Development of New Concepts of an Advanced Small-Gas Turbine in Series Hybrid Electric Vehicle Systems Based on T700-GE-401C Turboshaft Engine and Optimization

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Abstract

In this study, a micro gas turbine (MGT) was used for a series of Hybrid electric vehicles (SHEV). at first, the effect of turbine inlet temperature (TIT) and compressor pressure ratio in the T700-GE-401C engine cycle were investigated on performance parameters such as thermal efficiency ($\boldsymbol{\eta}$) and output power) and specific fuel consumption (PSFC). to optimize $\boldsymbol{\eta}$ and the supplementary of 400W as P_{net}, the design variables parameters consist of TIT, compression pressure ratio, and inlet air mass flow rate were calculated. Accordingly, Concept-A was used to use in SHEV with the requirement output converted with an electrical generator based on turboshaft engine T700-GE-401C with additive Compression refrigeration cycle to optimize cycle for development at Concept-B rather than Concept-A a heat recovery was added. effect of $\boldsymbol{\eta}$, $P_{net,}$ and PSFC for both Concept-A and Concept-B for $\boldsymbol{\eta}$ was orderly 53.84,55,63 and 54.90 303K according to COP 2,4,6 respectively. also, the effects of diesel fuel and liquid natural gas on the cost of fuel Concept-B fuel consumption cost were investigated. the results showed that the use of diesel fuel in diesel Concept- B rather than the other fuels used in this study has the lowest cost.

Keywords: series Hybrid electric vehicles, Micro gas turbine, Compression refrigeration cycle, Heat recovery, biofuel

NOMENCLATURE

$A(m^2)$	Surface area	**Subscripts**	
$P(kPa)$	pressure	amb	ambient
$Q(kW)$	Heat rate	av	average
$FHV(Mj/Kg)$	Fuel heat value	b	burner
$\Delta T(K)$	Difference temperature	C	Cold reservoir
$m(kg/s)$	Mass flow rate	C	Compressor
$TIT(K)$	Turbine inlet temperature	Comb	combustor
$Cp(Kj/kg/K)$	Specific heat rate	E	Exhaust surrounding
k	the friction of special heat capacity	F	fuel consumption
$PSFC(g/KWh)$	specific fuel consumption	Hr	Heat recovery
COP	Coefficient of performance	Is	Isentropic
Ma	Mach number	JP4	jet Propellant 4
$H(m)$	altitude	LMTD	The logarithmic mean temperature difference
$IAT(K)$	Inlet air temperature	Net	net
$\dot{C}(\$/h)$	performance Cost	T	turbine
$FC(\$/Kg)$	Fuel price	Th	thermal
Greek Symbols		0	ambient
Π	Compression ratio	1	Inlet air
ρ	Density (kg/m^3)		
η	Thermal efficiency (%)		
ω	Rotational speed(rpm)		

1. Introduction

In recent years, by global warming, ecologists are doing any action, idea, or policy to reduce air pollution, destructive environmental emissions, and greenhouse gases that have become a global concern. The response of the world car industry to these concerns led to car production with a plug-in hybrid electric vehicle (PHEVs) technology and an electric hybrid vehicle. The hybrid system of PHEVs and Hybrid electric vehicles (HEVs)is considered as a possible vehicle technology to increase demand for vehicles with more fuel and reasonable fuel, due to the environmental effects. In order to improve the performance of fuel cell vehicles, the producer considers parameters such as fuel economy, radiation, the comfort of passengers, and safety. The PHEVs electric vehicles contain all-electric vehicles that do not use internal combustion engines (ICE) and are supplied by a large battery pack. In electric vehicles, challenges are associated with high performance, device integration, and low cost in power converters and electrical devices.

In the HEVs and the PHEVs, the efficiency increase of the system at the operational point is detected using internal combustion engine control (ICE). In a variety of hybrid systems, it is produced by an internal combustion engine or by regenerative braking, while in other systems, one can use an electrical output to recharge the battery. The other machines also have electrical propulsion, which can operate at the same time as the internal combustion engine, or it can be had the adjustments that switch between the two engines.[1-4]

While the vehicle's performance with a combination of a second-series hybrid electric combination for vehicle parameters. They were optimized in view to maximize the fuels and the economy using software consultants for the driving cycle of the Indian Highway. Proposed a new hybrid system of melt carbonate fuel cell (MCFC) and the gas turbine (GT), which developed with conditions of Ansaldo fuel cell and ENEA company. The MCFC has been discovered gas turbines with auxiliary boilers had the best performance on energy efficiency were compared to ICE, GT, and MCFC[5, 6].

The analysis of the electric hybrid vehicle using the AVL software to save fuel consumption and vehicle performance for high productivity reduces the consumption of fuel in the car sector under certain operational conditions in the driving cycle. The reduction of greenhouse gases and fossil fuels caused by energy generation is formed by the ICE [7, 8]. Due to environmental impacts, a parallel hybrid type of HEV system is investigated to reduce fuel consumption. The explanation for the parallel hybrid electric hybrid vehicles with mass power demand was gradually exploited. This explanation was new for vehicles outside the road to guide the control strategy engine on the high performance of an AC hybrid motor, which depends on the power demand and battery-state charging.[9-11]

Kim et al. [12] thermodynamically performed the power turbine cycle with a Rankine cycle (RC) to restore the heat flow of the exhaust gas turbine cycle for the marine propulsion. Also, the effect of the use of R32, r134a, R152a, and R1234yf as the Rankine cycle factor analysis showed that in the best case, the energy efficiency and exergy efficiency of the cycle have been increased by heat recovery,

Also, Pape and Parente [13] attempted to optimize a heat recovery system for the micro-gas turbine (MGT) on the performance of a gas turbine was investigated to use to advanced concept gas turbine for the multiplicative water injection system.

In a gas turbine to arrive at zero-emission with four novel novels and approximately no three-stage emission of gas propulsion, hybrid wind turbines are explained using fuel combustion (oxy), which means methane in an environment near pure oxygen as a fuel. It is also replaced by compressor working fluid compressor or pump working fluid as a supercritical fluid.[14, 15]

in the parallel hybrid system, this type of hybrid vehicle is suitable for use in urban areas and is not recommended for driving on highways and at high speeds. The important thing is that the batteries are small from this hybrid, so it is possible to achieve maximum load capacity in a short time. The series hybrid machines can be used as clean and clean as a clean and clean vehicle since a gasoline-powered engine is the only battery - charging and all tasks are carried out by the electric motor. The operating method in these machines is that the gasoline engine will transfer the power generated by the generator to the power controller, and then the generated force is used to charge the battery as well as the power of the electric motor and eventually the engine. Electricity is injected into the transmission system, causing the vehicle to move completely without fuel consumption and the gasoline engine operating only as a power generator. The striking point in the hybrid series machines is that this type of vehicle uses a more expensive and larger battery than other types. Because of the complexity of internal combustion generators, these cars are usually more expensive than pure electric cars.[16, 17]

The gas turbine engine is accompanied by a sample vehicle with a generator to assist a developed hybrid vehicle with a wide range. The new cooling effects of the gas turbine fitted with the use of turbo compressors are carried out to increase the performance of the gas turbine in the Iranian refinery with two practical air-cooling methods to be used in moist environments or a mechanical chiller[18, 19].

It has been presented a new combined cooling system and compared to other configurations of the gas turbine input air conditioning system. Their proposed system consists of the earth-air heat exchanger and the smoky cooling system. Another investigation studied the energy induced by the natural cooling temperature for the cooling system for the new input air cooling system to cleanse the energy plant's performance. They used the development of design parameters to maximize the power plant performance. A comparison of the adsorption of cooling air at 15 ° C for cooling the air at the inlet of the gas turbine in different climatic conditions over a year. The additional study investigated the effects of cooling the input air of a gas turbine plant by adding an absorptive chiller, thermal pump, and entrance systems for cities of Iran with different climatic conditions. It has been conducted a numerical study on the production of clean energy as fuel for the generation of electric energy on an ICE or in a gas turbine. They found that the benefits of using the micro – Gas turbine in the operation of plants in the power output are increasing.[20-23]

Hashmi et al.[24] was performed the inlet air-cooling effects on the gas turbine cycles performance (GT) by the gas turb12 software. More while, the effect of the moisture and inlet temperature on the fuel consumption and heat efficiency and the heat efficiency of the turbine in this study show that by reducing the temperature of inlet air to the cycle, thermal efficiency cycle and the output power increased and the fuel consumption cycle decreased. also, Santos and Andrade have studied the effect of air temperature on the performance of a gas turbine cycle studied by increasing the temperature of the thermal efficiency and the output power of the gas turbine cycle.[25, 26]

In addition, performed a thermo - economic analysis of a gas turbine cycle with a cooling air cooling system. the results of this study show that by reducing the temperature of the input temperature to the gas turbine cycle, the thermal efficiency of the fuel is decreased.[27]

The results of the investigation showed that the addition of thermal recovery to a gas turbine cycle causes an increasing flow temperature of the combustor. the reduction of the fuel consumption is reduced to the inlet temperature of the turbine, so by adding the thermal recovery system of the consuming fuel mass flow rate and specific fuel consumption (PSFC) and the thermal efficiency of the cycle is increased [28].

According to previous studies, it has been proposed that instead of an internal combustion engine in a hybrid system, the gas turbine has not been investigated so far. in this study, a new hybrid vehicle has been proposed, the performance of the system and fuel consumption are investigated. by using a hybrid internal combustion engine (ICE). also, the effect of reducing the inlet temperature to the gas turbine cycle in the proposed system performance is explored. to cool the inlet air, the compression refrigeration cycle was used. also, to improve the performance of the thermal efficiency by adding a heat recovery converter to a gas turbine cycle causes an increase in the temperature of the combustor input current is considered. In addition, the fuel consumption is reduced to the inlet temperature of the turbine, so by adding the thermal recovery for fuel consumption is decreased and the thermal efficiency of the cycle is increased.

2. Problem description

The arrival air enters the compressor from ambient, the departure air enters the combustor for heating the air for the possibility to enter the turbine in the sample gas turbine cycle.

gas turbine modeling of a gas turbine in fig.1.

the gas turbine includes the combustion chamber, compressor, turbine, and

exhaust. modeling is done for steady-state.

schematic of the simple gas turbine cycle is also illustrated in Fig.1.

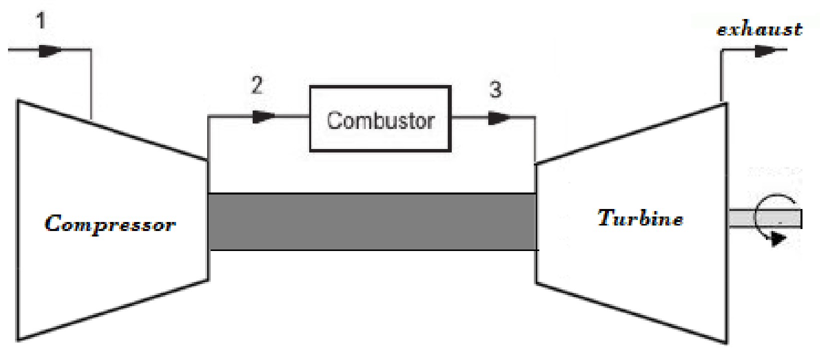

Fig.1. schematic of the simple gas turbine cycle.

Also, the schematic of the Concept- A cycle recommendation is shown in Fig.2.

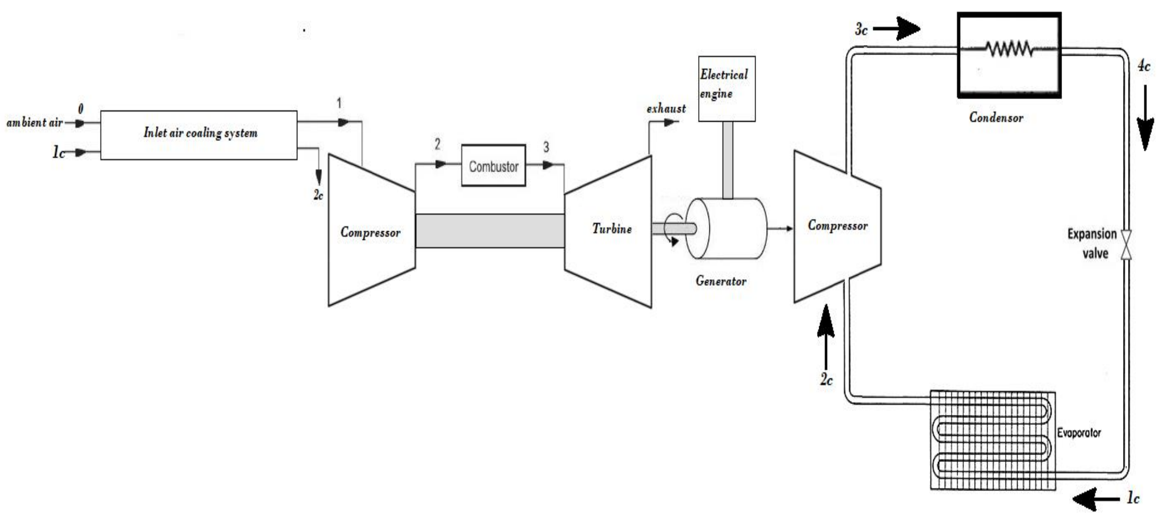

Fig.2. schematic of the concept - A cycle recommendation.

The schematic diagram of the concept - B cycle in Fig.3. is shown in Fig.

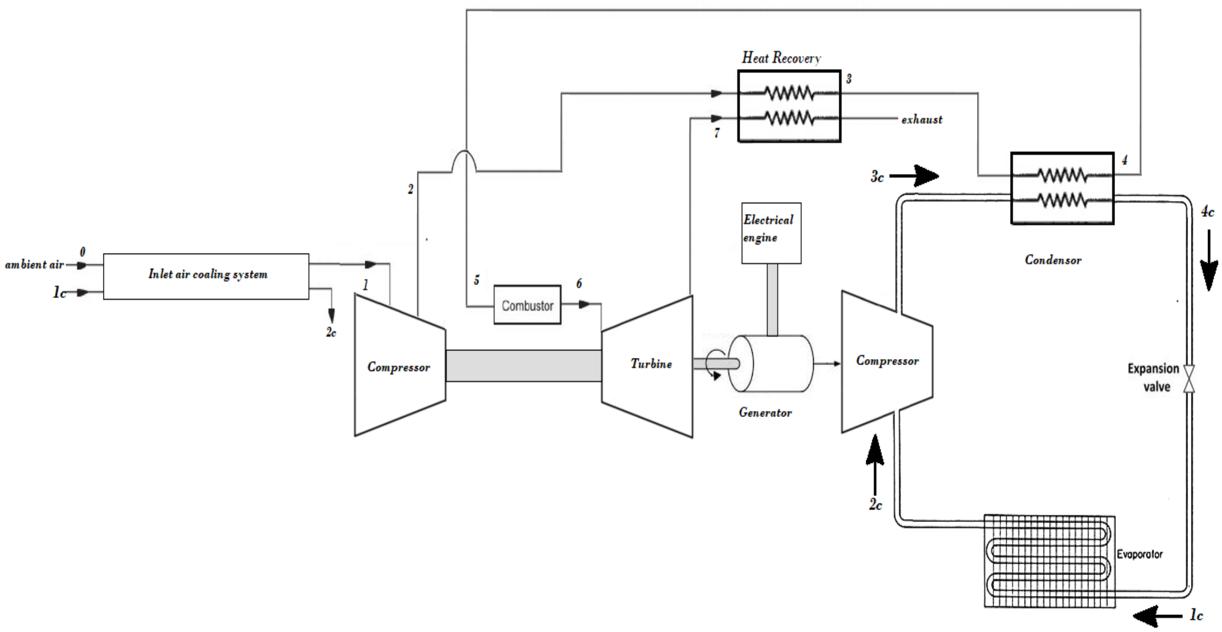

Fig.3. schematic of the concept - B cycle recommendation.

air cooling in the inlet and cooling system is negligible, it can be approved for pressure in eq (1);

$$P_0 = P_1 \tag{1}$$

$$T_0 = T_1 - \Delta T \tag{2}$$

Which P_0 and T_0 , orderly, are the pressure and temperature of air ambient, also P_1 and T_1 are the pressure and temperature of the inlet air compressor, P_2 and T_2 are the pressure and temperature of the inlet air to the burner. ΔT The temperature difference between the inlet air temperatures varies to the temperature of the environment. The compressor outlet pressure is calculated as:

$$P_2 = \pi_c P_1 \tag{3}$$

Which, π_C is pressure ratio of compressor and P_2 is outlet pressure.

The outlet temperature of the compressor is presented as;

$$T_2 = \frac{T_1}{\eta_c}\left[\left(\pi_c\right)^{\frac{k_c-1}{k_c}} - 1\right] + T_1 \qquad (4)$$

Which, η_C is isentropic compressor efficiency, T_2 is the outlet temperature of the compressor and k_c is the friction of special heat capacity in pressure constant process to the special heat capacity in volume constant process in the compressor, which is the function of the mean temperature of the passing air from in along of the compressor, also the consumed power of compressor can be calculated as;

$$W_c = \frac{m_a T_1}{\eta_c}\left[\left(\pi_c\right)^{\frac{k_c-1}{k_c}} - 1\right] \qquad (5)$$

The real mass flow rate of inlet air to the engine considered as:

$$m_a = \rho V_0 A \qquad (6)$$

Where A is the surface of the air inlet, ρ is air density and V_0 is inlet airflow velocity

The Fuel heat value Q_h is calculated.

$$Q_h = m_a C_{av}\left(T_3 - T_2\right) \qquad (7)$$

Where, C_{av} is the friction of specific heat capacity of air in pressure constant process which can be defined as a function of mean temperate along of combustor. Also, T_3 is the outlet flow temperature in turbine and T_2 is the inlet flow temperature in the turbine

In the following equation the mass flow rate of consumed fuel can be calculated as;

$$m_f = \frac{Q_h}{FHV\eta_{comb}} C_{av}\left(T_3 - T_2\right) \qquad (8)$$

Where FHV is the Fuel heat value per kilograms of fuel and $\eta_{combustor}$ is the thermal combustor efficiency and Q_h is the heat value which, has been defined respectively.

The passing mass flow rate of the turbine can be defined as the summation of the mass flow rate of air and the mass flow rate of fuel consumption

$$m_{tot} = m_a + m_f \tag{9}$$

Where, m_f is the mass flow rate of fuel consumption. The generated power can be calculated as the following equation.

$$W_T = m_T C_{av,T} \left(T_3 - T_4\right) \tag{10}$$

Where, m_T is the mass flow rate of passing from turbine and C_{pav} specific heat capacity means pressure constant which, can be calculated as a function of mean temperature along of turbine. It is assumed that the ignorance of loss of mechanical power in shafts according to the principle of conservation of energy, the turbine power is equal to the total power consumption of the compressor.

$$W_{net} = W_T - W_c \tag{11}$$

Where, $T_4$is turbine outlet temperature, T_3 is turbine inlet temperature, m_T is mass flow rate passing the turbine andC_{pavt} mean heat capacity of pressure constant which, is a function of mean temperature along of turbine. Also, η_c isentropic thermal efficiency of compressor and k_c is the friction of heat capacity of pressure constant to volume constant of air which is the mean temperature of air along of compressor.

Brake-specific fuel consumption (PSFC) is the specific consumed fuel of an engine, which defined as the friction of mass flow rate of fuel consumption ($\mathrm{m_f}$) to out power in the following equation.

$$PSFC = \frac{m_f}{W_{net}} \tag{12}$$

$$\eta_{th} = \frac{W_{net}}{Q_h} \tag{13}$$

Where, η_{th} is the thermal efficiency of the engine and m_a is the mass flow of air passing through the engine. In the refrigeration cycle, the heat transfer rate of cooling can be defined as;

2.1 thermodynamic modeling of the compressive refrigeration cycle

In this case, to improve the performance of the micro turboshaft – engine for the recommended concept, heat recovery has been added to the turbine exhaust flow. also, the cooling system is added to the system based on the compression refrigeration cycle.

$$Q_C = m_a Cp_a \Delta T_c \tag{14}$$

Where,$m_a$is the mass flow rate of inlet air in the refrigeration cycle, C_{p_a} is specific heat rate capacity in pressure constant process and ΔT_c is difference temperature of the inlet air to the refrigeration cycle in the engine to the ambient temperature.

The inlet power to the compressor in the refrigeration cycle can be calculated.

$$\dot{W}_C = \frac{Q_C}{COP} \tag{15}$$

Where, Q_c is heat rate of cooling source in refrigeration cycle of inlet airflow to the engine and COP is the coefficient of performance of refrigeration cycle.

The heat gained from heat recovery (hr) installed in the refrigeration(ref) cycle can be calculated base on LMTD (Logarithmic mean temperature difference). the deference temperature in heat recovery base on the LMTD method defined as,

$$\Delta T_{LMTD} = \frac{\Delta T_{hr1} - \Delta T_{hr2}}{\ln\left(\frac{\Delta T_{hr1}}{\Delta T_{hr2}}\right)} \tag{16}$$

Where, ΔT_{hr1} is the hot source temperature changes and ΔT_{hr2} is the cold source temperature changes. The heat rate of heat recovery is calculated in eq (17).

$$Q_{hr} = m_a Cp_a \Delta T_{LMTD} \tag{17}$$

2.2 Optimization Method

in this study, we used a genetic algorithm to optimize the performance of the T700 micro turboshaft engine. Genetic Algorithm A general purpose is the population-based search algorithm which is a purpose for the reorganization of the sample of all probabilities, whether they are solutions in the problem space, strategy for the game, rules in classifier systems, or arguments for problems of performance optimization problems. Individuals have evolved to form better people by sharing and combining their information about space.

the functions used to evaluate the performance of the distributed genetic algorithm and the traditional algorithm are called Walsh polynomials, which are based on Walsh functions. the Walsh polynomial can be classified as functions with each class having different degrees of difficulty. by generating large numbers of samples from different classes, the difference between distributed and traditional genetic algorithms can be statistically analyzed.

the first part of this thesis studies the segmented genetic algorithm, a version of distributed genetic algorithm. experiments on four different groups of Walsh polynomial show that the partitioned algorithm continuously operates from the traditional algorithm as a function. this is because the physical part of the population allows each other to examine space independently. also, good people are likely to be identified in smaller areas than the diverse and diverse population. the second part of this research studies the effects of migration on the performance of the distributed genetic algorithm. experiments show that with an average migration rate, the distributed genetic algorithm finds better people than the traditional algorithm while preserving the overall population proportion. Theoretical fundamentals of genetic algorithms are investigated, which include the schema theorem and the exact and accurate models of the standard genetic algorithm. [29, 30]

3. Validation

The inlet data from the T700 turboshaft engine cycle input and functional modules are shown in table 1.

validated

At first, the T700 turboshaft engine

Secondly,

The calculations are shown in table 2. with reference values by expressing the percentage difference [31].

It can be proved which, the results of modeling with reference results are adequate.

Table 1

The input data of the engine cycle engine.

Parameters	Description	Value	Unit
m_a	Air mass flow rate	4.573	Kg/s
π_c	Compression ratio	17.53	-
TIT	Turbine inlet temperature	1525	K
$\eta_{is,c}$	compressor isentropic efficiency	0.89	-
$\eta_{is,T}$	turbine isentropic efficiency	0.85	-
η_b	burner efficiency	0.99	-
P_1	Inlet air pressure	101.300	Kpa
T_1	Inlet air temperature (IAT)	288	K
FHV_{JP4}	JP4 Fuel heat value	43.323	MJ/Kg

Table 2

The performance parameters of T700 in return for JP4 fuel compared to reference values.

Parameters	Net power (KW)	PSFC(g/KWh)	Fuel mass flow rate(g/s)
Simulated value	1215.4	304.207	102.87
Reference values [31]	1318.32	274.207	100.42
Error percent (%)	7.80	10.94	2.44

3. Result and discussion

3.1 Power output and specific fuel consumption variation

The inlet temperature effects on the performance of the T700 gas turbine has been investigated. the output power of the T700 engine is represented by the inlet temperature of the turbine for JP4 as fuel combustion according to equation (8) in figure (4). by increasing the inlet temperature of the turbine, the supplied power of the turbine is increased and the power is constant. the increased turbine power consumption is constant. Accordingly, the output power of the turbine is increased. On the other side, the output power compressor. Consequently, the net output power is correspondingly increased.

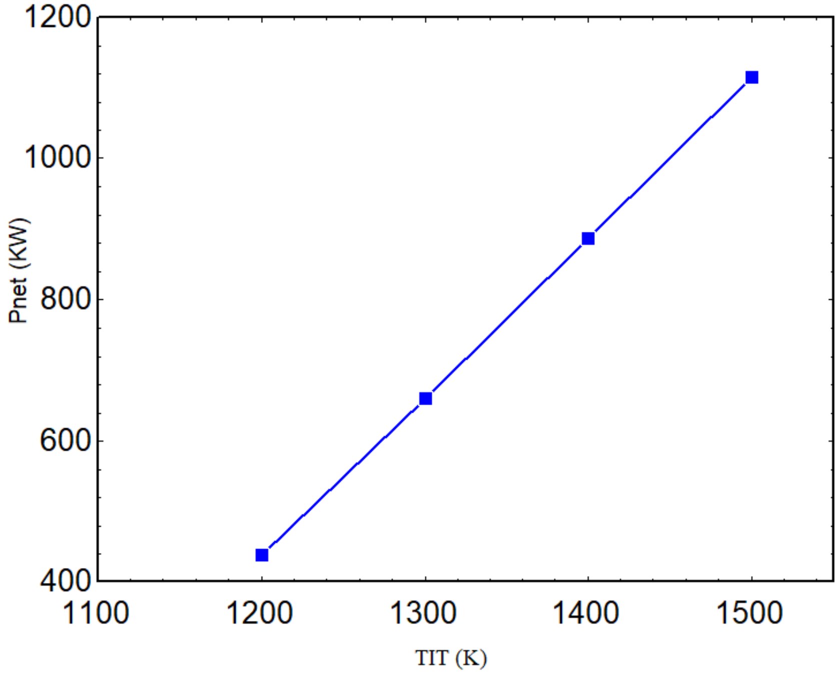

Fig.4. the output power of the T700 engine with the input temperature of the JP4 fuel.

The changes of PSFC in the T700 engine is represented by the temperature of the turbine in return for JP4 fuel shown in Figure 4. with increasing the temperature of the turbine due to increasing the heat transfer rate, the fuel consumption flow rate increased, the intensity of the increase in fuel mass flow rate is less than increasing the output temperature of the turbine. Therefore, the specific consumption fuel (PSFC) due to the increase in the turbine input power decreased.

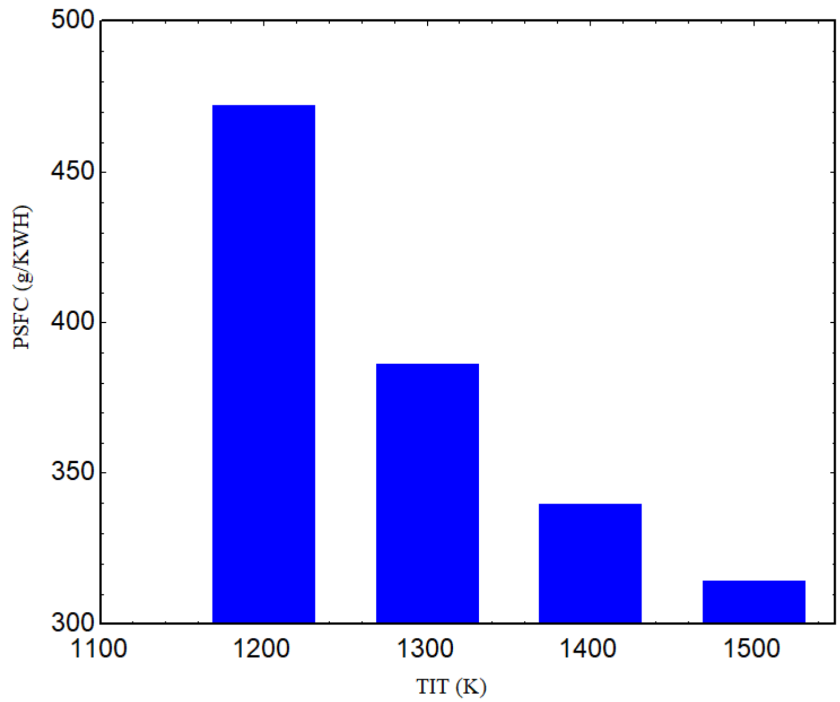

Fig.5. PSFC variation of the T700 engine with the input temperature for JP4 fuel.

the effect of the Compression ratio on the performance of the T700 gas turbine is investigated. The output power of the T700 engine is represented by the ratio of the compression ratio for JP4 fuel in Fig.6.

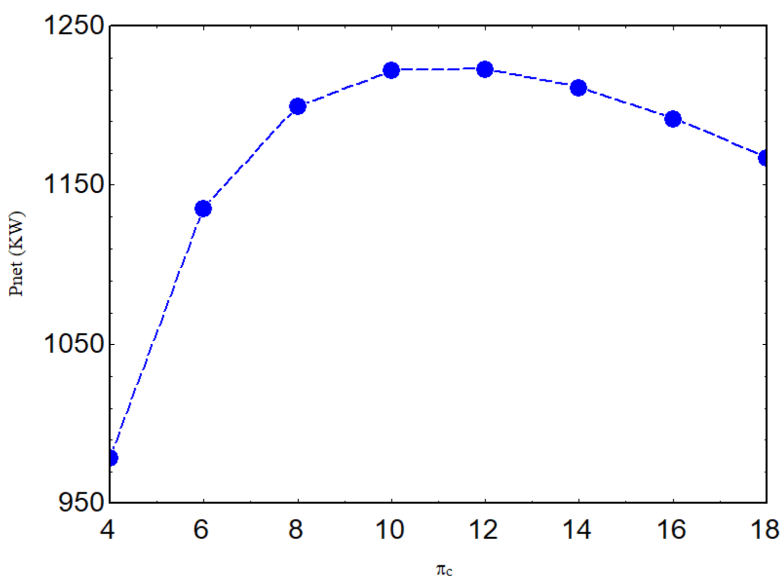

Fig.7. changes of the output power of the T700 engine with disparity of the compression ratio. in return for JP4 fuel.

The T700 engine is displayed with the ratio of the compressor in the form of JP4 fuel in Fig.7. It is observed that with the increase in the compression ratio as proportion of the density of the PSFC.

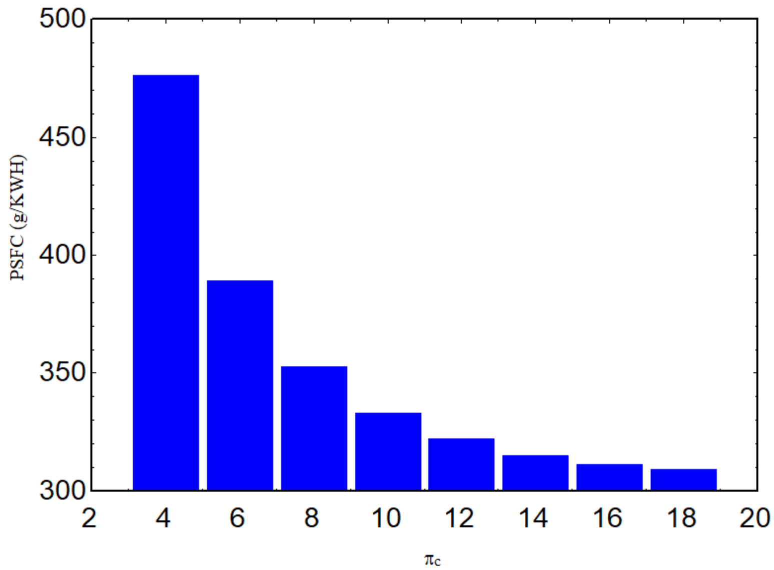

Fig.8. PSFC variation of the T700 engine for the compressor in return for JP4 fuel.

The output power variations of the T700 engine are represented by the temperature of the inlet air in Fig.11.Since the incoming air density decreases with decreasing air temperature, the real mass velocity of the input air is increased. so, according to eq (10), the supplied power of the turbine is increased. so, by the inlet air temperature, the output power of the cycle is increased.

The output power of the T700 motor is represented by the temperature of the inlet air in Fig.11.

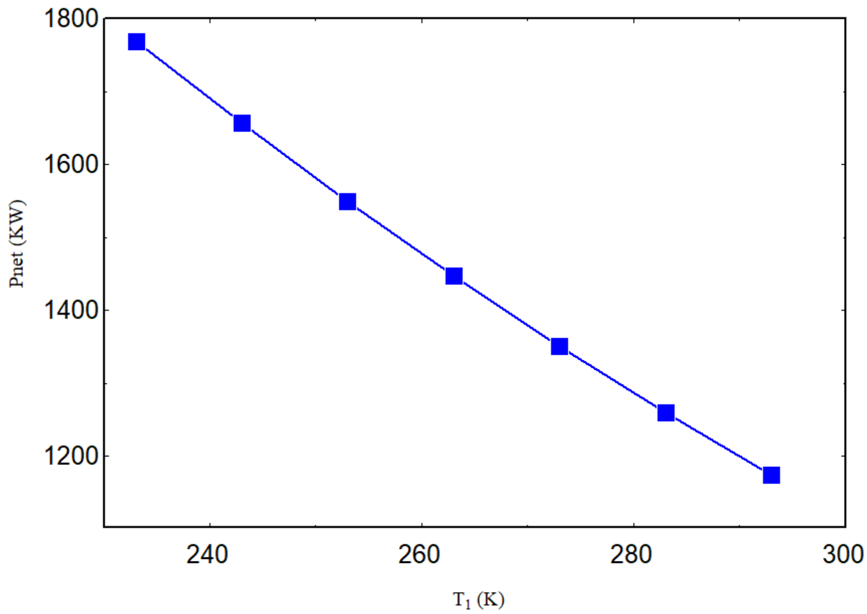

Fig.11. The variations output power of the T700 term engine in of inlet air temperature.

The PSFC variation of the T700 engine is represented by the temperature of the incoming air in Fig.12. Since the inlet temperature decrease of the input velocity of the input is increasing, the excess heat energy is required to give a flow to the temperature of the turbine due to the temperature of the turbine. so, the heating rate and the fuel flow rate increase by reducing the inlet air temperature. However, since the increase in the output power increase due to the decrease inlet air temperature is higher than the intensity of the increase in the temperature of the intake due to the reduction in the temperature of the input air temperature, PSFC decreased.

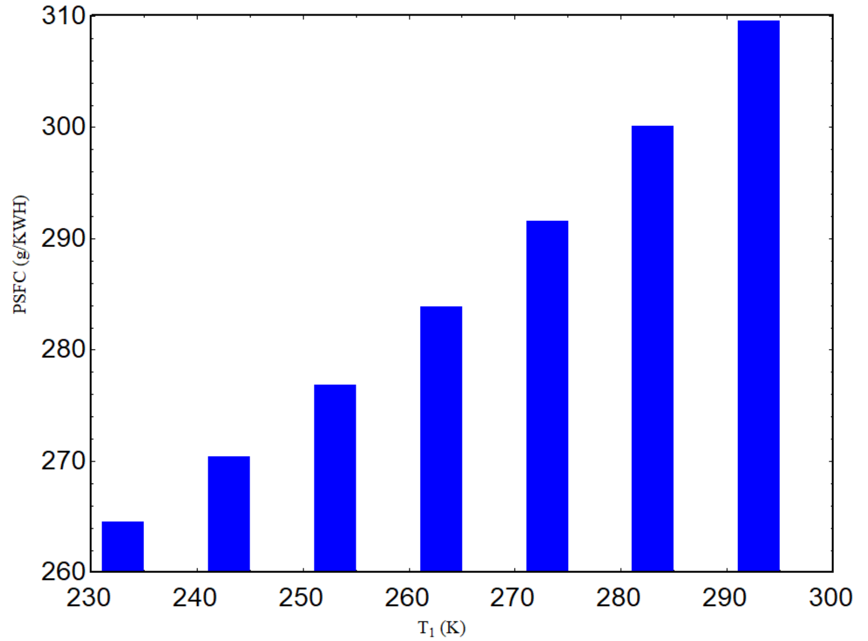

Fig.12. The variation of the T700 engine with the temperature of the inlet air temperature. in return for JP4 fuel

3.2 the effect of inlet temperature on the performance

The thermal efficiency changes of the T700 engine with the input temperature for JP4 fuel are shown 4.2 in Fig5. since the increase in heat rate due to the increase in the turbine input temperature is less than the intensity of increasing output cycle power, subsequently the heat efficiency increases with increasing the temperature of the turbine.

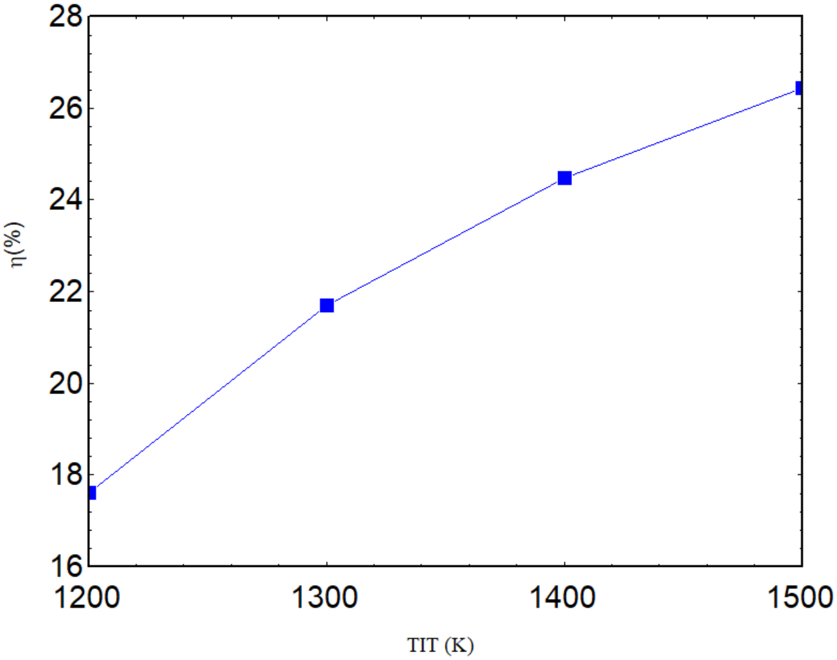

Fig.6. the thermal efficiency of the T700 engine with the input temperature of the JP4 fuel

3.3 the effect of the Compression ratio on the performance

The thermal efficiency of the T700 engine is represented by the ratio of the compression ratio for JP4 fuel in Fig.8. It is observed that increasing the compression ratio increases the thermal efficiency.

Fig.9. shows the thermal efficiency changes of the T700 engine to the compression ratio. in return for JP4 fuel.

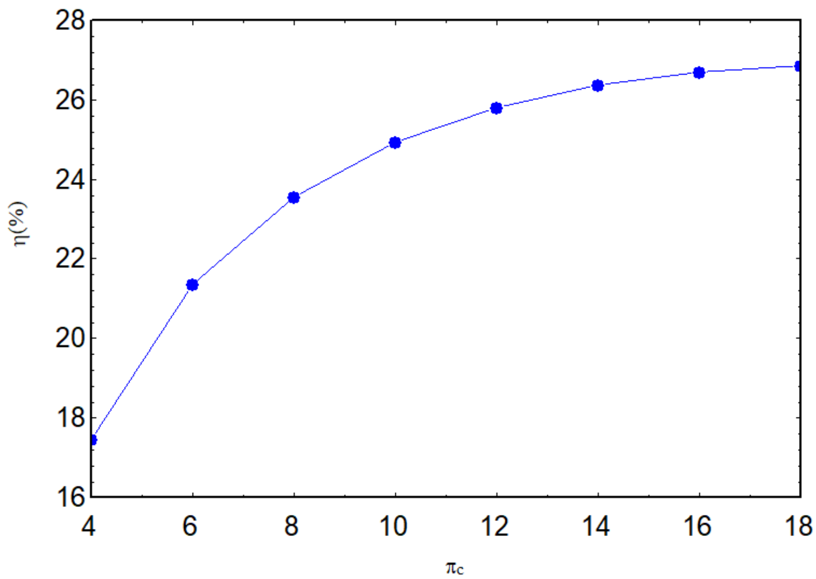

Fig. 9. the thermal efficiency changes of the T700 engine for the compression ratio. in return for JP4 fuel.

3.4 effect of inlet air temperature changes on T700 gas turbine performance

The real air mass flow rate changes in the input temperature of the T700 engine are shown in Fig.10. by reducing the temperature of the input temperature, the density of the incoming air is increased with increasing air density of the input air.

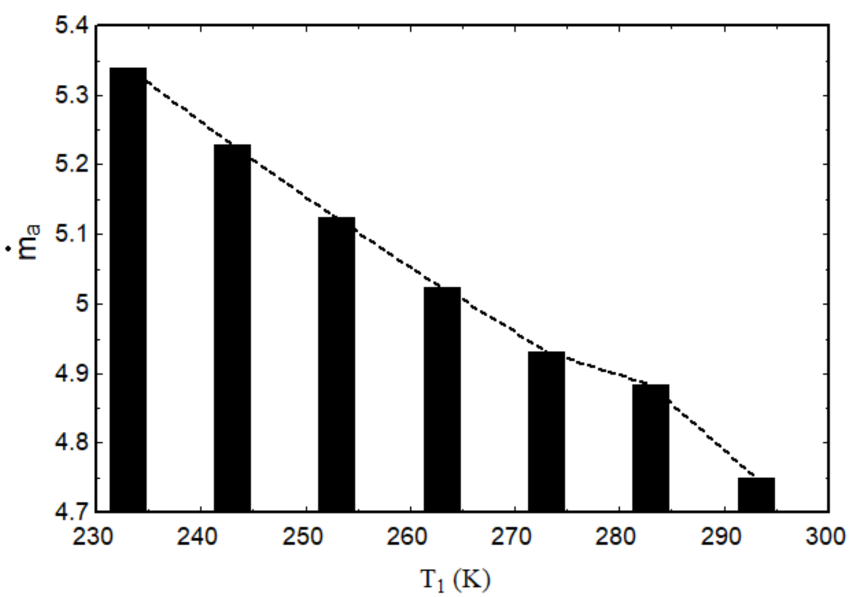

Fig.10. the changes of real inlet airflow rate in return for JP4 fuel in the T700 engine.

The thermal efficiency changes of the T700 engine by using JP4 as fuel are represented by the temperature of the incoming air temperature in Fig.13. since the increase in heat rate due to reduced input temperature decrease is less than increasing the output power of the cycle which is the effect of reduction inlet air temperature. therefore, the thermal efficiency increases with decreasing air temperature.

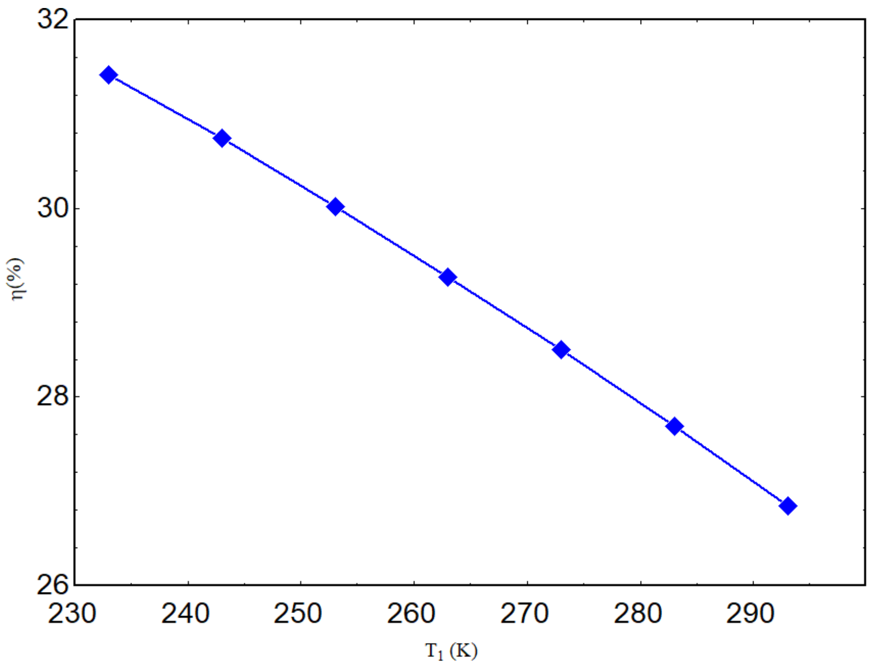

Fig. 13 shows the efficiency of the T700 with the inlet air temperature in return for JP4 fuel.

3.5 Optimization

in this study, the optimization of the T700 engine with the aim of maximum thermal efficiency is carried out to provide 400 KW the output power has done. The variable parameters such as compression ratio, the inlet air temperature, the inlet temperature to the turbine, and the inlet air mass flow rate are considered as design variables. The upper bound of these variables is shown in Table 3. also, the optimized cycle constraints based on the T700 engine cycle with the object parameter of thermal efficiency are shown in Table 4.

the results of optimization of optimized cycle design variables based on the T700 motorcycle with the aim of thermal efficiency are presented in table 5.

It has been explained about the performance parameters and their change in the initial state.

Table 3

design parameters. the optimized cycle is based on the T700 engine cycle, with the aim of thermal efficiency

Parameters	Description	Limits	Values	Unit
π_c	Compression ratio	Upper bound	17.56	-
		Lower bound	4	-
TIT	Turbine inlet temperature	Upper bound	1535	K
		Lower bound	800	K
T_1	Inlet air temperature	Upper bound	353	K
		Lower bound	273	K
m_a	Inlet air mass flow rate	Upper bound	1.5	Kg/s

	Lower bound	0.5	Kg/s

Table 4

the optimized cycle optimization constraints are based on the T700 engine cycle with the objective of thermal efficiency

Parameters	Limits	Values	Unit
P_{net}	Upper bound	420	KW
	Lower bound	380	KW
$PSFC$	Upper bound	250	g/kWh
	Lower bound	100	g/kWh

Table 5

The optimized constraint of the optimized cycle based on the T700 motorcycle with the objective of thermal efficiency for the description of the new concepts

Parameters	Values	Unit
TIT	1400	K
T_1	273	K
m_a	1.222	Kg/s
π_c	15	-

3.6 concept-A

according to use series Hybrid electric vehicle (SHEV) to improve the heat efficiency of

a simulated annealing cycle to reduce the

inlet air temperature from ambient air temperature to the nominal air temperature of the cycle in terms of the T700 engine cycle, the thermal efficiency was increased by objective maximization of thermal efficiency with genetic algorithm. in addition, to use the output power of the

optimized cycle based on the T700 engine cycle to optimize the thermal efficiency of the hybrid

electric vehicle in electric motor, an electrical generator has been converted mechanical output power requirement of the optimized cycle to electrical power. also, the required input power for the compression refrigeration cycle is from the output electric power to the generator. this cycle was called concept - A. the changes of net power of the refrigeration cycle component in concept-
A with ambient air temperature and COP for the refrigeration cycle are shown in fig14. by increasing amb ient air temperature and constant COP due to increased heat from inlet airflow according to equation (14), the required power for cooling is increased, therefore the net output power of the cycle decreases. also, for constant ambient air temperature because with the COP increasing according to equation (15), the output power increased.

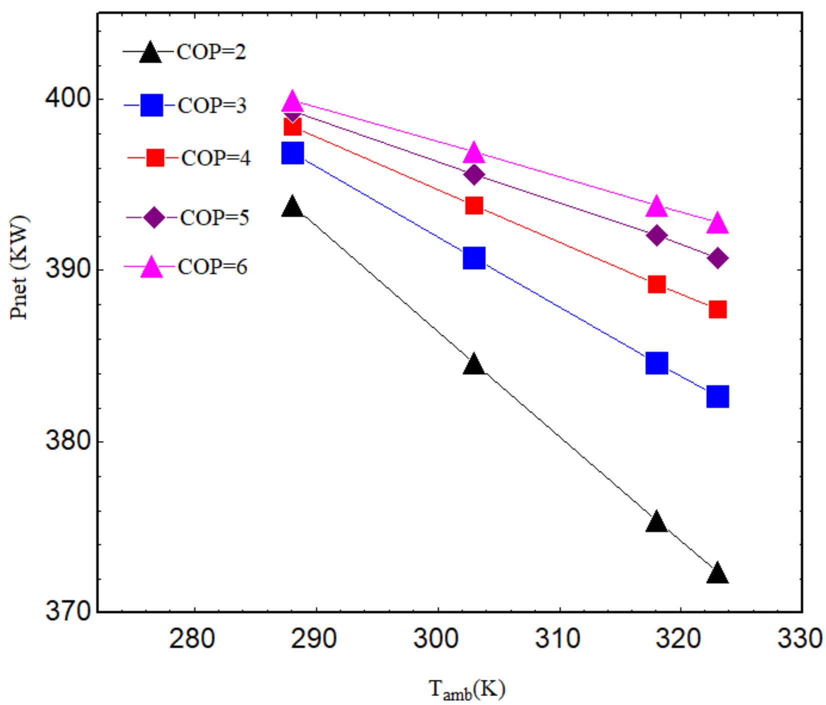

Fig.14. The useful changes to the output power of the concept - A cycle with the temperature of the environment and the COP for the chill cycle. in return for fuel.

also, the thermal efficiency of the concept - A cycle with the different COP in the chill cycle and the ambient temperature for JP4 fuel is shown in Fig 15. since, in return for a constant COP with increasing the temperature of the ambient air, more power is needed for cooling the flow of the input air, so the output power of the cycle decreased because of decreasing the thermal efficiency of the cycle. Also, in return for the constant ambient temperature, where the COP requires less work for the cooling of the input airflow, the efficiency of the output increases with increasing the COP, and increasing the efficiency of the output leads to an increase in the thermal efficiency of the cycle.

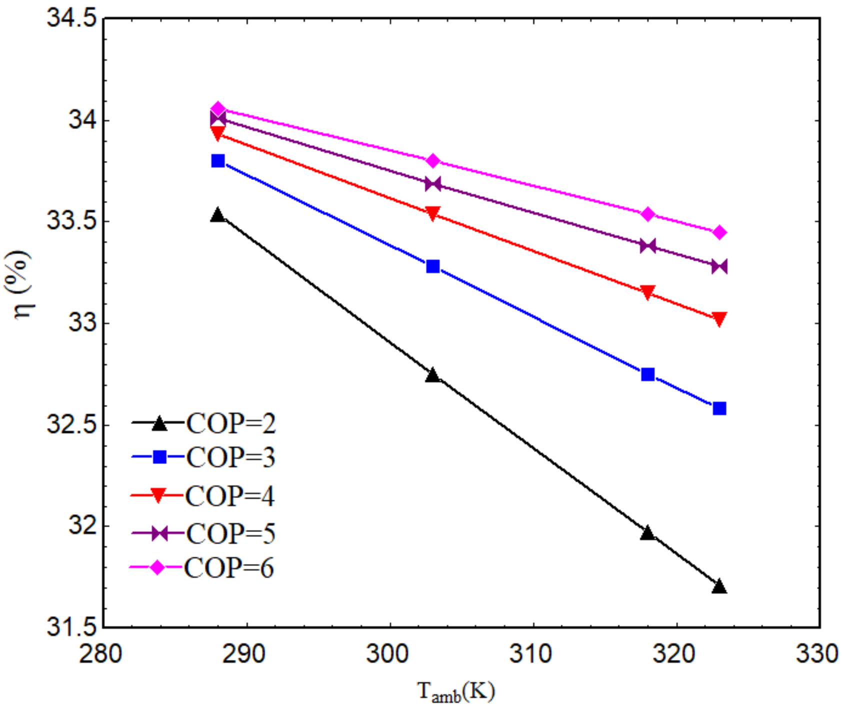

Fig.15. the changes of thermal efficiency of the concept - A cycle with the COP in the chill cycle and the ambient temperature in return for JP4 fuel.

Also, the variation of the concept - A cycle with the COP in the chill cycle and with ambient temperature is shown in Fig. As the temperature of the ambient temperature increases for the COP at a constant temperature, the work required for the cooling is increased and the effectiveness of the output is reduced. It is also consistent with the temperature of the constant environment with the COP increased because the work required to cool the input airflow decreases and the efficiency of the output increases. Therefore, the PSFC decreases.

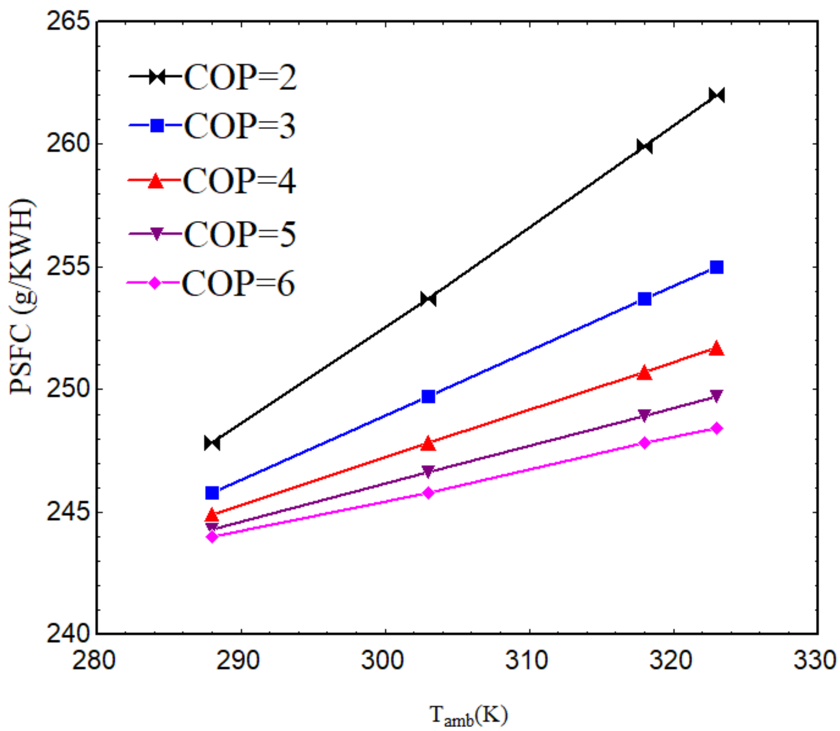

Fig.16. The PSFC cycle of the concept - A cycle with the variation COP in the chill cycle and at a temperature of ambient per fuel.

3.7 concept-B

in this study, in another mode, to the two-heat exchanger was added the concept - A cycle. a heat exchanger is carried out by the condenser to transfer the compressive chill to the flow airflow. another heat exchanger as heat recovery is used for the heat absorbed from the exhaust flow of the power turbine to the airflow of the cycle output of the converters enters the combustor. This cycle was called the concept - B cycle. The changes of the concept - B cycle output power with the ambient temperature of the medium and the COP for the mango production refrigeration cycle at Te= 400 k in terms of the exhaust temperature to the ambient which, is the outlet of heat recovery is displayed for the JP4 fuel in Figure (17). by increasing the temperature of the ambient air in returns of JP4 fuel and a fixed COP due to the heat increase from the incoming airflow according to equation (14) the required power for cooling is increased, hence, the net output power of the cycle decreases. Also, in return for the constant ambient temperature due to the increase in the COP with the function of equation (15) for the input temperature, the output power will be increased.

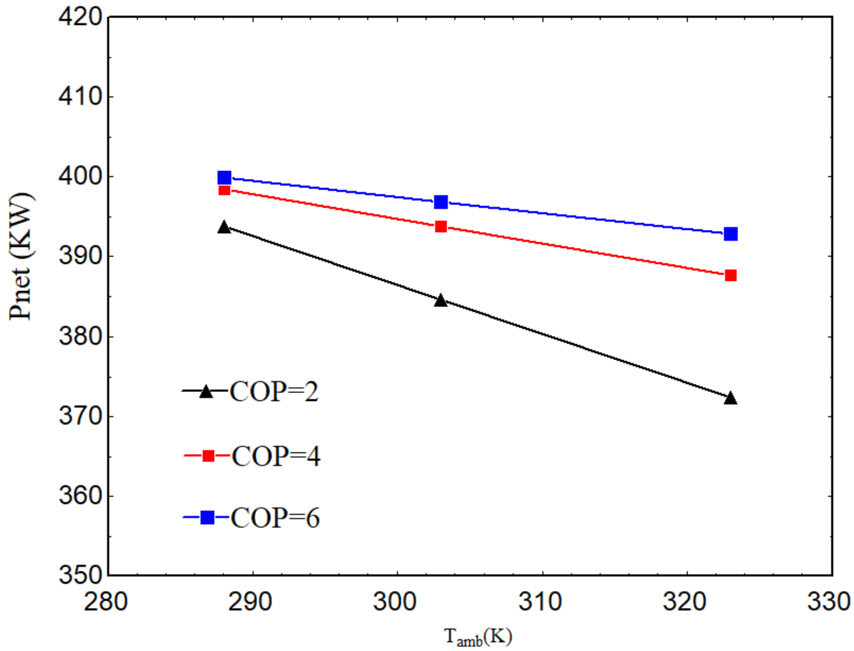

Fig. 17. The useful changes to the output power of the concept - B cycle with temperature and COP for the chill cycle. For the output temperature of Te = 400 k for JP4 fuel

Also, the thermal efficiency of the concept - B cycle with the COP in the chill cycle and the ambient temperature for the JP4 output temperature = 400 k is shown in Fig. 17.

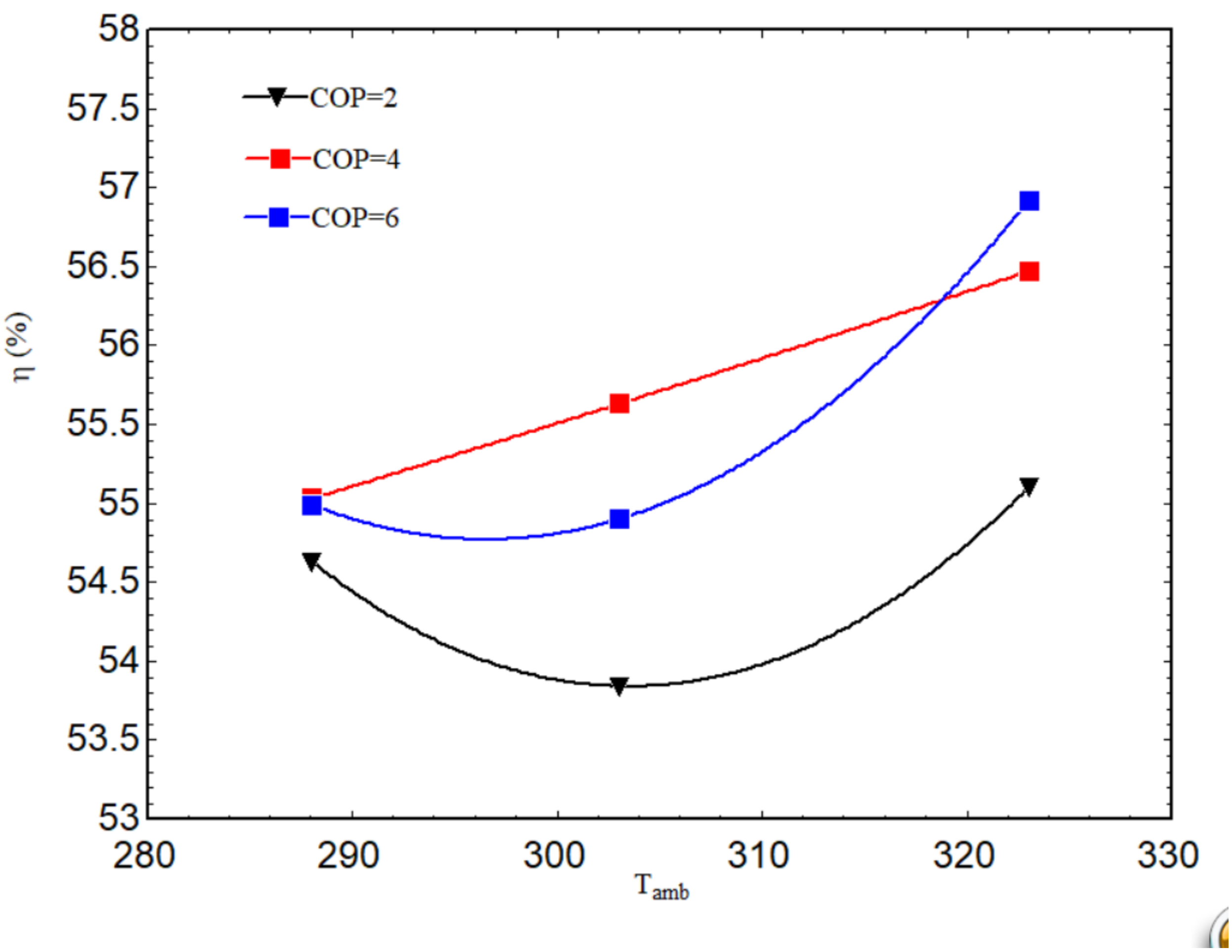

Fig. 18. also showed the thermal efficiency of the concept - B cycle with the COP in the chill cycle and temperature of Te = 400 k in return for fuel.

Also, the PSFC cycle of the concept - B cycle with the COP in the chill cycle and the temperature of the ambient temperature = 400 k is displayed in Fig. 400 k for the JP4 fuel in Fig.

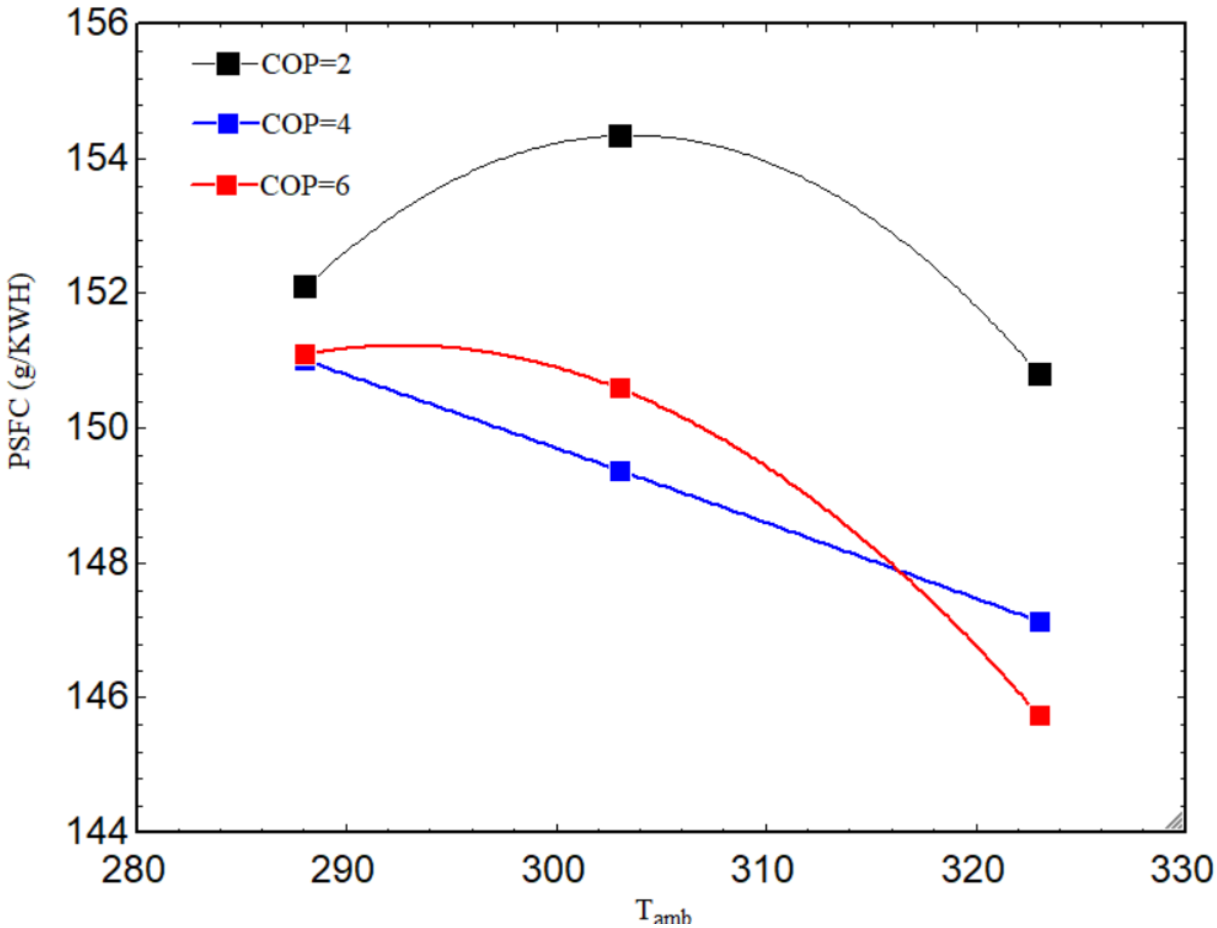

Fig. 19. The PSFC cycle of the concept - B cycle with the COP in the chill cycle and at a temperature of Te = 400 k for JP4 fuel.

in this part, thermal efficiency and heat efficiency were compared for the motorcycle in the F (Ma = 0, h = 0) and the concept - B cycle and the XU7 / L3 cycle.

The performance parameters of the XU7 / L3 engine are shown in Table 5.[32]

Table 5

The performance parameters of the XU7 / L3 engine for JP4 fuel according to different rotational speeds.

	Rotational Speed ω (RPM)							
	1500	2000	2500	3000	3500	4000	4500	5000
$P_{net}\,(KW)$	19.68	26.67	37.40	44.10	51.08	57.54	61.89	64.57

$m_f\,(g/s)$	1.28	1.74	2.49	2.91	3.54	4.18	4.72	5.05
$PSFC(g/KWh)$	235	236	240	238	250	262	275	282
$\eta(\%)$	35.48	35.37	34.66	34.97	33.30	31.76	30.26	29.50

The efficiency of the concept - B cycles per different amount of the COP and T700 in the take-off state Mach =0 and altitude H=0 and the XU7 / L3 engine in Fig. 20 show the thermal efficiency of concept - B cycle for different values of COP in the T700 motorcycle and the XU7 / L3 cycle.

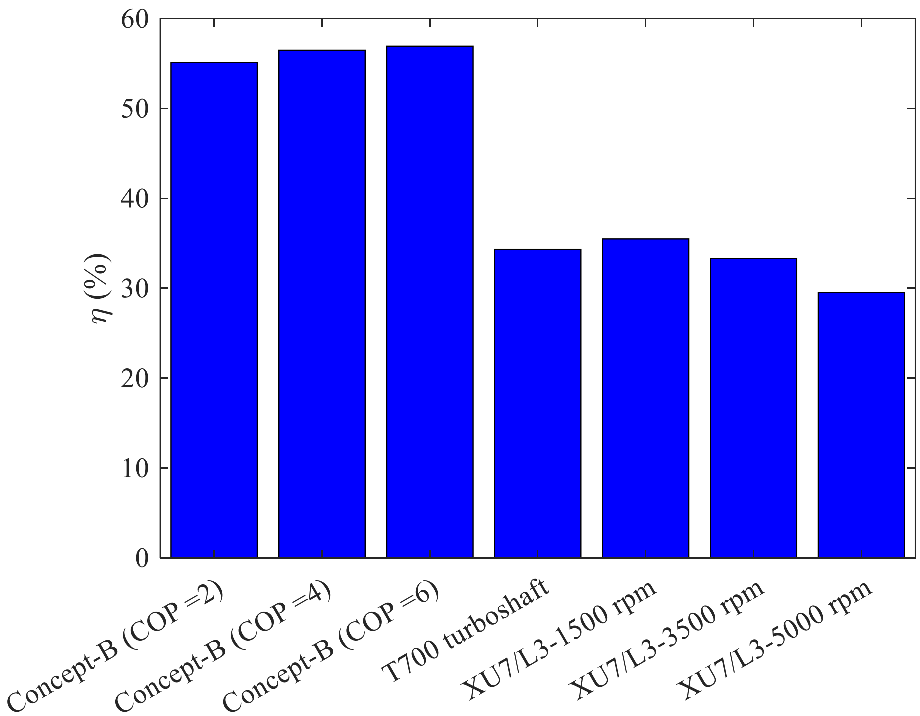

Fig. 20. The efficiency of cycles of concept - B cycles per different amount of COP and T700 in the (Ma = 0, h = 0) and the XU7 / L3 engine in rotational speed revolutions for JP4 fuel and ambient temperature of 323 K.

The PSFC of concept – B cycle for a different amount of COP, T700 in the take-off position with condition Mach =0 and altitude H=0 and the XU7 / L3 engine in different rotational speed revolutions for JP4 fuel in the condition of 323 K ambient temperature is shown in Fig 21. the results show that the concept - B cycle in return for different values of COP is less than the T700 motorcycle and the XU7 / L3 cycle.

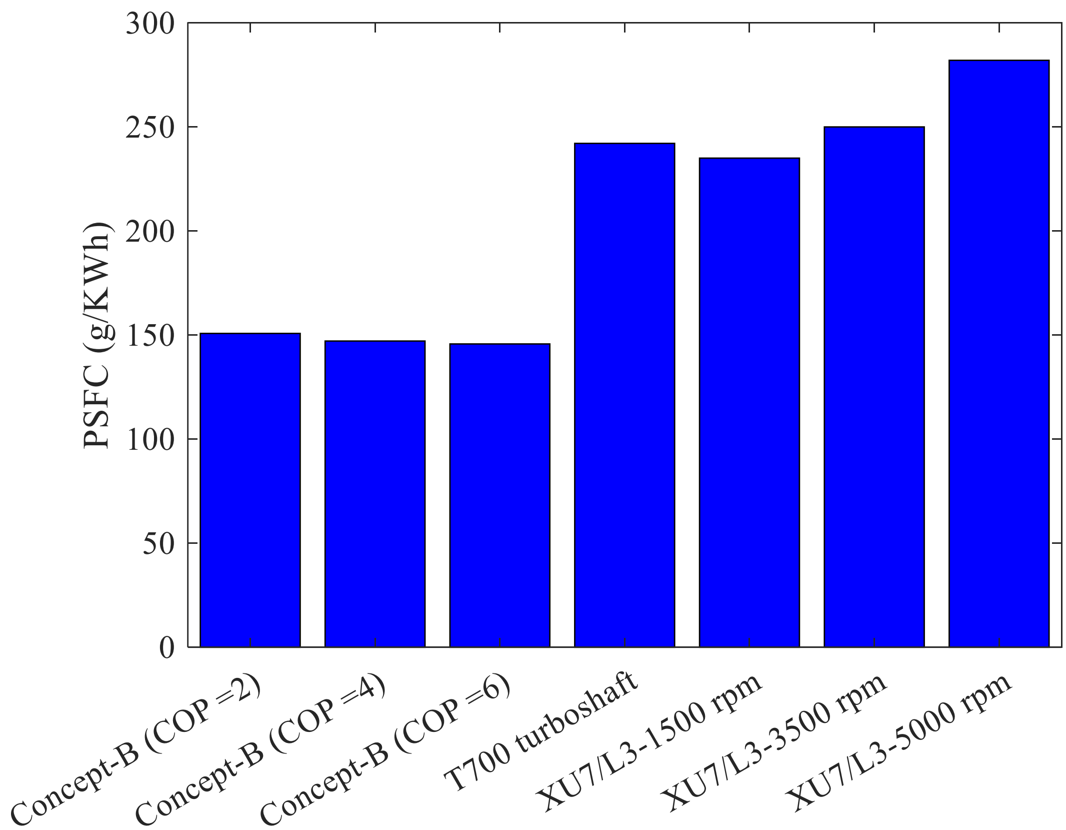

Fig. 21. cycles of concept - B cycles per different amount of COP and T700

3.8 Effect of Biofuels

in this study, the cost of fuel consumption per hour function of concept - B cycle in return for the use of three types of fuel Liquid natural gas, ethanol, and Diesel were compared.

The cost of fuel consumed per hour of the concept - B cycle function for the COP = 2 and the ambient temperature of 302 K for Liquid natural gas ,ethanol, and Diesel in table (6) are presented.

Table 5

the fuel cost per hour of the concept - B cycle function for the COP = 2 and the ambient temperature of 302 K for Liquid fuels, gas and ethanol, and Diesel[33]

Cycle	Fuel type	$FHV(Mj/Kg)$ [33]	$FC(\$/Kg)$	$m_f\,(g/s)$ [33]	$\dot{C}_f\,(\$/h)$
concept-B	Liquid natural gas	49.736	1.348	14.361	69.68
	Diesel	42.734	0.697	16.715	39.60
	Ethanol	26.810	0.489	26.643	46.80

the results showed that the minimum fuel cost consumption was diesel fuel using and the minimum mass flow rate was diesel fuel Liquid natural fuel between chosen fuel diesel.

4. Conclusion

in this study, first, the T700 engine cycle was simulated. effect of inlet air temperature, turbine inlet temperature (TIT), and a compression ratio of the compressor, separately on engine performance parameters such as the output power, PSFC (specific fuel consumption), and thermal efficiency were investigated. thus, in terms of the T700 engine cycle. thermal stability analysis was carried out to optimize the thermal efficiency and to generate intended for 400 KW.

the output power, and optimization is enhanced with various design parameters. compression ratio to the compressor, inlet air temperature (TIT), inlet air flow rate, and inlet air temperature to the compressor was measured. to use the hybrid Concept-A and Concept-B, the optimized cycle was investigated.

to cool the inlet air to the nominal temperature limit of the optimized cycle Concept-A considered. also, converting mechanical output power to hybrid-electric power can be used for Hybrid electric vehicles. the power required for the compression refrigeration cycle is supplied from the output electric power in the component of a generator. the effect of ambient air temperature and COP of the compression refrigeration cycle for performance parameters consist of PSFC, the output power, and thermal efficiency were investigated. The results showed that for each constant ambient temperature and by increasing COP of refrigeration cycle in Concept-A cycle the thermal efficiency and output power decreased. also, for each COP constant with increasing ambient air temperature, the specific fuel consumption (PSFC) increased.

At Concept-B, exhaust heat recovery was added to Concept-A. in addition, the expelled heat was also heated. the heat absorbed from the two transducers as heat exchangers are added to the airflow of the outlet air, and the airflow enters the combustor. the effect of ambient air temperature and cop of the compression refrigeration cycle in performance parameters including PSFC, the output power, and thermal efficiency were investigated. The results showed that the Concept - B cycle was achieved at constant ambient temperature by increasing COP of the refrigeration cycle, the output power increases with increasing COP. as well, in COP constant by increasing the ambient temperature, the output power was decreased. The heat efficiency of the thermal efficiency at the cycle of the concept-B for the COP=4 and COP=6 at the ambient temperature 288K, orderly the were 54.64%,55.03%, and 54.99% also, at the concept-B for the COP=4 and COP=6 at the ambient temperature 323K, orderly the thermal efficiency was 55.10%,56.47% and 56.92%.

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