AIRO JOURNALS
Free / Unpaid Peer Reviewed Multidisciplinary International

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

THEORETICAL EVALUATION OF HEAT RECOVERY SYSTEMS IN GASTURBINE POWER CYCLES: THERMODYNAMICS ANALYSIS FOR ENHANCED ENERGY RECOVERY AND PERFORMANCE OPTIMIZATION

Dr. Arvind Kumar Tiwari 1
aktiwarijmi@gmail.com
Dr. Arvind Tiwari:
Assistant professor,
Applied Science Department
(Mechanical Engineering)
Rajkiya Engineering College Sonbhadra
Anupama 2,
Associate Professor,
Mathematics,
Darbhanga College of Engineering, Darbhanga, Bihar, anupamadce@gmail.com corresponding author aktiwarijmi@gmail.com

Abstract

Modern applications like processing and petrochemical plants rely on gas turbine power cycles for a number of reasons. This article presents the results of a thermodynamic analysis of a gas turbine power plant's open cycle. We calculated and double-checked the outcomes and strategies for each endeavour. Gas turbine plant high temperature productivity, power output, and heat rate are analyzed to determine the impact of operating restrictions including ambient temperature, relative humidity, fan pressure fraction, turbine bay temperature (TIT), and isentropic fan and turbine efficiency. The in-house code was written in MATLAB. The information generated by the code was used to draw various important figures. The findings demonstrate that thermal efficiency can be considerably impacted by pressure ratio, ambient temperature, air-fuel ratio, and isentropic efficiency.

Keywords: *Gas Turbine Power Cycles, Thermodynamics Analysis, Heat Recovery, Optimization.*

AIRO JOURNALS
Free / Unpaid
Peer Reviewed
Multidisciplinary
International

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

1. Introduction

The major component of the present power plant is the gas turbine, which, despite its size and weight, produces a lot of energy. In the last 40 years, the gas turbine has been increasingly popular in the power sector, used by utilities, commercial facilities, the petrochemical industry, and utilities all over the world. Gas Turbine Technology has developed noticeably over the past 20 years. New coatings, cooling techniques, and improved material technologies all contribute to the rise. This has prompted an ascent in gas turbine warm effectiveness from around 15 to more than 45%, which is satisfactory for power plants, when joined with an expansion in blower pressure proportion. Enormous coal and thermal energy stations prevailed the creation of power previously. In any case, because of their dark beginning abilities, higher efficiencies, lower capital expenses, speedier establishment times, prevalent emanation qualities, and ample petroleum gas supplies, flammable gas terminated turbines as of now prevail the area of power age.

Gas turbine power plants might be worked for around half however much equivalent ordinary petroleum product steam power plants, which until the mid 1980s were the primary power sources. Soon, gas turbine or blended gas-steam turbine types are expected to make up the greater part of all power establishes that will be fabricated. The most preferred fuels for gas turbines at the moment are clean gaseous fuels like natural gas, kerosene, and diesel due to the low price of crude oil. These fuels will soon run out and increase in price significantly. Therefore, preparations for burning alternate fuels must be created. Today, a variety of applications use gas turbines. The two main fields in which gas turbine engines are used are:

- Airplane drive.
- Electric power age

Future power age might depend intensely on gas turbines to take care of the issues of making perfect, proficient, and fuel-adaptable power. A gas turbine is the component in a gas-powered engine that burns the air-fuel mixture to create hot gases, which in turn rotate to power the turbine. Gas turbines are widely used in a variety of modern environments, such as petrochemical and processing plants, as well as aircraft propulsion and power generation. They are excellent for airplane propulsion due to their high power-to-weight ratio, compact design, and simplicity of installation**.**

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject : Mechnical & Engineering

1.1.Gas Turbine Cycles

John Hare, an English barber, invented and accepted the Brayton cycle, a thermodynamic cycle that describes the operation of a constant-voltage heat engine, in 1791. This cycle is named after George Bailey Brayton. The Joule cycle is one more name for it. There are two unique kinds of Brayton cycles, one of which utilizes an inward burning chamber and is presented to the climate. Its schematic outline and T-S portrayal are represented in Fig. 1. The subsequent kind purposes a heat exchanger in a shut cycle**.**

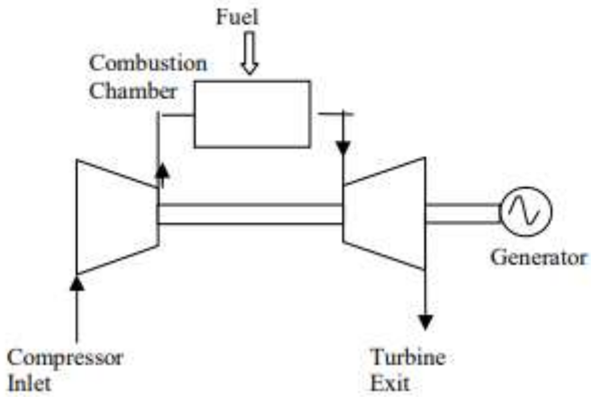

Figure 1: Open gas turbine cycle diagram

2. Literature Review

Use of heat that is economical and efficient is becoming more crucial as the world's energy demand rises. Particularly considering that significant volumes of waste heat are generated when primary energy is transformed into secondary energy, like electricity (Forman et al., 2016). High (>480 °C or 753.15 K), medium (240-480°C or 513.15-753.15 K), and low (<240 °C or 513.15 K) grade waste heat are the three categories into which wasted energy can be divided. According to Forman et al. (2016), the latter makes up 63% of the total amount of available waste heat. The narrow temperature differential between the heat source and the heat sink, which poses a significant challenge to the cycle efficiency, is a difficult for thermodynamic cycles involving low grade waste heat.

The Organic Rankine Cycle (ORC) is key to the reuse of low- and medium-grade waste heat. The technology uses working fluids with an organic C-H basis, such as hydrocarbons, halogens, or siloxanes. The reduced evaporation pressure that these synthetically created working fluids provide is advantageous for the operation of the power cycle (Alfani et al., 2021). In compared

AIRO JOURNALS
Free / Unpaid Peer Reviewed Multidisciplinary International
ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject : Mechnical & Engineering

to traditional ORC refrigerants, carbon dioxide (CO2) as a working fluid is chemically more stable (Calm and Hourahan, 2001). With a global warming potential (GDP) of 1 over 100 years and an ozone depletion potential (ODP) of zero, it is also more environmentally benign (Sarkar, 2015).

The fact that (CO_2) can operate at low critical temperatures (30.98 °C or 304.04 K) and moderate critical pressures (73.77 bar) during a power cycle is another attribute of the gas (Li et al., 2014). Thermodynamic properties change dramatically with temperature close to the supercritical point. These advantageous characteristics result in a high fluid density and, as a result, a significant reduction in system size. When operated with CO2 instead of steam, Sandia Research Institute (Wright, 2012) claims a 30 times smaller turbine. This is especially true for turbomachinery. However, all kinds of system components struggle under high operation pressures. Transcritical operation mode is used when CO2 cycles between supercritical and subcritical pressures during a power cycle. While the Brayton cycle only operates in the gaseous state, the Rankine cycle's working fluid undergoes a phase transition and is therefore transcritical (Li et al., 2014).

Fluid characteristics are crucial, particularly for the recovery of low grade waste heat. Each working fluid provides a unique set of challenges. On the other hand, the advantages of the low critical temperature of (CO_2) might be constrictive: A cooling down below (31 °C) must be ensured when employing ambient air as a heat sink in a Rankine cycle in order to assure a reliable operation.

3. Research Methodology

3.1. Thermodynamic Modelling of Gas Turbine

A gas turbine power plant has four main components: the blower, ignition chamber, turbine, and generator.

The definition of the compressor pressure ratio is:

$$r_p = \frac{P_2}{P_1} \qquad (1)$$

where the compressor's intake and exit air pressures are p1 and p2, respectively.

AIRO JOURNALS

Free / Unpaid
Peer Reviewed
Multidisciplinary
International

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

The validity of isentropy in design analysis validates the actual presentation of the device at the trade show achieved under ideal conditions of similar entry and exit conditions.

The isentropic effectiveness for blower is communicated as:

$$n_c = \frac{T_{2s} - T_1}{T_2 - T_1} \qquad (2)$$

The accompanying recipe is utilized to decide the temperature of the air leaving the blower:

$$T_2 = T_1 \left\{ 1 + \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right\} \qquad (3)$$

For disentanglement, two new factors and have been characterized as:

$$R_{pa} = \left\{ \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right\} \qquad (4)$$

$$R_{pg} = \left\{ 1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right\} \qquad (5)$$

Equation (3) can now be expressed as:

$$T_2 = T_1 \left(1 + R_{pa} \right) \qquad (6)$$

You can calculate the compressor's work as**:**

$$W_c = \frac{C_{pa} \times T_1 \left[r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1 \right]}{\eta_c \times \eta_m} \qquad (7)$$

$$W_C = \frac{C_{pa} \times T_1 \times R_{pa}}{\eta_m} \qquad (8)$$

where m is the mechanical blower productivity and dad c is air's particular heat. Because of the ignition chamber's energy balance:

$$m_a C_{pa} T_2 + m_f \times LHV = (m_a + m_f) C_{pg} \times TIT \qquad (9)$$

the low heat output LHV, the fuel mass flow rate mf, the air mass flow rate mama, and the turbine delta temperature t.

By reworking Eq, the turbine delta temperature can be communicated as:

$$T_3 = \frac{n_{com} \times LHV \times f \times C_{pa} \times T_2}{C_{pg} \times (1 + f)} \qquad (10)$$

where the combustion efficiency is denoted by com.

The temperature of the gas turbine's fumes gases, T4, is given by:

$$T_4 = T_3 \times (1 - \eta_t \times R_{pg}) \qquad (11)$$

where t is the turbine's isentropic efficiency.

To calculate the turbine's shaft work (Wt), we use the formula:

$$W_t = \frac{(C_{pg} \times T_3 \times n_t \times R_{PG})}{n_m} \qquad (12)$$

Next, the gas turbine-produced network is

$$w_{net} = W_t - W_c \qquad (13)$$

The turbine's total output power (P) is calculated as follows:

$$p = m_a \times W_{net} \qquad (14)$$

The amount of fuel used is calculated as

$$SFC = \frac{3600 \times f}{W_{net}} \qquad (15)$$

You may compute the heat provided as

$$Q_{add} = C_{pg} \times \left(T_3 - T_1 (1 + R_{pa})\right) \qquad (16)$$

The effectiveness of the gas turbine is determined as:

$$n_{th} = \frac{W_{net}}{Q_{add}} \qquad (17)$$

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

The gas turbine cycle's heat rate is calculated as:

$$\boldsymbol{HR = \frac{3600}{n_{th}}} \tag{18}$$

The appearance of gas turbine systems is greatly affected by the humidity of the surrounding air. The ratio of saturated vapor density to actual vapor density is called relative humidity. Mathematically,

$$\boldsymbol{\varphi = \frac{actual\ vapour\ density}{saturated\ vapour\ density}}$$

The thermodynamic model and MATLAB code were created using the aforementioned equations. Data have been produced for various air fuel ratios, compressor intake temperatures, and turbine inlet temperatures.

During the recreation interaction, sure of the boundaries are held steady at a given worth. Table 1 shows the upsides of the multitude of boundaries utilized in this exploration.

Table 1: Operational Parameter Summary

S\N	**Operating parameters**	**Value**	**Unit**
1	Mass flow rate of air through compressor (m_a)	125	Kg/s
2	Temperature of inlet air to compressor (T_1)	268-328	K
3	Compression pressure ratio	4-32	-
4	Air -fuel ratio (on mass basis)	40-56	-
5	Inlet temperature to gas turbine (T_3)	1219	K

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

6	Lower heating value (LHV)	47622	kl\kg
7	Specific heat of air	1.005	kl\kg
8	Specific heat of gas	1.8083	kl\kg
9	Isentropic efficiency of compressor (n_c)	85	%
10	Isentropic efficiency turbine (n_t)	90	%
11	Efficiency of combustor (n_{com})	95	%
12	Mechanical efficiency (n_m)	90	%

4. Results And Discussion

The exhibition of a gas turbine is examined in this work corresponding to the surrounding temperature, relative stickiness, pressure proportion, turbine channel temperature, isentropic proficiency of parts, and air fuel proportion. Energy balance conditions using the MATLAB program are utilized to decide the effects of working conditions on the organization yield, explicit fuel utilization, heat rate, and warm proficiency. The code's output is numerical data that is plotted to show how different operational parameters affect gas turbine performance.

Table 2: Impact of Temperature and Compression Ratio on Compressor Operation

	5	**10**	**15**	**20**	**25**	**30**	**35**	**40**
CIT =268K	100	205	300	350	390	400	450	500
CIT=278K	110	215	310	365	400	430	485	545
CIT=288K	115	220	325	375	420	480	520	560
CIT=298K	125	235	335	390	455	515	565	600
CIT=308K	130	250	350	405	475	535	595	635
CIT=318K	140	275	385	425	495	565	625	665
CIT=328K	150	295	400	450	510	600	645	690

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject : Mechnical & Engineering

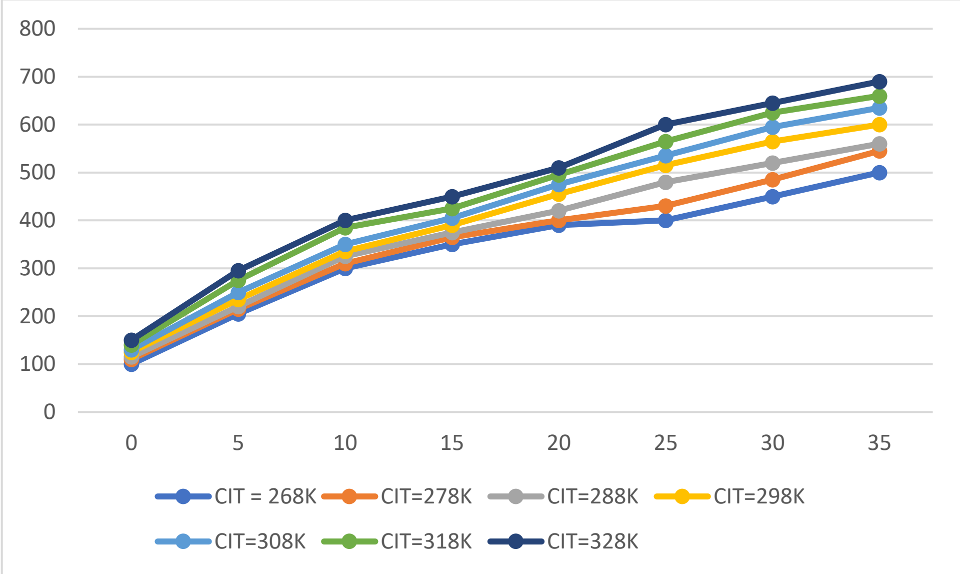

Figure 2: Impact of Temperature and Compression Ratio on Compressor Operation

Fig. 2 portrays the distinctions in blower activity with pressure proportion for different blower bay temperatures (CIT) going from 268K to 328K. From the diagram, it tends to be seen that blower power input ascends as blower channel temperature and pressure proportion rise. The way that blower exertion is straightforwardly connected with admission air temperature and pressure proportion gives a simple clarification to this. The diagram shows that for a blower pressure proportion of 15, the blower work input ascends by roughly 5.6% for each 12°C ascent in the encompassing temperature. Moreover, when the pressure proportion is expanded from 7 to 17 at 298 K surrounding temperature, the power contribution to the blower increments by around 25%**.**

Table 3: Impact of Compressor Isentropy and Air Fuel Ratio on Thermal Efficiency

	0.75	**0.80**	**0.85**	**0.90**	**0.95**	**1.00**
AFR=40kg	0.028	0.029	0.030	0.031	0.032	0.033
AFR=44kg	0.027	0.028	0.029	0.030	0.031	0.032
AFR=48kg	0.026	0.027	0.028	0.029	0.030	0.031
AFR=52kg	0.025	0.026	0.027	0.028	0.029	0.030
AFR=56kg	0.024	0.025	0.026	0.027	0.028	0.029

AIRO JOURNALS
Free / Unpaid
Peer Reviewed
Multidisciplinary
International

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

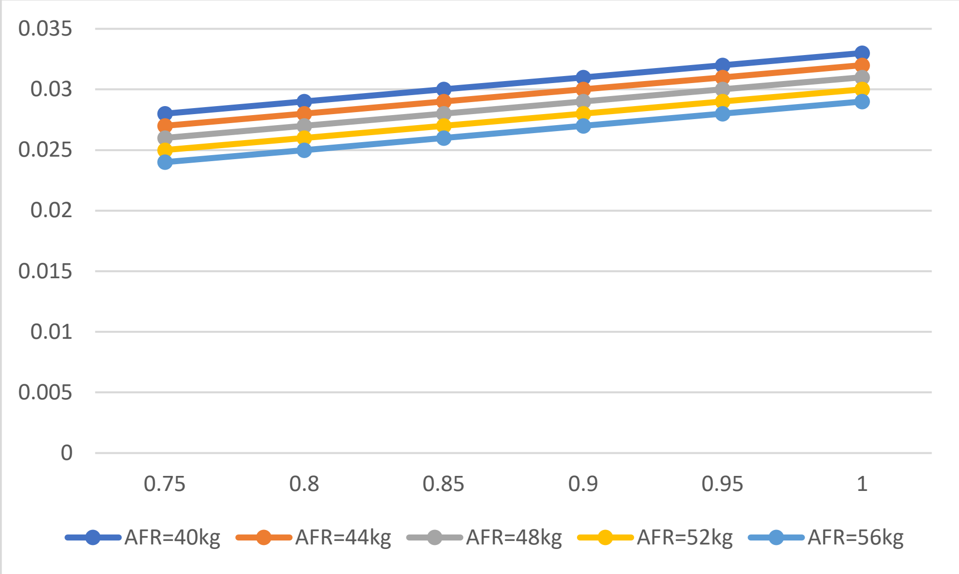

Figure 3: Impact of Compressor Isentropy and Air Fuel Ratio On Thermal Efficiency

In Figure 3 we have a scatterplot of thermal efficiency vs isentropic efficiency versus air-fuel ratio. It is evident that as the compressor's isentropic efficiency rises, thermal efficiency does as well. It is well known that the influence of friction or irreversibility in the system reduces as isentropic compression efficiency rises, resulting in less energy loss in the compressor. On the other hand, as the air to fuel ratio rises, thermal efficiency falls. It is seen that raising the isentropic blower effectiveness by 6% outcomes in a generally 5% improvement in warm productivity, while expanding the air fuel proportion from 40 to 44 outcomes in a 4.3% drop in warm proficiency.

Table 4: Thermal Efficiency Is Affected by Compression Pressure Ratio and Air Fuel Ratio**.**

	0	**5**	**10**	**15**	**20**	**25**
AFR=40kg	0.019	0.027	0.031	0.032	0.034	0.038
AFR=44kg	0.018	0.025	0.030	0.031	0.033	0.037
AFR=48kg	0.017	0.024	0.029	0.030	0.032	0.036
AFR=52kg	0.016	0.023	0.028	0.029	0.031	0.035

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

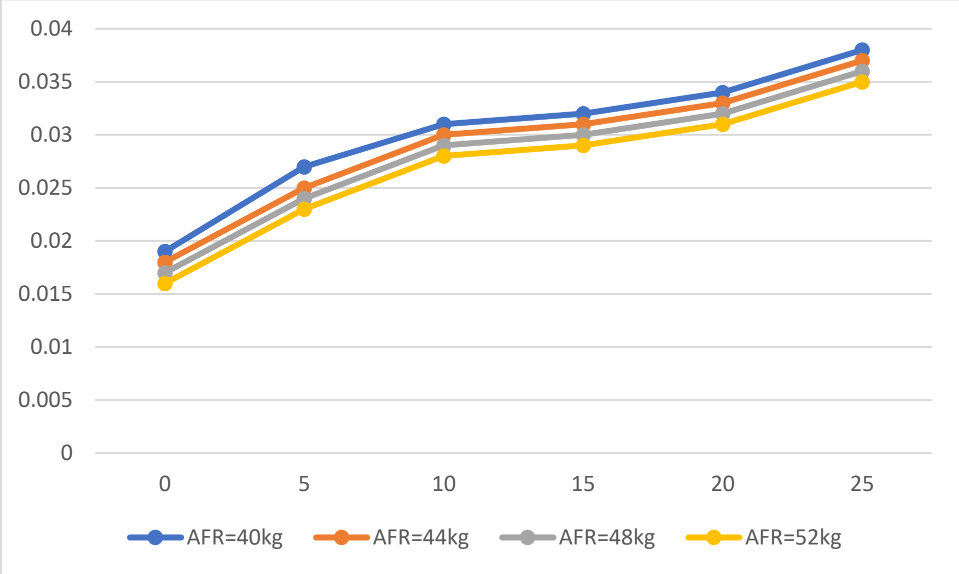

Figure 4: Thermal Efficiency Is Affected by Compression Pressure Ratio and Air Fuel Ratio.

Figure 4 shows how compression pressure ratio affects gas turbine thermal efficiency at various air-fuel ratios. As the compression ratio is raised, the pressure at the compressor's outlet rises. As a result, more heat may be transported because the matching temperature at the exit (TIT) is similarly high. The graphic shows that for an air fuel ratio of 44, thermal efficiency rises by 15% when the compression pressure ratio is increased from 7 to 17. Additionally, it should be remembered that the thermal efficiency declines as the air fuel ratio rises.

5. Conclusion

Ambient temperature and relative stickiness, fan pressure fraction, air-fuel ratio, turbine delta temperature, and isentropic efficiency of the part all had an impact on the high temperature output, power output, heat rate, and explicit fuel utilization of the gas turbine power plant, which was assessed through an analysis of a simplified gas turbine presentation.

Ambient conditions have a great influence on the performance of gas turbines. Evaporative coolers, mechanical coolers and assimilation coolers are therefore very useful in enhancing presentations by cooling information air.

References

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

1. A.K.Tiwari, M.M.Hasan and Mohd. Islam "Effect of Ambient Temperature on the Performance of a Combined Cycle Power Plant" Trans CSME Canada, issue 4, volume 37, 2013.
2. A.K.Tiwari, M.M.Hasan and Mohd. Islam "Exergy analysis of combined cycle power plant- NTPC Dadri, India" International Journal of Thermodynamics, Vol. 16, No. 1 pp 36-42, 2013.
3. Forman, C., Muritala, I. K., Pardemann, R., & Meyer, B. (2016). Estimating the global waste heat potential. *Renewable and Sustainable Energy Reviews*, *57*, 1568-1579.
4. Alfani, D., Binotti, M., Macchi, E., Silva, P., & Astolfi, M. (2021). sCO2 power plants for waste heat recovery: design optimization and part-load operation strategies. *Applied Thermal Engineering*, *195*, 117013.
5. Calm, J. M., & Hourahan, G. C. (2001). Refrigerant Data Summary. *Engineered systems*, *18*(11), 74-77.
6. Sarkar, J. (2015). Review and future trends of supercritical CO2 Rankine cycle for low-grade heat conversion. *Renewable and Sustainable Energy Reviews*, *48*, 434-451.
7. Wright, S. (2012). Mighty Mite. *Mechanical Engineering*, *134*(01), 40-43.
8. Li, X., Huang, H., & Zhao, W. (2014). A supercritical or transcritical Rankine cycle with ejector using low-grade heat. *Energy conversion and management*, *78*, 551-558.
9. Bade, M. H. and Bandyopadhyay, S. 2014. Analysis of gas turbine integrated cogeneration plant: Process integration approach. Applied Thermal Engineering. 78, 118-128.
10. Gell, M., et al., Bond strength, bond stress and spallation mechanisms of thermal Barrier coatings. Surface and Coatings Technology, 1999. 120-121: p. 53-60
11. Ghaderi, R. and Damircheli, M. 2014. Investigating factors affecting the efficiency of gas turbine power cycle. Scientific Journal of Pure and Applied Sciences. 3, 5, 271–279.
12. Kanoglu M, Dincer I, Rosen MA. Understanding energy and exergy efficiencies for improved energy management in power plants. *Energy Pol*. 2007; 35(7): 3967- 3978.
13. Kaviri AG, Jaafar MNM, Lazim TM, Barzegaravval H. Exergoenvironmental optimization of Heat Recovery Steam Generators in combined cycle power plant through energy and exergy analysis. *Energy Convers Manag*. 2013; 67: 27- 33.

ISSN: 2320-3714
Volume 3 Issue 2
August 2023
Impact Factor: 10.2
Subject: Mechnical & Engineering

14. Kaviri AG, Jaafar MNM, Lazim TM. Modeling and multi-objective exergy based optimization of a combined cycle power plant using a genetic algorithm. *Energy Convers Manag*. 2012; 58: 94- 103.
15. Sengupta S, Datta A, Duttagupta S. Exergy analysis of a coal-based 210 MW thermal power plant. *Int J Energy Res*. 2007; 31(1): 14- 28.

Author's Declaration

Dr. Arvind Kumar Tiwari
Dr. Arvind Tiwari
Anupam