TECHNICAL FACTORS AFFECTING ON THE INTERNAL COMBUSTION ENGINE PERFORMANCE AND ECONOMICAL FOR IRRIGATION PUMP OPERATION

By

HAYDER ABDULHUSSAIN SHANAN

B.Sc. Agric.Sc. (Machines and Equipment), Fac. of Agric., Univ. of Basrah, Iraq.(2008)
M.Sc. Agric.Sc. (Agric. Engineering), Fac. of Agric, Mansoura Univ., Egypt. (2016)

A Thesis Submitted in Partial Fulfillment
Of
The Requirement for the Degree of

DOCTOR OF PHILOSOPHY
in
Agriculture Sciences
(Agricultural Engineering)

Department of Agricultural Engineering
Faculty of Agriculture
Ain Shams University

2022

Approval Sheet

TECHNICAL FACTORS AFFECTING ON THE INTERNAL COMBUSTION ENGINE PERFORMANCE AND ECONOMICAL FOR IRRIGATION PUMP OPERATION

By

HAYDER ABDULHUSSAIN SHANAN

B.Sc. Agric.Sc. (Machines and Equipment), Fac. of Agric., Univ. of Basrah, Iraq.(2008)
M.Sc. Agric.Sc. (Agric. Engineering), Fac. of Agric, Mansoura Univ., Egypt. (2016)

This thesis for ph.D degree. degree has been approved by:

Dr. Mohamed Mahmoud Ibrahim **………………….**
Prof. of Agricultural Engineering, Faculty of Agriculture, Cairo University.

Dr. Mustafa Fahim Mohamed **………………….**
Prof. of Agricultural Engineering, Faculty of Agriculture, Ain shams University.

Dr. Abdullah Mahmoud Abdelmaqsoud **………..…...…**
Prof. of Agricultural Economics, Faculty of Agriculture, Ain shams University.

Dr. Khaleed Farran El Bagoury **……….……...…**
Prof. of Agricultural Engineering, Faculty of Agriculture, Ain shams University.

Date of Examination: / / 2022

TECHNICAL FACTORS AFFECTING ON THE INTERNAL COMBUSTION ENGINE PERFORMANCE AND ECONOMICAL FOR IRRIGATION PUMP OPERATION

By

HAYDER ABDULHUSSAIN SHANAN

B.Sc. Agric.Sc. (Machines and Equipment), Fac. of Agric., Univ. of Basrah, Iraq.(2008)
M.Sc. Agric.Sc. (Agric. Engineering), Fac. of Agric, Mansoura Univ., Egypt. (2016)

Under the supervision of:

Dr. Khaleed Farran El Bagoury
Prof. of Agricultural Engineering, Faculty of Agriculture, Ain shams University.
(Principle supervisor).

Dr. Abdullah Mahmoud Abdelmaqsoud
Prof. of Agricultural Economics, Faculty of Agriculture, Ain shams University.

Dr. Waleed Kamel El-Helew
Associate Prof. of Agricultural Engineering, Faculty of Agriculture, Ain Shams University.

ABSTRACT

Hayder Abdulhussain Shanan: Technical Factors Affecting on The Internal Combustion Engine Performance and Economical for Irrigation Pump Operation. Unpublished Ph.D. Dissertation, Department of Agricultural Engineering, Faculty of Agriculture, Ain Shams University, 2022.

This study presents the experimental for the types of mixers used in mixing natural gas with air to operate the irrigation pump to save energy, many of measurements were carried 2021 year in workshops of Agricultural Engineering Department, Faculty of Agriculture, Ain Shams University, Egypt. Used engine air-cooled 4-stroke, single cylinder.. A new pump with a discharge diameter of 2 inches, which was an Egyptian manufacture. Several types of mixers were manufactured to mix natural gas with air before entering the engine. seven types of mixers were used (30 angle T mixer - 45 angle T mixer - 90 angle T mixer - venture mixer - 8cm perforated inner tube mixer - 10cm perforated inner tube mixer - Mixer With a perforated inner tube of length 12 cm). selected determine the four shaft speeds (1750, 2300, 2900 and 3500 rpm). The results here dealt with study the analysis of technical indicators for the types of mixers. where the actual power (Braking power) is superior to all types when operating with gasoline was (3.07 kW), A comparison with the use of natural gas, where the mixer type (T45) gave the highest power compared to the types of mixers T (2.83 kW) at an engine speed of (3500rpm) was 7.5% less than gasoline. The lowest Specific fuel consumption (s. fc) was gasoline (219.025 gm/kw.hr) and the highest specific fuel consumption when the mixer type (VM) was (668.52 gm/kw.hr). The percentage of CO_2 exhaust gases when operating with gasoline exceeding all types it was (7.96%) and the lowest percentage CO_2 was (2.77%) using mixer type T90. The highest pump discharge was with the T45 mixer (33.09 m^3/h), an increase of 2.1 % over gasoline at engine speed of (3500 rpm). As for the economic indicators the lowest Internal Rate of Return (IRR) was Gasoline (0.44). The highest Net Present Value (NPV) when carrying the mixer type (L10) was (108893.8). lowest Benefit Cost Ratio (B/C) was the mixer type (VM) was (1.38), and the highest (B/C) when carrying the mixer type (L10) was (1.56). The lowest payback period was the type of mixer (L10) was (1.66 year),and the highest payback period When the (VM) mixer, it was (2.37 year).

keywords: Power, Specific fuel consumption, irrigation pump, natural gas, gasoline, mixer, Engine speed and economic.

ACKNOWLEDGMENT

All greatest gratitude and deepest appreciation to Allah who enabled me to overcome any problems which faced me during the course of these investigation and helped me to achieve the aims of my thesis and accomplish this work.

I would like to express my deep thanks and gratitude to **Dr. Khaleed Farran El Bagoury,** Prof. of Agricultural Engineering, Faculty of Agriculture, Ain shams University, for his faithful supervision of my work, his endless help, advice and invaluable comments which were a great help to me.

Special thanks for **Dr. Abdullah Mahmoud Abdelmaqsoud,** Prof. of Agricultural Economics, Faculty of Agriculture, Ain shams University, for his support, advice help and continuous encouragement throughout the preparation of this thesis.

I wish to express my deepest gratefulness to **Dr. Waleed Kamel El-Helew,** Agricultural Engineering Department, Faculty of Agriculture, Ain Shams University, for supervising the work, suggesting, the topics of this research, continuous support, his guidance and invaluable criticism that lead to the final successful completion of this work.

Thanks to all staff members of Agricultural Engineering Department and postgraduate staff, Faculty of Agriculture, Ain shams University, for their help and providing facilities.

Lastly, My full respect and deep thanks to my father, my mother, my brothers, my wife for creating the suitable circumstances, encouragement and assistance during this study.

CONTENTS

Title	Page
LIST OF TABLES	**III**
LIST OF FIGURES	**IV**
LIST OF ABBRIVATION	**VIII**
1. INTRODUCTION	**1**
2. REVIEW OF LITERATURE	**4**
2.1. Characteristics of gas as a fuel for internal combustion engines	4
2.2. Carburetors	5
2.3. Specific fuel consumption	6
2.4. Dual fuel engines	7
2.5. Engine power	9
2.6. Fuel combustion (gasoline - natural gas - biogas)	11
2.7. Calorific value of fuel	14
2.8. Exhaust gas emissions	15
2.9. Engine efficiency	18
2.10. The costs and economic indicators	21
2.11. Biogas processing	24
2.12. Water pumps	25
2. 13. Combustion equations	27
2.14. Mixing chambers	28
3. MATERIALS AND METHODS	**32**
3.1 Materials	32
3.1.1. Engine	32
3.1.2 The pump	33
3.1.3 The gas cylinder and pressure regulator	33
3.1.4 Gas mixer	34
3.2. Methods	38
3.2.1. Discharge measuring tank	38
3.2.2.Water pressure gauge	38
3.2.3.Engine speed meter	39
3.2.4.Anemometer	40

3.2.5. Exhaust gases measuring device	41
3.2.6. Stop watch	42
3.2.7. Vernier calipers	42
3.3. Experimented measurements	42
3.3.1. Measuring the fuel consumption rate	43
3.3.2. Measuring the engine power	44
3.3.3. Measuring the performance of the pump	45
3.3.4. Measurement of engine exhaust gases	46
3.3.5. Economic indicators:	46
3.3.6. The studied factors	47
4. RESULTS AND DISCUSSION	**48**
4.1. Engine technical indicators	48
4.1.1. Engine power	48
4.1.2. Specific fuel consumption	50
Exhaust gases (CO_2 , CO , HC)	55
4.2. Pump indicators	59
4.2.1. Power and discharge of the pump using gasoline	59
4.2.2. Power and discharge of the pump using mixer type (T30)	60
4.2.3. Power and discharge of the pump using mixer type (T45)	61
4.2.4. Power and discharge of the pump using mixer type (T90)	62
4.2.5. Power and discharge of the pump using mixer type (VM)	63
4.2.6. Power and discharge of the pump using mixer type (L8)	64
4.2.7. Power and discharge of the pump using mixer type (L10)	65
4.2.8.Power and discharge of the pump using mixer type (L12)	66
4.3.Economic Indicators	68
5. SUMMARY AND CONCLUSION	**76**
6. REFERENCES	**80**
7. APPENDIX	**91**

LIST OF TABLES

Table		Page
1	Comparison of combustion types of fuels	14
2	Typical exhaust emissions from gasoline and diesel engines	18
3	Engine specifications	32
4	Specifications of the water pump	33
5	Specifications of the tachometer	39
6	Specifications of the air velocity meter	40
7	Specifications of the exhaust gas meter	41

List of Figures

Figures		Page
1	Schematic combination of air-oil-biogas mixer with carburetor	6
2	Energy conversion and losses in a pumping system	26
3	T-joint mixer	28
4	T-joint mixer with oblique, protruding gas inlet	29
5	Simple mixing chamber with hand - controlled valve	30
6	Mixing chamber with gas distribution pipe and wire mesh for intensive mixing	30
7	Venturi mixer with gas supply through several bores.	31
8	Single cylinder engine	32
9	Natural gas cylinder	33
10	Types of gas mixers	34
11	T-type mixer 90	35
12	T-type mixer 45	35
13	Mixer T-angle 30	36
14	Venture mixer	36
15	Mixer with a perforated inner tube	37
16	Simplified diagram of natural gas engine for powering water irrigation pump	37
17	A tank to measure water drainage	38
18	Water pressure gauge	39
19	Tachometer	39
20	Wind speed measuring device	40
21	Exhaust gas measuring device	41

22	Stop watch	42
23	Vernier calipers	42
24	Engine speed gauge slot	43
25	Gasoline consumption measuring tube	44
26	Simplified diagram of Prony brake.	45
27	The relationship between engine speed and the braking power for all types of mixers (gasoline – T30- T45- T90- VM)	49
28	The relationship between engine speed and the braking power for all types of mixers (gasoline – L8 – L10 - L 12 cm)	50
29	The relationship between engine speed and specific fuel consumption for each of (gasoline, T30, T45, T90, VM)	51
30	Specific fuel consumption for (gasoline, T30, T45, T90, VM) with a load at an engine speed of (2900, 3500) rpm	52
31	The relationship between engine speed and specific fuel consumption for each of (gasoline, L8, L10, L12)	53
32	Specific fuel consumption for (gasoline, L8, L10, L12) with a load at engine speed (2900- 3500) rpm	54
33	Specific fuel consumption for all mixer at an engine speed of (2900 rpm)	54
34	Exhaust gases average CO_2% for all types of mixers (Gasoline, T30, T45, T90, VM)	55
35	Exhaust gases average CO_2% for all types of mixers (Gasoline, L8, L10, L12)	56
36	Exhaust gases average CO% for all types of mixers (Gasoline, T30, T45, T90, VM)	57
37	Exhaust gases average CO% for all types of mixers (Gasoline, L8, L10, L12)	57

38	Exhaust gases average HC ppm for all types of mixers (Gasoline, T30, T45, T90, VM)	58
39	Exhaust gases average HC ppm for all types of mixers (Gasoline, L8, L10, L12)	59
40	The relationship between discharge, pressure and the hydraulic power of pump operating with gasoline at engine speed (3500 rpm)	60
41	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T30 at engine speed (3500 rpm)	61
42	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T45 at engine speed (3500 rpm)	62
43	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T90 at engine speed (3500 rpm)	63
44	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type VM at engine speed (3500 rpm)	64
45	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L8 at engine speed (3500 rpm)	65
46	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L10 at engine speed (3500 rpm)	66
47	The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L12 at engine speed (3500 rpm)	67
48	The hydraulic power for all types of mixers at engine speed (3500 rpm)	68
49	Net present value (NPV) for all types of mixers (gasoline – T30 – T45 – T90 - VM)	69
50	Net present value (NPV) for all types of mixers (gasoline – L8 – L10 – L12)	69
51	Benefit cost ratio (B/C) for all types of mixers	70

	(gasoline – T30 – T45 – T90 - VM)	
52	Benefit cost ratio (B/C) for all types of mixers (gasoline – L8 - L10 – L12)	71
53	Internal rate of return (IRR) for all types of mixers (gasoline – T30 – T45 – T90 - VM)	71
54	Internal rate of return (IRR) for all types of mixers (gasoline – L8 – L10 – L12)	72
55	Payback period (year) for all types of mixers (gasoline – T30 – T45 – T90 - VM)	73
56	Payback period (year) for all types of mixers (gasoline – L8- L10- L12)	73
57	The economic efficiency for all types of mixers (gasoline – T30- T45- T90- VM)	74
58	The economic efficiency for all types of mixers (gasoline – L8- L10- L12)	75

LIST OF ABBREVIATION

CNG	Compressed natural gas
H/C	Hydrogen / nitrogen
SI	Spark ignition
RON	Research octane number
MRG	Methanol reformed gas
ON	Octane number
ICEs	Internal combustion engines
A/F	Air/ fuel
LPG	liquefied petroleum
S. fc	Specific fuel consumption
BSFC	Brake specific fuel consumption (gm/Kw.h)
FIE	fuel injection equipment
NEDC	New european driving cycle
RPM	Revolution per minutes
NGV	Natural gas volume
GHG	Greenhouse gases
LFG	Landfill gas
EGR	Exhaust gas recirculation
DI	Diesel engine
T90	Mixer T-angle 90°
T45	Mixer T-angle 45°
T30	Mixer T-angle 30°
VM	Venture mixer
L8	Mixer with a perforated inner tube of 8cm
L10	Mixer with a perforated inner tube of 10cm
L12	Mixer with a perforated inner tube of 12cm
NPV	Net present value
B/C	Benefit cost ratio
IRR	Internal rate of return

INTRODUCTION

Energy has a crucial role in the rural course of events. However, the price of oil has already surpassed $100 a barrel on the global market and is expected to continue to rise. In this way, the use and accessibility of environmentally friendly power is currently the main concern for most academics (sunlight based - wind energy - natural gas - biogas - and etc) The significant and expanding energy demand in rural Egypt is a direct effect of the economy's turnaround and population increase. Oil makes up a sizable share of the energy consumed, but the substation does not prevent the use of partial or full energy. Environmentally and financially sound elective energy sources are ideal. Egypt's rural areas are seeing a high and rising demand for energy as a result of recent economic growth and population growth. Find alternatives to oil so that you can reduce your energy use completely or partially. Oil makes up a large portion of your energy use. Both ecologically responsible and commercially effective alternative energy sources are required **(Abdel-Galil *et al.*, 2008).**

The world is currently dealing with two crises: the depletion of fossil fuels and the destruction of the environment. The amount of underground-based carbon resources has decreased as a result of indiscriminate fossil fuel extraction and extravagant use. It has become crucial to look for alternative fuels that promise to be compatible with sustainable development, energy efficiency, and environmental preservation **(Faizala *et al.*, 2009).**

Alternatives to petroleum-derived fuels for internal combustion engines must be developed due to the depletion of fossil fuel resources. Since compressed natural gas (CNG) is far more plentiful than petroleum, it can be a great alternative fuel **(Andrei *et al.*, 2019).** Due to its relatively high octane rating, natural gas has recently been seen as a clean alternative fuel for Spark Ignition (SI) engines. In comparison to gasoline, the thermal efficiency and emissions would have increased with the mild burning of natural gas, which is primarily composed of methane, in SI engines. Natural gas enables higher pressure ratio combustion without banging thanks to its high research octane number (RON) of over 120. Due to its high hydrogen to carbon ratio, it also emits far less CO_2 than conventional hydrocarbon fuels **(Saad *et al.*, 2014).**

The combustion cycle is the most significant and complicated step in the SI four-stroke engine. The fact that the chemical energy of the fuel is transformed into thermal energy during this cycle has a substantial impact on engine performance and pollutant emissions, when the combustion time is too short, the fuel will not be entirely burned, preventing the fuel's chemical energy from being fully transformed into heat energy **(Nguyen *et al.*, 2019).**

Mixing devices for gases used in gas engines generally referred to as carburetor, for mixing air and gaseous fuels are commonly attached to the intake manifold of an internal combustion engine .For a specific engine load and speed, the ratio of air to gaseous fuels in a gas carburetor must be correct. To ensure simple adjustment and repeatable performance, simplicity and toughness have always been prioritised in the design of the production gas carburetor **(Ani *et al.*, 2006).** In comparison to gasoline, CNG was proven to have lower CO and CO_2 emissions. Internal combustion engine power and torque production are primarily influenced by the mass of the cylinder mixture and, obviously, the type of fuel utilised. Therefore, compared to other engine parameters, volumetric efficiency plays one of the most crucial roles when dealing with different fuel qualities with the same engine **(Sulaiman *et al.*, 2013).**

When the distance between the mixer and the input manifold is small and the mixing time is sufficient, mixing chambers with a higher volume than just a T-Joint tube allow longer air and fuel retention time within the chamber. Because the flow velocities are relatively lower, more time may be spent mixing, allowing for the use of a straightforward mixing chamber. The use of cleaner alternative fuels, like natural gas, methanol reformed gas (MRG), and hydrogen, has become much more widespread in recent years as a response to environmental issues like the effects of global warming and the depletion of crude oil reserves worldwide **(Fathollah *et al.*, 2013).**

Natural gas is made up of a variety of substances, including methane (98% of the dry gas is methane), other substances like ethane (C_2H_6), propane (C_3H_8), and butane (C_4H_{10}), as well as carbon dioxide CO_2, hydrogen H_2, and nitrogen N_2 **(Miqdam *et al.*, 2016)**. The use of mixtures of natural gas and biomass-generated syngas is crucial for real applications, where a shortage of primary sources or power dating issues may result in the need to complement syngas with natural gas **(Carlo *et al.*, 2021)**. Furthermore, CNG can reduce greenhouse gas emissions by up to 25% due to methane's low C:H ratio. **(Konigsson *et al.*, 2011).**

Alternatives to petroleum-derived fuels for internal combustion engines must be developed due to the depletion of fossil fuel resources. Since compressed natural gas (CNG) is far more plentiful than petroleum, it can be a great alternative fuel **(Andrei *et al.*, 2019)**, because of its high H/C ratio and high research octane rating, it produces cleaner exhaust gases than when conventional fuels are burned. Along with having strong anti-knocking properties, it also has a slower flame and a smaller flammability range. **Saravanan *et al.*, (2013)** Because CNG has an 8-times lower carbon content than gasoline, fewer carbon-based emissions are produced. Complete combustion also occurs in the gaseous fuel, lowering HC emissions. Compared to liquid fuels, gaseous fuels form more homogeneous mixtures. Dedicated CNG engines with higher compression ratios can reduce HC even more.

Through the implementation of a programmer for water rationalization in many sectors, especially agriculture, and the adoption of policies that encourage water cost recovery, Egypt's Vision 2030 **(MPMAR, 2016)** aims to sustainable consumption patterns of water and natural resources. Thermal efficiency continued to decline, and the rate of fuel consumption rose. Simply utilizing fuel with a higher octane number (ON) than gasoline could address this issue. Gasoline can be replaced with natural gas in this situation. Natural gas usually has an (ON) between 120 and 130, thus knocking should be avoided. Additionally, natural gas has a high mining capacity akin to "shale gas," and it is very convenient to utilize as a fuel for automobiles **(Wang and Krupnick, 2015)**. For this reason, some researchers have investigated the use of natural gas in SI engines.

There are many advantages to using this alternative fuel for lowering air and environmental pollution **(Cho *et al.*, 2007).** In the case of a gasoline leak, it is seen to be a considerably safer alternative to conventional fuels from an environmental standpoint. Natural gas disperses quickly when it is spilled or leaking because it is lighter than air **(Tilagone *et al.*, 2005).** Consequently, the purpose of this study is to:

1. Design a gas mixing device natural gas into air stream.
2. Convert SI engine to power a water pump for irrigation by using natural gas.
3. Compare the output power of using natural gas with fuel gasoline.
4. Reducing environmental pollution resulting from exhaust gas.
5. Reducing the costs needed to operate the engine and irrigation pump.

REVIEW OF LITERATURE

The available literature dealing with study Technical factors affecting the performance of an internal combustion engine and its economics for operating an irrigation pump these subject can be reviewed under the following topics:

2.1. Characteristics of gas as a fuel for internal combustion engines:

Natural gas engines can be categorized based on numerous parameters including: mixture preparation(premixed or non-premixed), ignition (spark ignition or diesel pilot) and the dominant engine cycle(Otto or Diesel). One common categorization **(Harldson, 2010)**:

- Premixed charge, spark ignition, natural gas only.
- Premixed charge, diesel pilot ignition, natural gas/diesel dual fuel.
- High pressure direct injection of natural gas, diesel pilot ignition, natural gas/diesel dual fuel.

The use of cleaner alternative fuels, like natural gas, methanol reformed gas (MRG), and hydrogen, has become much more widespread in recent years as a response to environmental issues like the effects of global warming and the depletion of crude oil reserves worldwide. One of the major issues with switching from a gasoline engine to a CNG one is power reduction **(Fathollah *et al.*, 2013).**

Hannu, (2020) Reported that most industrial and commercial natural gas engines are said to fall into one of four technical categories: stoichiometric Otto cycle engines, low burn Otto cycle engines, dual fuel mixed cycle (Otto and Diesel combined) engines, and diesel cycle natural gas engines. The thermal efficiency, performance, and post-treatment needs of these methods vary.

Natural gas is particularly appealing for urban buses and local transportation in highly populated regions since it produces low amounts of smoke and particulate matter, and its contribution to the development of pollution is negligible compared to SI-engines fueled with gasoline and diesel. **(Dhyani and Subramanian, 2019 ; Corbin *et al.*, 2020).**

Thermal efficiency continued to decline, and the rate of fuel consumption rose. Simply utilizing fuel with a higher octane number (ON) than gasoline could address this issue. Gasoline can be replaced with natural gas in this situation. Natural gas usually has an ON between 120 and 130, thus knocking should be avoided. Additionally, natural gas has a high mining capacity akin to "shale gas," and it is very convenient to utilize as a fuel for automobiles. **(Wang and Krupnick , 2015)**. For this reason, several researchers have attempted to use natural gas in SI engines.

Natural gas is ideal for city transportation since it produces fewer pollutants and soot particles. Additionally, using natural gas in spark ignition (SI) engines allows for cylinder pressure to exceed 90 bar and spark plug gap tension to practically triple at full load in a supercharged engine. SI engines typically have compression ratios between 10 and 12. It is preferable to guarantee ignition by using a modest amount of diesel pilot injection to solve this issue. 12 Additionally, efficiency declines as a result of the partial load use of natural gas in SI engines, the presence of a throttle in Otto engines, and low compression ratios **(Umierski and Stommel, 2000).**

2.2. Carburetors:

A carburetor is typically described as a device that forces fuel to flow into the air stream due to a flow-induced pressure drop. In an ideal world, the carburetor would supply the engine with the proper air-to-fuel (A/F) ratio throughout the engine's entire operating range, from no load to full load. Carburetors should be repeatable and have clear adjustment processes to ensure proper performance. **(Ani *et al.*, 2006).**

Ani *et al.*, (2006) Reported that the intake manifold of an internal combustion engine is frequently attached with gas mixing devices, sometimes known as carburetors, for combining air and gaseous fuels. For a specific engine load and speed, the ratio of air to gaseous fuels in a gas carburetor must be correct. To ensure simple adjustment and repeatable performance, simplicity and toughness have always been prioritized in the design of the production gas carburetor.

Tjokorda *et al.*, (2021) Found that based on the creative integration of the carburetor with an air-oil-biogas mixer, it was discovered that a typical two-stroke engine could be modified to run on biogas as well as gasoline Figure (1). The 63 cc single-cylinder engine has air cooling.

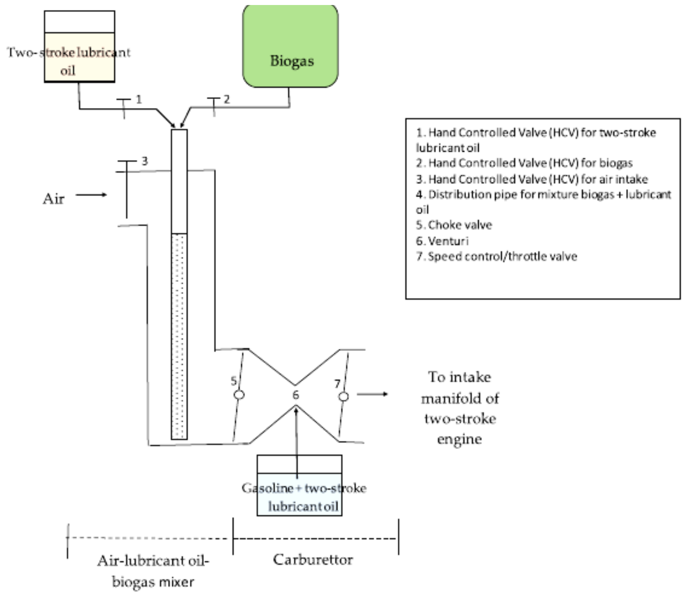


Figure (1): Schematic combination of air-oil-biogas mixer with carburetor

Surata *et, al.*, (2014) Showed that the gasoline engine's carburetor was changed, and only the fuel and air mixer's single component was employed. The biogas fuel intake should be finished with a valve that can be opened automatically by vacuum of the engine's suction stroke. Liquefied petroleum gas (LPG), up to 80% biogas and 20% LPG, was added to the mixture to improve the engine's performance.

2.3. Specific fuel consumption:

(Heywood, 2018) Found that when engine speed and load are close to the maximal torque operation point, the engine's specific fuel consumption (S. fc) normally falls to its lowest value. Losses rise as the proportional amount of crank train friction increases due to increases in low loads. Because friction increases and decreases at high speeds, combustion efficiency similarly decreases at high speeds. The two fuels' consumption rates (BSFC) were high in every instance. This outcome was caused by CNG's higher energy density (49.3 MJ/kg) compared to gasoline (42.8 MJ/kg), The major benefit of using CNG in a SI engine is a superior BSFC of CNG combustion than that of gasoline combustion since fuel weight is essential **(Jeongwoo *et al.*, 2020).**

Consumption of biogas as fuel: The rate of biogas fuel was calculated by timing when the engine consumed a specific volume of biogas while simultaneously measuring the inner tube's pressure at the start and end of the test. Under the identical starting and ending pressures, the volume of biogas used was measured **(House, 1981)** Then the rate of biogas fuel consumption was calculated by using the following equation:

$$F_{Bc} = (V/t) \times 0.0036 \quad \text{.........(1)}$$

V = Volume of consumed biogas fuel (cm^3)

t = Time of operation (S)

FBc = Rate of biogas fuel consumption (m^3/h)

Jahirul *et al.*, (2010) Reported that examined the operation and exhaust emissions of a spark-ignited engine powered by gasoline and Compressed Natural Gas (CNG) at 50 and 80 percent throttle positions. When the engine was running on CNG instead of gasoline, comparative study revealed 19.25 and 10.86 percent less brake power and 15.96 and 14.68 percent less brake specific fuel consumption (BSFC) at 50 and 80 percent throttle settings, respectively **(Jahirul *et al.*, 2010).**

The gaseous fuel methane enriched biogas is as excellent as natural gas mix ratios of city gas, according to the engine test results that were achieved in terms of brake power output, specific gas consumption, and thermal efficiency on methane enriched biogas that contains 95% methane **(Papacz, 2011)**. Natural gas cars can run on biogas as fuel.

Specific fuel consumption (S.fc) for LPG-fueled engines drops to 28.38%. Additionally, compared to ULP engines, LPG engines offer lower energy costs with a difference of up to 47.40%. **(Sulaiman *et al.*, 2013)**

2.4. Dual fuel engines:

changed the fuel injection equipment (FIE) systems completely to convert a gasoline SI engine to a CNG SI engine **(Kichiro et al., 1999).** Due to the lower C/H ratio of CNG, the results demonstrated a 20% reduction in carbon dioxide (CO_2) emissions when compared to those of a gasoline engine. However, the CNG SI engine's maximum load was 10% less than the gasoline SI engine's. Under the New European Driving Cycle (NEDC) conditions, Paola et al. also showed a considerable reduction in CO_2 emissions while using natural gas in place of gasoline in a 1.4-L SI engine. **(Paola and Jose 2009).**

(Bade and Narayanan , 2008) Showed that one instance is the study by where it was shown that running on landfill gas with a compression ratio of 12 resulted in comparable performance for the identical natural gas engine with a compression ratio of 8.5 running on pure methane (the engine used had variable compression ratios).

Saad *et al.*, (2014) Reported that :

• On average, gasoline and CNG produce more brake power, brake torque and brake mean effective pressure at wide throttle position

• Compared to gasoline, CNG operation results in lower brake power, torque, and mean effective pressure.

• Results also show that BSFC of gasoline is less than CNG

• The emission of NO and NO, of CNG is lower at low speeds and high speed but the emission increases at 3000-3500 rpm

• The emissions of CO, and CO_2 were found less of CNG compared to gasoline

Gasoline can be replaced with natural gas in this situation. Natural gas usually has an ON between 120 and 130, thus knocking should be avoided. Additionally, natural gas has a high mining capacity akin to "shale gas," and it is very convenient to utilize as a fuel for automobiles **(Wang and Krupnick, 2015)** This is why a number of researchers have tried using natural gas in SI engines.

(Kichiro *et al.*, 1999) Showed that transformed the fuel injection equipment (FIE) systems entirely, converting a gasoline SI engine to a CNG SI engine. Due to the lower C/H ratio of CNG, the results demonstrated a 20% reduction in carbon dioxide (CO_2) emissions when compared to those of a gasoline engine. However, the CNG SI engine's maximum load was 10% less than the gasoline SI engine's.

(Paola and Jose, 2009) Reported that when natural gas is used in place of gasoline in a 1.4-L SI engine under the new european driving cycle (NEDC) conditions, there is a considerable reduction in CO_2 emissions. Thermal efficiency continued to decline, and the rate of fuel consumption rose. Simply utilizing fuel with a higher octane number (ON) than gasoline could address this issue.

In a review study that examined emissions and performance for compression- and spark-ignited natural gas engines. The findings indicated that both engines could run on natural gas, but that the engines needed to be improved **(Korakianitis *et al.*, 2011).**

Research has been done to modify biogas combustion qualities to more closely mirror those of natural gas, either at the production stage or by employing additives, because biogas is frequently utilized as the fuel in engines made to run on natural gas **(Chandra *et al.*, 2011).**

Next to being less expensive, hydrogen-fueled ICEs offer a number of other benefits of which the most practical one is the ability to run in bi-fuel or flex-fuel operation. These benefits, and experimental research on hydrogen-fueled ICEs is reviewed by the authors elsewhere **(Verhelst S *et al.*, 2006).**

According to reports, a spark ignition engine with a higher compression ratio could replace a gasoline-fueled engine with one that runs on biogas. However, when operating with biogas that contains significant amounts of inert gases like CO_2 and N_2, the performance of the engine will be lower than when running on natural gas or gasoline. It's conceivable that NO_x emissions will rise **(Crookes, 2006).**

2.5. Engine power:

Khan *et al.*, (2016) Natural gas, which is regarded as the most promising alternative fuel, has been thoroughly reviewed by the authors as a transportation fuel. Despite being a tried-and-true technology, port-injected CNG has not gained as much traction as conventional gasoline. The lower torque and power output of bivalent cars compared to gasoline is one of the reasons, in addition to a lack of infrastructure and recharging facilities. This is because the gaseous fuel no longer effectively replaces the incoming fresh air, which results in lower peak torque and power.

It was discovered that 1500 rpm is the highest RPM (revolutions per minute) that can be produced using only biogas. This makes sense given that the energy of biogas is lower than that of gasoline, which ranges between 6.0 and 6.5.kW/m^3 **(Deublin and Steinhauser, 2008).**

Internal combustion engine power and torque production are primarily influenced by the mass of the cylinder mixture and, obviously, the type of fuel utilized. Therefore, compared to other engine parameters, volumetric efficiency plays one of the most crucial roles when dealing with different fuel qualities with the same engine. **(Sulaiman *et al.*, 2013).**

Stefan, (2004) Showed that biogas Otto engines modified from Otto engines using petrol fuel are found to produce less power than the petrol version. The cause is a decrease in volumetric efficiency as a gaseous fuel occupies a larger portion of the mixture's volume sucked into the engine than liquid fuel and displaces air accordingly. The liquid fuel, however, has a higher volumetric energy content than biogas and also cools the air/fuel mixture when evaporating in the intake manifold. The amount of air/fuel mixture that is actually pulled into the engine on a mass basis is higher due to the cooling's impact on an increase in density.

When compared to ULP, LPG fuels have a higher octane value, allowing for a higher engine compression ratio. **(Mustafa and Gitano, 2009)**, and hence, engine performance might be enhanced. Some of these alternatives, which can be used to overcome this power loss as technology advances, include the use of turbocharging, supercharging, and/or raising the compression ratio when operating an engine with naturally aspirated fuel **(Kutlar *et al.*, 2005).**

Dashti *et al.*, (2012) Showed that have predicted the engine performance and emissions using a thermodynamic cycle simulation of a standard four-stroke SI engine running on gasoline and CNG. The findings indicated that due to increased volumetric efficiency, the power of a CNG-fueled engine is approximately 11% less than a gasoline-fueled engine over the speed range of 1500–4000 rpm. Over this speed range, the ISFC is typically lowered by about 16% when the engine runs on CNG fuel. The specific emissions of CO_2, CO and UHC concentration are, however, significantly reduced for this engine speed range, by around 33, 60, and 53%, respectively, while NO concentration is raised by 50%.

Saad *et al.*, (2014) Showed that results Compared to CNG, gasoline has more power, torque, and brake mean effective pressure (BMEP), but less brake specific fuel consumption (BSFC).

The researchers found that, although it reduced brake torque and thermal efficiency, a reduction in CO_2 concentration to 30% of the fuel by volume was best for reducing hydrocarbon and NO emissions. **(Porpatham *et al.*, 2007).**

When biogas is used as a fuel for a spark ignition engine, **(Porpatham *et al.*, 2007)** examination is into the influence of the methane concentration in the fuel revealed that when CO_2 levels were reduced, the brake power output and thermal efficiency increased across the whole range of equivalency ratios examined (0.6 to 1.2).

Chaichan, (2012) Reported that the engine is fueled with natural gas instead of gasoline, the ensuing engine brake power is reduced by around 15%. The study found that the flame speed drop in the lean mixture for both gases causes the optimal spark time for the best torque to increase as the equivalent ratio decreases. Due to methane's slower flame speed than gasoline, natural gas' ideal spark timing for the best torque was larger by roughly 6 degrees than those of gasoline.

Chaichan, (2014) showed that natural gas causes an additional 17% power loss in the engine as compared to gasoline since it has a lower volumetric efficiency.

Pipitone and Genchi, (2014) Reported that due to the low volumetric efficiency of methane, the engine brake power is reduced as compared to gasoline. As opposed to using gasoline, where the fuel cools the air entering the cylinder, gas displaces some of the air entering the cylinder.

2.6. Fuel combustion (gasoline - natural gas - biogas):

The most widely used and dispersed gaseous fuel today and one of the primary alternatives to traditional liquid fossil fuels like gasoline and diesel is natural gas (NG). NG is a fossil fuel that may either be mined independently or derived from other fossil fuels (such as crude oil in oil fields or coal in coal beds) **(Korakianitis *et al.*, 2011).**

Taggart *et al.*, (2006) Found that natural gas composition varies by nation, with CH_4 making up 90% to 96% of total natural gas (methane). The remaining ingredients are 2.41% ethane, 0.74% propane, 0.37% butane, 0.78% nitrogen, 0.16% pentane, and 0.08% carbon dioxide. Furthermore, while being a fossil fuel, CNG can reduce greenhouse gas emissions by up to 25% due to methane's low C:H ratio **(Konigsson *et al.*, 2011).**

An alternative fuel that can be used in place of gasoline, diesel, or propane fuel is compressed natural gas (CNG). This alternative fuel has numerous benefits for reducing environmental and air pollution **(Cho *et al.*, 2007)** . It is considered to be an environmentally ‘‘clean’’ alternative to those fuels and it is much safer in the event of a fuel leakage. Natural gas disperses quickly when it is spilled or leaking because it is lighter than air **(Tilagone *et al.*, 2005).**

Yasin *et al.*, (2016) Showed that natural gas is a significant alternative fuel that can be used as a fuel substitute in internal combustion engines since it meets several of these requirements (ICEs). Natural gas is a very practical alternative fuel since it is readily available, has greater reserves than oil, is less expensive, has better combustion properties, produces fewer car emissions, and is supported by distribution systems.

Miqdam *et al.*, (2016) Reported that natural gas is made up of a variety of substances, including methane (98% of the dry gas is methane), other substances like ethane (C_2H_6), propane (C_3H_8), and butane (C_4H_{10}), as well as carbon dioxide CO_2, hydrogen H_2, and nitrogen N_2.

Carlo *et al.*, (2021) Reported that for practical applications where the need to supplement syngas with natural gas may arise due to a lack of primary sources or power-related constraints, the utilization of mixtures of natural gas and biomass-generated syngas is essential.

Thermal efficiency continued to decline, and the rate of fuel consumption rose. Simply utilising fuel with a higher octane number (ON) than gasoline could address this issue. Gasoline can be replaced with natural gas in this situation. Natural gas usually has an ON between 120 and 130, thus knocking should be avoided. Additionally, natural gas has a high mining capacity akin to "shale gas," and it is very convenient to utilise as a fuel for automobiles **(Wang and Krupnick, 2015)**. This is why a number of researchers have tried using natural gas in SI engines.

The technique developed by introduces a mechanism for blending biogas with municipal gas **(Yamasaki *et al.*, 2013)** A gas engine system that can operate steadily with any mixture of city gas and biogas was created. A spark-ignition gas engine, an additional electronic fuel throttle valve, and a control algorithm make up the gas engine system. The fuel and air ratio might be adjusted by the management algorithm to increase generation efficiency and decrease NO_x emissions.

Faizala *et al.*, (2009) Reported that carbon and hydrogen make up the majority of gasoline's composition, with a few traces of other species. By weight, it fluctuates between 83% and 87% carbon and 11% and 14% hydrogen. This enables the fuel to burn freely and generate a significant amount of heat energy. Numerous combinations of carbon and hydrogen can result in a wide variety of chemical molecules. Leaded and unleaded gasoline are the two varieties. Leaded gasoline is more efficient and has a higher octane rating than unleaded gasoline.

Nippon, (2015) Showed that although the calorific value of biogas is only approximately half that of fossil fuels, it is comparable to that of LPG and natural gas. When applied to adapted engine-generators or modified cooking stoves, biogas can be used in the same way as LPG or natural gas.

IRENA, (2018) Showed that natural gas or bio methane must be liquefied, which is an expensive and energy-consuming process. Bio methane is actually a "drop-in" replacement for fossil natural gas because it shares many characteristics with natural gas. There are no restrictions on its use in an NGV. Vehicles powered by natural gas, both light and heavy duty, can be loaded with biomethane.

Chandra *et al.*, (2011) compared the evaluated brake-specific gas consumption, brake thermal efficiency, and power created for an engine using CNG, methane-enriched biogas (95 percent CH_4, 3 percent CO_2, and 2 percent other gases), and raw biogas (65 percent CH_4, 32 percent CO_2, and 2 percent other gases). According to their findings, methane-enriched biogas can provide 1.43 times as much power as raw biogas under the same operating circumstances.

The engine test results on methane-enriched biogas with a 95% methane content showed that the engine performance is almost identical to that of compressed natural gas in terms of brake power output, specific gas consumption, and thermal efficiency. As a result, the gaseous fuel methane-enriched biogas is comparable to city gas's natural gas mix ratios. **(Chandra *et al.*, 2011).**

Abdel-Galil *et al.*, (2008) Reported that bio methane will directly power natural gas engines. Although biomethane is a cleaner fuel than biogas, it will probably still be more economical to utilise biogas to generate electricity to power pumps rather than converting the pumps to run on biomethane.

IRENA, (2018) Showed that Gas quality and biomethane often need to meet natural gas criteria as well as threshold values for specific trace chemicals found in biogas. In order to surpass the pressure in NGV (natural gas volume) fuel tanks, biomethane must be compressed to at least 200 bars at the refilling station before being injected into a vehicle.

Alternatives to petroleum-derived fuels for internal combustion engines must be developed due to the depletion of fossil fuel resources. Since compressed natural gas (CNG) is far more plentiful than petroleum, it can be a great alternative fuel **(Andrei *et al.*, 2019).** Because of its high H/C ratio and high research octane rating, it produces cleaner exhaust gases than when conventional fuels are burned. Additionally, it has strong anti-knocking properties, although with a slower flame and a smaller flammability range **(Ma *et al.*, 2010).**

Yasin *et al.*, (2016) Reported that natural gas is a significant alternative fuel that can be used as a fuel substitute in internal combustion engines since it meets several of these requirements (ICEs). Natural gas is a very practical alternative fuel since it is readily available, has greater reserves than oil, is less expensive, has better combustion properties, produces fewer car emissions, and is supported by distribution systems.

Stefan, (2004) Showed that because of their impact on engine combustion, biogas' technical characteristics are crucial. These characteristics include:

- CH_4 is potential to ignite when mixed with air:

CH_4 : 5...15 Vol % .

air : 95...85 Vol % .

Stefan, (2004) Found that spark ignition cannot properly ignite mixtures containingless than 5 vol% of CH_4 or more than 15 vol% of CH_4.

-Combustion velocity in a mixture with air at p = 1 bar:

cc = 0.20 m/s at 7% CH_4

cc = 0.38 m/s at 10% CH_4

The volume proportion of the component that can burn, in this case CH_4, determines the combustion velocity. The largest value of cc is seen at surplus air ratios between 0,8 and 0,9, which is close to the stoichiometric air/fuel ratio. At higher temperatures and pressures, it sharply rises **(Stefan, 2004)**

Natural gas is ideal for city transportation since it produces fewer pollutants and soot particles. Additionally, using natural gas in spark ignition (SI) engines allows for normal compression ratios between 10 and 90 bar in the cylinder, approximately tripling spark plug gap tension at full load in a supercharged engine. It is preferable to guarantee ignition by using a modest amount of diesel pilot injection to solve this issue. 12 Additionally, efficiency declines as a result of the partial load use of natural gas in SI engines, the presence of a throttle in Otto engines, and low compression ratios **(Umierski and Stommel, 2000)**

2.7. Calorific value of fuel:

Faizala *et al*, (2009) Found mentioned that the components of fuels are in Table (1).

Table (1): Comparison of combustion types of fuels

			Emissions (at 3000 rpm)		
Fuel	**LHV(KJ/Kg)**	**RON**	**CO (%)**	**NO_X(ppm)**	**UHC(ppm)**
Gasoline	43.000	95	0.41	364	47
Diesel	42.500	-	0.02	800	15
Natural gas	49.770	120	0.12	194	29

Research Octane Number (RON)

Biogas calorific value: The efficiency of the appliances affects the net calorific value of the biogas. The valuable element in the context of using biogas as a fuel is methane. As a result, the calorific value of biogas may be determined using the findings from the biogas study **(Mitzlaff, 1988)**

Stefan, (2004) Reported that the real calorific value of biogas depends on three variables that vary from situation to situation: the temperature, the absolute pressure, and the CH_4 %. The following equation can be used to determine the true calorific value of biogas, a crucial factor in an engine's performance:

$$H_{u,act} = \frac{V_{CH_4}}{V_{tot}} \cdot \rho_{CH4,act} \cdot H_{u,n} \; . \quad \ldots\ldots\ldots\ldots(2)$$

2.8. Exhaust gas emissions:

With CNG operation, the volumetric efficiency is decreased by 4–10%, and the HC, CO, and CO_2 exhaust emissions are greatly reduced as compared to gasoline by 40–66%, 5–87%, and 28–30%, respectively **(Saravanan *et al.*, 2013)**

Jeongwoo *et al.*, (2020) CNG combustion produced better results than gasoline combustion did. The low CO_2 emissions from CNG combustion are due to a number of factors. Under the same conditions of low heating value, CNG has less carbon in the fuel than gasoline.

That a rise in oxygen levels in the combustion chamber causes the formation of HC emissions from combustion products. More hydrogen and oxygen reactions occur when oxygen levels are higher than hydrogen interactions with carbon **(Li *et. al.*, 2016).**

Abbas *et al.*, (2017) Showed that when compared to gasoline, CNG produced exhaust emissions that were 30–91% lower than those from gasoline. CO is a byproduct of incomplete combustion, which occurs when an engine is operated with a rich mixture or when correct air–fuel mixing is not achieved. Since CNG (mostly CH_4) has a higher hydrogen to carbon ratio than gasoline and has a simpler chemical composition, it should emit less CO than gasoline.

Wayan *et al.*, (2014) Reported that natural gas reduces CO_2 emissions by 20–30% when utilised as a car fuel. When manure-derived biogas is used as a car fuel, a reduction above 100% can be attained. Thus, using biogas as a vehicle fuel can reduce both the emissions of fossil carbon dioxide and the leaking of methane from manure. Another benefit is that automobiles powered by enhanced biogas or natural gas emit fewer particulates, NO_x, and SO_x into the atmosphere **(Papacz, 2011).**

Saravanan *et al.*, (2013) Showed that because CNG has an 8-fold lower carbon content than gasoline, fewer carbon-based emissions are produced. Complete combustion also occurs in the gaseous fuel, lowering the amount of HC emissions. In comparison to liquid fuels, gaseous fuels form more homogeneous mixtures. Dedicated CNG engines with a greater compression ratio can reduce HC even more.

That a rise in oxygen levels in the combustion chamber causes the formation of HC emissions from combustion products. More hydrogen and oxygen reactions occur when oxygen levels are higher than hydrogen interactions with carbon **(Li *et al.*, 2016)**.

IRENA, (2018) Found that when used as a substitute for gasoline, natural gas emits 124g CO_2-eq./km, which is a 24% decrease. Bio methane has a huge impact on the outcomes. The feedstock type affects the GHG emissions of biomethane consumption. Compared to gasoline, GHG emissions can be decreased by about 60% with bio methane made from crops like maize.

Natural gas cars can run on biogas as fuel. The advantage is that fossil fuels may be replaced, including gasoline and diesel. With biogas, greenhouse gas emissions can be reduced by as much as 100%. Natural gas reduces CO_2 emissions by 20–30% when utilised as a car fuel. When manure-derived biogas is used as a car fuel, a decrease of more than 100% can be attained **(Wayan, *et al.*, 2014).**

By employing natural gas as fuel, **(Zarante and Sodre, 2009)** examined carbon emissions, According to the findings, utilising natural gas instead of gasoline decreased CO and CO_2 levels. no burn Hydrocarbons (HC) and carbon dioxide (CO), as well as the proportion of nitrogen oxides (NO), dropped as the hydrogen fraction increased.

Saad *et al.*, (2014) demonstrates that carbon dioxide (CO) and carbon monoxide (CO) emissions from CNG are less than those from gasoline. CNG emits less nitric oxide (NO) and mono-nitrogen oxides (NO) than gasoline.

The comparison of the HC exhaust emissions for the two fuels reveals that CNG consistently emits fewer unburned hydrocarbons than gasoline across the entire speed range. Due to a more complete combustion of CNG than gasoline, the emission of HC is greatly reduced by 25-72% when using CNG **(Abbas *et al.*, 2017)**

Another experiment involved retrofitting a 1.5 L, 4-cylinder Proton Magma. The results showed that CNG has higher FCE and lower CO, CO_2, and HC emissions than gasoline, but higher NO emissions **(Kalam *et al.*, 2005).**

Using biogas as a vehicle fuel can reduce both the emissions of fossil carbon dioxide and the leaking of methane from manure. Another benefit is that automobiles powered by enhanced biogas or natural gas emit fewer particulates, NO_x, and SO_x into the atmosphere **(Papacz, 2011).**

The injection time, natural gas mix, and beginning temperature have the biggest impacts on emission. Increasing ethane could accelerate ignition and increase NO, while delaying fuel injection timing could decrease NO **(Zheng *et al.*, 2005).**

According to study findings in general, hydrogen enrichment at a volume of 20% yields the best benefits. Simulated analysis of NO_x emissions from HCNG engines shows that the equivalency ratio and compression ratio have a significant impact on NO_x generation **(Pranav *et. al.*, 2017).**

Carbon monoxide emissions were shown to diminish with rising excess air ratios and, once they exceeded a certain point, often settled to extremely low levels. The CO emissions decrease to these insignificant levels at higher engine speeds and lower surplus air ratio values. Another pattern that was seen was a decrease in CO emissions as the fuel's H_2 fraction rose. However, it should be mentioned that the study by **(Zhunqing and Zhang, 2011)** found that CO emissions rise as the hydrogen fraction rises. As the surplus air ratio rises, carbon dioxide emissions rise as well, reaching excess air ratio levels of roughly 1.05 to 1.11. Ratios greater than this result in lower CO_2 emissions. CO_2 emissions fall as the H_2 proportion rises.

Robert, (2014) Showed that because a higher combustion temperature promotes greater oxidation, it was demonstrated that hydrocarbon emissions reduced as CO_2 concentrations rose. For the same reason, an inverse link between NO emissions and CO_2 levels was discovered. CO and CO_2 emissions behaved similarly at ratios close to stoichiometric equivalence.

When engine-out specific emissions from gasoline tests were compared to post-catalyst specific emissions from biogas tests, it was found that NO_x emissions could be reduced to levels that were significantly lower than those achieved with gasoline and that THC emissions could also be reduced to levels that were closer to those of engine-out gasoline emissions. With the catalyst, CO emissions, which were already much lower with NG than with gasoline, were further decreased **(Robert, 2014).**

Hannu, (2020) Reported that stoichiometry and natural gas engines burning a "stoichiometric" air-fuel mixture are often calibrated to operate slightly rich. Stoichiometry to minimise NO_x and oxidise CO and hydrocarbons in the exhaust. The "rich-burn" engines for natural gas are those that use a nearly stoichiometric mixture.

Abo-Qudais and Qdais, (2005) Showed that the fuel composition varies according to crude source, used refining technique, and type and amount of additives. This variation in fuel composition will affect the emitted pollutants, Table (2) shows comparison of different emissions types emitted from both engines.

Table (2): Typical exhaust emissions from gasoline and diesel engines

missions	Gasoline engine (%)	Diesel engine (%)
CO	0.8 – 5.2%	0.1–1.6%
CO_2	9.0 – 12.5%	8.0–11.0%
HC	0.03 -0.04%	0.002–0.004%
O_2	3 – 5%	7–10%

2.9. Engine efficiency:

Saad *et al,* (2014) Reported that because natural gas has a significantly higher octane number than other clean fuels, it has recently been considered as an alternative for Spark Ignition (SI) engines. In comparison to gasoline, lean natural gas combustion in SI engines offers the potential to increase thermal efficiency and lower emissions. Natural gas is primarily composed of methane. Natural gas may burn at a higher compression ratio without banging because of its high Research Octane Number (RON), which is more than 120. Due to its higher hydrogen to carbon ratio compared to other hydrocarbon fuels, it also offers significantly reduced CO_2 emissions.

As the ULP is a liquid fuel, it enters the combustion chamber as vapour, increasing the density of the intake mixture and, as a result, the volumetric efficiency. It also serves as an engine coolant. In contrast to gaseous fuels, such as LPG, which are vapour at room temperature and have no cooling impact, a larger volume of fuel in the input mixture can also reduce volumetric efficiency **(Ali and Amir, 2009).**

Bade and Narayanan, (2008) Showed that examined the impact of hydrogen addition on combustion using a test fuel combination of 53% CH_4, 42% CO_2, and 5% N_2 to simulate landfill gas. When hydrogen was added to landfill gas, it was discovered that H_2 significantly increased the peak cylinder pressure under all operating conditions, significantly decreased the coefficient of variation of peak cylinder pressure, and significantly decreased ignition lag and combustion duration (but only up to an H2/(Landfill gas (LFG)+H_2) ratio of 5%, after which it decreased).

Researchers looked at how adding hydrogen and using an exhaust gas recirculation (EGR) system affected the efficiency of the power plant and the pollutants were studied by **(Lee *et al.,* 2010)** In an engine powered by a biogas-hydrogen blend, their research looked at the connections between generating efficiency, spark timing, (EGR) rate, and emissions (NO_x and CO_2).

(Andreass , *et al.*, 2001) Report that the amount of inlet air mass to the engine can also be increased if we can raise the amount of air pressure entering the engine. To compensate for CNG engines seems to be important since the mass density of air entering the engine will cause the combustion to be complete, improving thermal efficiency.

For the same amount of power, the brake thermal efficiency was significantly higher when hydrogen was introduced (31% vs. 27.5% at 24 kW), and the advantage hydrogen offered in terms of brake thermal efficiency grew with higher engine speeds and larger excess air ratios. This was attributed to the fact that H_2 addition allows for greater flame speeds and hence shorter combustion duration as well as less cyclic variability, both of which are problems with lean-burn engines powered by natural gas (NG) **(Marie, 2007).**

The combustion cycle is the most significant cycle and difficult process in the four-stroke SI engine. Because the chemical energy of the fuel is turned into thermal energy during this cycle, it has a significant impact on engine performance and pollutant emissions. In the combustion cycle, the combustion duration is a crucial factor that determines the best burning process. The chemical energy of the fuel cannot be completely transformed into thermal energy if combustion time is too short, and conversely, if combustion time is too long, more thermal energy is wasted since it takes longer for heat to move to the cylinder and piston **(Nguyen *et al.*, 2019).**

Nafiz *et al.*, (2009) Discovered that higher hydrogen concentration in the gasoline increased the thermal efficiency of the brakes. Additionally, as excess air ratio rises, ηth rises as well, however the effect varies depending on engine speed, which also has a proportional impact on ηth.

Park *et al.*, (2011) Sought to more clearly show how the addition of H_2 affects a biogas engine's thermal efficiency. The researchers chose a mixture of only natural gas and N_2 to imitate biogas since they found that the general tendencies of N_2 dilution are comparable to those of CO_2 dilution. The thermal efficiency at each composition was computed using several gas compositions, ranging from 0% to 80% N_2 as fuel and at an equivalency ratio of unity. At 80% N_2, the maximum thermal efficiency that was seen was 31.1%.

In contrast to the proportional relationship that occurs between the concentration of H_2 and thermal efficiency in the 0% to 5% range, cooling losses became so substantial at

H_2 concentrations larger than 5% that thermal efficiency declined. Researchers also looked at the impact of adding hydrogen under lean running conditions, and they discovered that 10% hydrogen addition and 30% extra air produced the test's highest thermal efficiency. **(Park *et al.*, 2011).**

It has been discovered that both natural gas and biogas have a strong resistance to knocking. Because they have higher octane ratings than gasoline natural gas at 120 and biogas, which is composed of 65% CH_4 and 35% CO_2 at 136 **(Zhang, 1998)** engines may run at higher compression ratios for greater efficiency.

Effects on the thermal efficiency were also examined, and the greatest brake thermal efficiency for the methane-enriched biogas was 26.2% at a brake load of 59%, compared to 23.3% for the raw biogas at a brake load of 53.5% **(Chandra *et al.*, 2011).**

The two primary factors utilised to determine the load control technique are the engine efficiency and the nitrogen oxide (NO_X) emission. Throttled operation with a constant equivalency ratio has been utilised, although mostly for demonstrations **(Tang *et al.*, 2002).**

The authors also state that employing the equivalency ratio to manage load (WOT, qualitative control approach) rather than throttling results in a relative increase of 15 to 20% in brake thermal efficiency at the lower loads (quantitative control strategy). Last but not least, Ford reports that a single cylinder DI engine it tested produced an estimated peak brake thermal efficiency of 45% in a joint publication with Westport Innovations and Pacific Northwest National Laboratory **(Welch *et al.*, 2008).**

Berckmüller *et al.*, (2003) Included the wide open throttle strategy, throttled stoichiometric, and supercharged stoichiometric strategies in its indicated thermal efficiency map for a port fuel injected engine. At low load, indicated thermal efficiency were 40%, while at high load, they were 32%.

Verhelst *et al.*, (2009) Showed that the efficiency advantages of hydrogen compared to gasoline are particularly substantial at low loads, which are crucial in drive cycles as well as for daily driving (up to 60% relative increase in the BTE). If the BTE on hydrogen was compromised when a gasoline engine was modified to run on hydrogen, one might anticipate that the changes in BTE for a dedicated H2ICE will be significantly greater.

The liquid fuels (kerosene and gasoline) and biogas were contrasted. According to the findings, at 2500 rpm engine speed, specific energy consumption while operating in the biogas mode was reduced on average by 80.3% and 79% when compared to operating in the gasoline and kerosene modes, respectively. About 58% of the total pumps used in biogas operations were efficient **(Abdel-Galil *et al.*, 2008).**

Robert, (2014) Reported that the ratio of CH_4 to CO_2 in the gas used to approximate biogas was 60% to 40%. Because their research indicated that this was the most potent composition for raising the producing efficiency, 5% of the fuel's total heating value (or roughly 9% of the total gas flow volume) was added for the experiments with the hydrogen additive.

2.10. The costs and economic indicators:

Through the implementation of a programme for water rationalisation in many sectors, especially agriculture, and the adoption of policies that encourage water cost recovery, Egypt's Vision 2030 **(MPMAR, 2016)** aims to sustainable consumption patterns of water and natural resources.

Awulachew *et. al.*, (2009) Showed that in reality, this means that diesel engines are more durable and have more precise fuel injection parts than gasoline engines of similar power. They are also roughly three times heavier. A diesel engine costs more to purchase than a comparable petrol engine, however although though petrol engines run for fewer hours each day, their operational lives (measured in years) will typically be longer than diesel engines. Additionally, a diesel engine is better suited for continuous, long-term operation. Small petrol engines, on the other hand, are made to run for just a few hours (up to five) per day. Diesel engines often require less routine maintenance than gasoline ones. However, issues with injectors or injection are present when a diesel engine suffers a catastrophic failure.

Awulachew *et. al.*, (2009) Showed that when it comes to pumping technology, extension officers' key responsibility is assisting farmers in making a decision that is well-suited to their requirements and limitations. Recommended selection standards include:

- Best match between engine power and necessary discharge and head
- Low purchase cost
- Long working life
- High efficiency of human or fuel energy
- Low operating costs

The cost of administering, running, and maintaining the irrigation system puts further strain on the federal budget. Around LE 1.39 billion of the Egyptian government's budget was, on average, allotted to irrigation and drainage infrastructure investments between 2012 and 2017 **(MPMAR, 2017)**. Another difficulty is raising enough money to run and maintain the irrigation system effectively.

Fragoso and Marques, (2013) Showed that both fixed and variable costs are included in the price of delivering irrigation water. The first is known as capital operations and maintenance (O&M) and consists of taxes, insurance, interest on investments (the opportunity interest cost of investing in a water lifting system is the rate of return capital would earn in its next best alternative), permanent labor (such as pump guard), costs of wear out depreciation, some fixed O&M for administration and rehabilitation, and typically includes the costs of energy for pumping and pressurization (diesel or electricity), grease, oil, repairs, and tem (e.g. pump operator). While the fixed cost is independent of the quantity of water delivered, the variable cost is. Most nations, including Egypt, greatly subsidies the fixed cost **(Tsur and Dinar, 1995).**

Ban *et.al.*, (2011) The economic evaluation process is the method that enables planners to verify the economic feasibility of projects, and we have no other choice than profit as a tool that can show the extent of the economic efficiency of the performance and development of industrial facilities, and thus their effective contribution to the economic and the profit or loss achieved by any facility, the criterion of the return of the pound from energy costs means the total return of one pound of the cost of energy used in the irrigation process, and this criterion is used when comparing the returns on investment costs from the items of partial costs of the irrigation process, and it is calculated by dividing The total value of the production return in pounds over the total value of the energy costs used in irrigation in pounds.

Mccauley *et al.*, (2002) Claimed that because farmers do not pay for irrigation water as a limited natural resource, the anticipated price of irrigation water under Egypt's current water policy would represent the shadow price of water rather than a market price. The incremental crop yield produced by the most recent cubic metre of irrigation water added multiplied by the crop price is hence the marginal return to irrigation water (as a shadow demand price of water for agriculture). The shadow price of irrigation water of the survey (2016/2017) for wheat, clover, sugar beet, and rice, respectively, reached roughly LE 1.32, 1.67, 1.47, and 1.84 for one cubic metre as a result, and the residual approach was used to estimate it.

Fragoso and Marques, (2013) Showed that the choice depends primarily on the implementation costs, which vary from region to region due to climate issues, demography, social structure, water rights, water facilities, history, and economic conditions. Each of these water cost recovery schemes is associated with varying levels of welfare and net benefits. The cost recovery strategy that yields the greatest benefit should be chosen. The most effective strategies are volumetric ones where there are no implementation costs.

Awulachew *et al.*, (2009) Reported that since there are moving parts, water, and pipes, there are energy losses that result from friction and are often lost as heat energy. As gasoline is burned to produce power, an engine heats up. Pumping systems can experience significant energy losses, which can be expensive in terms of fuel use.

Between 50 and 75 USD dollars per m^3 of capacity are spent on the total cost of a biogas plant, which includes all necessary facilities but excludes land. Average Cost of a Biogas Plant The digester accounts for 35 – 40% of the total costs. As volume increases, the cost per cubic meter of digester volume drops. **(Pruthviraj, 2016).**

IRENA, (2018) Reported that For facilities with a size greater than 500 m^3/hour (h) of raw biogas using residues and wastes as feedstock, the total supply cost for biogas as car fuel (including distribution cost) ranges between USD 0.22/m^3 and USD 0.88/m^3 methane. Costs for small-scale plants producing 100 m^3/h of raw biogas range from $1.00 to $1.55 per m^3 of methane. It is very difficult to compete with natural gas costs that are so low (USD 0.13/m^3 in 2016) **(NYMEX, 2018)**.

Net present value (NPV) used to evaluate a project's or investment's profitability. It is determined by subtracting the difference between the current value of cash inflows and cash outflows throughout the course of the project (from the project over its life) **(Willis *et al.*, 2018)**.

Benefit cost ratio(B/C) used to evaluate a project's or investment's profitability. It is computed by dividing the project's lifetime's present value of cash inflows by its lifetime's present value of cash outflows **(Willis *et al.*, 2018)**. The following equations were used to calculate (B/C, NPV):

B/C = Total Present Benefits / Total Present Costs

N.P.V = Total Present Benefits - Total Present Costs

$$\mathbf{P.V. = F.V. \times (1/ (1+r)\ n\ )} \quad \text{………….(3)}$$

P.V. =Present Value

F.V. = Future Value

r = Discount rate

n = Number of Years

(1/ (1+r) n) = The Present value coefficient.

Internal rate of return (IRR) a rate of interest that makes the net present value (NPV) of all cash flows from a specific project equal to zero and is used to examine the profitability of a project or investment, is the interest rate that matches the present value of benefits to the present value of costs. The same NPV method is used to calculate the internal rate of return**(Willis *et al.*, 2018).**

Payback period is the amount of time required for the project to pay back the capital investment cost and generate a net profit (benefits). The payback period, or the length of time it will take for the total cash flows to equal the investment expenses **(Willis *et al.*, 2018)**, was calculated using the following equations:

Payback period = investment costs / annual net return (profit)

2.11. Biogas processing:

The composition of the substrate from which the gas was created affects the concentrations of the contaminants (methane, carbon dioxide, water, hydrogen sulphide, nitrogen, oxygen, ammonia, siloxanes, and particles). However, it may be useful to clean the gas before the upgrading in order to prevent corrosion and mechanical wear of the upgrading equipment itself **(Papacz, 2011).**

Stefan, (2004) Reported that between 50% and 70% of biogas is composed of CH_4, 2% of H_2, and up to 30% of CO_2. This gas becomes a reasonably homogeneous fuel after being cleaned of carbon dioxide, containing up to 80% methane and having a calorific value of more than 25 MJ/m^3. Methane, or CH_4, is the most significant component of biogas in terms of calorific value.

By precipitating in the digester liquid or treating the gas in a freestanding vessel or while extracting carbon dioxide, hydrogen sulphide concentrations in biogas can be reduced. When Fe^{2+} or Fe^{3+} ions are added to the digester in the form of $FeCl_2$, $FeCl_3$, or $FeSO_4$, the virtually insoluble iron sulphide is precipitated and removed along with the digestate **(Papacz, 2011).**

Desulfurizing the biogas as the first stage, followed by dehumidification, and putting the gas in the bag of gas holder can be used to convert an engine from a gasoline-fueled one to one that runs on biogas. The biogas should then be compressed into a gas container for simple mixing with airborne oxygen **(Wayan *et al.*, 2014).**

Since CH_4 and hydrogen are the combustible gases, a larger CH_4 content is preferable since it gives the gas more energy, whereas carbon dioxide and nitrogen are inert and have no energy. Only trace amounts of harmful substances like hydrogen sulphide and ammonia, which must be treated to be removed, are present **(Cheolwoong *et al.*, 2011).**

Desulfurizers can improve biogas to have zero H_2S contaminants, which prevents the lubricant from becoming more acidic and, as a result, corrosion in the combustion chamber **(Nindhia, 2012).** Prior until now, H_2S in biogas was eliminated by more frequent engine oil changes, which would raise the cost of operation **(Huanga and Crookesb, 1998).**

Since CH_4 and CO_2 make up 98% or 99% of all biogas, it is possible to estimate the CH_4 percentage in biogas by simply extracting the CO_2, which occurs when biogas is passed through an alkaline solution ($Ca(OH)_2$) 1N. CO_2 reacts with $Ca(OH)_2$ to form $CaCO_3$, and since the volume of the biogas before passing through the alkaline solution (Vb) is smaller than the final volume (Ve), the percentage of CO_2 can be calculated **(House, 1981):**

$$\%CO_2 = \left(\frac{V_b - V_e}{V_b}\right) \times 100 \qquad \text{............(4)}$$

And CH_4 could be approximated by using (% CH_4= 100 – CO_2 %)

2.12. water pumps:

Natural slopes, raising to a higher location, pumps, and pressured pipelines can all be used to move water. Devices for lifting water include very effective pumps that are powered by electric, gasoline, or diesel motors as well as traditional, native water lifts **(Garg, 1989; Michael, 1990**) A pump is a machine that lifts, moves, or compresses gases and liquids. Pumps can be divided into four general categories: reciprocating, centrifugal, jet, and other pumps.

Awulachew *et al.*, (2009) Showed that positive displacement pumps and variable displacement pumps are the two main categories for engine-powered pumps. The positive displacement pumps are once more separated into rotary and reciprocating pumps. Lift or force pumps are two different types of reciprocating pumps. Pumps can be single-acting or double-acting for both lift and force.

Awulachew *et. al.*, (2009) Reported that the rotating impeller, often referred to as a blade, of centrifugal pumps, also referred to as rotary pumps, is submerged in the liquid.

Near the axis of the impeller, liquid enters the pump. At high pressure, the revolving impeller sweeps the liquid out toward the ends of the impeller blades. The diffuser, a stationary component of the pump, can convert the liquid's relatively high velocity into pressure because of the impeller. A number of impellers may be used in sequence in high-pressure pumps, and the diffusers after each impeller may have guiding vanes to gradually slow the flow of liquid. The diffuser for lower-pressure pumps is often a spiral tunnel known as a volute, with its cross-sectional area gradually increasing to effectively lower the velocity. Before the impeller can start working, it must be primed, which means that when the pump is turned on, liquid must surround the impeller. This can be accomplished by installing a check valve in the suction line, which keeps the liquid within the pump even while the impeller is not turning. If this valve leaks, it could be necessary to prime the pump by adding liquid from an external source, like the discharge reservoir. The discharge line of a centrifugal pump typically incorporates a valve to regulate flow and pressure.

Awulachew *et al.*, (2009) showed that energy is changed three times before it is used by water in a conventional petrol-powered pumping system. The engine burns the chemical energy in the fuel to create mechanical energy. In the case of centrifugal pumps, this is transferred to the pump via a driving shaft and then to the water via an impeller. Figure (2).

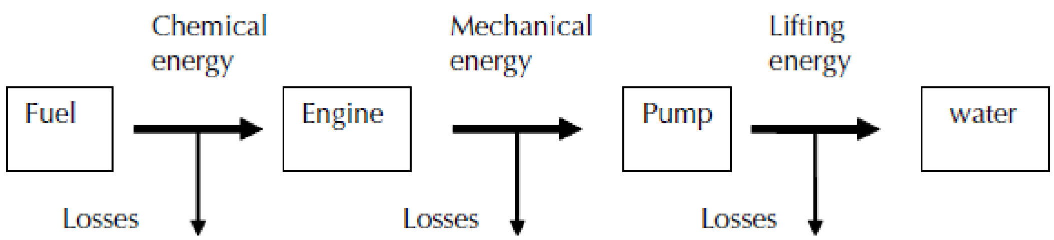


Figure (2): Energy conversion and losses in a pumping system.

Awulachew *et. al.*, (2009) Reported that power and energy are frequently confused. Although they are linked, their meanings are different. The ability to lift water is energy. Power is the rate at which energy is used and is typically expressed in Watts (W) or kilowatts (kW), where 1 kW equals 1000 W. Horsepower (hp) is a different way to express power (1 kW = 1.36 hp). Power is determined by:

$$P = \frac{E}{t} \quad \text{.............(5)}$$

Where (P) stands for power in Watts, (E) for energy in Watt Hours, and (t) for time in Hours. Water flow volume divided by time equals discharge. Equation is created using this relationship:

$$P = 9.81\ Q\ H \quad \ldots\ldots\ldots\ldots(6)$$

Where (Q) is discharge in liters per second (liters/sec).

The majority of engine-powered pumps used for small-scale irrigation are centrifugal pumps. They are incredibly simple to maintain and reasonably inexpensive. The impeller of the centrifugal pump has blades and spins rapidly inside the pump casing. The diameter (in mm) of the delivery connection pipe, to which the hose is connected, is used to characterise centrifugal pumps **(Awulachew *et al.*, 2009)**

Awulachew *et al.*, (2009) It was demonstrated that the characteristic curve for pumps provided by their makers assumes that the pump is operated at its ideal design speed. The optimal design speed for centrifugal pumps is typically three-quarters of the maximum throttle on diesel or gasoline engines running them. Because it's challenging for them to control flow in the field, farmers frequently run their pumps at extremely low pressures to limit the discharge. In this situation, inefficiency (fuel waste) gets significantly worse because the pump will use more fuel despite pumping less water. When operating at its ideal design speed, a pump performs at its peak level.

2.13. Combustion equations:

khurmi and Gupta, (1998) Found that there are many chemical equations that represent the chemical combustion of hydrocarbons in oxygen as follows:

a- Combustion of carbon:

1- When carbon burns in sufficient quantity of oxygen :

$C + O_2 \rightarrow CO_2 + \text{heat}$

$12kg + 32kg \rightarrow 44kg + 410000\ KJ$

$1kg + 8/3\ kg \rightarrow 11/3\ kg + 102500/3\ KJ$

2- When carbon burns in insufficient quantity of oxygen :

$C + 1/2\ O_2 \rightarrow CO + \text{heat}$

$12\ kg + 32/2\ kg \rightarrow 28\ kg + 124500\ KJ$

$1\ kg + 4/3\ kg \rightarrow 7/3\ kg + 10375\ KJ$

3- If carbon monoxide is burnt further it converted into carbon dioxide :

$CO + 1/2\ O_2 \rightarrow CO_2 + \text{heat}$

$28\ kg + 32/2\ kg \rightarrow 44\ kg + 285000\ KJ$

i.e. $1kg + 4/7\ kg \rightarrow 11/7\ kg + 71250/7\ KJ$

b- Combustion of hydrogen:

$H_2 + 1/2\ O_2 \rightarrow H_2O + \text{heat}$

2kg + 32/2 kg → 18 kg + 242000 KJ

i.e. 1kg + 8 kg → 9 kg + 121000 KJ

c- Combustion of methane:

$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$

$1m^3 + 2m^3 \rightarrow 1m^3 + 2m^3$ (by volume)

(12+4)kg + (2×32)kg → 44kg + (2×18)kg (by mass)

i.e. 1kg + 4kg → 11/4 kg + 9/4 kg

El-Ashry (2003) and El-Emam (2004) Reported that atom balances on C, H, and N atoms result in the general formula shown below when a fuel composed of carbon, hydrogen, and oxygen is burned completely with a stoichiometric amount of air:

$$\mathbf{C_\alpha H_\beta O_\gamma + (\alpha + \beta/4 - \gamma/2)(O_2 + 3.76\ N_2) \rightarrow \alpha\ CO_2 + (\beta/2)\ H_2O}$$
$$\mathbf{+\ 3.76\ (\alpha + \beta/4 - \gamma/2)\ N_2 \ldots\ldots\ldots\ldots\ldots (7)}$$

where:

α, β and γ : the number of carbon, hydrogen and oxygen atoms in molecule of fuel.

3.76 = the nitrogen to oxygen ratio in air.

2.14. Mixing chambers:

Klaus, (1988) shows that when the distance between the mixer and the inlet manifold is small, mixing chambers with a larger volume than just a T-Joint tube provide longer air and fuel retention time within the chamber and a more homogeneous mixture becomes necessary. As a result, the mixing time is sufficient, as shown in the figure (3).

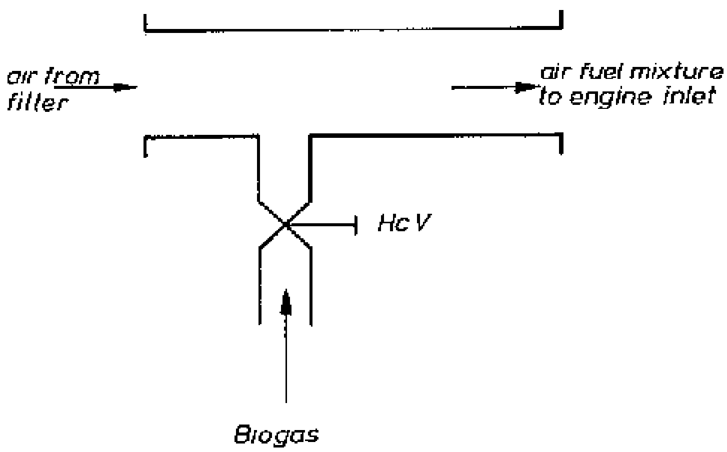


Figure (3): T-joint mixer

Klaus, (1988) Showed that demonstrated that the location where the gas tube (the nozzle) is cut diagonally (30–45°) with the orifice facing the inlet of the engine is where the T with the gas tube extending into the mixing device is connected. The cross-sectional area of the air flow is decreased and slightly decreased by a slightly projecting gas tube, increasing the active pressure drop of the gas flow to the mixing mechanism. Since the pressure drop increases with engine speed and suction, more gas is also absorbed. The operation resembles a Venturi plane in certain ways. Figure illustrates how the mixing performance is better than a blunt T-joint (4).

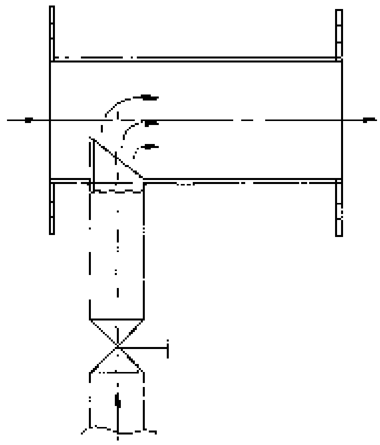


Figure (4): T-joint mixer with oblique, protruding gas inlet

Mixing chambers with larger volumes the comparatively low flow velocities provide more time for mixing. But if the gas pipe extends into the chamber and disperses the gas through a number of holes, as indicated in the pictures **(Klaus, 1988)**, it is good for the mixing (5 , 6).

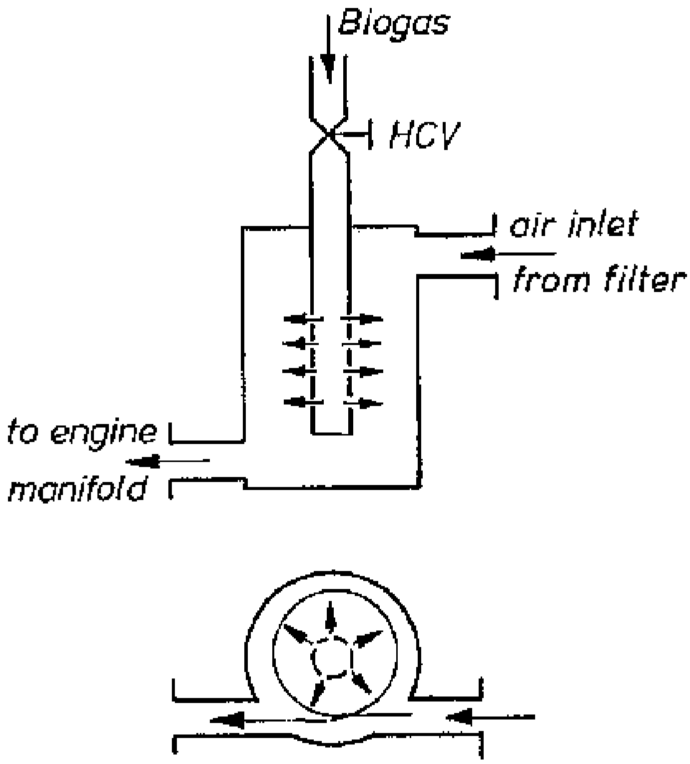


Figure (5): Simple mixing chamber with hand - controlled valve (HCV)

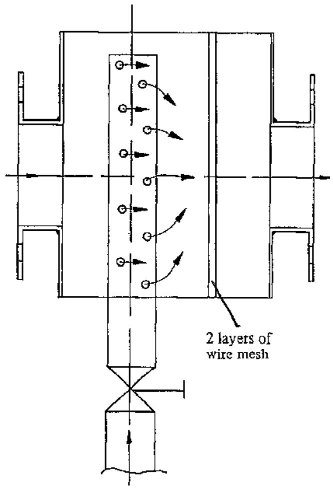


Figure (6): Mixing chamber with gas distribution pipe and wire mesh for intensive mixing

Klaus, (1988) demonstrated that a Venturi mixer uses the same fluid mechanical effect as a typical carburetor, where a change in air flow quantity and velocity results in a change in pressure as the duct contracts, which in turn results in a change in the flow of a different medium (fuel) to join and mix with the flow. The principal air in the necessary ratios, as depicted in the image (7). The following is how the venturi principle works:

-Air velocity is high.

- At the cross-section that has constricted, air pressure is low.

-There is a significant pressure difference between the airstream and fuel gas.

- A lot of fuel gas passes through the holes and combines with the airflow .

For low air volume flow :

- Low air velocity.

-The cross-section that has contracted has a high air pressure.

-There is little pressure difference between the airstream and fuel gas.

- Only a small amount of fuel gas enters the airstream through the apertures.

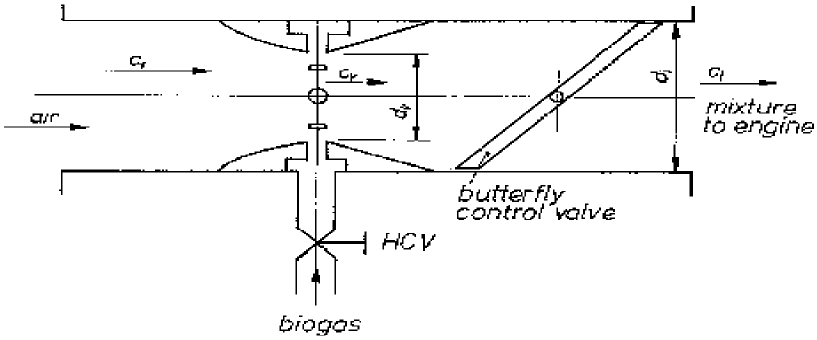


Figure (7): Venturi mixer with gas supply through several bores.

MATERIALS AND METHODS

3.1 Materials:

This study presents the experimental for the types of mixers used in mixing natural gas with air to operate the irrigation pump to save energy, many of measurements were carried 2021 year in workshops of Agricultural Engineering Department, Faculty of Agriculture, Ain Shams University, Egypt.

3.1.1 Engine:

A new single-cylinder, air-cooled, engine was used, which was purchased from the local market, as shown in Figure (8), made in Japan, and the engine specifications are shown in Table (3).

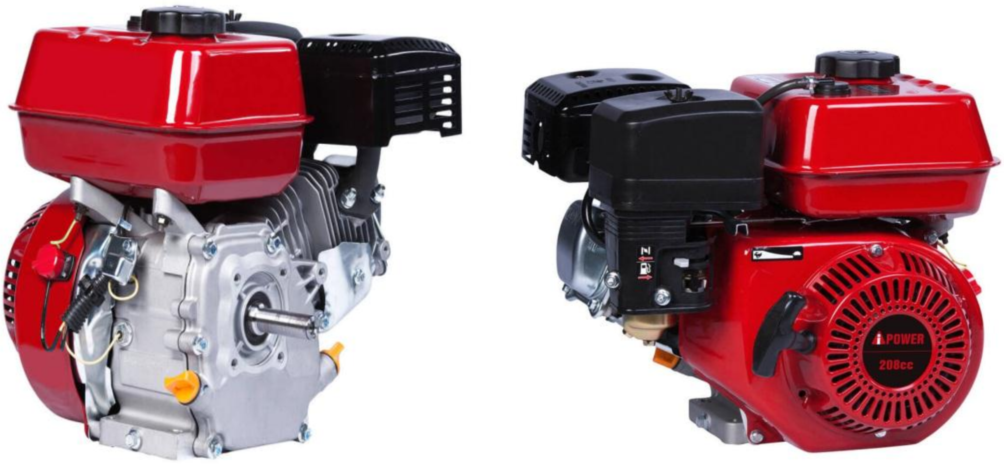


Figure (8): Single cylinder engine

Table (3): Engine Specifications

Model	*AP17OF*
Engine Type	*Air-cooled 4-stroke OHV Single Cylinder*
Bore x Stroke	*70 x 54*
Displacement	*208cc*
Compression Ratio	*8.5:1*
Rated Output (Kw at 3600 rpm)	*4.4 kw at 3600 rpm*
Output Shaft	*3/4 in. dia. x 2-7/16 in. L*
Carburetor	*Butterfly*
Fuel Consumption	*≤ 374g/kw/h*
Fuel Tank Capacity	*0.95 gal. (3.6L)*

3.1.2 The pump:

A new pump with a discharge diameter of 2 inches was used, which was an Egyptian manufacture, as indicated in the pump specifications in Table (4).

Table (4): Specifications of the water pump

MODEL	SE-50X
TYPE	: SE-50X-BDM-0
CONNECTION DIA	: 50 mm 2 inch
DELIVERY VOLUME	: 560 l/min 147 gal/min
TOTAL HEAD	: 30 m 98 ft

3.1.3 The gas cylinder and pressure regulator:

Two types of fuel were used to run the engine, namely (gasoline - natural gas), where a high-octane type (92) gasoline was used. As for natural gas, the gas cylinder used in cars was purchased with a volume of (70L) and was filled from natural gas filling stations for cars. As shown in Figure (9). A gas pressure regulator was purchased in order to determine the pressure of the gas leaving the cylinder and reduce it to suit the operation of the engine. The regulator measured high pressure (300 bar) and low pressure (16 bar).

Figure (9): Natural gas cylinder

3.1.4 Gas mixer:

Several types of mixers were manufactured to mix natural gas with air before entering the engine. They were manufactured in the workshop of the Agricultural Engineering Department / Faculty of Agriculture, Ain Shams University using iron pipes of different diameters (1 cm, 2.54 cm, 3.81 cm) as shown in Figure (10, a, b, c) The types were as follows:

Figure (10): Types of gas mixers

- **Mixer T-angle 90**

It is an iron tube with a diameter of (2.54 cm), length (20 cm) and thickness (0.27 cm), where a pipe of diameter (1 cm) and thickness (0.27 cm) was welded in the middle perpendicular to it at an angle of (90) for the entry of natural gas from the cylinder and the process of mixing air with gas takes place As shown in Figure (11), this mixer is fixed to the motor using screws.

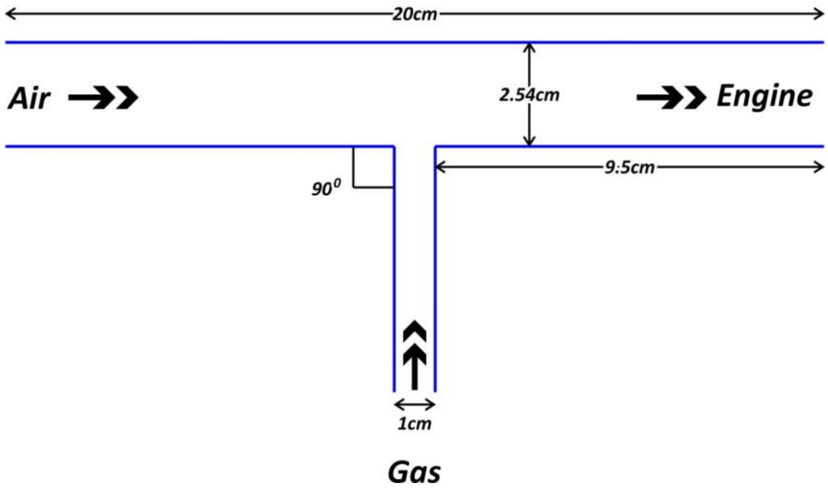


Figure (11): T-type mixer 90

- **Mixer T-angle 45**

It is an iron pipe with a diameter of (2.54 cm) and a length of (20 cm) where a pipe with a diameter of (1 cm) was welded in the middle perpendicular to it at an angle of (45) for the entry of natural gas from the cylinder and the process of mixing air with gas takes place as shown in Figure (12) and this is installed Motorized mixer with screws.

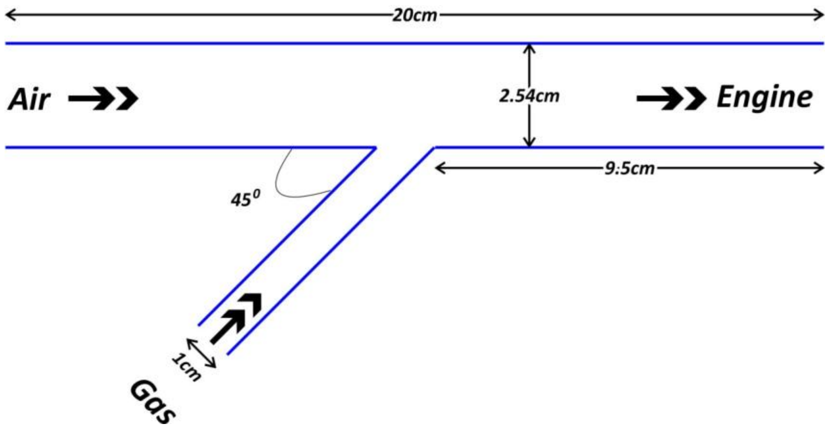


Figure (12): T-type mixer 45

- **Mixer T-angle 30**

It is an iron pipe with a diameter of (2.54 cm) and a length of (20 cm), where a pipe with a diameter of (1 cm) was welded in the middle perpendicular to it at an angle of (30) for the entry of natural gas from the cylinder and the process of mixing air with gas takes place as shown in Figure (13) and this is installed Motorized mixer with screws.

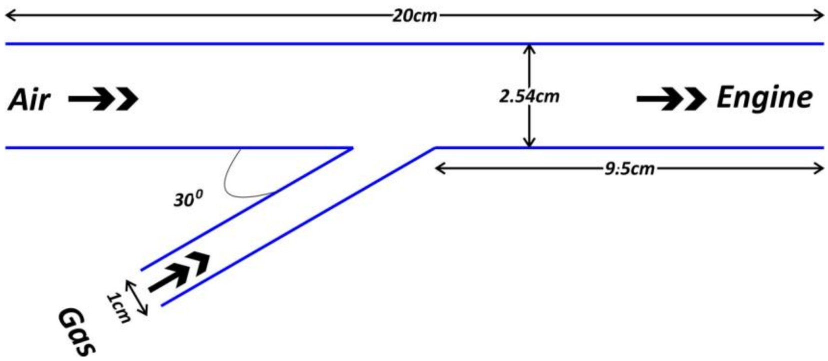


Figure (13): Mixer T-angle 30

- **Venture mixer**

It is a mixer made of plastic with a length of (27.8 cm) and a diameter of (5.08 cm) that contains a waistline in the middle with a diameter of (2.54 cm) and a tube with a diameter of (2.54 cm) is installed in this waistline as shown in Figure (14) where this mixer is installed in the motor to mix Entry of gas with air, where the gas enters through the small tube in the middle.

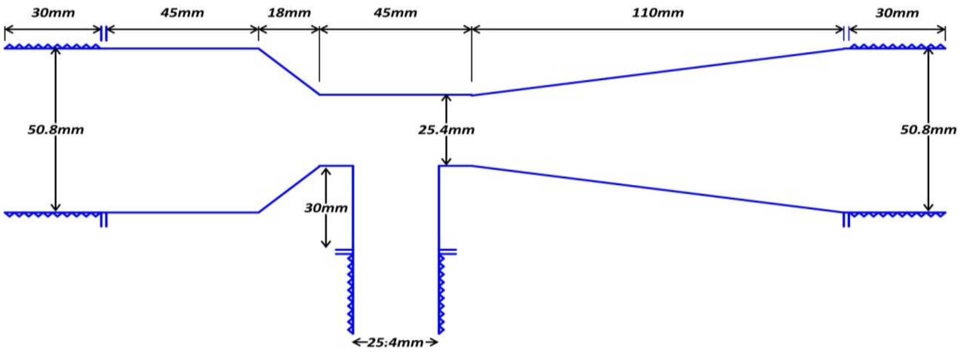


Figure (14): Venture Mixer

- **Mixer with perforated inner tube**

It is an iron pipe with a diameter of (3.81 cm), a length of (13 cm) and a thickness of (1 mm). A pipe with a diameter of (2.54 cm) was welded perpendicular to it, one from the top and the other from the bottom, so that one direction is opposite to the air entry from the air filter and the second connects with an inlet hole Air to the engine, as for the gas, an iron tube was made of three lengths (8-10-12 cm) perforated with a number of holes (15) perforations on its perimeter and it is fixed from the bottom of the mixer by screws as shown in Figure (15). whereby the gas is entered into the mixer and then exits From the holes to mix with the air and thus enter the mixture into the engine As shown in the experiment diagram in Figure (16).

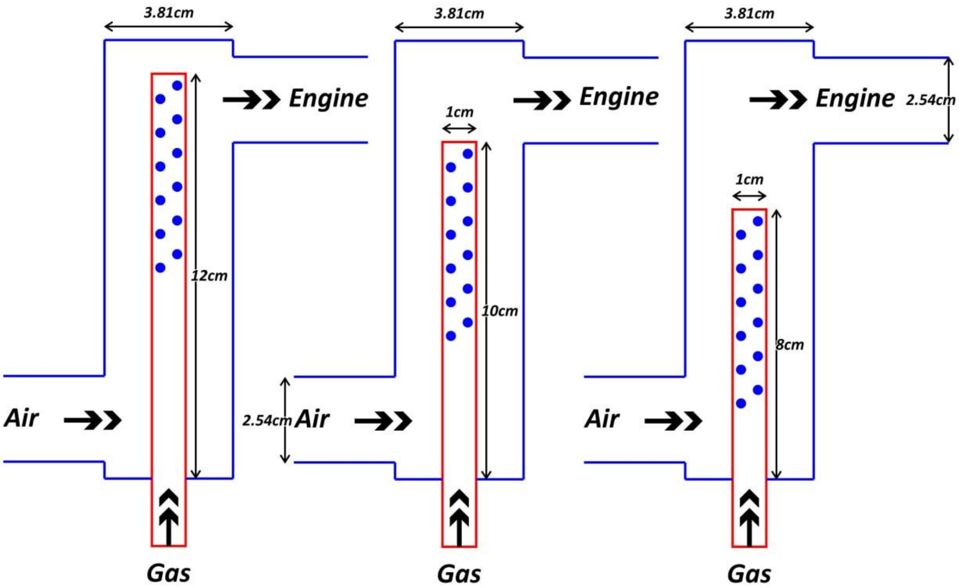


Figure (15): Mixer with a perforated inner tube

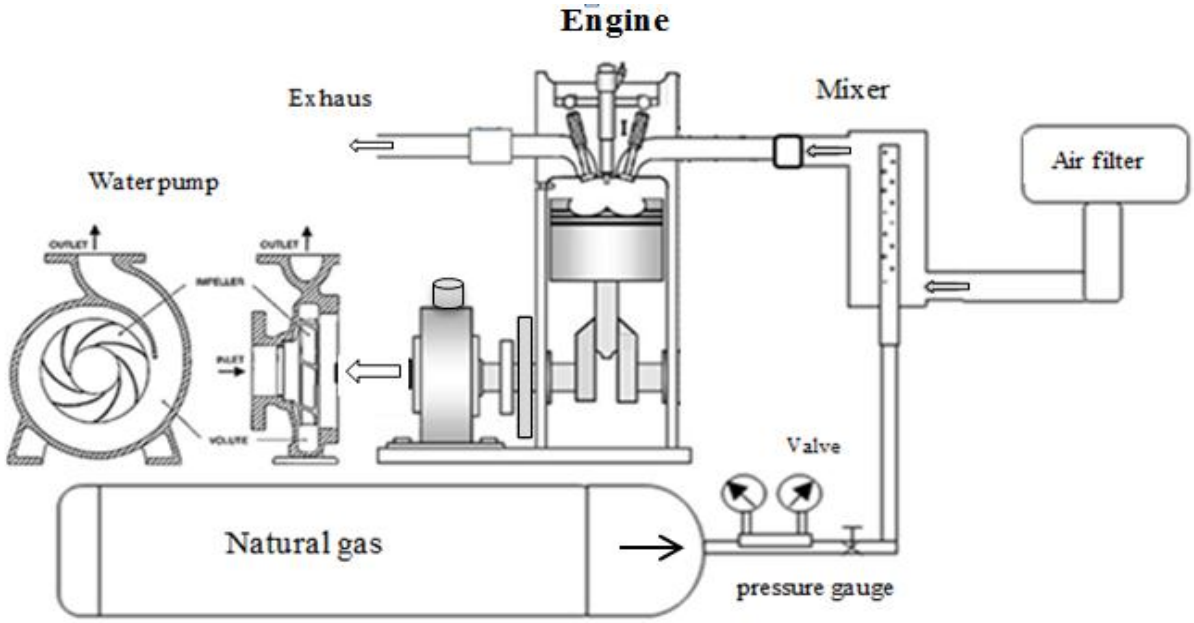


Figure (16): Simplified diagram of natural gas engine for powering water irrigation pump

3.2. Methods:

3.2.1. Discharge measuring tank:

The tank was manufactured inside the workshop of the Agricultural Engineering Department at the Faculty of Agriculture / Ain Shams University of iron sheets with a thickness of (2 mm) and the dimensions of the tank were (25-25- 60 cm) as shown in Figure (17) and a calibration was carried out for the amount of water entering the tank by placing a hose External transparent to determine the level of water inside the tank and make a gradation for each liter of water, where the capacity of the tank was about (40L).

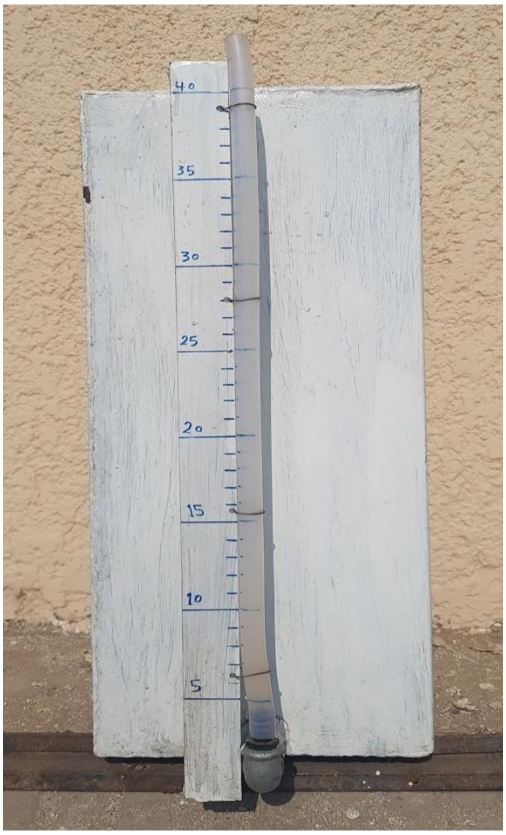


Figure (17): A tank to measure water drainage

3.2.2.Water pressure gauge:

A mercury-type 4 bar pressure gauge was used, which was attached to the water pump to calculate the pump pressure as shown in Figure (18).

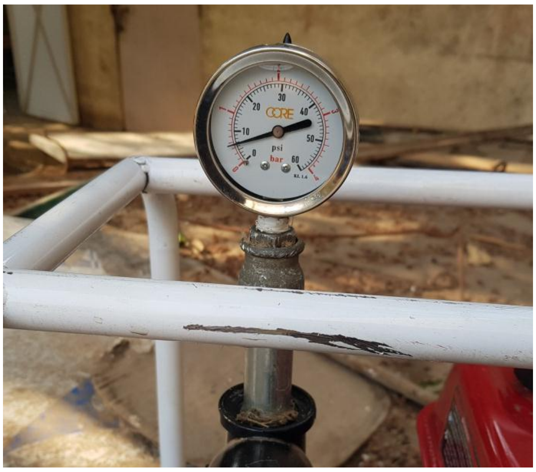


Figure (18): Water pressure gauge

3.2.3.Engine speed meter:

An engine speed meter was used to determine the speed of the engine shaft for four speeds (1750 - 2300 - 2900 - 3500 rpm) as shown in Figure (19), As shown in the device specifications in the Table (5).

Where the measurement of rotational speed is carried out from a distance and depends on the return of laser light and as its specifications .

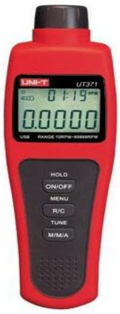


Figure (19): Tachometer

Table (5): Specifications of the tachometer

Measurement	10 to 99,999 RPM
Range Selection	10 to 99 RPM, 100 to 999RPM, 1000 to 9999 RPM, 10000 to 99999 RPM
Best Accuracy	0.04% ± 2dgt
Display	99999 counts
Target Distance	50 mm to 200 mm
Start / Hold	Start and stop Measurement, Data Hold
MAX /MIN / AVE / ZERO	Maximum Value, Minimum Value, Average Value, Zeroing
Low Battery Display	4.8 V

3.2.4.Anemometer:

The wind speed measurement device was used to measure the amount of natural gas entering the engine was also measured by calculating the gas flow velocity with the calculation of the cross-sectional area where the device is placed between the gas pipe from the source (gas cylinder) to the gas inlet to the mixer as shown in Figure (20), the accuracy of the device's measurement (0.01), the device is made in Taiwan, model 731A, as shown in the specifications of the device in Table (6).

Figure (20): Wind speed measuring device

Table (6): Specifications of the air velocity meter

Units	Resolution	Threshold	Range	Accuracy (±% reading + dgts)
m / s	0.01	0.3	0.00 – 30.00	± 3% +0.1
ft/ min	1	60	0 - 5900	± 3% +10
knots	0.1	0.6	0.0 – 58.0	± 3% +0.1
Km / hr	0.1	1.1	0.0 – 108.0	± 3% +0.1
mph	0.1	0.7	0.0 – 67.0	± 3% +0.1
Sensor : Thermistor temperature sensor				
Range : -20°c to 60°c (-4°F to 140°F) - Resolution : 0.1°c / 0.1°F				

3.2.5. Exhaust gases measuring device:

An exhaust gas measuring device model (Gas & Smoke AUTOCHECK) was used to measure engine exhaust gases (HC, CO, CO_2, O_2 as shown in Figure (21) and the device specifications in Table (7).

Figure (21): Exhaust gas measuring device

Table (7) Specifications of the exhaust gas meter

MODEL	AUTOCHECK	
Measurement Range	HC : 0~10,000ppm(hexane)	NOx : 0~5000ppm(Option)
	0~20,000ppm(propane)	Lambda : 0.5~3.0
	CO : 0~10%	AFR : 5~25
	O_2 : 0~25%	RPM : 600~9,000(Option)
	CO_2 : 0~20.00%	Oil Temp : 0~150˚C (Option)
Accuracy	HC, CO, CO_2, O_2 : 3% rel	
Response Time	HC, CO, CO_2 : less than 8 seconds	
	O_2, NO : less than 12 seconds	
Warming Up Time	Less than 20 seconds	
Operating Environment	Temperature : 0~50˚C	
	Humidity : Up to 95%(non-condensing)	
	Altitude : -300~2,500m(-1,000~8,000ft)	

3.2.6. Stop watch:

A stopwatch was used while measuring the water flow from the pump to the tank, as well as to measure the time required for fuel consumption (gasoline - natural gas) as shown in Figure (22)

Figure (22): Stop watch

3.2.7. Vernier calipers:

The diameter and thickness gauge was used to measure the diameters of the mixers pipes and the thickness of the disposal measuring tank with a measurement accuracy of (0.1) as shown in Figure (23).

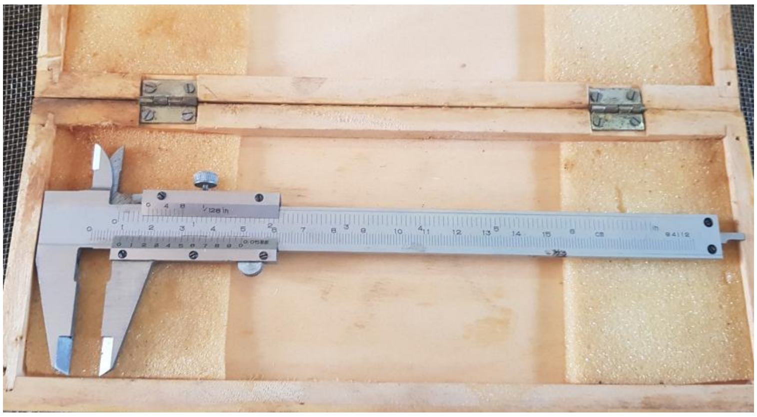


Figure (23): Vernier calipers

3.3. Experimented measurements:

After preparing the engine and filling it with oil, preparing each of the water barrel with a capacity of (200L) and the tank of disposal measurement (40L) and installing water hoses (the intake hose, the push hose) with a length of (3m) for each of them with a valve with a diameter of (5.08cm) at the beginning of the push hose To control the amount of water leaving the pump.

Starting the engine with gasoline to determine the four shaft speeds (1750 - 2300 - 2900 - 3500 rpm) using the engine speed measuring device after making a hole for the outer casing of the engine from the side of the starting lever as shown in Figure (24) where a paper carton was installed on the body. The engine is below the lever to determine the engine speed and determine the required speed.

Figure (24): Engine speed gauge slot

To assess the effect of experimental factors and the performance of the engine and pump work using types of fuel (gasoline - natural gas) and for different engine speeds and for all types of mixers, with respect to operating the engine with natural gas, where the operation was started with gasoline fuel and then closed the gasoline valve and opened the gas valve gradually until the operation of the engine with gas stabilized Naturally, many measurements were made in the experimental workshop at the Agricultural Engineering Department, Faculty of Agriculture, Ain Shams University, Egypt, and they were as follows:

3.3.1. Measuring the fuel consumption rate:

The gasoline tank was opened and replaced with a graduated glass tube, which was installed on the engine chassis, and the gasoline hose was connected to the caliper, as shown in Figure (25) in order to measure the rate of fuel consumption (gasoline). The rate of gasoline fuel consumption was measured by filling the glass tube listed and then the pump was run with load for all engine shaft speeds and calculating the amount of fuel (gasoline) consumed per unit time.

Figure (25): Gasoline consumption measuring tube

The rate of gas fuel consumption (natural gas) was measured by placing a wind speed measuring device on the gas hose between the gas cylinder and the gas entry hole to the mixer for all types of mixers, with a load for all engine speeds, where the cross-sectional area was calculated with the flow speed Gas To determine the amount of gas entering the engine, the following equations were used to calculate the fuel consumption rate **(Macmillan, 2002)**:

$$\mathbf{F_{BC} = (V / t) \times 0.0036 \;\ldots\ldots\ldots\ldots(8)}$$

$$\mathbf{F_{C} = (V / t) \times 3.6 \;\ldots\ldots\ldots\ldots(9)}$$

V = Volume of consumed natural gas fuel (cm^3)

t = Time of operation (s)

F_{Bc} = Rate of natural gas fuel consumption (m^3/h)

3.3.2. Measuring the engine power:

The power developed was measured by a prony brake at various speeds, loads and throttle valve positions, The net power of an engine is the power delivered at crankshaft, torque brake was used in the measurement of engine at all engine speed. As shown in a simplified diagram of prony brake in Figure (26). The power developed from engine is calculated using by the following equation:-

$$P= (2\pi\ n\ W\ L)/1000 \quad \ldots\ldots\ldots\ldots(10)$$

$$P=T\ .\omega \quad \ldots\ldots\ldots\ldots\ldots(11)$$

Where: P : power (kW),

n : speed of the pulley in (rpm),

W : load registered by the spring balance in (N),

L - length of the brake arm in (m),

T : torque arm in (N)

ω : angular speed (rev. /s).

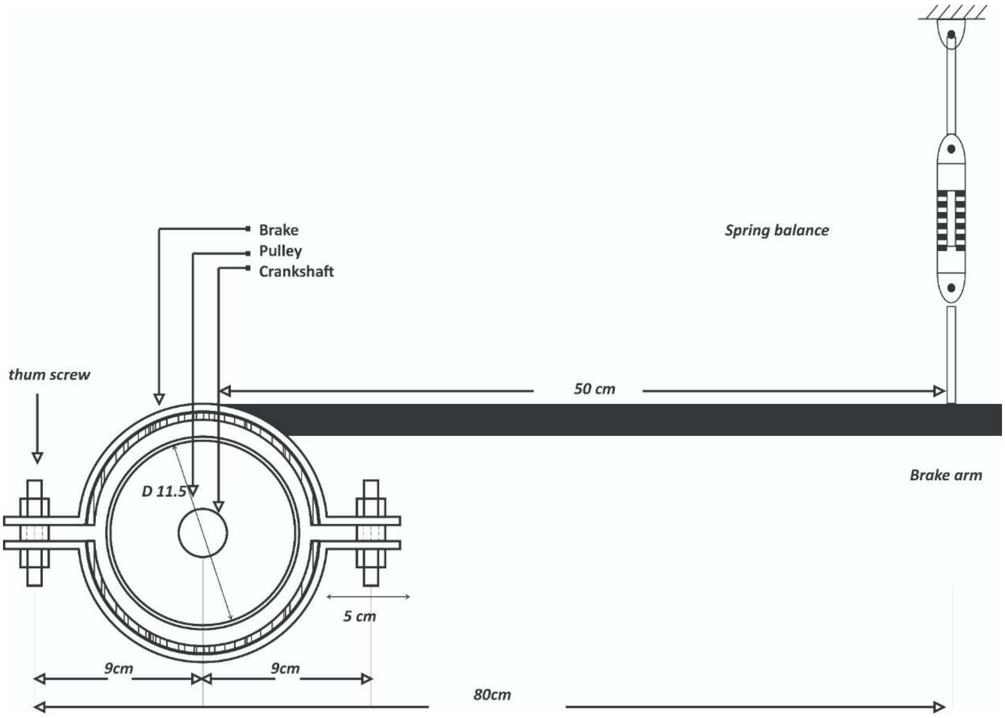


Figure (26): Simplified diagram of prony brake.

3.3.3. Measuring the performance of the pump:

The pump performance was measured using fuel types (gasoline - natural gas) and for all engine shaft speeds (1750 - 2300 - 2900 – 3500 rpm) by placing the intake hose in a barrel filled with water with a capacity of (200 L) and the exit hose (thrust) was placed in discharge measurement tank and controlling the amount of water coming out through the valve and taking a reading for each (0.1bar) per minute.

Where the actual hydraulic power was measured for types of fuel (gasoline - natural gas) for all engine speeds and for all types of mixers and for each pump pressure (0.1 bar), where after calculating the pump's discharge Q (m^3/s) and pump pressure P (bar) and using the following equation to calculate the actual hydraulic power:

$$HP = \rho . g . Q . h / \eta \quad \ldots\ldots\ldots\ldots.(12)$$

HP = Hydraulic power or useful water power (kW)

ρ = Water density (kg/ m^3)

g = Gravitational acceleration (m/s^2)

Q = Pump discharge (m^3/s)

H = Total head of the system (m)

η = Efficiency

3.3.4. Measurement of engine exhaust gases:

Where the exhaust gas measurement device (AUTO CHEK GAS) was used to measure the percentage of gases (CO_2 - CO - HC) for types of fuel (gasoline - natural gas) and for all engine speeds and for all types of mixers.

3.3.5. Economic indicators:

- Net present value (NPV) is used to analyze the profitability of a project or investment. It is calculated by the difference between the present value of cash inflows and the present value of cash outflows during the life of the project (from the project over its life). Note has been appreciated Value of marginal product (or shadow price) of irrigation water (1.84 LE/m^3).

- Benefit cost ratio(B/C) is used to analyze the profitability of a project or investment. It is calculated by dividing the present value of the cash inflows and the present value of the cash outflows during the life of the project **(Willis *et al.,* 2018)**. the following equations were used to calculate (B/C, NPV):

B/C = Total Present Benefits / Total Present Costs

N.P.V = Total Present Benefits - Total Present Costs

P.V. = F.V. X (1/ (1+r) n)……….(13)

P.V. =Present Value

F.V. = Future Value

r = Discount rate

n = Number of Years

(1/ (1+r) n) = The Present value coefficient.

- Internal rate of return (IRR) used to analyze the profitability of a project or investment this is the rate of interest that equates the present value of benefits to the present value of costs, IRR the discount rate that makes the net present value (NPV) of all cash flows from a given project equal to zero. The calculation of the internal rate of return is based on the same NPV formula **(Willis *et al.*, 2018).**

- Payback period is the time needed for the project to recover the net return (benefits) the cost of the capital investment for the project. In other words, the time needed for the cumulative cash flows to equal investment costs, the following equations were used to calculate (Payback period):

Payback period = investment costs / annual net return (profit)

3.3.6. The studied factors:

Through the preliminary experiments, the experimental factors can be determined in the average of the experiments, and the factors were as follows:

1- Type of fuel: two types of fuel were used (gasoline - natural gas) to run the engine and the water pump.

2- Operating the engine: the pump was operated once with a load and once without a load for all used speeds and for all types of mixers and types of fuel (gasoline - natural gas).

3- Engine speed: where four speeds were used for the motor shaft (1750-2300-2900-3500 rpm).

4- Gas mixing method: seven types of mixers were used (90 angle T mixer - 45 angle T mixer - 30 angle T mixer - venture mixer - 8cm perforated inner tube mixer - 10cm perforated inner tube mixer - Mixer With a perforated inner tube of length 12 cm (and it can be expressed by the following symbols:

- ✓ Mixer T-angle 90 → (T90)
- ✓ Mixer T-angle 45 → (T45)
- ✓ Mixer T-angle 30 → (T30)
- ✓ Venture Mixer → (VM)
- ✓ Mixer with a perforated inner tube of 8cm → (L8)
- ✓ Mixer with a perforated inner tube of 10cm → (L10)
- ✓ Mixer with a perforated inner tube of 12cm → (L12)

RESULTS AND DISCUSSION

The results here dealt with two main parts of the study, where the first part dealt with the analysis of technical indicators for the types of mixers used in mixing natural gas with air to operate the irrigation pump to save energy ،while the second part dealt with the analysis of the economic indicators for operating the irrigation pump. The results are discussed under the following headlines as bellow:

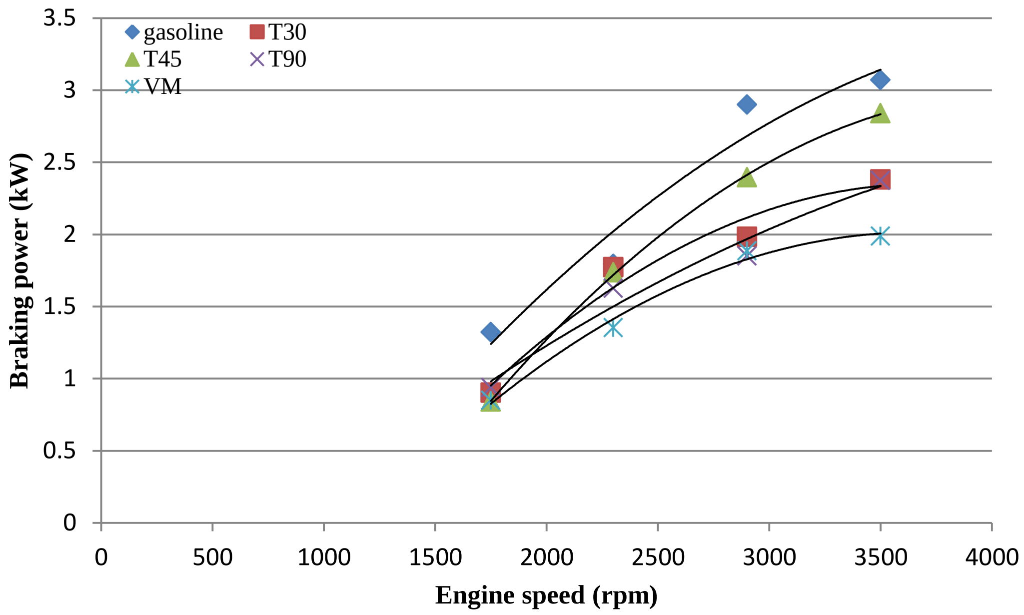
4.1. Engine technical indicators:

This part deals with technical indicators of engine power, specific fuel consumption and exhaust gases analysis. The results are discussed as follows:

4.1.1. Engine power:

A. Engine power (T30, T45, T90, VM):

Figure (27) and appendix (1) shows the relationship between engine speed and the actual power (Braking power) for each of (Gasoline, T30, T45, T90, VM) Where results guarantee that The relationship between engine speed and actual power is a quadratic relationship, As we note with the increase in the engine speed, the power increases in all types of mixers. The actual power is superior to all types when operating with gasoline was (3.07 kW) at engine speed (3500 rpm) A comparison with the use of natural gas, where the mixer type (T45) gave the highest power compared to the types of mixers (2.83 kW) at an engine speed of (3500rpm) was 7.5% less than gasoline. The higher the engine speed the higher the actual pump power in all types of mixers this is consistent with **(Ahmet and Rasim, 2021).**

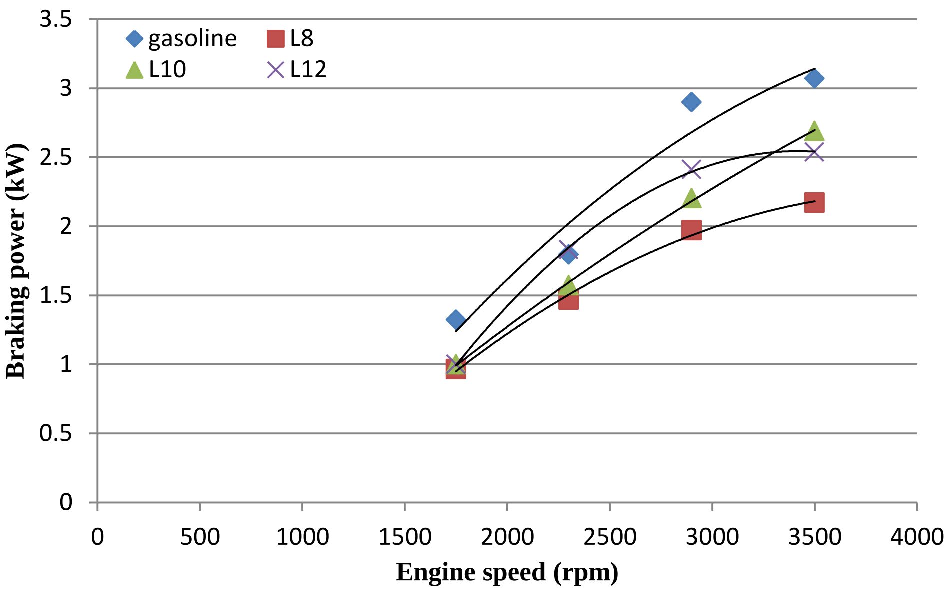


Types mixers	Regression equation	R^2
gasoline	Power = 3E-07X^2 + 0.0026X – 2.37	0.949
T30	Power = -4E-07X^2 + 0.002X – 2.361	0.996
T45	Power = -4E-07X^2 + 0.0031X – 3.457	0.999
T90	Power = -1E-07X^2 + 0.0015X – 1.239	0.999
VM	Power = -3E-07X^2 + 0.0024X – 2.36	0.990

Figure (27): The relationship between engine speed and the braking power for all types of mixers (gasoline – T30- T45- T90- VM)

A. Engine power (gasoline , L8, L10, L12):

Figure (28) and appendix (2) shows the relationship between engine speed and the actual power (Braking power) for each of (Gasoline, L8, L10, L12) Where results guarantee that The relationship between engine speed and actual power is a quadratic relationship, As we note with the increase in the engine speed, the power increases in all types of mixers. The actual power is superior to all types when operating with gasoline was (3.07 kW) at engine speed (3500 rpm) A comparison with the use of natural gas, where the mixer type (L10) gave the highest power compared to the types of mixers (2.69 kW) at an engine speed of (3500rpm) was 10% less than gasoline. this is due to reduced volumetric efficiency as the gaseous fuel displaces incoming fresh air resulting in reduced peak torque and power This is consistent with **(Khan *et al.*, 2016).** The higher the engine speed the higher the actual pump power in all types of mixers this is consistent with **(Ahmet and Rasim, 2021).**



Types mixers	Regression equation	R^2
gasoline	Power = 3E-07X2 + 0.0026X – 2.37	0.949
L8	Power = -3E-07X2 + 0.002X – 1.836	0.996
L10	Power = -1E-07X2 + 0.0015X – 1.333	0.999
L12	Power = -6E-07X2 + 0.0038X – 3.69	0.999

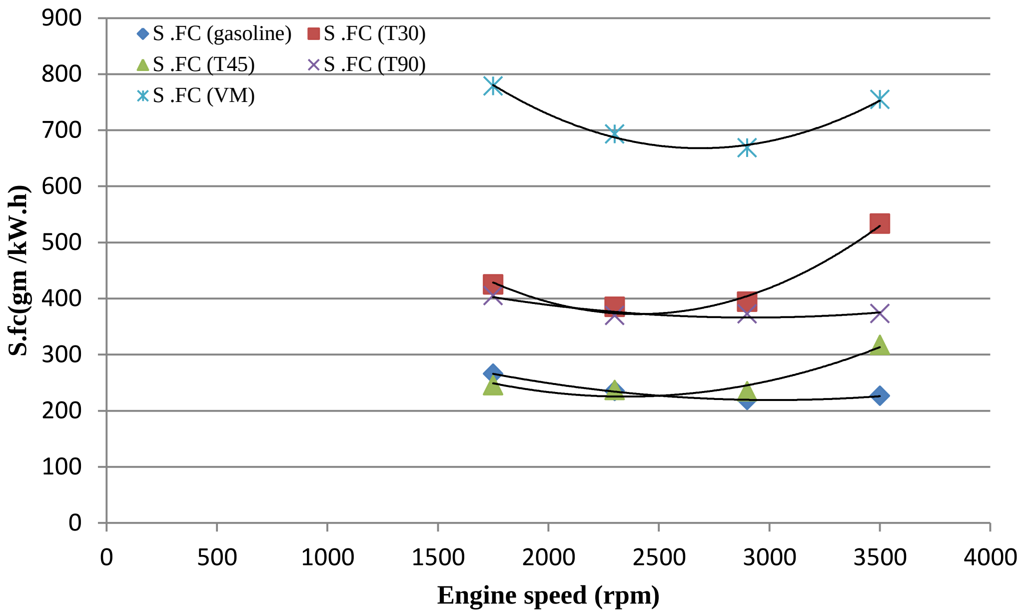
Figure (28): The relationship between engine speed and the braking power for all types of mixers (gasoline – L8 – L10 - L 12 cm)

4.1.2. Specific fuel consumption :

A. Specific fuel consumption (gasoline , T30, T45, T90, VM):

Figure (29) and appendix (3) shows the relationship between engine speed and specific fuel consumption (S.fc) for each of (Gasoline, T30, T45, T90, VM) at deferent engine speed . The relationship between engine speed and specific fuel consumption is of a quadratic function type. When the engine speed is increased, the specific fuel consumption decreases and the values were (226.109 , 533.12, 316.89, 373.2, 754.59 gm/kW.h) When using mixers (Gasoline, T30, T45, T90, VM) respectively at engine speed (3500 rbm).

The specific fuel consumption rate ranged (265.727- 219.025 gm/kW.h) for gasoline compared to (754.59 - 234.61gm/kW.h) with natural gas. We note a decrease in fuel consumption with increasing engine speed this is consistent with **(Mika *et al.*, 2022).** The lowest (S.fc) for gasoline was (219.025 gm/Kw.h) at an engine speed of (2900 rpm), The lowest (S.fc) for the types of mixers was the mixer (T45) was (234.612 gm/Kw.h) at an engine speed of (2900 rpm) and an increase of 6.6 % over gasoline.



Types mixers	Regression equation	R^2
gasoline	S.fc = 3E-05X2 - 0.1757X + 484.27	0.999
T30	S.fc = 0.0001X2 - 0.6318X + 1132.1	0.981
T45	S.fc = 7E-05X2 - 0.3119X + 591.46	0.940
T90	S.fc = 3E-05X2 - 0.1558X + 593.36	0.888
VM	S.fc = 0.0001X2 - 0.6892X + 1593.9	0.991

Figure (29): The relationship between engine speed and specific fuel consumption for each of (gasoline, T30, T45, T90, VM)

Figure (30) shows specific fuel consumption (S.fc) for each of (Gasoline, T30, T45, T90, VM) at engine speed (2900- 3500 rbm). Where results guarantee that Specific fuel consumption using using VM mixer was (754.59 gm/kW.h) more than that of gasoline by 70 %at engine speed (3500 rbm). The lowest specific fuel consumption at the T45type mixer compared to other typeT mixers was (316.898 gm/kW.h) and an increase of 28.6 % over gasoline at engine speed (3500 rbm). The lowest (S.fc) at an engine speed of 2900 using gasoline was (219.025 gm/kW.h) and using natural gas when the type of mixer T45 was (234.61 gm/kW.h) an increase of 6.6 % over gasoline.**(Jeongwoo *et al.*, 2020)** Reported that result due to the higher energy density of CNG (49.3 MJ/kg) than that of gasoline (42.8 MJ/kg), Since the weight of fuel is important, a better S.fc of CNG combustion than that of gasoline combustion is the biggest advantage of using CNG in a SI engine.

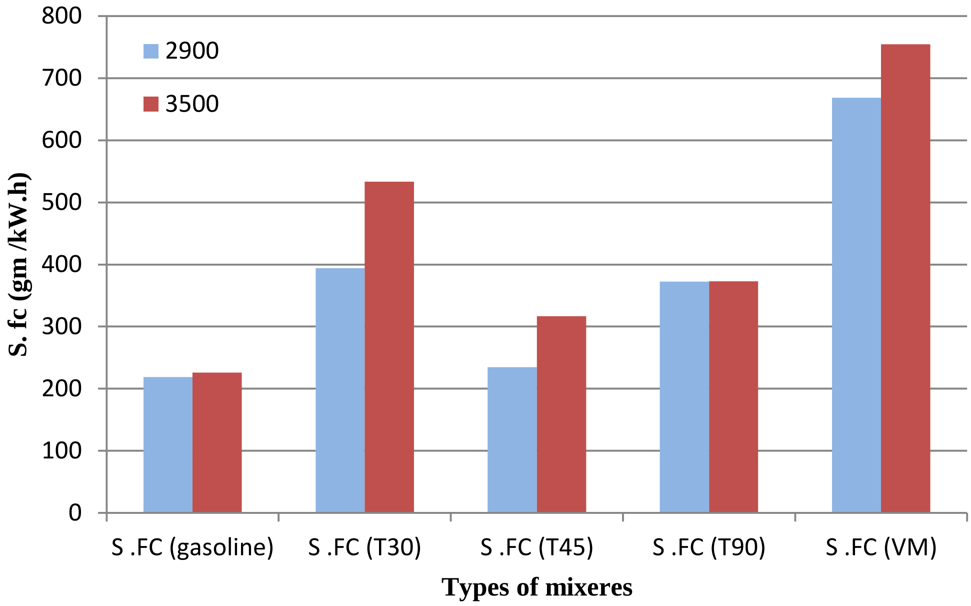
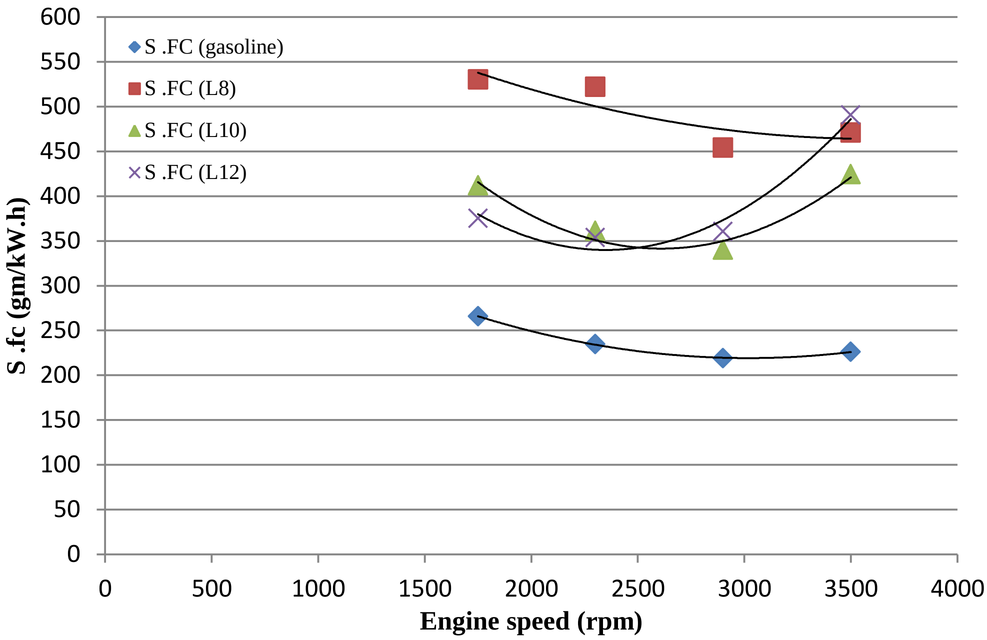


Figure (30): Specific fuel consumption for (gasoline, T30, T45, T90, VM) with a load at an engine speed of (2900, 3500) rpm

B. Specific fuel consumption (L8, L10, L12):

Figure (31) and appendix (4) shows the relationship between engine speed and specific fuel consumption (S.fc) for each of (Gasoline, L8, L10, L12) at deferent engine speed. The relationship between engine speed and specific fuel consumption is of a quadratic function type. When the engine speed is increased, the specific fuel consumption decreases and the values were (226.109 , 471.039, 424.314 , 490.3871gm/kW.h) When using mixers (Gasoline, L8, L10, L12) respectively at engine speed (3500 rbm).

The specific fuel consumption rate ranged (265.727- 219.025 gm/kW.h) for gasoline compared to (530.218 - 340.144 gm/kW.h) with natural gas. We note a decrease in fuel consumption with increasing engine speed this is consistent with **(Mika *et al.*, 2022).** The lowest (S.fc) for gasoline was (219.025 gm/Kw.h) at an engine speed of (2900 rpm), The lowest (S.fc) for the types of mixers was the mixer (L10) was (340.144 gm/Kw.h) at an engine speed of (2900 rpm) and an increase of 35.6. % over gasoline.



Types mixers	Regression equation	R^2
gasoline	S.fc = 3E-05X2 - 0.1757X + 484.27	0.999
L8	S.fc = 2E-05X2 - 0.1558X + 744.18	0.765
L10	S.fc = 0.0001X2 - 0.5237X + 1024.7	0.953
L12	S.fc = 0.0001X2 - 0.5198X + 950.82	0.968

Figure (31): The relationship between engine speed and specific fuel consumption for each of (gasoline, L8, L10, L12)

Figure (32) shows specific fuel consumption (S.fc) for each of (Gasoline, L8, L10, L12) at engine speed (2900- 3500 rbm). Where results guarantee that Specific fuel consumption using gasoline (226.109 gm/kW.h), and using L12 mixer was (490.387 gm/kW.h) more than that of gasoline by 53.89 %at engine speed (3500 rbm). The lowest specific fuel consumption at the L10 type mixer compared to other type L mixers was (424.314 gm/kW.h) and an increase of 46.7 % over gasoline at engine speed (3500 rbm). The lowest (S.fc) at an engine speed of 2900 using gasoline was (219.025 gm/kW.h) and using natural gas when the type of mixer L10 was (340.144 gm/kW.h) an increase of 35.6 % over gasoline at engine speed (2900 rbm). Results show gasoline is more power, torque and Brake Mean Effective Pressure (BMEP) than CNG but Brake Specific Fuel Consumption (BSFC) of gasoline is not as much as CNG **(Saad *et al.*, 2014).**

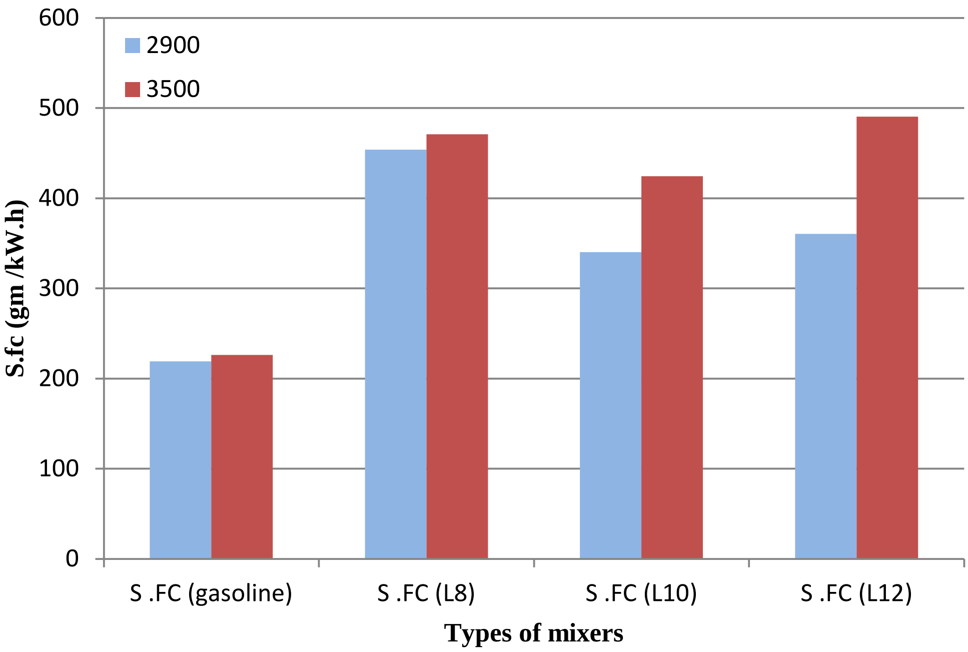


Figure (32): Specific fuel consumption for (gasoline, L8, L10, L12) with a load at engine speed (2900- 3500) rpm

Figure (33) shows the specific fuel consumption with load for all mixers at engine speed (2900 rpm), where we note that the lowest (S. fc) was gasoline with a load (219.025 gm/kw.hr) and the highest specific fuel consumption when the mixer type (vm) was (668.52 gm/kw.hr). the lowest (S. fc) when the mixer type (T45) It was (234.61gm/kw.hr) compared to all mixers at engine speed (2900 rpm).

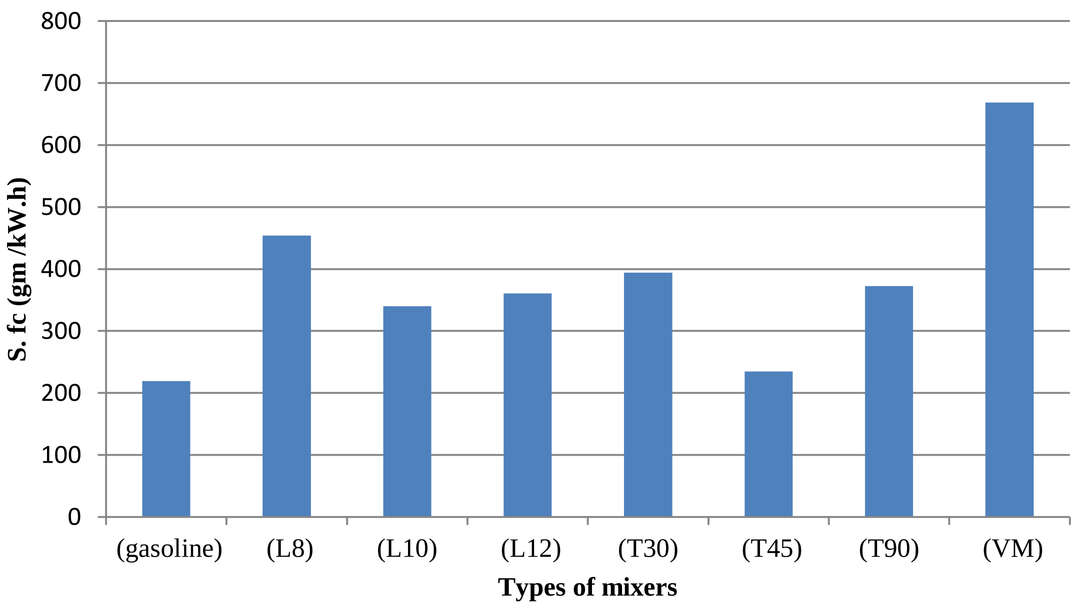


Figure (33): Specific fuel consumption for all mixer at an engine speed of (2900 rpm)

4.1.3. Exhaust gases (CO_2 , CO , HC):

Figure (34) and appendix (A) shows the average of carbon dioxide exhaust gases for operating the pump with types of mixers (Gasoline, T30, T45, T90, VM), where we note the percentage of CO_2 exhaust gases when operating with gasoline exceeding all types it was (7.96%) and the lowest percentage carbon dioxide was (2.77%) using mixer type T90. These results are in agreement with **(Wayan, et al., 2014)** reported that natural gas used as a vehicle fuel gives 20-30 % lower CO_2 emissions.

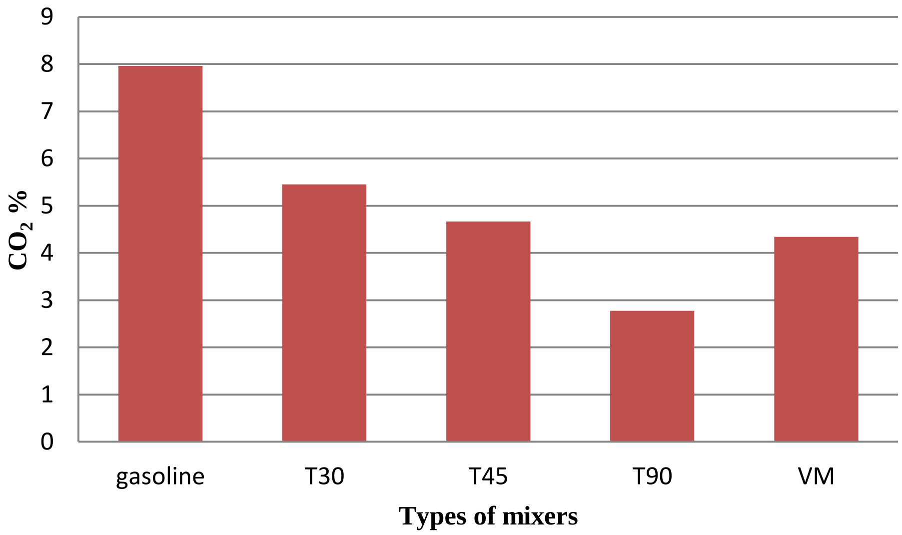


Figure (34): Exhaust gases average CO_2% for all types of mixers (Gasoline, T30, T45, T90, VM)

Figure (35) and appendix (6) shows the average of carbon dioxide exhaust gases for operating the pump with types of mixers (Gasoline, L8, L10, L12), where we note the percentage of CO_2 exhaust gases when operating with gasoline exceeding all types it was (7.96%) and the lowest percentage carbon dioxide was (4.22%) using mixer type L12. These results are in agreement with **Jeongwoo et al., (2020)** CNG combustion produced better results than gasoline combustion did. Under the same conditions of low heating value, CNG has less carbon in the fuel than gasoline.

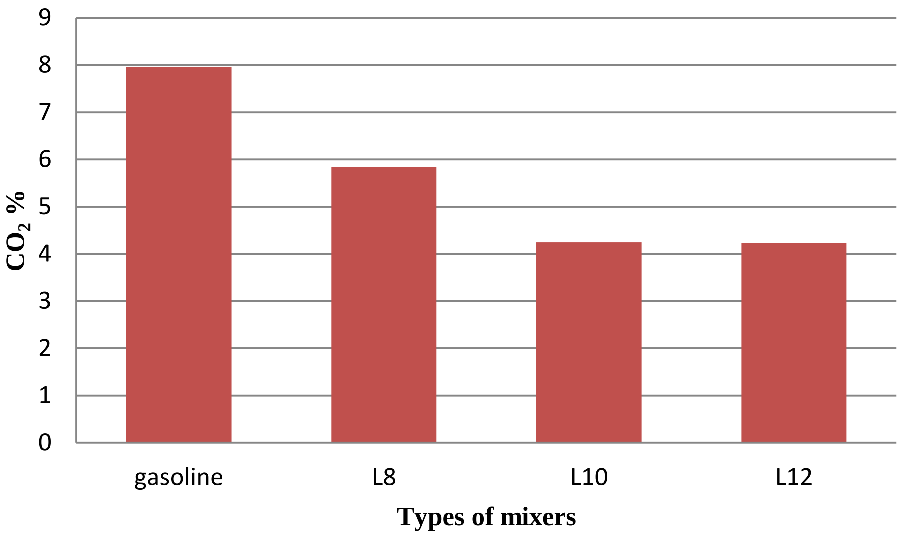


Figure (35): Exhaust gases average CO_2% for all types of mixers (Gasoline, L8, L10, L12)

Figure (36) and appendix (5) shows the average of carbon monoxide exhaust gases for operating the pump with types of mixers (Gasoline, T30, T45, T90, VM), where we note the percentage of CO exhaust gases when operating with gasoline exceeding all types it was (0.53%) and the lowest percentage carbon monoxide was (0.18%) using mixer type T30. These results are in agreement with **(Saad *et al,* 2014)** Reported that due to its high research octane number (RON) which is higher than 120, natural gas allows combustion at a higher compression ratio without knocking. It also offers much lower CO, gas emissions compared with other hydrocarbon fuels as a result of its higher hydrogen to the carbon ratio.

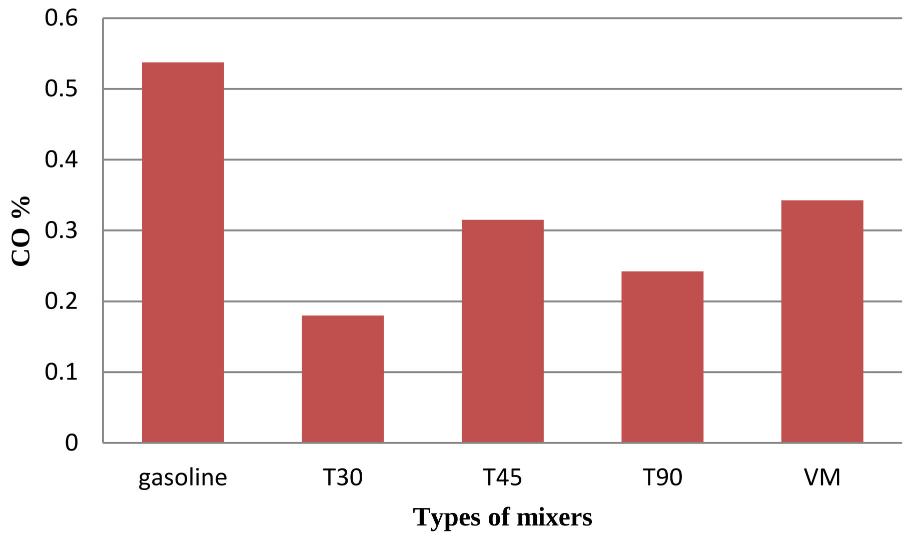


Figure (36): Exhaust gases average CO% for all types of mixers (Gasoline, T30, T45, T90, VM)

Figure (37) and appendix (6) shows the average of carbon monoxide exhaust gases for operating the pump with types of mixers (Gasoline, L8, L10, L12), where we note the percentage of CO exhaust gases when operating with gasoline exceeding all types it was (0.53%) and the lowest percentage carbon monoxide was (0.22%) using mixer type L8. These results are in agreement with (**Saad *et al.*, 2014)** Reported that The emissions of CO_2, and CO were found less of CNG compared to gasoline.

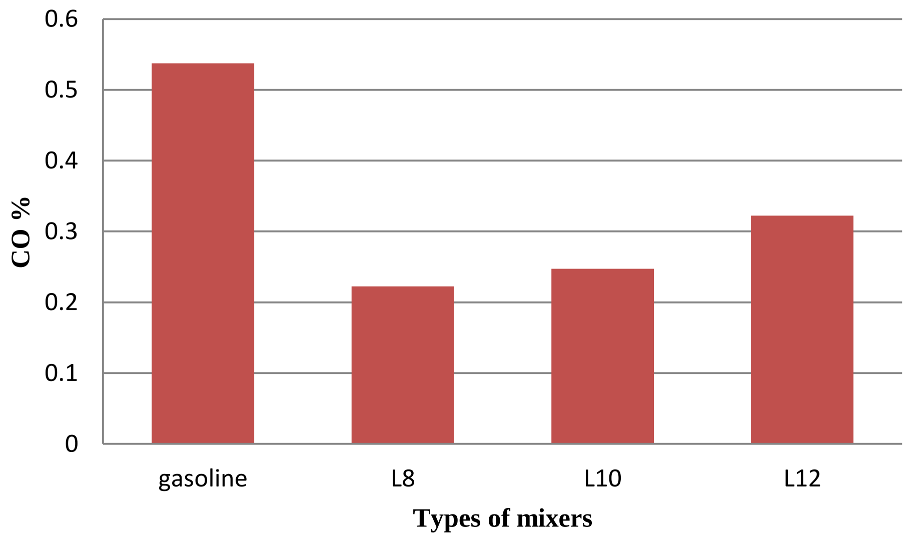


Figure (37): Exhaust gases average CO% for all types of mixers (Gasoline, L8, L10, L12)

Figure (38) and appendix (5) shows the average of HC ppm exhaust gases for operating the pump with types of mixers (Gasoline, T30, T45, T90, VM), where we note the average of HC exhaust gases when operating with gasoline exceeding all types it was (48.25 ppm) and the lowest percentage HC was (17.25 ppm) using mixer type T90. These results are in agreement with **Saravanan *et al.*, (2013)** Showed that because CNG has an 8-fold lower carbon content than gasoline, fewer carbon-based emissions are produced. Dedicated CNG engines with a greater compression ratio can reduce HC even more.

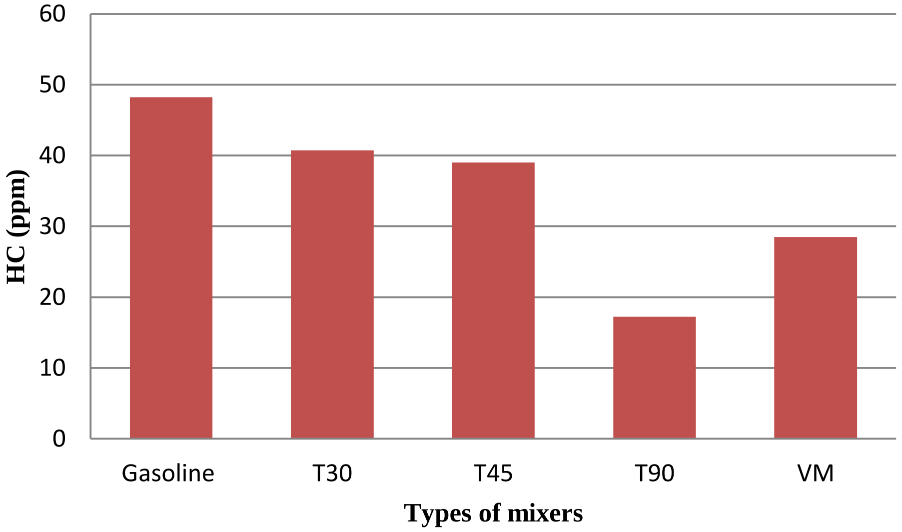


Figure (38): Exhaust gases average HC ppm for all types of mixers (Gasoline, T30, T45, T90, VM)

Figure (39) and appendix (6) shows the average of HC exhaust gases for operating the pump with types of mixers (Gasoline, L8, L10, L12), where we note the average of HC exhaust gases when operating with gasoline exceeding all types it was (48.25 ppm) and the lowest percentage HC was (23.75 ppm) using mixer type L10. These results are in agreement with **(Abbas *et al.*, 2017)** Due to a more complete combustion of CNG than gasoline, the emission of HC is greatly reduced by 25-72% when using CNG

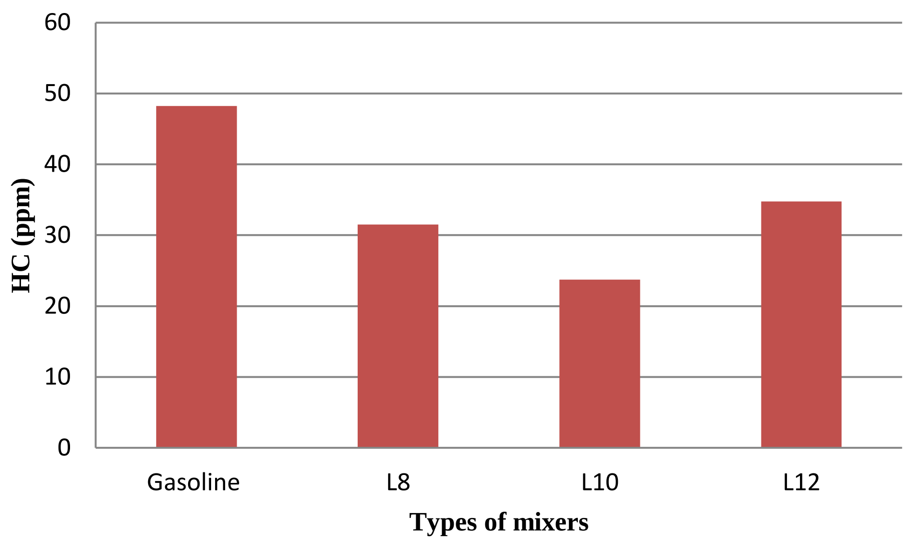


Figure (39): Exhaust gases average HC ppm for all types of mixers (Gasoline, L8, L10, L12)

4.2. Pump indicators:

This part deals with pump indicators of pump discharge, actual pump power. The results are discussed as follows:

4.2.1. Power and discharge of the pump using gasoline:

Figure (40) and appendix (7) shows the relationship between pump discharge and power and pump pressure for gasoline operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using gasoline (32.38 m^3/h) at pressure (0.2bar) and minimum discharge (5.66 m^3/h) at pressure (1.8 bar).

Figure (40) shows the relationship between pump discharge and the actual hydraulic power to operate using Gasoline at engine speed 3500 rpm Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.89 kW) at pump discharge of (21.97 m^3/h).

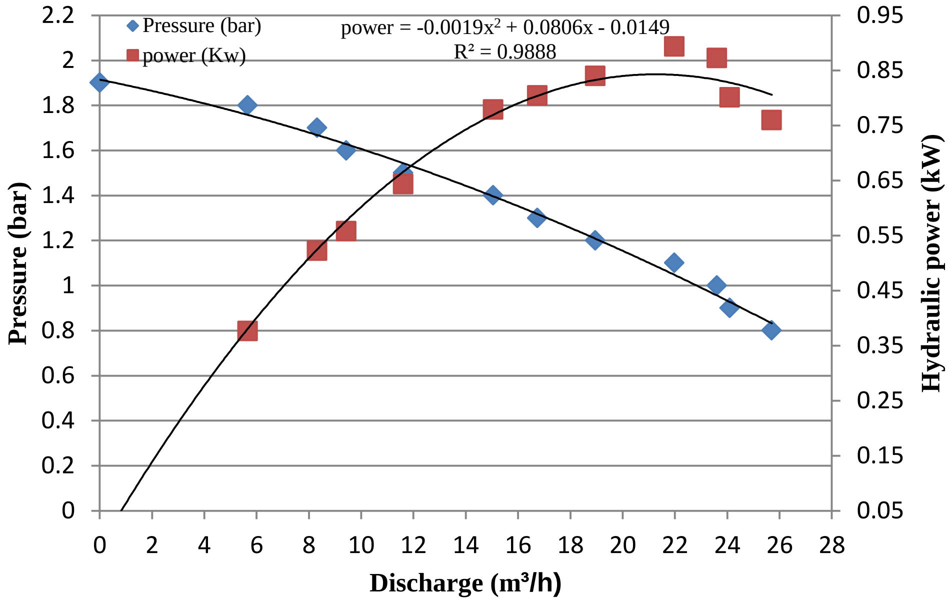


Figure. (40): The relationship between discharge, pressure and the hydraulic power of pump operating with gasoline at engine speed (3500 rpm)

4.2.2. Power and discharge of the pump using mixer type (T30):

Figure (41) and appendix (8) shows the relationship between pump discharge and power and pump pressure for mixer T30 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using T30 mixer (32.46 m^3/h) at pressure (0.2bar) and minimum discharge (5.01m^3/h) at pressure (1.8bar).

Figure (41) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer T30, Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.76 kW) at pump discharge of (17.24 m^3/h).

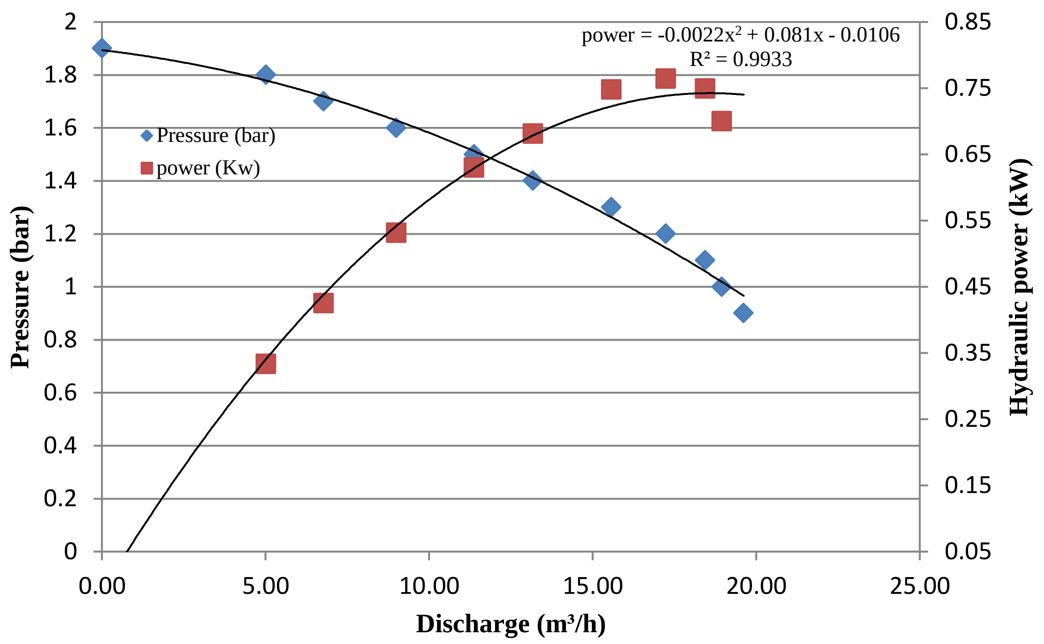


Figure. (41): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T30 at engine speed (3500 rpm)

4.2.3. Power and discharge of the pump using mixer type (T45):

Figure (42) and appendix (9) shows the relationship between pump discharge and power and pump pressure for mixer T45 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using T45 mixer (33.09 m^3/h) at pressure (0.2bar) and minimum discharge (3.60 m^3/h) at pressure (1.8bar).

Figure (42) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer T45, Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.79 kW) at pump discharge of (19.65m^3/h).

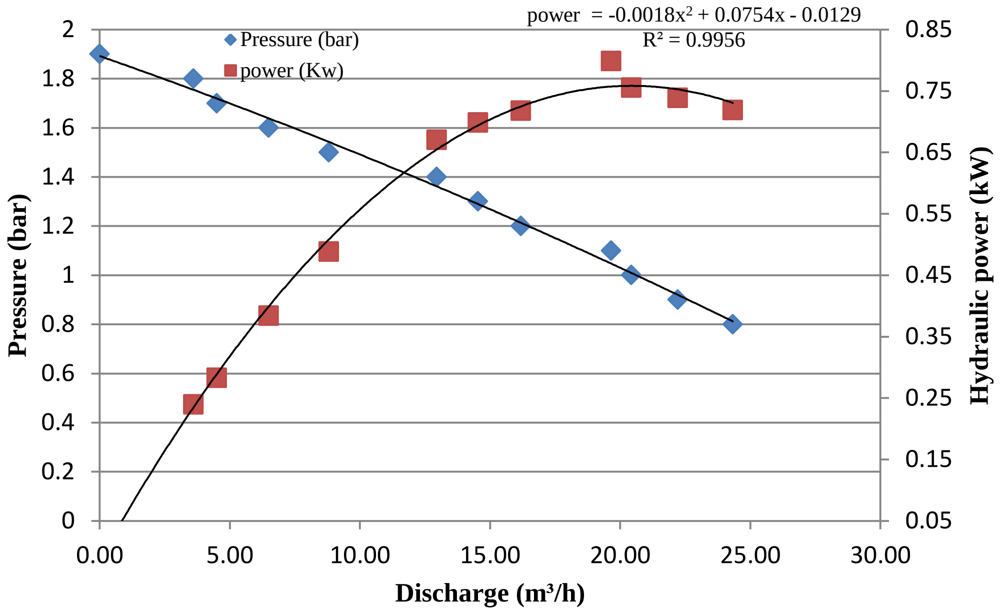


Figure. (42): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T45 at engine speed (3500 rpm)

4.2.4. Power and discharge of the pump using mixer type (T90):

Figure (43) and appendix (12) shows the relationship between pump discharge and power and pump pressure for mixer T90 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using T90 mixer (29.88 m^3/h) at pressure (0.2bar) and minimum discharge (2.08m^3/h) at pressure (1.8bar).

Figure (43) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer T90, Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.73 kW) at pump discharge of (18.17 m^3/h).

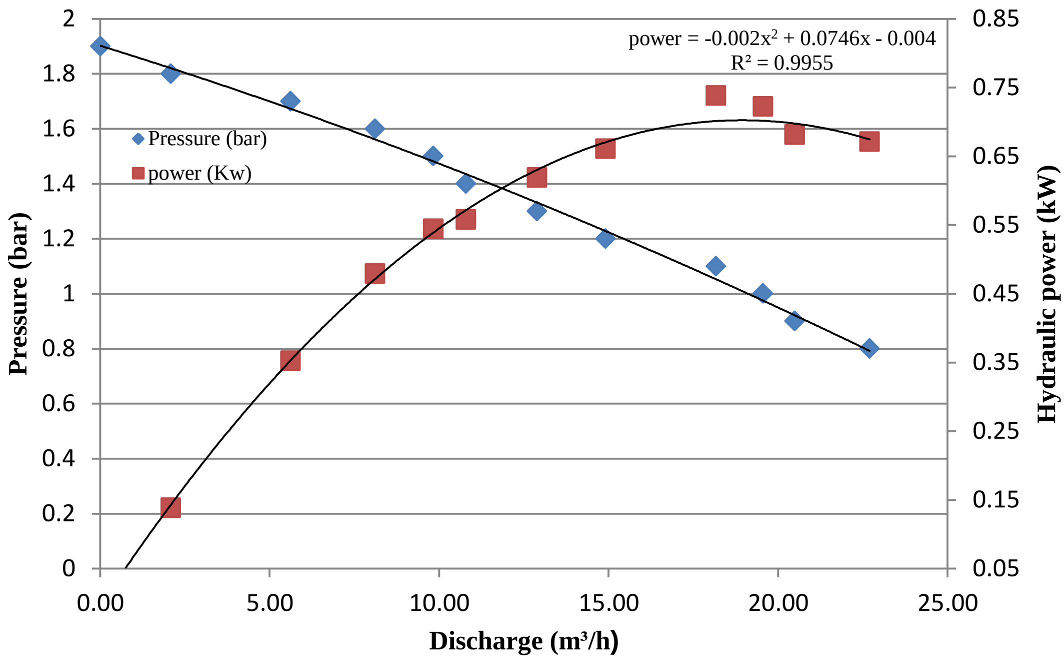


Figure. (43): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type T90 at engine speed (3500 rpm)

4.2.5. Power and discharge of the pump using mixer type (VM):

Figure (44) and appendix (11) shows the relationship between pump discharge and power and pump pressure for mixer VM operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using VM mixer (31.69 m^3/h) at pressure (0.2bar) and minimum discharge (1.85 m^3/h) at pressure (1.8bar).

Figure (44) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer VM, Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.705 kW) at pump discharge of (23.88 m^3/h).

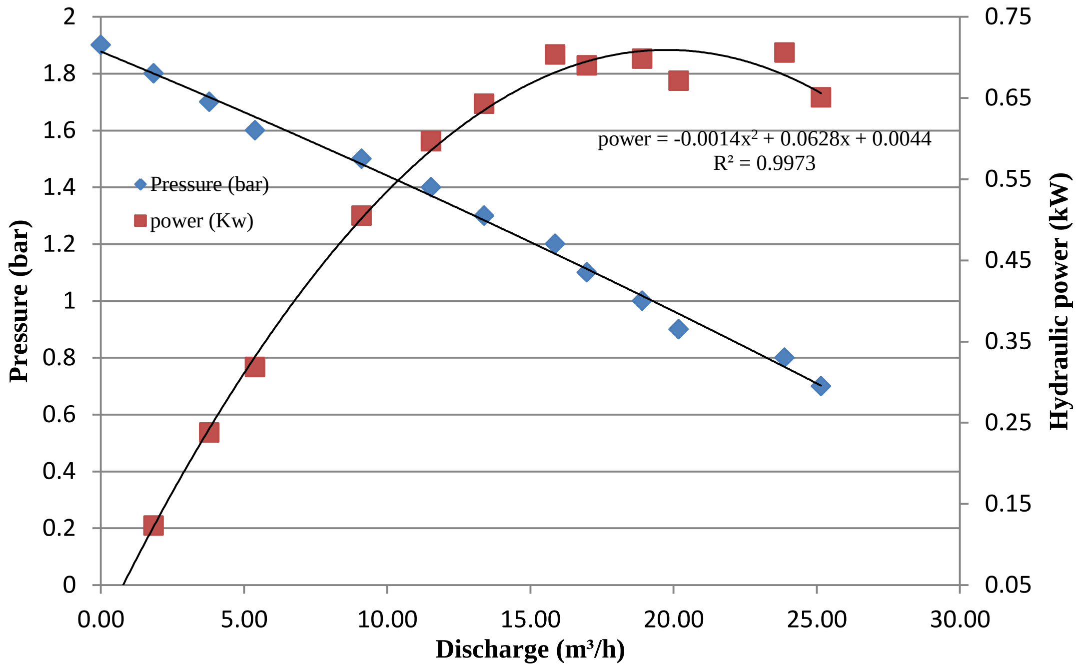


Figure. (44): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type VM at engine speed (3500 rpm)

4.2.6. Power and discharge of the pump using mixer type (L8):

Figure (45) and appendix (12) shows the relationship between pump discharge and power and pump pressure for mixer L8 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using L8 mixer (32.98 m^3/h) at pressure (0.2bar) and minimum discharge (3.62 m^3/h) at pressure (1.8bar).

Figure (45) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer L8 at engine speed 3500 rpm Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.717 kW) at pump discharge of (19.42 m^3/h)

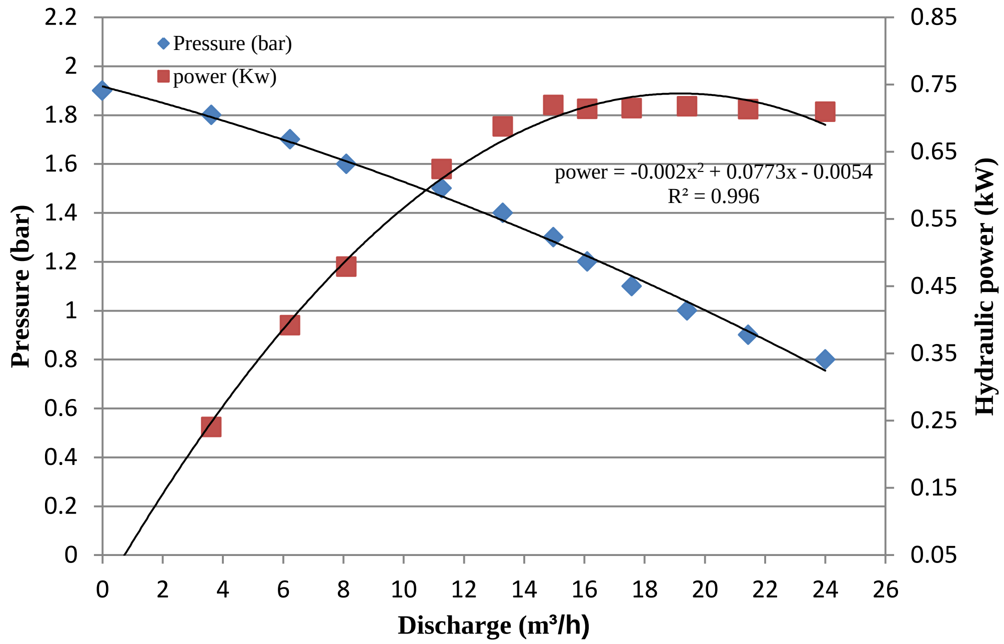


Figure. (45): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L8 at engine speed (3500 rpm)

4.2.7. Power and discharge of the pump using mixer type (L10):

Figure (46) and appendix (13) shows the relationship between pump discharge and power and pump pressure for mixer L10 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using L10 mixer (31.37 m^3/h) at pressure (0.2bar) and minimum discharge (2.64 m^3/h) at pressure (1.8bar).

Figure (46) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer L10 at engine speed 3500 rpm Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.782 kW) at pump discharge of (23.54 m^3/h).

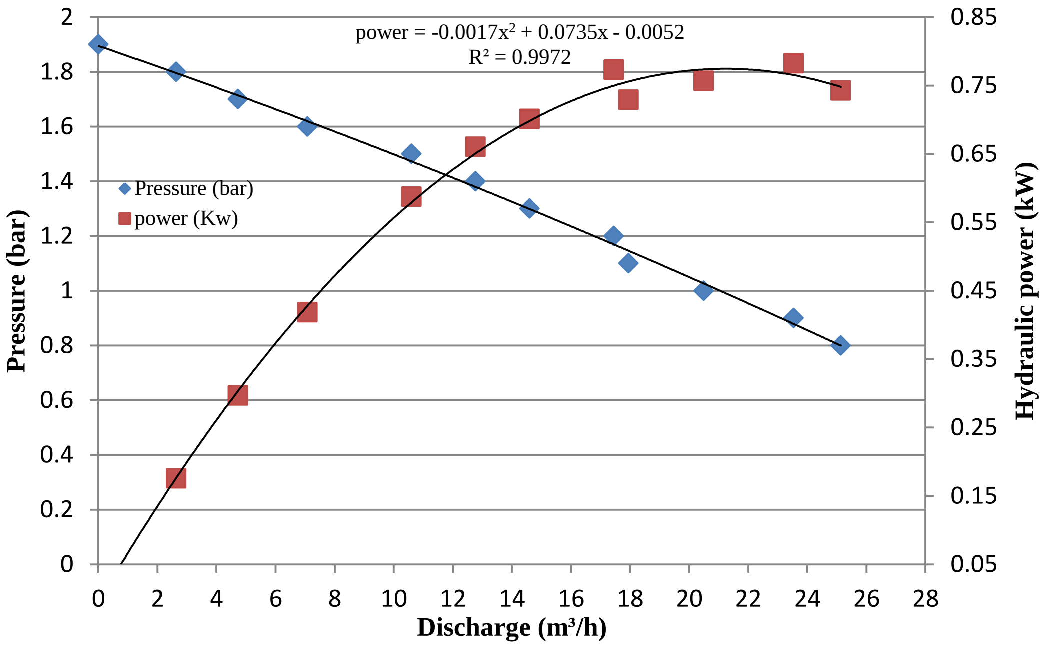


Figure. (46): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L10 at engine speed (3500 rpm)

4.2.8. Power and discharge of the pump using mixer type (L12):

Figure (47) and appendix (14) shows the relationship between pump discharge and power and pump pressure for mixer L12 operating at engine speed (3500 rpm) Where results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using L12 mixer (29.51 m^3/h) at pressure (0.2bar) and minimum discharge (2.18 m^3/h) at pressure (1.8bar).

Figure (47) shows the relationship between pump discharge and the actual hydraulic power to operate using mixer L12 at engine speed 3500 rpm Where results guarantee that the relationship between pump discharge and actual power is a quadratic relationship. As the pump discharge increases, The actual hydraulic power increases. The maximum actual hydraulic power was (0.473 kW) at pump discharge of (12.81 m^3/h).

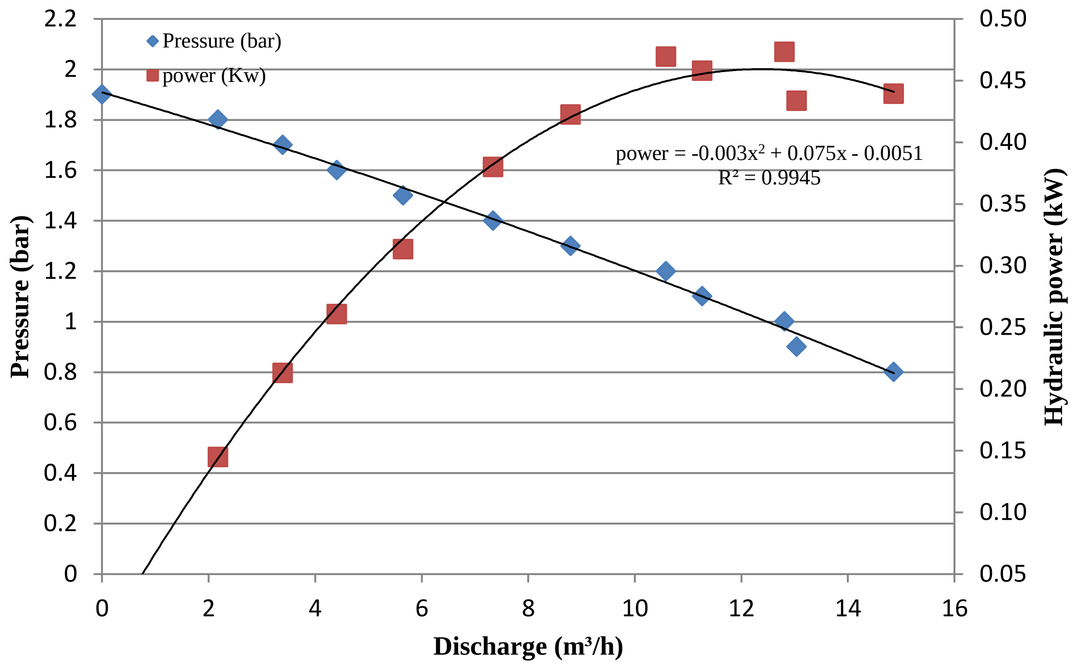


Figure. (47): The relationship between discharge, pressure and the hydraulic power of pump operating with mixer type L12 at engine speed (3500 rpm)

Discharge the pump with the L12 mixer (29.51 m^3/h) was 8.8 % lower than that of gasoline at an engine speed of (3500 rpm). The highest pump discharge was with the L8 mixer (32.98 m^3/h), an increase of 1.8% over gasoline at an engine speed of (3500 rpm). The actual hydraulic power with L12 mixer (0.473 kW) was 47% lower that of gasoline. The highest actual hydraulic power with L10 type mixer compared to other type L mixers (0.782 kW) was 12 % lower than that of gasoline.

Figure (48) shows the hydraulic power when the engine is operating with gasoline and natural gas for all types of mixers at an engine speed of (3500 rpm), where we notice the highest hydraulic power was when operating with gasoline (0.89 kW), then the mixer T45 was (0.798 kW), while lowest the hydraulic power was at the L12 mixer (0.473kW) This is consistent with **(Khan *et al.*, 2016)** this is due to reduced volumetric efficiency as the gaseous fuel displaces incoming fresh air resulting in reduced peak torque and power.

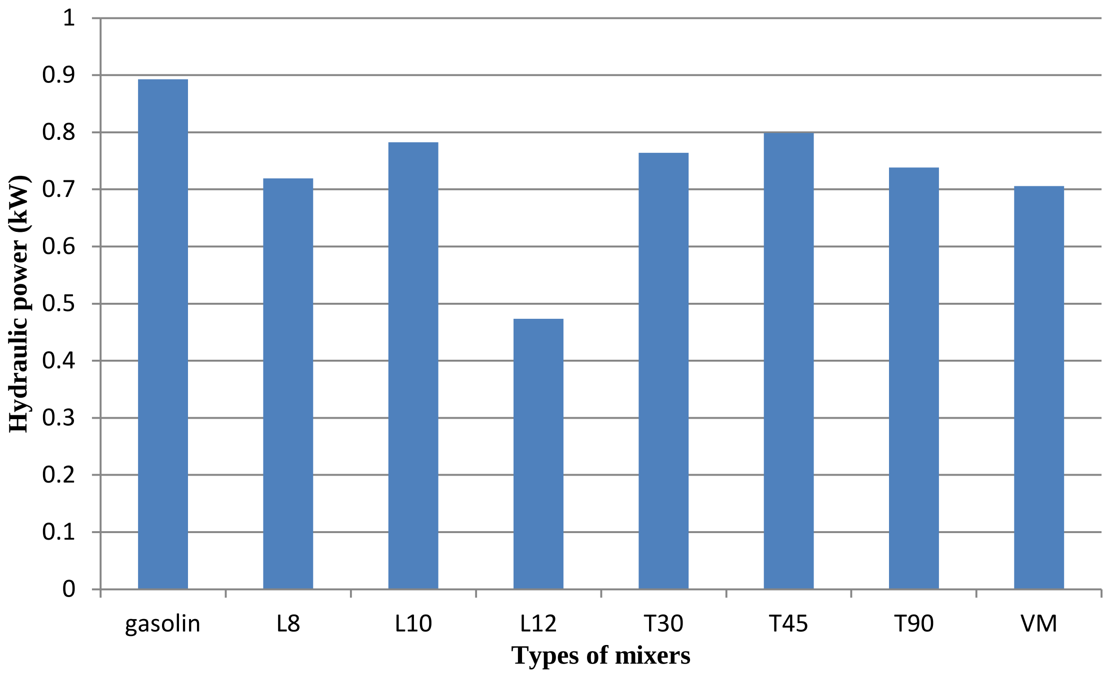


Figure. (48): The hydraulic power for all types of mixers at engine speed (3500 rpm)

4.3.Economic indicators :

A. Net present value (NPV):

Net present value is used to analyze the profitability of a project or investment. It is calculated by the difference between the present value of cash inflows and the present value of cash outflows during the life of the project (from the project over its life). Figure (49) shows net present value (NPV) with a load for each of (Gasoline, T30, T45, T90, VM) , where we note that the lowest (NPV) was the mixer type (T90) was(77219.5), and the highest (NPV) when carrying the mixer type (T45) was (106900.7).

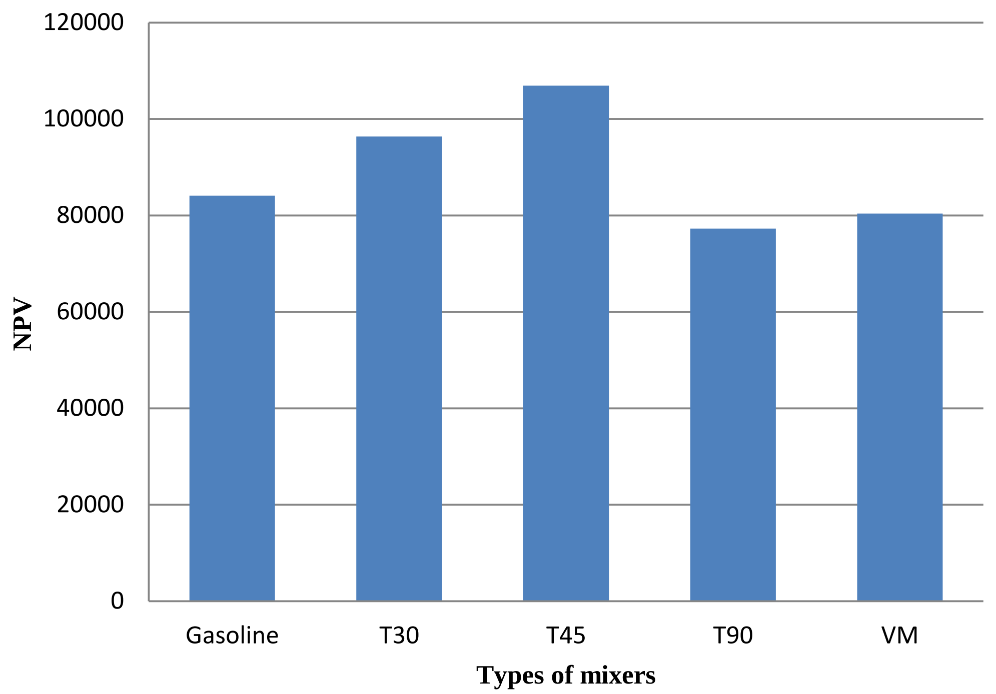


Figure (49): Net present value (NPV) for all types of mixers (gasoline – T30 – T45 – T90 - VM)

Figure (50) shows net present value (NPV) with a load for each of (Gasoline, L8, L10, L12) , where we note that the lowest (NPV) was the mixer type (L12) was(78791.4), and the highest (NPV) when carrying the mixer type (L10) was (108893.8).

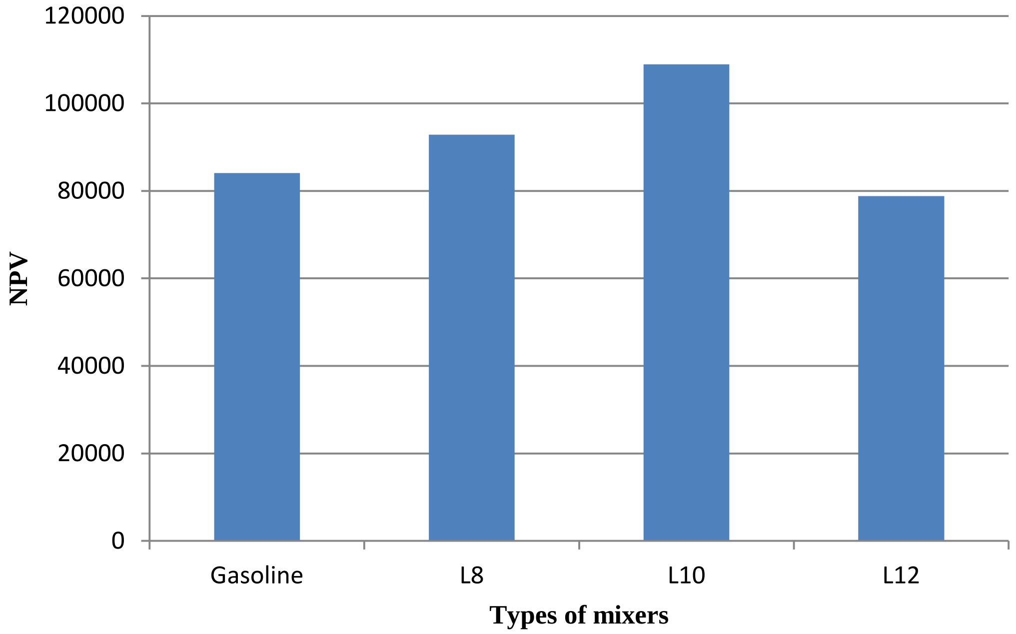


Figure (50): Net present value (NPV) for all types of mixers (gasoline – L8 – L10 – L12)

B. Benefit cost ratio (B/C):

Benefit cost ratio is used to analyze the profitability of a project or investment. It is calculated by dividing the present value of the cash inflows and the present value of the cash outflows during the life of the project (from the project over its life). Figure (51) shows benefit cost ratio (B/C) with a load for each of (Gasoline, T30, T45, T90, VM) , where we note that the lowest (B/C) was the mixer type (VM) was(1.38), and the highest (B/C) when carrying the mixer type (T45) was (1.54).

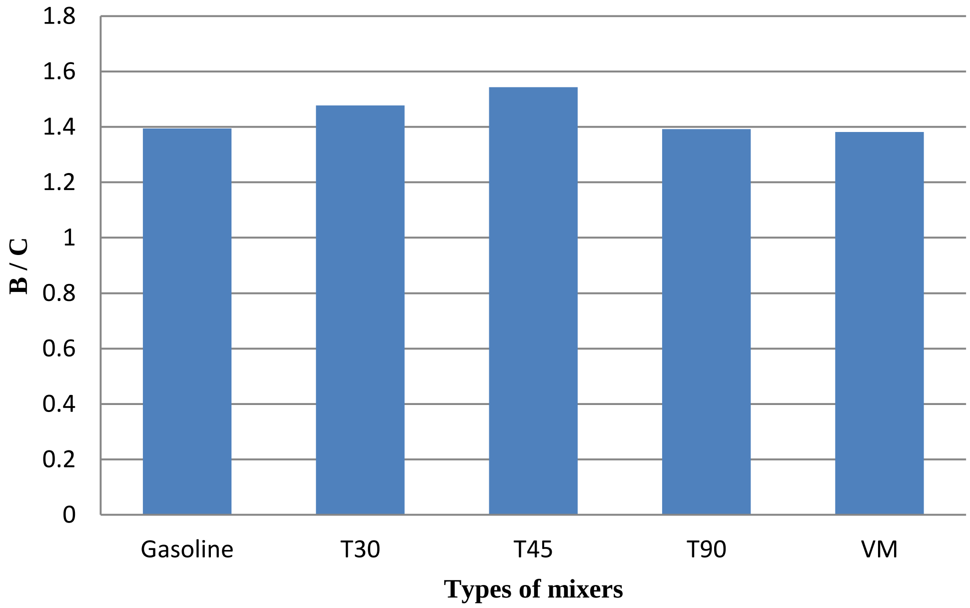


Figure (51): Benefit cost ratio (B/C) for all types of mixers (gasoline – T30 – T45 – T90 - VM)

Figure (52) shows benefit cost ratio (B/C) with a load for each of (Gasoline, L8, L10, L12) , where we note that the lowest (B/C) was the mixer type (Gasoline) was(1.39), and the highest (B/C) when carrying the mixer type (L10) was (1.56).

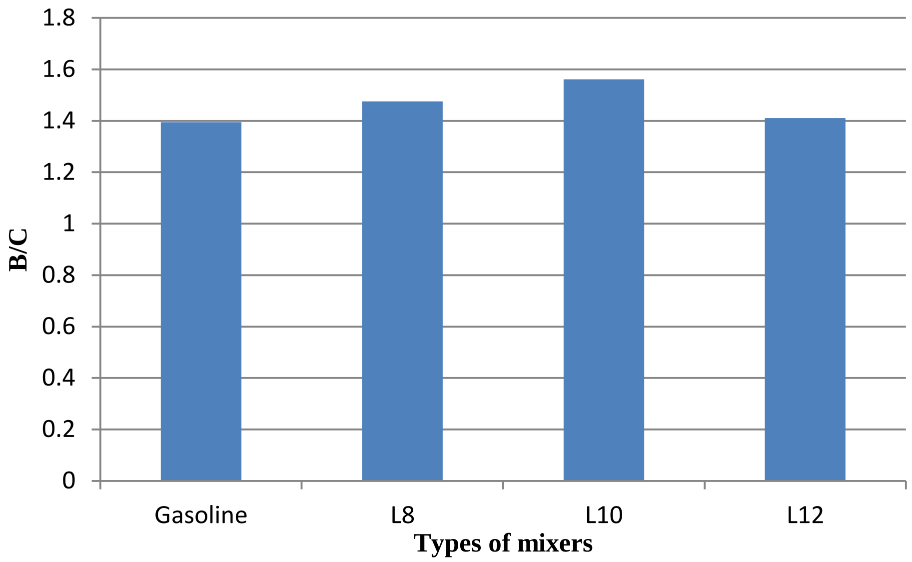


Figure (52): Benefit cost ratio (B/C) for all types of mixers (gasoline – L8 - L10 – L12)

C. Internal rate of return (IRR):

Economic metric is the internal rate of return this is the rate of interest that equates the present value of benefits to the present value of costs. Figure (53) shows internal rate of return (IRR) with a load for each of (Gasoline, T30, T45, T90, VM) , where we note that the lowest (IRR) was the mixer type (VM) was (0.42), and the highest (IRR) when carrying the mixer type (T45) was (0.59).

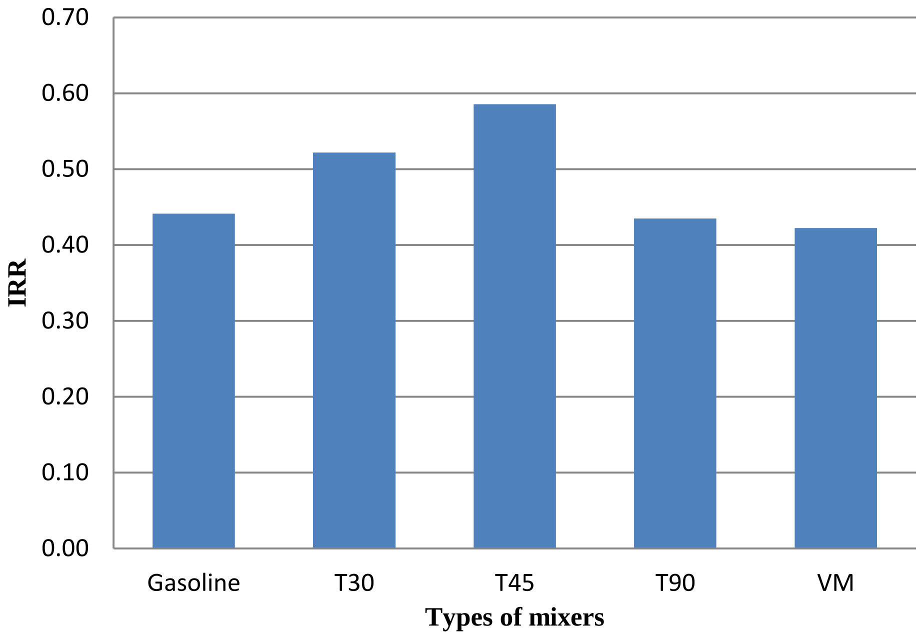


Figure (53): Internal rate of return (IRR) for all types of mixers (gasoline – T30 – T45 – T90 - VM)

Figure (54) shows internal rate of return (IRR) with a load for each of (Gasoline, L8, L10, L12) where we note that the lowest (IRR) was the mixer type (Gasoline) was(0.44), and the highest (IRR) when carrying the mixer type (L10) was (0.60).

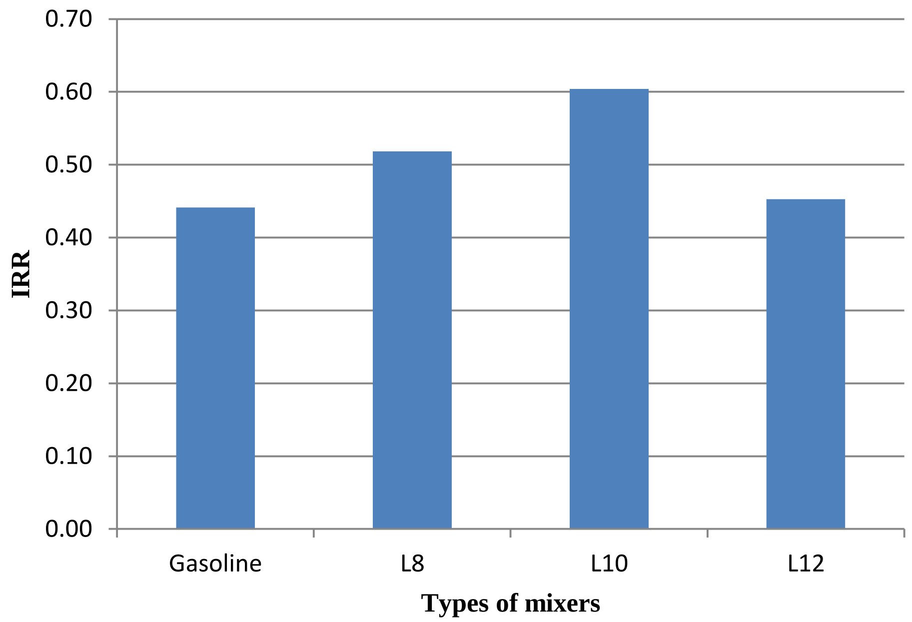


Figure (54): Internal rate of return (IRR) for all types of mixers (gasoline – L8 – L10 – L12)

D. Payback period:

The time needed for the project to recover the net return (benefits) the cost of the capital investment for the project. In other words, the time needed for the cumulative cash flows to equal investment costs. Figure (55) shows the capital payback period with a load for each of (Gasoline, T30, T45, T90, VM), where we note that the lowest payback period was the type of mixer (T45) was (1.7 year), and the highest payback period When the (VM) mixer, it was (2.37 year).

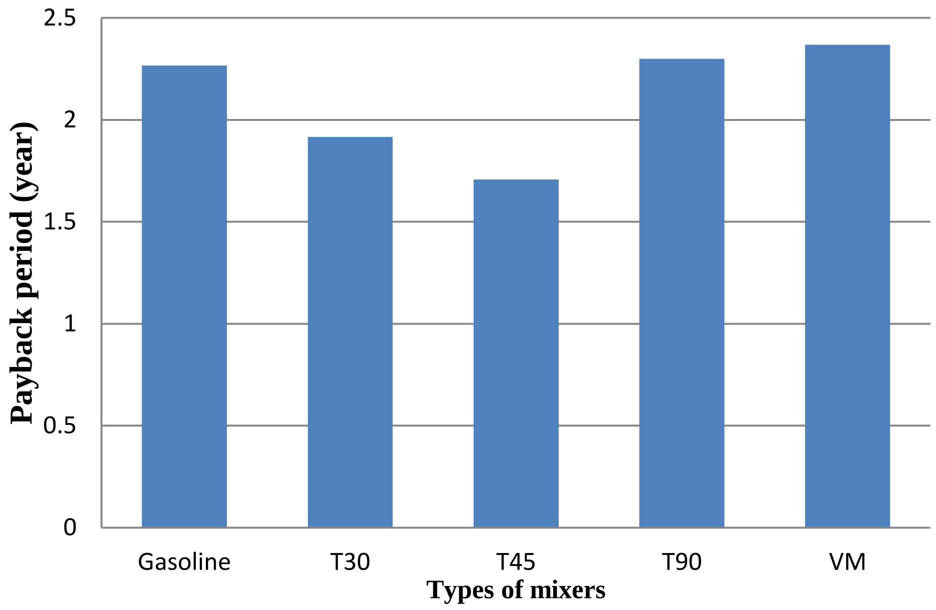


Figure (55): Payback period (year) for all types of mixers (gasoline – T30 – T45 – T90 - VM)

Figure (56) shows the capital payback period with a load for each of (Gasoline, L8, L10, L12) where we note that the lowest payback period was the type of mixer (L10) was (1.66 year), and the highest payback period When the (Gasoline) mixer, it was (2.27 year).

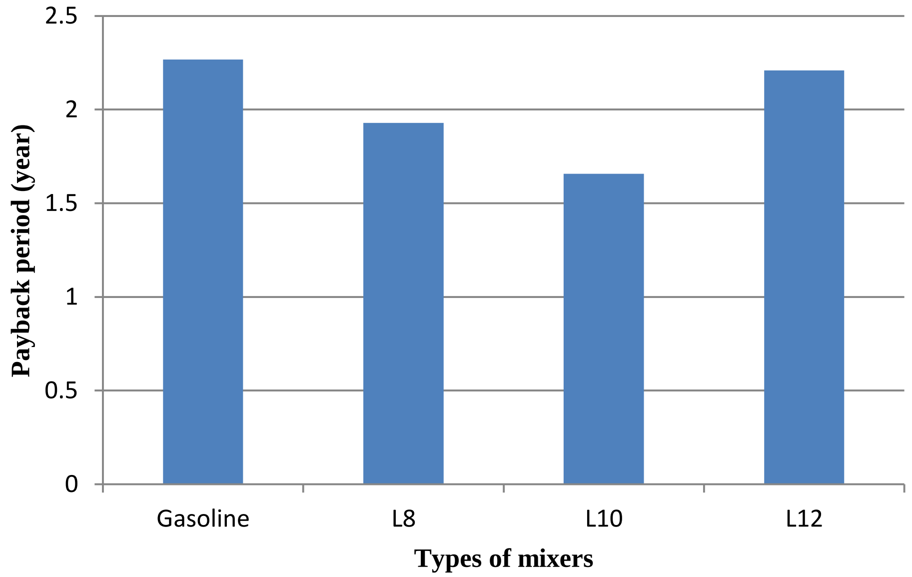


Figure (56): Payback period (year) for all types of mixers (gasoline – L8- L10- L12)

E. Economic efficiency:

One of the economic efficiency criteria is the criterion of return of the pound from energy costs, which means the total return of one pound of the cost of energy used in the irrigation process, and this criterion is used when comparing the returns on investment costs from the items of partial costs of the irrigation process It is calculated by dividing the total value of production revenue in pounds by the total value of the energy costs used in irrigation in pounds. Figure (57) shows the economic efficiency with a load for each of (Gasoline, T30, T45, T90, VM), where we note that the lowest economic efficiency was the type of mixer (T90) and gasoline was (1.65), and the highest economic efficiency When the (T45) mixer, it was (1.83).

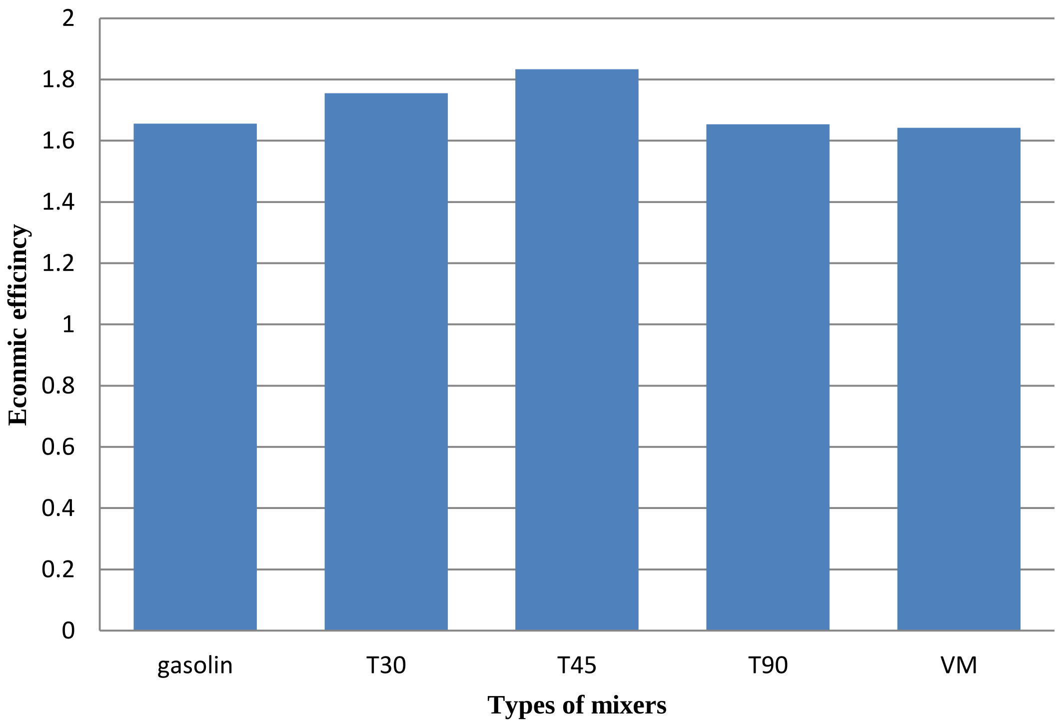


Figure (57): The economic efficiency for all types of mixers (gasoline – T30- T45- T90- VM)

Figure (58) shows the economic efficiency with a load for each of (Gasoline, L8, L10, L12), where we note that the lowest economic efficiency was the type of gasoline was (1.65), and the highest economic efficiency When the (L8) mixer was (1.85).

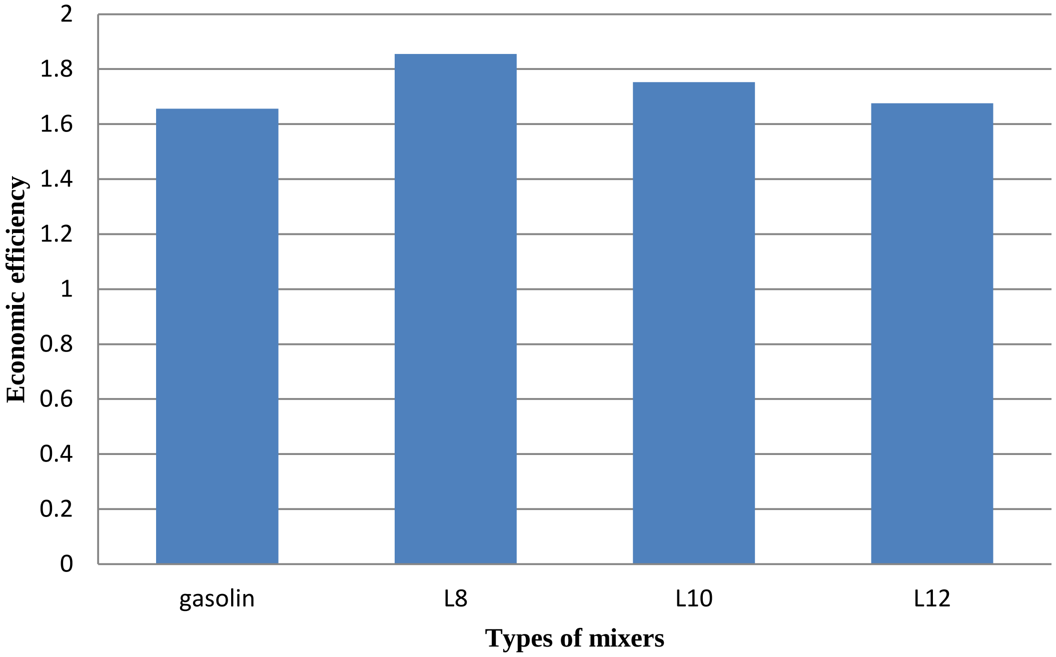


Figure (58): The economic efficiency for all types of mixers (gasoline – L8- L10- L12)

SUMMARY

Energy demand in the Egyptian countryside is high and growing as a direct result from economic development and population growth. Because of depleting fossil fuel resources, alternatives to petroleum derived fuels for the internal combustion engines need to be found. Compressed natural gas (CNG) can be such an alternative fuel because is much more abundant than petroleum **(Andrei *et al.*, 2019).** In recent years, natural gas has been viewed as a clean alternative fuel for Spark Ignition (SI) engines due to its relatively high octane number. The light combustion of natural gas, which contains mostly methane, in SI engines would have improved thermal efficiency and reduced emissions compared to gasoline.

5.1. Objective:

1. Design a gas mixing device natural gas into air stream.
2. Convert SI engine to power a water pump for irrigation by using natural gas.
3. Compare the output power of using natural gas with fuel gasoline.
4. Reducing environmental pollution resulting from exhaust gas.
5. Reducing the costs needed to operate the engine and irrigation pump.

5.2. Experimental factors:

Through the preliminary experiments, the experimental factors can be determined in the average of the experiments, and the factors were as follows :

1. Type of fuel: two types of fuel were used (gasoline - natural gas) to run the engine and the water pump.
2. Operating the engine: the pump was operated once with a load and once without a load for all used speeds and for all types of mixers and types of fuel (gasoline - natural gas).
3. Engine speed: where four speeds were used for the motor shaft (1750-2300-2900-3500 rpm)
4. Gas mixing method: seven types of mixers were used (90 angle T mixer - 45 angle T mixer - 30 angle T mixer - venture mixer - 8cm perforated inner tube mixer - 10cm perforated inner tube mixer - Mixer With a perforated inner tube of length 12 cm)and it can be expressed by (T90, T45, T30, VM, L8, L10, L12).

5.3. Experimented measurements:

Many measurements were made in the experimental workshop at the Agricultural Engineering Department, Faculty of Agriculture, Ain Shams University, Egypt.

5.3.1. Engine power:

Where results guarantee that the relationship between engine speed and actual power is a quadratic relationship, As we note with the increase in the engine speed, the power increases in all types of mixers. The actual power is superior to all types when operating with gasoline A comparison with the use of natural gas, where the mixer type (T45) gave the highest power compared to the types of mixers (2.83 kW) at an engine speed of (3500 rpm) was 7.5% less than gasoline.

5.3.2. Specific fuel consumption:

Where results guarantee that he lowest (S.fc) at an engine speed of 2900 using gasoline was (219.025 gm/kW.h) and using natural gas when the type of mixer T45 was (234.61 gm/kW.h) an increase of 6.6 % over gasoline. The lowest (S.fc) at an engine speed of 2900 rpm using natural gas when the type of mixer L10 was (340.144 gm/kW.h) an increase of 35.6 % over gasoline.

5.3.4. Exhaust gases:

The results guarantee the percentage of CO_2 exhaust gases when operating with gasoline exceeding all types it was (7.96%) and the lowest percentage carbon dioxide was (4.22%) using mixer type L12. The results the percentage of CO exhaust gases when operating with gasoline exceeding all types it was (0.53%) and the lowest percentage carbon monoxide was (0.18%) using mixer type T30.

5.3.5.Power and discharge of the pump:

The hydraulic power when the engine is operating with gasoline and natural gas for all types of mixers at an engine speed of (3500 rpm), the highest hydraulic power was when operating with gasoline (0.89 kW), then the mixer T45 was (0.798 kW), while lowest the hydraulic power was at the L12 mixer (0.473kW).

The results guarantee the relationship between pressure and pump discharge is inverse, as the higher the pressure, the lower the pump discharge. Maximum discharge using gasoline

($32.38\ m^3/h$) at pressure (0.2bar) and minimum discharge ($5.66\ m^3/h$) at pressure (1.8 bar) at engine speed (3500 rpm)

The results guarantee maximum discharge using T45 mixer ($33.09\ m^3/h$) at pressure (0.2bar) and minimum discharge ($3.60\ m^3/h$) at pressure (1.8bar). Maximum discharge using L12 mixer ($29.51\ m^3/h$) at pressure (0.2bar) and minimum discharge ($2.18\ m^3/h$) at pressure (1.8bar). at engine speed (3500 rpm)

5.4. Economic indicators:

- **Net present value (NPV):**

Net present value (NPV) with a load for each of (Gasoline, T30, T45, T90, VM) that the lowest (NPV) was the mixer type (T90) was(77219.5), and the highest (NPV) when carrying the mixer type (T45) was (106900.7). Net present value with a load for each of (Gasoline, L8, L10, L12) that the lowest (NPV) was the mixer type (L12) was(78791.4), and the highest (NPV) when carrying the mixer type (L10) was (108893.8).

- **5.6.2.Benefit cost ratio (B/C):**

Benefit cost ratio (B/C) with a load for each of (Gasoline, T30, T45, T90, VM) that the lowest (B/C) was the mixer type (VM) was(1.38), and the highest (B/C) when carrying the mixer type (T45) was (1.54). Benefit cost ratio (B/C) with a load for each of (Gasoline, L8, L10, L12) that the lowest (B/C) was the mixer type (Gasoline) was(1.39), and the highest (B/C) when carrying the mixer type (L10) was (1.56).

- **Payback period:**

The time needed for the project to recover the net return (benefits) the cost of the capital investment for the project. the capital payback period for each of (Gasoline, T30, T45, T90, VM) that the lowest payback period was the type of mixer (T45) was (1.7 year), and the highest payback period When the (VM) mixer, it was (2.37 year). the capital payback period for each of (Gasoline, L8, L10, L12) where we note that the lowest payback period was the type of mixer (L10) was (1.66 year), and the highest payback period When the (Gasoline) mixer, it was (2.27 year).

- **Economic efficiency:**

The results guarantee the lowest economic efficiency was the type of mixer (T90) and gasoline was (1.65), and the highest economic efficiency When the (T45) mixer, it was (1.83). The economic efficiency When the (L8) mixer was (1.85).

CONCLUSION:

From obtained results of this study to solve the deterioration of fuel economy by using natural gas , The goal of tis to improve the fuel/air mixing and combustion process, this study and the main conclusions were summarized as below:

1. Gasoline engines can be converted to work with natural gas with an efficiency (Braking power) of up to 90% .
2. The highest pump discharge was with the T45 mixer (33.09 m3/h), an increase of 2.1% over gasoline at an engine speed of (3500 rpm) .
3. The highest actual power with L10 type mixer compared to other type L mixers (2.69 kW) was 10 % lower than that of gasoline.
4. The highest actual power with T45 type mixer compared to other type T mixers (2.83 kW) was 7.5% less than gasoline.
5. As for the economic indicators, the use of natural gas gave a good economic return for all mixers, and the best economic indicators was for the mixer (L10) an increase of 26.6 % over gasoline.

Recommendations:

We recommended that :

- Use (L10) mixer to reduce fuel consumption and increase the engine power and actual hydraulic power to mix natural gas with air.
- Use (T45) mixer to reduce fuel consumption and increase the engine power and actual hydraulic power to mix natural gas with air, It gave better results than the (L 10) mixer.
- We recommended that in further testing to measure engine life using natural gas due to its low environmental impact and cheap price.

REFERENCES

Abbas, R.T.; A.A. Hamidi and H. Ghadamian., (2017). Experimental investigation of CNG and gasoline fuels combination on a 1.7 L bi-fuel turbocharged engine. Int J Energy Environ Eng 2017, (8): 37–45.

Abdel-Galil, A.; M.M. Mostafa; M.A. Elnono and M.F. Mohamed., (2008). Biogas utilization for powering water irrigation pump biological engineering. Misr J. Ag. Eng., 25 (4): 1438-1453.

Abo-Qudais, S. and H.A. Qdais., (2005). Performance evaluation of vehicles emissions prediction models. Clean Techn Environ Policy. 2005, (7): 279–284.

Afgan, N.H.; D. Al-Gobaisi; M.G. Carvalho and C. Maurizio., (1998). Sustainable energy development. Renew Sustain Energy Rev 1998, (2): 186 –235.

Ahmet, Y. and B. Rasim., (2021). Effect of different types of fuels tested in a gasoline engine on engine performance and emissions. international journal of hydrogen energy 2021, (46): 33325 -33338.

Ali, M.P. and H. Amir., (2009). Shamekhi and Farhad Salimi. Alternative fuel and gasoline in an SI engine: A comparative study of performance and emissions characteristics. Elsevier, Issue 5 Fuel Vol. (89): 1056-1063.

Andreass, S.C.; V. Rocco; M. Gambino and S. Iannaccone., (2001). Analysis of Combustion Instability Phenomena in a CNG fueled heavy-duty turbocharged engine (No. 2001-01-1907). SAE Technical. pp. 1-12.

Andrei, L.N.; M. Lucian and R. Chiriaca., (2019). On the possibility to simulate the operation of a SI engine using alternative gaseous fuels. Energy Reports 2020, (6): 167–176 .

Ani, T.R; S.D. Ravi; M. Shashikanth; P.G. Tewari and N.K. Rajan., (2006). CFD Analysis of a Mixture Flow in a Producer Gas Carburetor'. International Conference on Computational Fluid Dynamics, Acoustics, Heat Transfer and Electromagnetics Andhra University, Visakhapatnam – 530003, India. 2006, (6): 20-25.

REFERENCES

Awulachew, S.B.; P. Lemperiere and T. Tulu., (2009). Pumps for small-scale irrigation". Improving Productivity and Market Success (IPMS) of Ethiopian farmers project International Livestock Research Institute (ILRI), Addis Ababa, Ethiopia. 2009, pp. 128- 165.

Bade, S.O. and G. Narayanan., (2008). Landfill gas with hydrogen addition – A fuel for SI engines". Fuel, 87 (17-18): 3616–3626.

Ban H.A.; A. J. Shatha and M. I. Feryal., (2011). Evaluation of performance efficiency using some productivity indicators. Alustath Journal for Human and Social Science. Issue 140, Volume (6): 169-192.

Berckmüller, M.; H. Rottengruber; A. Eder; N. Brehm; G. Elsässer; G. Müller-Alander, and C. Schwarz., (2003). Potentials of a charged SI-hydrogen engine. SAE, (No. 2003-01-3210). SAE Technical Paper. 11.p.

Carlo, C.; U.Z. Baškovi; M. Renzi; T. Seljak; S.R. Oprešnik; M. Baratieri and T. Katrašnik., (2021). Complementing syngas with natural gas in spark ignition engines for power production: Effects on emissions and combustion. Energies, 14 (12): 1-12.

Chaichan, M.T., (2012). Characterization of lean misfire limits of alternative gaseous fuels used for spark ignition engines. Tikrit Journal of Engineering Sciences. 19 (1): 50 - 61.

Chaichan, M.T., (2014). Evaluation of the effect of cooled EGR on knock of SI engine fueled with alternative gaseous fuels. Sulaimani Journal for Engineering Science, 1 (1): 7-15.

Chandra, R.; V.K. Vijay; P.M. Subbarao and T.K. Khura., (2011). Performance evaluation of a constant speed engine on CNG, methane enriched biogas and biogas. Applied Energy, 88 (11): 3969 –3977.

Cheolwoong, P.; S. Park; Y. Lee; C. Kim; L. Sunyoup and Y. Moriyoshi., (2011). Performance and emission characteristics of a SI engine fueled by low calorific biogas blended with hydrogen. International Journal of Hydrogen Energy, 36 (16): 10080–10088.

Cho, H; M. He and B.Q., (2007). Spark ignition natural gas engines—a review. Energy Convers. Manag. 2007, (48): 608–618.

Corbin, J.C.; W. Peng; J. Yang; D.E. Sommer; U. Trivanovic; P. Kirchen; J.W. Miller; S. Rogak; D.R. Cocker and G.J. Smallwood., (2020). Characterization of particulate matter emitted by a marine engine operated with liquefied natural gas and diesel fuels. Atmospheric Environment, 117030. 220, 29.p.

Crookes, R. J. (2006). Comparative bio-fuel performance in internal combustion engines. Biomass and Bioenergy, 30 (5): 461-468.

Dashti, M.; H.A. Asghar and M.A. Asghar., (2012). Thermodynamic model for prediction of performance and emission characteristics of SI engine fuelled by gasoline and natural gas with experimental verification. J. Mech. Sci. Technol. 26 (7): 2213 –2225.

Deublein, D. and A.Steinhauser., (2008). Attachment I: typical design calculation for an agricultural biogas plant. In Biogas from Waste Renew. Resour. Wiley-VCH Verlag GmbH & Co. KGaA Weinheim, Germany. 2008, pp. 407- 414.

Dhyani, V. and K.A. Subramanian., (2019). Experimental based comparative exergy analysis of a multi-cylinder spark ignition engine fuelled with different gaseous (CNG, HCNG, and hydrogen) fuels. Int. J. Hydrogy Energy 2019, (44): 20440–20451.

El-Ashry, E.S., (2003). Biogas generation from agricultural residues, Misr J. Ag. Eng. 18, (3): 741-753.

El-Emam, S.H., (2004). An introduction to combustion engineering. Text of handbook, Department of Mechanical Power Engineering, Mansoura University. Egypt. 2004, 178.p.

Faizala, M.; M. H. Hamzah and A. Navaretsnasinggam., (2009). Review of fuels for internal combustion engines in the aspect of economy, performance, environment and sustainability. 2009, pp. 1-47.

Fathollah, O.; M. Ehsan and N. Kouros., (2013). Comparing performance characteristics of a gasoline and cng engines and increasing volume efficiency and power using designed turbocharger. ACTA TECHNICA CORVINIENSIS – Bulletin of Engineering 2013. Fascicule 2 (4–6): 101- 108.

Fragoso, R. and C. Marques., (2013). The Economic Impact of Alternative Water Pricing Policies in Alentejo Region (No. 2013_02). University of Evora, CEFAGE-UE (Portugal). https://ideas.repec.org/p/cfe/wpcefa/2013_02.html.

Garg, S.K., (1989). Irrigation engineering and hydraulic structures. Khama Publishers, New Delhi, India. (8). 1291.p.

Hannu, J., (2020). Natural Gas Engines. DieselNet Technology Guide. Copyright © ECOpoint Inc. Revision 2020, pp: 1-6.

Harldson, L., (2010). Engine technical adjustment, EffShip Workshop on alternative marine fuels, 2010, (16): 1- 20.

Heywood, J.B., (2018). Internal Combustion Engine Fundamentals; McGraw-Hill Education: New York, NY, USA, 2018. 78.p.

House, D., (1981). Biogas Handbook. peace press, Inc Culver City, California 90230 USA. 1981, 135.p.

Huanga, J. and R.J. Crookesb., (1998). Assessment of Simulated Biogas as a Fuel for the Spark Ignition Engine. Fuel; 77, (15): 1793-1801.

IRENA., (2018). Biogas for road vehicles: Technology brief. International Renewable Energy Agency, Abu Dhabi. 2018, 125.p.

Jahirul, M.I.; H.H. Masjuki; R. Saidur; M.A. Kalam; M.H. Jayed and M.A. Wazed., (2010). Comparative engine performance and emission analysis of CNG and gasoline in a retrofitted car engine". Applied Therm. Eng., 2010, (30): 2219-2226.

Jeongwoo, L.; P.Cheolwoong; B. Jongwon; K. Yongrae; L. Sunyoup and K. Changgi., (2020). Comparison between gasoline direct injection and compressed natural gas port fuel injection under maximum load condition. Journal Pre-proof. Energy. 2020, (2): 1-37.

Kalam, M.A.; H.H. Masjuki; M.A. Maleque; M.A. Amalina and H. Abdesselam., (2005). Power improvement of a modified natural gas engine. Department of Mechanical Engineering, University of Malaya, (2004), pp. 1-9.

Khan, M.I.; T. Yasmeen; M.I. Khan; M. Farooq and M. Wakeel., (2016). Research progress in the development of natural gas as fuel for road vehicles: A bibliographic review (1991–2016). Renewable and Sustainable Energy Reviews, 2016, (66): 702-741.

khurmi, R.S. and J.K. Gupta., (1998). A textbook of thermal engineering. S . Chand and Company LTD. Ram Nagar, New Delhi- 110055. pp. 289-311.

Kichiro, K.; I. Kohei; M. Michihiko; O. Katsuji; Y. Akio and T. Keiso., (1999). Development of Engine for Natural Gas Vehicle. Journal of Engines-V108-3 SAE Technical paper. 1999, pp. 939 -947.

Klaus, V.M., (1988). Engines for Biogas. Theory, modification, economic operation. A Publication of Deutsches Zentrum für Entwicklungstechnologien - GATE in: Deutsche Gesellschaft für Technische Zusammenarbeit (GTZ) GmbH, 1988, 133.p.

Konigsson, F.; P. Stalhammar and H.E. Angstrom., (2011). Combustion modes in a diesel-CNG dual fuel engine. SAE paper 2011, (1): 1962-2011.

Korakianitis, T.; A.M. Namasivayam and R.J. Brookes., (2011). Natural-gas fueled Spark-Ignition (SI) and Compression-Ignition (CI) engine performance and emissions". Prog. Energy Combust. Sci., 37 (1): 89 -112.

Kutlar, O.A.; H. Arslan and A.T. Calik., (2005). Method to improve efficiency of four stroke, spark ignition engine at part load. Elsevier, Energy Conversion & Management, 2005, (46): 3202-3220.

Lee, K.; K. Taesoo; C. Hyoseok; S. Soonho and M.C. Kwang., (2010). Generating efficiency and NO_x emissions of a gas engine generator fueled with a biogas–hydrogen blend and using an exhaust gas recirculation system. International Journal of Hydrogen Energy, 35 (11): 5723–5730.

Li, Y.; K. Nithyanandan; T. H. Lee; R.M. Donahue; Y. Lin; C.F. Lee and S. Liao., (2016). Effect of water-containing acetone–butanol–ethanol gasoline blends on combustion, performance, and emissions characteristics of a spark-ignition engine. Energy Conversion and Management, 2016, (117): 21-30.

Ma, F.; N. Naeve; M. Wang; L. Jiang; R. Chen and S. Zhao., (2010). Hydrogen-enriched compressed natural gas as a fuel for engines, in: Natural gas, 2010, pp. 307-333.

Macmillan, R.H., (2002). The Mechanics of Tractor – Implement Performance: Theory and Worked examples. A textbook for students and engineers. agricultural Engineering International Development Technologies Centre University of Melbourne. 2002, (44): 1-4.

Marie, B., (2007). Engine characteristics of emissions and performance using mixtures of natural gas and hydrogen. Energy, 32, (4): 482–489.

Mccauley, D.S.; R. Anderson; R. Bowen; I. Elassiouty; E. Mahdy; I. Solima and H. Shehab., (2002). Economic Instruments for Improved Water Resources Management in Egypt: Summary Progress Report, Exploring the Potential of Applying Economic Instruments to Water Resources Management in Egypt. Cairo, Egypt. 2002, 173.p.

Michael, A.M., (1990). Irrigation: Theory and practice. Vikas Publishing House, New Delhi, India. 1990, 801.p.

Mika, L.; G. Giota; T. Anna; J. Suraj; L. Tuomo and S. Jussi., (2022). General-Purpose and Scalable Internal-Combustion Engine Model for Energy-Efficiency Studies. Machines 2022, (10): 1- 26.

Miqdam, T.C.; A.K. Jaafar and S.R. Khalid., (2016). Spark Ignition Engine Performance When Fueled with NG, LPG and Gasolin. Saudi J. Eng. Technol. Iss-3, (1): 105-116.

Mitzlaff, V., (1988). Engines for biogas. German Appropriate Technology Exchange (GATE), Deutsche Gesellschaft für Technische Zusammenarbeit (GTZ). 1988, pp. 1-132.

MPMAR., (2016). Egypt Vision 2030 – Sustainable Development Strategy (SDS). Ministry of Planning, Monitoring and Administrative Reform (MPMAR), Cairo, Egypt. 2016. pp. 171-216.

MPMAR., (2017). The National Socio-Economic Development Plan for Fiscal Year 2016/2017, Ministry of Planning and Follow-up and Administrative Reform (MPMAR). Cairo, Egypt. 2017, pp. 1-59.

Mustafa, K.F. and H.W. Gitano-Briggs., (2009). Liquid Petroleum Gas (LPG) as an alternative fuel in spark ignition engine: Performance and emission characteristics. IEEE Energy and Environment . 2009, pp. 189-194.

Nafiz, K.; C. Bilge; S.O. Akansu and A. Kadir., (2009). Investigation of combustion characteristics and emissions in a spark-ignition engine fueled with natural gas-hydrogen blends. International Journal of Hydrogen Energy, 34 (2): 1026–1034.

Nguyen, X.K.; K. Yujin and L. Ocktaeck., (2019). The effects of combustion duration on residual gas, effective release energy and engine power of motorcycle engine at full load.10th International Conference on Applied Energy (ICAE2018), 22-25 August 2018, Hong Kong, China. Energy procedia. 2019, (158) 1835 –1841.

Nindhia, T.G., (2012). Removal of Hydrogen Sulfide (H_2S) contaminant in Biogas by Utilizing Solid Waste Steel Chips from The Process of Turning'. The Twenty-seventh International Conference on Solid Waste Technology and Management, Philadelphia, 2012, pp. 4-11.

Nippon, K., (2015). Guideline for Biogas Generation. The Project for Establishment of Rural. Electrification Model Using Renewable Energy. Ministry of Energy and Petroleum. Project Completion Report. 2015, 154.p.

NYMEX., (2018). Natural gas price at the New York Mercantile Exchange, converted by 0.035315. $MMBTU/m^3$: See: www.finanzen.net/rohstoffe/Erdgas-Preis-Natural-Gas.

Paola, H. and R. Jose., (2009). Evaluating carbon emissions reduction by use of natural gas as engine fuel. Journal of Natural Gas Science and Engineering 2009, (1): 216–220.

Papacz, W., (2011). Biogas as Vehicles Fuel. J Kones Powertrain and Transport 2011, (18): 11- 403.

Park, C.; P. Seunghyun; L. Yonggyu; K. Changgi; L. Sunyoup and M. Yasuo., (2011). Performance and emission characteristics of a SI engine fueled by low calorific biogas blended with hydrogen. International Journal of Hydrogen Energy, 36 (16): 10080–10088.

Pipitone, E. and G. Genchi., (2014). Experimental Determination of Liquefied Petroleum Gas– Gasoline Mixtures Knock Resistance. Journal of Engineering for Gas Turbines and Power, 136 (12): 1-7.

Porpatham, E.; A. Ramesh and A. Nagalingam., (2007). Investigation on the effect of concentration of methane in biogas when used as a fuel for a spark ignition engine. Fuel, 87 (8-9):1651–1659.

Pranav, V.; Kherdekar and B. Divesh., (2017). Simulation of a spark ignited hydrogen engine for minimization of NO_x emissions, International Journal of Hydrogen Energy, 42. (7): 4579- 4596.

Pruthviraj, N.B., (2016). Introduction To Biogas & Applications. International Journal of Advanced Research in Mechanical Engineering and Technology (IJARMET). Issue 4, (2): 2454-8723.

Robert, A., (2014) A Study on Biogas-Fueled SI Engines: Effects of Fuel Composition on Emissions and Catalyst Performance. A thesis for the degree of Master. Department of Mechanical and Industrial Engineering University of Toronto.2014, 90.p.

Saad, A.; F.M. Mohd; A. Shahrir and A. Yusoff., (2014). Comparison of Performance and Emission of a Gasoline Engine Fuelled by Gasoline and CNG Under Various Throttle Positions. Journal of Applied Sciences 14 (4): 386-390.

Saravanan, V.S.; P.S. Utgikar and S.L Borse., (2013). Experimental Study on Conversion of 4 Stroke Gasoline Internal Combustion Engine into Enriched Compressed Natural Gas Engine To Achieve Lower Emissions. Engineering Research and Applications (IJERA) ISSN: 2248-9622 www.ijera.com., Issue 4, (3): 1103-1110.

Stefan, M., (2004). Biogas fuel for internal combustion engines. Annals of the faculty of engineering hunedoara – 2004. Fascicole 3. (2): 179-190.

Sulaiman, M.Y.; M.R. Ayob and I.a. Meran., (2013). Performance of Single Cylinder Spark Ignition Engine Fueled by LPG. Procedia Engineering, 2013. (53): 579 – 585.

Surata, I.W.; T.N. Tjokorda; I.A. Ketut; N.N. Dewa and I. Wayan., (2014). International Conference on Alternative Energy in Developing Countries and Emerging Economies. Energy Procedia. 2014, (52): 626 – 632.

Taggart, C.G.; H. Jones and S. Rogak., (2006). Directinjected hydrogen-methane mixtures in a heavy-duty compression ignition engine. SAE Technical Paper. 2006, (65): 1-13.

Tang, X.; D.M. Kabat; R.J. Natkin; W.F. Stockhausen and J. Heffel., (2002). Ford P2000 hydrogen engine dynamometer development. SAE Transactions, 2002, pp. 631-642.

Tilagone, R.; S. Venturi and G. Monnier., (2005). Natural gas-an environmentally friendly fuel for urban vehicles: the smart demonstrator approach. Oil & gas science and technology, 61 (1): 155-164.

Tjokorda, G.T.; M. Morag and S. David., (2021). Greenhouse gas mitigation and rural electricity generation by a novel two-stroke biogas engine. Journal of Cleaner Production. 2021, 280. pp. 1- 12.

Tsur, Y. and A. Dinar., (1995). Efficiency and Equity Considerations in Pricing and Allocating Irrigation Water (No. 1460), World Bank Policy Research Working Paper. Washington, D.C., U.S.A. 1995, 42.p.

Umierski, M. and P. Stommel., (2000). Fuel efficient natural gas engine with common-rail micro-pilot injection. SAE paper. 2000, pp. 3137-3144.

Verhelst, S.; P. Maesschalck; N. Rombaut and R. Sierens., (2009). Efficiency comparison between hydrogen and gasoline, on a bi-fuel hydrogen/gasoline engine. International Journal of Hydrogen Energy., Issue 5, (34): 2504-2510.

Verhelst, S.; S. Verstraeten and R. Sierens., (2006). A critical review of experimental research on hydrogen fueled SI engines. SAE technical paper nr 2006-01-0430. Also in SAE 2006 Transactions Journal of Engines, 2006, pp. 264-274.

Wang, Z. and A. Krupnick., (2015). A Retrospective Review of Shale Gas Development in the United States: What Led to the Boom? Economics of Energy & Environmental Policy, 4. (1): 5 –18.

Wayan, S.; T.N. Tjokorda; I.A. Ketut; P.N. Dewa and I. Wayan., (2014). Simple Conversion Method from Gasoline to Biogas Fueled Small Engine to Powered Electric Generator. Energy Procedia. 2014, (52): 626–632.

Welch, A.; D. Mumford; S. Munshi; J. Holsbery; D. Boyer; M. Younhins and H. Jung., (2008). Challenges in developing hydrogen direct injection technology for internal combustion engines. SAE, 2008, 14.p.

Willis, H.T.; W. Raymond; J. David; E. Michael., (2018). The Basics of Project Evaluation and Lessons Learned, Thase, CRC Press, 2018, 214.p.

Yamasaki, Y.; M. Kanno; Y. Suzuki and S. Kaneko., (2013). Development of an Engine Control System Using City Gas and Biogas Fuel Mixture 2013, (101): 465-474.

Yasin, K.; S. Tarkan; U.O. Koylu; S.D. Ahmet and W. Somchai., (2016). Effect of the use of natural gas–diesel fuel mixture on performance, emissions, and combustion characteristics of a compression ignition engine. Advances in Mechanical Engineering. 4, (8): 1–13.

Zarante, P.H.B. and J.R. Sodre., (2009). Evaluating carbon emissioris reduction by use of riatural gas as engine fuel. J. Natural Gas Science and Engineering, 1 (6): 216-220.

Zhang, D., (1998). A numerical study of natural gas combustion in a lean burn engine. Fuel. 77 (12): 1339-1347.

Zheng, Q.P.; H.M. Zharig and D.F. Zharig., (2005). A computational study of combustion in compression ignition natural gas engine with separated chamber. Fuel, 84 (12-13): 1515-1523.

Zhunqing, H. and X. Zhang., (2011). Experimental study on performance and emissions of engine fueled with lower heat value gas–hydrogen mixtures. International journal of hydrogen energy, 37 (1): 1080-1083.

A. APPENDIX

Appendix (1): Engine speed and power (braking power) for each of (Gasoline, T30, T45, T90, VM).

RPM	Gasoline (kW)	T30 (kW)	T45 (kW)	T90 (kW)	VM (kW)
1750	1.3209	0.902317	0.83895	0.937125	0.847875
2300	1.79469	1.77123	1.73604	1.624605	1.352469
2900	2.89884	1.98186	2.39598	1.84875	1.885725
3500	3.0702	2.38	2.83815	2.372265	1.986705

Appendix (2): Engine speed and actual power (braking power) for each of (Gasoline, L8, L10, L12).

RPM	Gasoline (kW)	L8 (kW)	L10 (kW)	L12 (kW)
1750	1.3209	0.9639	0.9996	0.9996
2300	1.79469	1.468596	1.57182	1.82988
2900	2.89884	1.970028	2.20371	2.41077
3500	3.0702	2.17	2.69	2.536

Appendix (3): Engine speed and specific fuel consumption (S. fc) for each of (Gasoline, T30, T45, T90, VM).

RPM	(gasoline) (gm/kW.h)	(T30) (gm/kW.h)	(T45) (gm/kW.h)	(T90) (gm/kW.h)	(VM) (gm/kW.h)
1750	265.7279	424.8041	244.9825	404.8941	778.584
2300	234.6923	384.7252	236.778	369.7972	692.789
2900	219.0255	393.9799	234.613	372.4716	668.527
3500	226.109	533.1213	316.8983	373.2094	754.5967

Appendix (4): Engine speed and specific fuel consumption (s. fc) for each of (Gasoline, L8, L10, L12).

RPM	(gasoline) (gm/kW.h)	(L8) (gm/kW.h)	(L10) (gm/kW.h)	(L12) (gm/kW.h)
1750	265.7279	530.2185	411.8661	374.9402
2300	234.6923	522.0063	361.278	353.7745
2900	219.0255	453.9965	340.1448	360.396
3500	226.109	471.0393	424.3147	490.3873

Appendix (5): The average of exhaust gases for operating the pump with types of mixers (Gasoline, T30, T45, T90, VM).

Gas	gasoline	T30	T45	T90	VM
CO_2 %	7.963	5.453	4.665	2.778	4.340
CO %	0.538	0.180	0.315	0.243	0.343
HC pmm	31.500	40.750	39.000	17.250	28.500

Appendix (6): The average of exhaust gases for operating the pump with types of mixers (Gasoline, L8, L10, L12).

Gas	gasoline	L8	L10	L12
CO_2 %	7.963	5.835	4.243	4.225
CO %	0.538	0.223	0.248	0.323
HC pmm	31.500	48.250	23.750	34.750

Appendix (7): Pump pressure and discharge (m3/h) and hydraulic power when the engine is operating with gasoline at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
32.38	0.2	0.2392163
30.71	0.3	0.3403775
30.27	0.4	0.4473068
29.46	0.5	0.5441065
28.90	0.6	0.6405335
27.26	0.7	0.7049769
25.70	0.8	0.7595828
24.09	0.9	0.8010831
23.60	1	0.8720413
21.97	1.1	0.8929826
18.95	1.2	0.8400526
16.74	1.3	0.8040894
15.05	1.4	0.7786126
11.61	1.5	0.6431727
9.43	1.6	0.5575507
8.32	1.7	0.522482
5.66	1.8	0.3765586

Appendix (8): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with T30 mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
32.46	0.2	0.2398358
29.26	0.3	0.3242527
28.29	0.4	0.4179995
27.23	0.5	0.5030193
24.72	0.6	0.5478576
22.94	0.7	0.5931261
20.05	0.8	0.5926498
19.62	0.9	0.6522573
18.95	1	0.7000943
18.44	1.1	0.7494437
17.24	1.2	0.7642845
15.57	1.3	0.7478905
13.17	1.4	0.681312
11.37	1.5	0.6302006
8.99	1.6	0.5316276
6.77	1.7	0.4252526
5.01	1.8	0.3333166

Appendix (9): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with T45 mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
33.09	0.2	0.24451252
31.53	0.3	0.349412945
30.29	0.4	0.447623027
28.35	0.5	0.523746242
26.79	0.6	0.59385963
24.98	0.7	0.646081415
24.33	0.8	0.719038787
22.21	0.9	0.73852506
20.44	1	0.755052083
19.65	1.1	0.7986517
16.19	1.2	0.71785952
14.55	1.3	0.698628385
12.96	1.4	0.670158627
8.81	1.5	0.488399275
6.49	1.6	0.383873467
4.50	1.7	0.282730513
3.60	1.8	0.2394

Appendix (10): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with T90 mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
29.88	0.2	0.220792857
27.82	0.3	0.308367815
25.51	0.4	0.377047907
24.80	0.5	0.458181675
24.03	0.6	0.53259717
23.34	0.7	0.60372557
22.70	0.8	0.670954853
20.49	0.9	0.681270555
19.56	1	0.722580133
18.17	1.1	0.738271252
14.91	1.2	0.66082646
12.89	1.3	0.618858088
10.79	1.4	0.558094157
9.82	1.5	0.54438895
8.10	1.6	0.479016347
5.61	1.7	0.352391923
2.08	1.8	0.13851684

Appendix (11): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with VM mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
31.69	0.2	0.234138963
30.24	0.3	0.335203225
28.13	0.4	0.415708347
28.03	0.5	0.517761242
27.65	0.6	0.61281745
25.15	0.7	0.650478838
23.88	0.8	0.70571396
20.18	0.9	0.67113795
18.90	1	0.698360833
16.98	1.1	0.689855705
15.87	1.2	0.7033971
13.39	1.3	0.642862372
11.53	1.4	0.596469977
9.11	1.5	0.504718375
5.39	1.6	0.318341707
3.79	1.7	0.23783459
1.85	1.8	0.12291993

Appendix (12): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with L8 mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
32.98	0.2	0.2436737
30.00	0.3	0.3325
27.70	0.4	0.4093235
26.83	0.5	0.4955724
26.40	0.6	0.5850909
26.03	0.7	0.6732433
24.00	0.8	0.7093333
21.45	0.9	0.7131866
19.42	1	0.7173244
17.58	1.1	0.7144439
16.10	1.2	0.7135955
14.98	1.3	0.7194225
13.29	1.4	0.6875342
11.26	1.5	0.6241424
8.10	1.6	0.4789773
6.24	1.7	0.3916693
3.62	1.8	0.2404773

Appendix (13) Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with L10 mixer at engine speed (3500 rpm)

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
31.37	0.2	0.23178974
28.73	0.3	0.31842328
28.37	0.4	0.419309987
28.26	0.5	0.521956283
28.21	0.6	0.62525561
25.99	0.7	0.672059418
25.13	0.8	0.742872387
23.54	0.9	0.782564685
20.49	1	0.756967283
17.95	1.1	0.729268925
17.44	1.2	0.77338968
14.60	1.3	0.701227648
12.77	1.4	0.6604514
10.60	1.5	0.587278125
7.08	1.6	0.418240667
4.73	1.7	0.29691452
2.64	1.8	0.17550414

Appendix (14): Pump pressure and discharge (m^3/h) and hydraulic power when the engine is operating with T30 mixer at engine speed (3500 rpm).

Discharge(m^3/h)	Pressure (bar)	Power (Kw)
29.51	0.2	0.218032663
27.70	0.3	0.30699991
24.13	0.4	0.356627973
22.31	0.5	0.412111583
20.36	0.6	0.45136875
17.82	0.7	0.460950513
14.86	0.8	0.439199693
13.04	0.9	0.43346961
12.81	1	0.473309317
11.27	1.1	0.457838535
10.59	1.2	0.4694102
8.80	1.3	0.42246386
7.34	1.4	0.3797549
5.65	1.5	0.313138525
4.41	1.6	0.260516853
3.39	1.7	0.212812857
2.18	1.8	0.14484498

الملخص العربي

الطلب على الطاقة في الريف المصري مرتفع ومتزايد كنتيجة مباشرة للتنمية الاقتصادية والنمو السكاني. بسبب استنفاد موارد الوقود الأحفوري ، يجب إيجاد بدائل للوقود المشتق من البترول لمحركات الاحتراق الداخلي. يمكن أن يكون الغاز الطبيعي المضغوط (CNG) وقودًا بديلاً لأنه أكثر وفرة من البترول **(Andrei *et al.*, 2019)**. في السنوات الأخيرة كان يُنظر إلى الغاز الطبيعي على أنه وقود بديل نظيف لمحركات Spark Ignition (SI) نظرًا لعدد الأوكتان المرتفع نسبيًا. كان من شأن الاحتراق الخفيف للغاز الطبيعي ، الذي يحتوي في الغالب على الميثان في محركات الاحتراق الداخلي يحسن الكفاءة الحرارية ويقلل من الانبعاثات مقارنة بالبنزين.

1-5 الهدف من الدراسة :

الهدف من هذا البحث هو:

1- تصميم جهاز خلط الغاز الطبيعي مع تيار الهواء.
2- تحويل محرك(SI) لتشغيل مضخة مياه للري باستخدام الغاز الطبيعي.
3- مقارنة الطاقة الناتجة لاستخدام الغاز الطبيعي مع وقود البنزين.
4- الحد من التلوث البيئي الناتج عن غازات العادم.
5- تقليل التكاليف اللازمة لتشغيل المحرك ومضخة الري.

2-5 عوامل الدراسة:

من خلال التجارب الأولية يمكن تحديد العوامل التجريبية في متوسط التجارب ، وكانت العوامل على النحو التالي:

1- نوع الوقود: تم استخدام نوعين من الوقود (بنزين - غاز طبيعي) لتشغيل المحرك ومضخة المياه.
2- تشغيل المحرك: تم تشغيل المضخة مرة مع حمل ومرة بدون حمل لجميع السرعات المستخدمة ولجميع أنواع الخلاطات وأنواع الوقود (بنزين - غاز طبيعي).
3- سرعة المحرك: حيث تم استخدام أربع سرعات لعمود المحرك (1750-2300-2900-3500 دورة في الدقيقة)
4- طريقة خلط الغاز: تم استخدام سبعة أنواع من الخلاطات (خلاط بزاوية 90 - خلاط بزاوية 45 - خلاط بزاوية 30 - خلاط فنشوري - خلاط أنبوب داخلي مثقب 8 سم - خلاط أنبوب داخلي مثقب 10 سم - خلاط مع أنبوب داخلي مثقب بطول 12 سم) ويمكن التعبير عنها بـ (T90، T45، T30، VM، L8، L10، L12).

5 -3 القياسات التجريبية :

تم إجراء العديد من القياسات في الورشة التجريبية بقسم الهندسة الزراعية ، كلية الزراعة ، جامعة عين شمس ، مصر.

5-3-1 قدرة المحرك :

حيث تضمن النتائج أن العلاقة بين سرعة المحرك والقوة الفعلية هي علاقة تربيعية ، فكما نلاحظ مع زيادة سرعة المحرك تزداد القوة في جميع أنواع الخلطات. تتفوق القدرة الفعلية على جميع الأنواع عند التشغيل بالبنزين مقارنة باستخدام الغاز الطبيعي حيث أعطى نوع الخلاط (T45) أعلى قوة مقارنة بأنواع الخلاطات (2.83 كيلو واط) بسرعة محرك (3500 دورة في الدقيقة)) كان 7.5٪ أقل من البنزين.

5-3-2 استهلاك الوقود المحدد :

حيث تضمن النتائج أن أقل استهلاك الوقود (S.FC) عند سرعة محرك 2900 باستخدام البنزين كانت (219.025 جم / كيلوواط ساعة) وباستخدام الغاز الطبيعي عند نوع الخلاط T45 كان (234.61 جم / كيلوواط ساعة) بزيادة قدرها 6.6٪ على البنزين. أقل (S.fc) عند سرعة محرك 2900 دورة في الدقيقة باستخدام الغاز الطبيعي عند نوع الخلاط L10 كانت (340.144 جم / كيلوواط ساعة) بزيادة قدرها 35.6٪ عن البنزين.

5-3-4 غازات العادم:

تضمنت النتائج نسبة غازات عادم ثاني أكسيد الكربون عند التشغيل بالبنزين تجاوزت جميع الأنواع كانت (7.96٪) وأقل نسبة لثاني أكسيد الكربون كانت (4.22٪) باستخدام الخلاط من النوع L12. وأظهرت النتائج أن نسبة غازات عادم احادي أكسيد الكربون عند التشغيل بالبنزين تجاوزت جميع الأنواع كانت (0.53٪) وأقل نسبة من أول أكسيد الكربون كانت (0.18٪) باستخدام الخلاط من النوع T30.

5 -3-5 قدرة وتصريف المضخة :

القدرة الهيدروليكية عند تشغيل المحرك بالبنزين والغاز الطبيعي لجميع أنواع الخلاطات بسرعة محرك (3500 دورة في الدقيقة) ، كانت أعلى قدرة هيدروليكية عند التشغيل بالبنزين (0.89 كيلو واط) ، ثم الخلاط T45 كان (0.798 كيلو واط) ، بينما كانت أقل قدرة هيدروليكية عند الخلاط (L12 0.473 كيلو وات).

تضمنت النتائج أن العلاقة بين الضغط وتصريف المضخة علاقة عكسية ، فكلما زاد الضغط انخفض تصريف المضخة. أقصى تصريف باستخدام البنزين (32.38 م^3 / ساعة) عند ضغط (0.2

بار) و اقل تصريف كان (5.66 م^3 / ساعة) عند الضغط (1.8 بار) عند سرعة المحرك (3500 دورة في الدقيقة)

تضمنت النتائج الحد الأقصى من التصريف باستخدام)خلاط T45 (33.09م^3 / ساعة) عند ضغط (0.2 بار) والحد الأدنى من التصريف (3.60 م^3 / ساعة) عند الضغط (1.8 بار). أقصى تصريف باستخدام خلاط L12 (29.51م^3 / ساعة) عند ضغط (0.2 بار) وأدنى تصريف (2.18 م^3 / ساعة) عند ضغط (1.8 بار). عند سرعة المحرك (3500 دورة في الدقيقة)

4-5 المؤشرات الاقتصادية:

- **صافي القيمة الحالية (NPV):**

صافي القيمة الحالية (NPV) لكل من (Gasoline، T30، T45، T90، VM) حيث أن أقل (NPV) كان نوع الخلاط (T90) كان (77219.5) وأعلى (NPV) عند حمل نوع الخلاط (T45) كان (106900.7). صافى القيمة الحالية مع حمل لكل من (بنزين، L8، L10، L12) حيث ان ادنى (NPV) كان نوع الخلاط (L12) كان (78791.4) واعلى (NPV) عند حمل نوع الخلاط (L10) كان (108893.8).

- **نسبة الفوائد للتكاليف (B / C):**

نسبة تكلفة المنفعة (B / C) لكل من (Gasoline، T30، T45، T90، VM) حيث أن أقل (B / C) كان نوع الخلاط (VM) كان (1.38) ، وأعلى (B / C) عند حمل لخلاط نوع (T45) كان (1.54). نسبة تكلفة المنفعة (B / C) مع حمل لكل من (Gasoline، L8، L10، L12) حيث أن أقل (B / C) كان باستخدام البنزين (1.39) ، وأعلى (B / C) عند حمل الخلاط نوع (L10) كان (1.56).

- **فترة الاسترداد:**

الوقت اللازم للمشروع لاسترداد صافي العائد (الفوائد) من تكلفة الاستثمار الرأسمالي للمشروع. فترة استرداد رأس المال لكل من (Gasoline، T30، T45، T90، VM) حيث كانت أقل فترة استرداد كانت من نوع الخلاط (T45) كانت (1.7 سنة) ، وأعلى فترة استرداد عند الخلاط (VM) ، كان (2.37 سنة). فترة استرداد رأس المال لكل من (بنزين ، L8، L10، L12) حيث نلاحظ أن أقل فترة استرداد كانت لنوع الخلاط (L10) كانت (1.66 سنة) ، وأعلى فترة استرداد عند استخدام البنزين كان (2.27 سنة).

- **الكفاءة الاقتصادية:**

تضمنت النتائج ان اقل كفاءة اقتصادية كانت نوع الخلاط (T90) والبنزين (1.65) واعلى كفاءة اقتصادية عند الخلاط (T45) كانت (1.83). الكفاءة الاقتصادية عندما كانت الخلاطة (L8) تبلغ (1.85).

❖ الاستنتاجات

من النتائج التي تم الحصول عليها من هذه الدراسة لحل تدهور الاقتصاد في الوقود باستخدام الغاز الطبيعي ، والهدف من هذه الدراسة هو تحسين عملية خلط الوقود / الهواء والاحتراق ، تم تلخيص هذه الدراسة والاستنتاجات الرئيسية على النحو التالي:

1- يمكن تحويل محركات البنزين للعمل بالغاز الطبيعي بكفاءة (قوة الكبح) تصل إلى 90٪.
2- كان أعلى تصرف للمضخة مع خلاط T45 (33.09 م^3 / ساعة) ، بزيادة قدرها 2.1٪ عن البنزين عند سرعة محرك (3500 دورة في الدقيقة).
3- كانت أعلى قدرة فعلية للخلاط من النوع L10 مقارنة بالخلاطات الأخرى من النوع L (2.69 كيلو واط) أقل بنسبة 10٪ من قدرة استخدام البنزين.
4- أعلى قدرة فعلية مع الخلاطة من النوع T45 مقارنة بالخلاطات الأخرى من النوع T (2.83 كيلو واط) كانت 7.5٪ أقل من البنزين.
5- أما بالنسبة للمؤشرات الاقتصادية ، فقد أعطى استخدام الغاز الطبيعي عائدا اقتصاديا جيدا لجميع الخلاطات ، وأفضل مؤشرات اقتصادية كانت للخلاط (L10) بزيادة قدرها 26.6٪ عن البنزين.

❖ التوصيات:

نوصي بما يلي:

- استخدم الخلاط (L10) لتقليل استهلاك الوقود وزيادة قدرة المحرك والطاقة الهيدروليكية الفعلية لخلط الغاز الطبيعي بالهواء.
- استخدام الخلاط (T45) لتقليل استهلاك الوقود وزيادة قدرة المحرك والطاقة الهيدروليكية الفعلية لخلط الغاز الطبيعي مع الهواء ، حيث أعطت نتائج أفضل من الخلاط (L10).
- نوصي بإجراء مزيد من الاختبارات لقياس عمر المحرك باستخدام الغاز الطبيعي نظرًا لتأثيره البيئي المنخفض ورخص ثمنه.

العوامل الفنية المؤثرة على أداء محرك احتراق داخلي واقتصادياته لتشغيل مضخة ري

رسالة مقدمة من

حيدر عبد الحسين شنان

بكالوريوس العلوم الزراعية (المكائن و الآلات الزراعية)، كلية الزراعة، جامعة البصرة، العراق (2008)

ماجستير العلوم الزراعية (الهندسة الزراعية)، كلية الزراعة، جامعة المنصورة، مصر (2016)

كجزء من متطلبات الحصول على

درجة دكتور الفلسفة في العلوم الزراعية

(الهندسة الزراعية)

قسم الهندسة الزراعية

كلية الزراعة

جامعة عين شمس

2022

صفحة الموافقة على الرسالة

العوامل الفنية المؤثرة على أداء محرك احتراق داخلي واقتصادياته لتشغيل مضخة ري

رسالة مقدمة من

حيدر عبد الحسين شنان

بكالوريوس العلوم الزراعية (المكائن و الآلات الزراعية)، كلية الزراعة، جامعة البصرة، العراق(2008)

ماجستير العلوم الزراعية (الهندسة الزراعية)، كلية الزراعة، جامعة المنصورة، مصر (2016)

كجزء من متطلبات الحصول على
درجة دكتور الفلسفة في العلوم الزراعية
(الهندسة الزراعية)

وقد تم مناقشة الرسالة والموافقة عليها

اللجنة

د. محمد محمود إبراهيم **....................................**
أستاذ الهندسة الزراعية، كلية الزراعة، جامعة القاهرة

د. مصطفى فهيم محمد **....................................**
أستاذ الهندسة الزراعية المتفرغ، كلية الزراعة، جامعة عين شمس

د. عبد الله محمود عبد المقصود **....................................**
أستاذ الاقتصاد الزراعي، كلية الزراعة، جامعة عين شمس

د. خالد فران طاهر الباجوري **....................................**
أستاذ الهندسة الزراعية، كلية الزراعة، جامعة عين شمس

تاريخ المناقشة : / /

جامعة عين شمس
كلية الزراعة

رسالة دكتوراه

اسم الطالب : حيدر عبد الحسين شنان
عنوان الرسالة : العوامل الفنية المؤثرة على أداء محرك احتراق داخلي واقتصادياته لتشغيل مضخة ري
اسم الدرجة : دكتور الفلسفة في العلوم الزراعية (الهندسة الزراعية)

لجنة الإشراف

د. خالد فران طاهر الباجوري
أستاذ الهندسة الزراعية، قسم الهندسة الزراعية، كلية الزراعة، جامعة عين شمس (المشرف الرئيسي)

د. عبد الله محمود عبد المقصود
أستاذ الاقتصاد الزراعي، قسم الاقتصاد الزراعي، كلية الزراعة، جامعة عين شمس.

د. وليد كامل محمد الحلو
أستاذ الهندسة الزراعية المساعد، قسم الهندسة الزراعية، كلية الزراعة، جامعة عين شمس.

تاريخ التسجيل 9 / 3 / 2020
الدراسات العليا

ختم الإجازة	**اجيزت الرسالة بتاريخ**
	/ / 2022
موافقة مجلس الكلية	**موافقة مجلس الجامعة**
/ / 2022	/ / 2022