification combined cycle
IGCHAT integrated gasification cascaded humidified advanced turbine
IGHAT integrated gasification humid air turbine
IHI-FLECS Ishikawajima-Harima Heavy Industries flexible electric cogeneration system
ind. industrial
LHV lower heating value [MJ/kg]
MCFC molten carbonate fuel cell
Mtoe million tonnes of oil equivalent (1 Mtoe is approximately 41.9 GJ [294])
NOₓ nitrogen oxides (nitrogen monoxide, NO, and nitrogen dioxide, NO₂)
REC recuperator
RH reheat
SAT saturated steam injection
SC simple cycle gas turbine
SI steam-injected gas turbine
SCO steam cooling
SOFC solid oxide fuel cell
Sprint spray intercooling system (General Electric)
ST topping steam turbine
STIG steam-injected gas turbine (General Electric)
TopHat recuperated gas turbine cycle with water injection in the compressor
WI recuperated water-injected gas turbine (evaporative aftercooling)

**Parameters**

m mass flow rate (kg/s)
P power output (MWₑ)
p pressure (bar)
PR pressure ratio
TIT  turbine inlet temperature (°C) (the definition of the turbine inlet temperature varies between different papers)

η  efficiency

Subscripts

e  electricity  
el  electrical  
exp  expander  
hp  high-pressure  
lp  low-pressure  
max  maximum

References

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SEA. Energiläget. Eskilstuna: Swedish energy agency (Statens energimyndighet); 2002. [in Swedish].