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**Analysis of power generation cycles & application of
compression-vapour cycle**

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1. Part A

(a)

(i)

Assuming the compression cycle is 8:1.

Standard pressure is 101 325 Pa.

$$P_1 = 101\,325 \text{ Pa}$$

Using A-20 Table:

$$C_p = 1.005 \text{ kJ/kgK}$$

$$C_v = 0.718 \text{ kJ/kgK}$$

$$\gamma = 1.4$$

Calculation of T_1

$$32^\circ\text{C} + 273 = 305\text{K}$$

Calculation of V_1

Specific gas constant for air = 8.314kJ/kWK

Molar mass of air = 98.97kg/kmol

$$8.314/28.97 = 0.287\text{kJ/kgK}$$

To calculate $V(\text{m}^3/\text{kg})$ the following equations are used:

$$pV = RT \quad [1]$$

$$V = \frac{RT}{p} \quad [2]$$

$$V_1 = (0.287 \times 305) / 101.325 = 0.864 \text{ m}^3/\text{kg}$$

Calculation of V_2

$$r = 8 = \frac{V_1}{V_2} = \frac{V_4}{V_3} \quad [3]$$

Therefore,

$$V_2 = 0.864 / 8 = 0.108 \text{ m}^3/\text{kg}$$

Calculation of V_3

$$V_2 = V_3 \quad [4]$$

$$V_3 = 0.108 \text{ m}^3/\text{kg}$$

Calculation of V_4

$$V_1 = V_4 \quad [5]$$

$$V_4 = 0.864 \text{ m}^3/\text{kg}$$

Calculation of T_2

The temperature at the end of compression process for the compression process, 1-2;

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^{\gamma-1} \quad [6]$$

$$T_2 = T_1 \times r^{\gamma-1} \quad [7]$$

$$T_2 = 305 \times 8^{1.4-1}$$

$$T_2 = 700.706 \text{ K}$$

Calculation of P₂

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2} \right)^{\gamma} \quad [8]$$

$$P_2 = 101.325 \times 8^{1.4} = 1862.27 \text{ kPa}$$

Calculation of T₃

$$pV = RTv \quad [9]$$

$$\frac{V}{R} = \frac{T}{p} \quad \text{therefore,} \quad \frac{T_2}{P_2} = \frac{T_3}{P_2} \quad [10] \& [11]$$

$$\Delta u + \Delta ke + \Delta pe = (q_{in} + w_{in}) - (q_{out} + w_{out}) \quad [12]$$

Assuming kinetic energy and potential energy are negligible,

$$q_{in} = u_3 - u_2 \quad [13]$$

$$q_{in} = C_v(T_3 - T_2) \quad [14]$$

$$T_3 = \frac{q_{in}}{C_v} + T_2 \quad [15]$$

$$T_3 = (1650 \text{ kJ/kg} / 0.718) + 700.706 = 2998.76 \text{ K}$$

Calculation of P₃

$$\frac{V}{R} = \frac{T}{p} \quad \text{therefore,} \quad \frac{T_2}{P_2} = \frac{T_3}{P_3} \quad [16]$$

$$P_3 = 1862.27 \times \left(\frac{2998.76}{700.706} \right) = 7969.82 \text{ kPa}$$

Calculation of T4

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} \quad [17]$$

$$T_4 = T_3 \times \left(\frac{1}{r}\right)^{\gamma-1} \quad [18]$$

$$T_4 = 2\,998.76 \times \left(\frac{1}{8}\right)^{\gamma-1} = 1\,1305.29K$$

Calculation of P4

$$P_4 = P_3 \times \left(\frac{1}{r}\right)^{\gamma} \quad [19]$$

$$P_4 = 7\,969.82 \times \left(\frac{1}{8}\right)^{1.4} = 433.63kPa$$

Table 1. Temperature and pressure at all points of an Otto cycle.

	T(K)	P(kPa)	V(m ³ /kg)
1	305	101.325	0.864
2	700.706	1 862.27	0.108
3	2 998.76K	7 969.82	0.108
4	1 305.29	433.63	0.864

During an isotropic compression process, as the volume decreases, the temperature and pressure should increase as well. The heat energy added per cycle also causes the pressure and the temperature to increase when (constant volume). Then through expansion power is drawn out of the gas which causes the temperature to drop (1305.29K) and so should the pressure (433.63kPa).

On the other hand during an isochoric cooling process, the temperature and pressure both drop again (1 305.29K → 305K; 433.63kPa →101.325kPa)

To check the calculations are correct, ignoring any discrepancy due to rounding errors..

$$\frac{T_4}{P_4} = \frac{T_1}{P_1} \text{ therefore, } P_1 = \frac{T_1}{T_4} \times P_4 \quad [20]$$

$$P_1 = \left(\frac{305}{1305.29} \right) \times 433.63$$

The result of the above equation is 101.324. This indicates that the calculations are error-free because the P_1 from the equation above and the initially obtained P_1 result match.

(ii)

Air standard thermal efficiency calculated from the temperature points

$$\eta = 1 - \frac{\text{heat rejected}}{\text{heat supplied}} \quad [21]$$

$$\eta = 1 - \frac{m.C_v(T_4 - T_1)}{m.C_v(T_3 - T_2)} \quad [22]$$

$$T_1 = 305K$$

$$T_2 = 305 \times 8.5^{1.4-1} = 307.35K$$

$$T_3 = \left(\frac{1650}{0.718} \right) + 307.35 = 2605.40K$$

$$T_4 = 2605.40 \times \left(\frac{1}{8} \right)^{1.4-1} = 1134.07K$$

$$\eta = 1 - \frac{(1134.07-305)}{(2605.40-1134.07)} = 0.44 = 44\%$$

Air standard thermal efficiency is 44%.

Air standard efficiency calculated from the compression ratio

$$\eta = 1 - \frac{1}{(V_1/V_2)^{\gamma-1}} \quad [23]$$

$$\eta = 1 - \frac{1}{\left(\frac{0.864}{0.108} \right)^{1.4-1}} = 0.56 = 56\%$$

Air Standard efficiency is 56 % when determined using compression ratio data. It is about a 10% increase when compared to calculating it using temperature points. This is due to the fact that the second efficiency is less precise since it is assessed through the lens of an ideal world rather than the real one.

(b)

$$C_p = 1.005 \text{ kJ/kgK}$$

$$C_v = 0.718 \text{ kJ/kgK}$$

$$\gamma = \left(\frac{C_p}{C_v} \right) = 1.4 \text{ [24]}$$

To find T_1 :

$$T_1 = 5 + 273 = 323K$$

To find T_2 :

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2} \right)^{\gamma-1} \text{ [25]}$$

$$T_2 = 323 \left(\frac{17.5}{1} \right)^{1.4-1} = 1014.89K$$

To find T_3 :

$$V_2 = V_3 \text{ [26]}$$

(e)

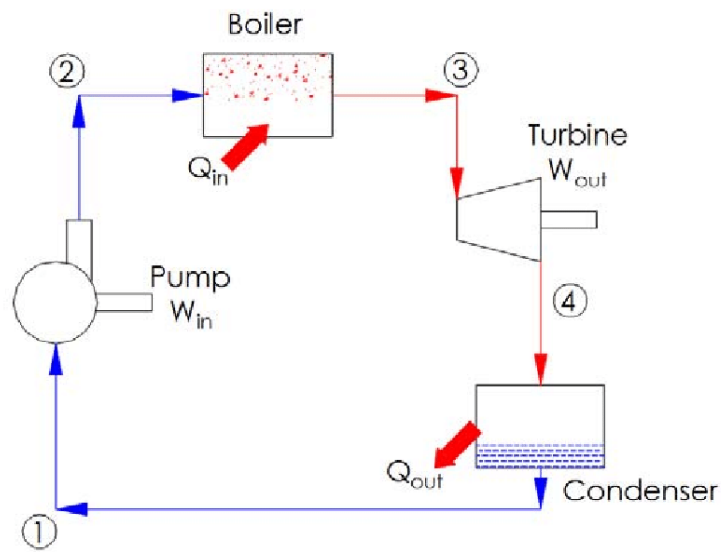


Image 1. Rankine cycle. (Omosanya, Abolaji & Akinlabi, Esther & Okeniyi, Joshua. 2019)

$$(1) P_1 = P_{low} = 0.04 \text{ mPa} \quad [27]$$

$$x_1 = 0$$

$$h_1 = 121.46 \text{ kJ/kg}$$

$$s_1 = 0.4226 \text{ kJ/kgK}$$

$$v_1 = 0.0010040 \text{ m}^3/\text{kg}$$

$$(2) P_2 = P_{high} = 4.4 \text{ mPa} \quad [28]$$

$$s_2 = s_1 \quad [29]$$

$$h_2 =$$

$$(3) P_3 = P_{high} \quad [30]$$

$$x_3 = 1$$

$$h_3 =$$

$$s_3 =$$

$$(4) P_4 = P_{low} \quad [31]$$

$$s_4 = s_3 \quad [32]$$

$$h_4 =$$

2. Part B

Gas turbines operate on a constant-volume cycle so the power production is proportional to air mass flow. High-temperature air is less dense than cold air, resulting in lower power output. Warm air also requires more energy to compress than cold air, requiring the compressor to perform more work and increasing internal losses. The compressor can consume a lot of the total turbine power. As a result, cooling the incoming air can enhance gas turbine efficiency, particularly useful in hot, dry regions (Ibrahim, Rahman and Abdalla, 2011). Turbine air-intake cooling can reduce (or eliminate) fuel efficiency loss by evaporative cooling the incoming air to the wet-bulb temperature (Ibrahim, Rahman and Abdalla, 2011).

Note, however, that chilling past the wet-bulb temperature will boost power while providing little or no net efficiency benefit (Amer, Boukhanouf and Ibrahim, 2015).

Mechanical and absorption chillers (Kwon et al., 2018) and evaporative cooling (wetted media, fogging, and wet compression/overspray) (Brun et al., 2013) are options for cooling gas turbine air inlets.

Water droplets can get sprayed into the intake air, effectively intercooling the compressor, reducing compression effort, and increasing efficiency (Muhammad Izzat Nor et al., 2019). The CHP plant also benefits from this strategy since the increased mass flow enhances the exhaust gas's specific heat. Liquid natural gas (LNG) vaporisation produces a cooling effect (Zhang et al., 2010), which is good if LNG gets used as plant fuel.

2.1 Gas turbine

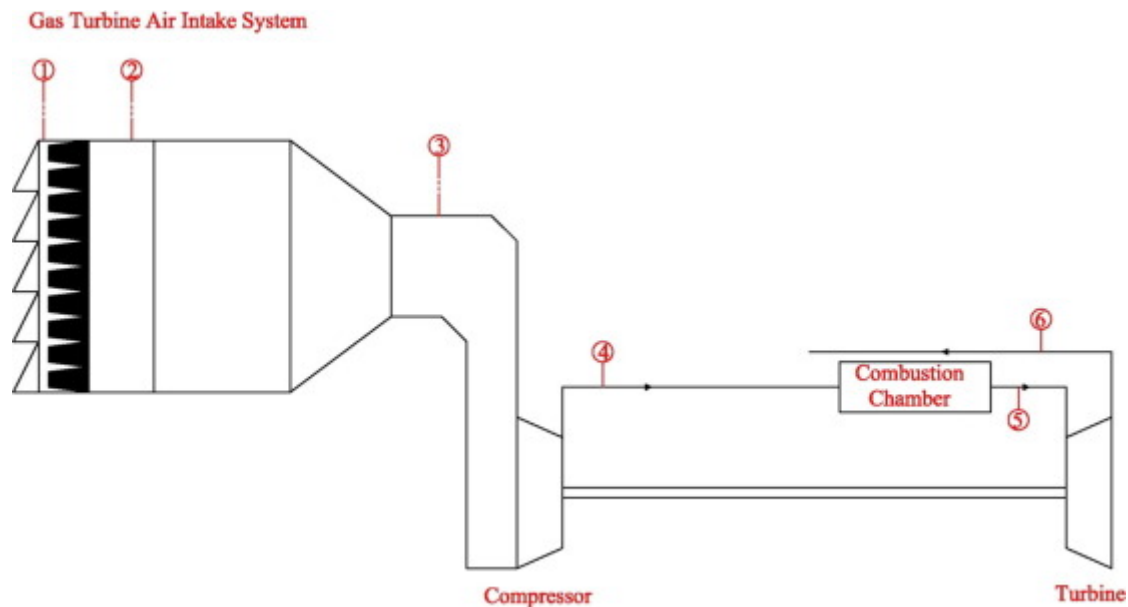


Image 2. Simple gas turbine cycle (El-Shazly et al., 2016).

Inlet air warming

The best plant efficiency does not necessitate the maximum gas turbine efficiency or pressure ratio in a combined cycle. To maximise the steam cycle production, a balance between the gas turbine firing (combustion) temperature and pressure ratio must get achieved. In such cases, scavenged heat inlet air warming can enhance CCGT (Combined Cycle Gas Turbines) efficiency on cold days. Inlet air warming is also frequently employed to defend against the mechanical limits of gas turbines.

Inlet air warming can also advantage gas turbine plants operating in cold locations to reduce de-icing requirements or maintain a high exhaust temperature (Ojo et al., 2010) for an efficient combined cycle or CHP plant operation.

Steam/water injection

Gas turbines can be fed with steam or water to boost output power, enhance part-load flexibility, and reduce exhaust pollutants considerably (Nikbakht Naserabad, Mehrpanahi and Ahmadi, 2019). Saturated or superheated steam may typically be pumped into the combustion chamber, although this will boost pressure significantly, perhaps exceeding the compressor's aerodynamic/mechanical design limitations. As a result, the steam/fuel ratio is likely to be restricted, making steam injection impossible for all gas turbines, which reduces efficiency. However, the increased mass flow rate through the turbine increases power output.

Steam-operated machines' energy-saving techniques have caused seasonal thermal/electric load changes. The adaptable heat and power gas turbine can react effectively to such fluctuating demands using thermal and electric energy. Saturated steam (from a heat recovery steam generator) is pumped into the gas turbine to boost production. Varying the amount of steam injection allows for variable heat and power operation. The turbine may run in water or steam injection modes and transition between them to decrease NO_x emissions. The adjustable heat and power range significantly expand when the steam gets fed. The use of superheated injected steam allows for even more efficient energy generation.

Filtering inlet air

Dirt from the intake air (rarely, the fuel supply) can damage the blades due to gas turbines' high and demanding working conditions. Particulates clinging to the blades (fouling), corroding them, or degrading them cause this. These negative consequences lower blade efficiency, impair performance and raise maintenance costs. Turbine blades have exact aerodynamic properties; any damage will diminish mass flow and the effective pressure ratio—both lower fuel economy. Inlet air filtration (and fuel filtration) is necessary to keep most particulates from the turbine (Brumbaugh, 2002). However, such systems cause a pressure decrease which is an inefficiency in and of itself. Even a slight reduction in pressure drop may save fuel expenses for big plants. As a result, an excellent filter design should balance filtering efficiency with minimum pressure drop to minimise pressure drop.

Because replacing gas turbine's air intake filter cartridges may be costly, pre-filters that catch bigger particles are frequently used to extend the life of the primary filter. However, if dirt builds up too much in pre-filters, pressure drops; thus, they should be able to be cleaned or replaced while the gas turbine is running. Pre-filters can go for years without being washed.



Image 3. Example of a filter `DNA Round Clamp 62mm Inlet Air Filter for BMW R9T (14-19) ONLY OV-6200-125-R9T` (e-dnafilters, 2022).

Filter monitoring

Thousands of filter components must get examined regularly or when scheduled for maintenance on air intake filter banks. Both activities are time-consuming, and electronically monitoring air filters is a better technique to maximise their efficiency. Modern filter monitoring systems give the precise quantitative measurements and cross-sectional mapping of air filter performance that is needed.

Parasitic energy losses

All gas turbine moving elements, such as bearings and seals, cause parasitic energy losses. Frictional losses in bearings lose energy, produce heat, and shorten bearing life, including lubricant life. Simply supplying oil to the bearings consumes a considerable amount of the total fluid film bearing losses in high-speed turbomachinery.

The plant's auxiliary equipment can absorb a significant portion of the overall power production. Major electrical demands are the enclosure ventilation fan motors and the liquid fuel pump motors. Losses can be reduced by carefully selecting, operating, and maintaining equipment. The fuel gas compressor for gas-fired turbines has a considerable load, lowering the plant's net output.

Fuels.

In the long run, the fuel utilised in a gas turbine might indirectly impact fuel efficiency (Curran et al., 2014). Many liquid fuels, for example, require pre-treatment to eliminate hazardous vanadium and sodium constituents (Navarro et al., 2007). The principal ash components generated at higher temperatures are vanadium pentoxide and sodium sulphate, which stick to blades and corrode them (Rocca, Steinmetz and Moliere, 2003). Natural gas, properly prepared syngas and distillate oils are the preferable fuels in this case.

Cogeneration plant

Some of the gas turbine exhaust energy is recovered in cogeneration facilities for drying, process heat, or process steam production, among other things. The gas turbine can be optimised for maximum thermal efficiency or reduced thermal efficiency with more useable heat in the exhaust, depending on the specific nature of the CHP plant.

Higher air intake or combustion temperatures boost exhaust temperature, but a higher pressure ratio is likely to lower exhaust temperature. High energy in the exhaust, rather than final gas turbine fuel efficiency, maybe more essential because a high-pressure ratio and exhaust temperature are required for a gas turbine to deliver good simple cycle and cogeneration efficiency. Even low-efficiency simple-cycle gas turbines can be very efficient when employed in a cogeneration facility.

Additional firing may be the solution if even more exhaust heat is necessary. Gas turbine exhaust includes up to 15% oxygen (Pavri, R. and Moore, G.D., 2001), which may burn more fuel and raise the gas temperature further before flowing to boiler flues in a combined cycle plant, making the process feasible.

2.2 Steam turbine



Image 4. Steam turbine (Turbomachinery international, 2020)

Increasing the average temperature of the steam turbine power plant's heat source would improve efficiency (Arsalis, 2008). On the high-temperature parts of the isobaric processes of initial and intermediate steam, superheating is done by mixing superheated steam with natural gas combustion in oxygen (Pribaturin et al., 2016). In the second scenario, the average temperature of the heat supply is raised even further due to retaining the temperature of steam around the initial temperature at the turbine's inlet during the bulk of its expansion in the turbine. This is achieved by providing additional heat to the steam as it expands in the turbine.

The steam temperature is highest at the turbine's input and drops during expansion during the steam turbine cycle. The steam temperature for most of the expansion process might get kept close to the starting temperature to maximise thermodynamic efficiency. Extra heat will be necessary during expansion. Multi-stage fuel burning in the working material stream as it expands in the turbine can accomplish this.

The early portion of the expansion process in the turbine will be generally isothermal rather than adiabatic due to the heat source. However, the turbine's second phase of expansion should be adiabatic to maintain the temperature of heat evacuation as low as feasible. A steam turbine power plant can get modified by mixing steam with products of natural gas combustion (methane) in oxygen after the boiler's overheater, which includes an excess of methane. The surplus methane gets burned by supplying oxygen to the working material during turbine expansion.

3. Part C

Vapour Compression Refrigeration Cycle is a compression method for increasing refrigerant pressure as it passes through an evaporator (Process Solutions, Inc., 2020). It includes a compressor, condenser, expansion valve/throttle valve and evaporator. The high-pressure refrigerant passes via a condenser/heat exchanger before returning to the evaporator at a lower pressure. The stages are outlined in greater detail further down.

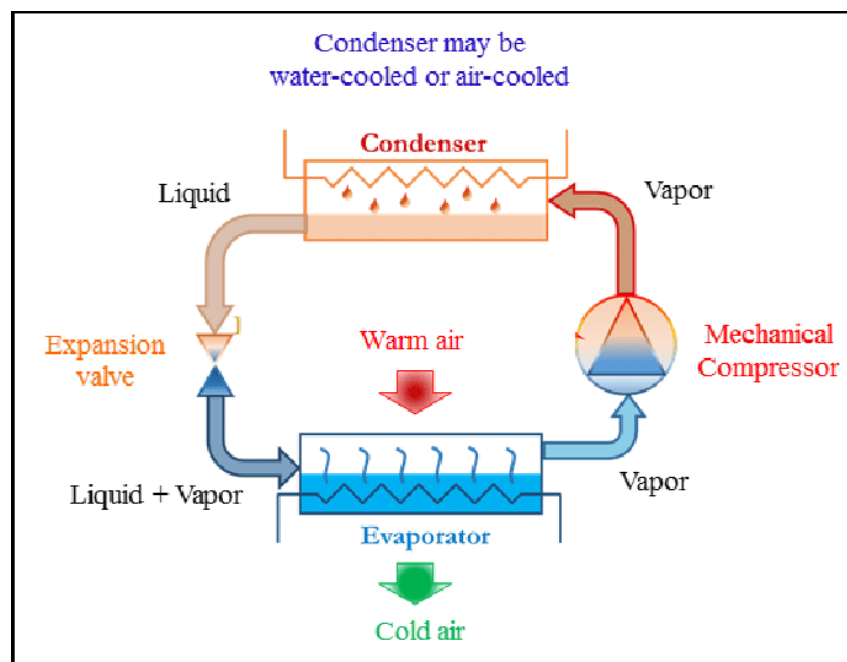


Image 5. Vapour Compression Refrigeration Cycle (Uddin, 2018).

Step 1: Compression

The refrigerant enters the compressor at low temperature and pressure. It's gaseous right now. Compression gets used to boost the temperature and refrigerant pressure in this area. The refrigerant enters the condenser after leaving the compressor. An electric motor might be employed because this procedure demands labour. Scroll, screw, centrifugal, and reciprocating compressors are all available.

Step 2: Condensation,

In essence, the condenser is a heat exchanger. The heat from the refrigerant gets transferred to a flow of water. In the case of water-cooled condensation, this water gets sent to a cooling tower for cooling. The refrigerant is under constant pressure as it passes through the condenser. Condenser safety and performance must get overlooked. Pressure control, in particular, is critical for both safety and efficiency. There are a variety of pressure-controlling devices available to meet this need.

Step 3: Throttling and Expansion

The refrigerant expands and releases pressure as it enters the throttling valve. As a result, the temperature is dropping at this point. The refrigerant exits the throttle valve as a liquid vapour combination. Throttling valves are essential in the vapour compression cycle for two reasons. They keep a pressure difference between the low and high-pressure sides for starters. They also regulate the amount of liquid refrigerant that enters the evaporator.

Step 4: Evaporation

The refrigerant is at a low temperature, and as a result, it evaporates and absorbs vaporisation heat. Low pressure and temperature get used to remove heat from the refrigerant, and the suction action of the compressor aids in maintaining low pressure. There are several types of evaporators on the market, but the two most common are liquid cooling and air cooling, which refer to whether they cool liquid or air.

4. References

- Amer, O., Boukhanouf, R. and Ibrahim, H.G. (2015). *Download Limit Exceeded*. [online] citeseerx.ist.psu.edu. Available at: <https://citeseerx.ist.psu.edu/viewdoc/download?doi=10.1.1.686.7367&rep=rep1&type=pdf> [Accessed 20 May 2022].
- Arsalis, A. (2008). Thermoeconomic modeling and parametric study of hybrid SOFC–gas turbine–steam turbine power plants ranging from 1.5 to 10MWe. *Journal of Power Sources*, 181(2), pp.313–326. doi:10.1016/j.jpowsour.2007.11.104.
- Brumbaugh, D. (2002). *Document - Gale Academic OneFile*. [online] go.gale.com. Available at: <https://go.gale.com/ps/i.do?id=GALE%7CA93793415&sid=googleScholar&v=2.1&it=r&linkaccess=abs&issn=00325961&p=AONE&sw=w&userGroupName=anon%7Eca7b37a2> [Accessed 20 May 2022].
- Brun, K., Kurz, R., Nored, M. and Thorp, J. (2013). INLET FOGGING AND OVERSPRAY IMPACT ON INDUSTRIAL GAS TURBINE LIFE AND PERFORMANCE. *oaktrust.library.tamu.edu*. [online] Available at: <https://oaktrust.library.tamu.edu/handle/1969.1/172614> [Accessed 20 May 2022].
- Curran, S.J., Wagner, R.M., Graves, R.L., Keller, M. and Green, J.B. (2014). Well-to-wheel analysis of direct and indirect use of natural gas in passenger vehicles. *Energy*, [online] 75(0360-5442), pp.194–203. doi:10.1016/j.energy.2014.07.035.
- e-dnfilters (2022). *DNA Round Clamp 62mm Inlet Air Filter for BMW R9T (14-19) ONLY OV-6200-125-R9T*. <https://www.e-dnfilters.co.uk/>. Available at: [https://www.e-dnfilters.co.uk/product/Motorcycle-Filters/Bmw/DNA-1332/OV-6200-125-R9T/DNA-Round-Clamp-62mm-Inlet-Air-Filter-for-BMW-R9T-\(14-19\)-ONLY-OV-6200-125-R9T](https://www.e-dnfilters.co.uk/product/Motorcycle-Filters/Bmw/DNA-1332/OV-6200-125-R9T/DNA-Round-Clamp-62mm-Inlet-Air-Filter-for-BMW-R9T-(14-19)-ONLY-OV-6200-125-R9T) [Accessed 20 May 2022].
- El-Shazly, A.A., Elhelw, M., Sorour, M.M. and El-Maghlany, W.M. (2016). Gas turbine performance enhancement via utilizing different integrated turbine inlet cooling techniques. *Alexandria Engineering Journal*, 55(3), pp.1903–1914. doi:10.1016/j.aej.2016.07.036.
- Ibrahim, T., Rahman, M. and Abdalla, A. (2011). Improvement of gas turbine performance based on inlet air cooling systems: A technical review. *International Journal of Physical Sciences*, [online] 6(4),

pp.620–627. doi:10.5897/IJPS10.563.

Kwon, H.M., Kim, T.S., Sohn, J.L. and Kang, D.W. (2018). Performance improvement of gas turbine combined cycle power plant by dual cooling of the inlet air and turbine coolant using an absorption chiller. *Energy*, 163(0360-5442), pp.1050–1061. doi:10.1016/j.energy.2018.08.191.

Muhammad Izzat Nor, M., Sathurshan, S., Shaheerthana, S., Yuli Panca, A. and Girma, T.C. (2019). The Development of an Aftermarket Intercooler Spray for Turbocharged Vehicles using Water. *INTI JOURNAL*, [online] 2019(17). Available at: <http://eprints.intimal.edu.my/1289/> [Accessed 20 May 2022].

Navarro, R., Guzman, J., Saucedo, I., Revilla, J. and Guibal, E. (2007). Vanadium recovery from oil fly ash by leaching, precipitation and solvent extraction processes. *Waste Management*, [online] 27(3), pp.425–438. doi:10.1016/j.wasman.2006.02.002.

Nikbakht Naserabad, S., Mehrpanahi, A. and Ahmadi, G. (2019). Multi-objective optimization of feed-water heater arrangement options in a steam power plant repowering. *Journal of Cleaner Production*, 220(0959-6526), pp.253–270. doi:10.1016/j.jclepro.2019.02.125.

Ojo, C.O., Schwillie, C., Nemet, A., Zierer, T. and Nicklas, M. (2010). *Optimization of Anti-Icing Limits for Alstom Gas Turbines Based on Theory of Ice Formation*. [online] asmedigitalcollection.asme.org. doi:10.1115/GT2009-59232.

Omosanya, Abolaji & Akinlabi, Esther & Okeniyi, Joshua. (2019). Overview for Improving Steam Turbine Power Generation Efficiency. *Journal of Physics: Conference Series*. 1378. 032040. 10.1088/1742-6596/1378/3/032040.

Pavri, R. and Moore, G.D., 2001. Gas turbine emissions and control. *Atlanta: GE Energy Services*.

Pribaturin, N.A., Fedorov, V.A., Alekseev, M.V., Bogomolov, A.R., Sorokin, A.L., Azikhanov, S.S. and Shevyrev, S.A. (2016). Experimental investigation on combustion of hydrogen–oxygen and methane–oxygen mixtures in the medium of low-superheated steam. *Thermal Engineering*, 63(5), pp.336–341. doi:10.1134/s0040601516050086.

Process Solutions, Inc. (2020). *How Does a Compression Refrigeration System Work?* [online] Process Solutions, Inc. Available at: <https://processsolutions.com/how-does-a-compression-refrigeration-system-work/#:~:text=The%20Vapor%20Compression%20Refrigeration%20Cycle&text=As%20the%20refrigerant%20circulates%20th>

rough [Accessed 20 May 2022].

Rocca, E., Steinmetz, P. and Moliere, M. (2003). Revisiting the Inhibition of Vanadium-Induced Hot Corrosion in Gas Turbines. *Journal of Engineering for Gas Turbines and Power*, [online] 125(3), pp.664–669. doi:10.1115/1.1456095.

Turbomachinery international (2020). *Doosan Škoda Power to supply steam turbine to Japan*. [online] Turbomachinery Magazine. Available at:
<https://www.turbomachinerymag.com/view/doosan-skoda-power-to-supply-steam-turbine-to-japan>
[Accessed 20 May 2022].

Uddin (2018). *Fig. 1. Schematic diagram of a typical vapor compression refrigeration...* [online] ResearchGate. Available at:
https://www.researchgate.net/figure/Schematic-diagram-of-a-typical-vapor-compression-refrigeration-cycle-17_fig1_326272160 [Accessed 20 May 2022].

Zhang, N., Lior, N., Liu, M. and Han, W. (2010). COOLCEP (cool clean efficient power): A novel CO₂-capturing oxy-fuel power system with LNG (liquefied natural gas) coldness energy utilization☆. *Energy*, 35(2), pp.1200–1210. doi:10.1016/j.energy.2009.04.002.