Design and Fabrication of Compressed Air Vehicle

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Abstract**: *Increase in the energy consumption rate in this present day's pose to cause a great number of hazards to man and his environment. With the aim of curbing this, researchers, engineers, scientist, and environmentalist around the world have taken great steps, but not all the steps have been applied to our physical environment. One of the least explored is compressed air powered vehicle, which can help in reducing a great deal of the hazards. Compressed air engines offer advantages of no pollution, light weight, reduced cost of maintenance and production over others engines using fossil fuels. The compressed air engine project involves modification of an internal combustion engine to operate on compressed air. A single cylinder four-stroke engine (Lifan 110 motorcycle) was selected alongside the designed frame considering design results. Air induction transmission systems were also designed. Several estimations of theoretical power were made along with estimated run time. The engine was fully assembled and incorporated into the frame. Although there were no measuring instruments within our capability to test for the flow rate giving the output power of the engine. The theoretical computations showed that efficiency of the engine running at a given pressure would be lower than that of engine running on fossil fuels. It was concluded that, while this type of engine may run on compressed air, it is likely that more powerful modified internal combustion engines would be more promising for accomplishing the project's goal of an efficient compressed air engine.***

Index Terms: Compressed air, Lifan 100, Fossil fuel, IC-engines.

I. INTRODUCTION

Since beginning of time, man has used energy from various sources of fossils fuels (that is petroleum, diesel, natural gas and coal) and non-fossils fuel. Fossil fuels which meet most of the world energy demand are being depleted rapidly and the product from combustion of these causes global problem such as the greenhouse gas (GHG) emissions, smog production, climate change, ozone layer depletion, acid rains and pollution). Due to the flammable nature of fossil fuels there are accidents associated with their exploration and production. The excessive use of these fuels is causing fast depletion of the fuel and increase in price of the fuel (U.S.E.P. agency, 2012). These factors affect the perception of current automotive technology and the increase in research to determine a more efficient means of transportation. In the United States GHG emissions from transportation increased by 20 percent between 1990 and 2010 (U. S. E. P. agency. 2012). This increase is attributed to a highway miles driven. The conventional internal combustion engine (ICE) produces carbon dioxide (CO2), methane (CH4), and nitrous oxide (N2O) during gasoline combustion (U. S. E. P. agency, 2012). Due to this GHG release, the U.S. Environmental Protection Agency (EPA) regulates the quality of gasoline to ensure less smog, by using reformulated gasoline and controlling the amount of sulfur in the gasoline, which is known to compromise the emissions control on cars (U. S. E. P. Agency. 2012). The U.S. Department of Transportation (DOT) and the EPA enacted regulations to limit the damageproduced by passenger vehicles in the U.S. by setting standards for the average fuel efficiency of cars and light trucks to 35.5 miles per gallon (mpg), by the model year 2016, which will be raised to 54.5 mpg by 2025. This increase in fuel efficiency will decrease GHG emissions and also reduce the frequency of refueling and the amount spent on gas (N. T. S H. adm., 2012). In order to meet fuel efficiency standards, automotive companies have increased development of alternatives to the ICE. Vehicle size reduction is a valid development concept, such as those vehicles currently available in Europe; however, these vehicles are not broadly available or welcomed in the sports utility vehicles (SUV). Currently the plug hybrid electric vehicle (PHEV), full electric vehicle (FEV), and hybrid electric vehicle (HEV) are the only alternatives to meet increased fuel efficiency needs (Michael 2013).

There are downfalls associated with each ICE alternative, including expensive batteries, using rare metals with possible toxic disposal issues and deferring the responsibility of GHG emissions to electricity production (Larson 2012). Automotive companies such as MDI and Peugeot Citroen are leading the development of vehicles using pressurized air. Advancements in the use of pressurized air could reduce tailpipe emissions, city smog production, and consumer fuel costs, while eliminating the need for corrosive batteries (Pollard. 2013). GHG emissions will be solely dependent on electricity production; therefore, compressed air vehicles will have lower net GHG levels than gasoline engines if electricity is sourced from renewable resources (Larson 2012). Many alternatives have been proposed and in some cases used to reduce these factors posing global problems by regulating the quality of fuels used. Controlling the amount of elements in the fuels, such as Sulphur and nitrogen results in the emission of less hazardous gases. Another such alternative is air, which is abundant, cost-effective, nonpolluting and can be stored. Compressed air utilization in the pneumatic applications has been long proven and was previously the basis of naval torpedo propulsion.

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Air motors, pneumatic engines, actuators and other equipment's are in use. Compressed air was also used in some of the vehicle for boosting the initial torque (Thipse, 2008). The major problem with these vehicles is the lack of torque produced by the engines. The costs involved in compressing the air to be used in the vehicle are inferior to that involved in a normal internal combustion engine. The technology involved with compressed air reduces the production costs of vehicles with 20% because it is not necessary to assemble a refrigeration system, a fuel tank, spark plugs or mufflers. (Thipse, 2008) Compressed air Vehicles may be left unused for longer periods of time than electric cars because there is no any risk of battery short circuit when the two terminals come in contact, It also have lower initial cost than electric vehicles when produced in mass. Compressed air is not subject to fuel tax. (Thipse, 2008) As pressurized air vehicles have not entered the market, the purpose of this project is to develop a method to convert a gasoline-fueled engine to a pressurized air engine in order to demonstrate the feasibility of utilizing the renewable resource of compressed air for vehicle propulsion

II. LITERATURE REVIEW

The view of automotive technology in the 21st century, have been hindered by a lot of factors and increase in research today, has drastically helped to improve on different means of transportation. More especially, when it comes to global concern over greenhouse gas (GHG) emissions, climate changes fossil fuel depletion, smog production, and higher petroleum products prices. Automotive companies drive towards producing efficient and economical passenger friendly vehicles (Thipse, 2008).In the early 19s, the use of air as an energy source to power machines may seem unrealistic, but today, with improvement in research, researchers have been able to come up with an idea of exploring the free air around us. All with the goal of rescuing man and his society from environmental degradation, high cost of living, persisting demand on energy and related catastrophes. Journeying to catch up with fuel efficiency standards, ICE alternatives have been a major goal of most automotive industries. From some research conducted, it was concluded that vehicle size reduction is a valid development concept, such as those vehicles currently available in Europe. However, these vehicles are not broadly welcomed in the sports utility vehicle (SUV) in the world. Automotive companies such as MDI and Peugeot Citroen are leading the development of vehicles using pressurized air (Pollard, 2013). Advancements in the use of pressurized air could reduce tailpipe emissions, city smog production, and consumer fuel costs, therefore, compressed air vehicles will have lower net GHG levels than gasoline engines if electricity is sourced from renewable resources (Huba, 2014).

III. BACKGROUND

Petrol engines are type of internal combustion engine. In an internal combustion engine, a fuel, in this case petrol is combusted with an oxidizer to form high temperature gases. These high temperature gases expand to do work and create mechanical energy (Benson, 2009). There are several types of internal combustion engines, which include two-stroke, four-stroke, and gas turbine engines. Four-stroke and two-stroke engines are the most common of these for small vehicle use. In a four-stroke engine, the combustion occurs in a cylinder and the high pressure gasses do work on a piston connected to a rotating crankshaft. Each cycle of a four-stroke engine can be broken down into four different parts: the four "strokes," which can be seen in figure 2.1.

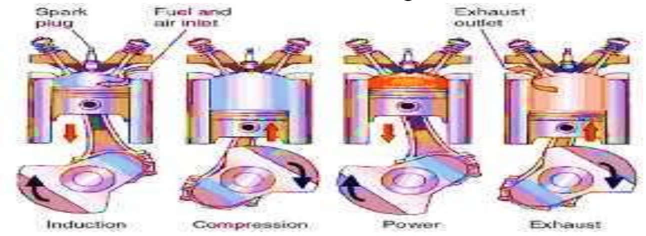


Fig 2.1. Stages of Four-stroke internal combustion engine These four strokes are indction,

compression, power and exhaust. The induction stroke begins at top dead center (TDC) and ends at bottom dead center (BDC). During this stroke, an induction (inlet) valve is opened at the top of the cylinder and a atomized mixture of petrol and air is drawn into the engine. The compression stroke begins at BDC and ends at TDC. During this stroke, the induction (inlet) valve is closed and the petrol and air mixture is compressed as the piston moves to TDC. The power stroke begins at TDC and ends at BDC. During this stroke, the compressed gasoline/air mixture is ignited by a spark at the top of the cylinder, causing combustion, forming carbon dioxide, water vapor and heat. As the gasoline/air mixture burns, the products expand and cool, forcing the cylinder downwards. The exhaust stroke begins at BDC and ends at TDC. During this stroke, an exhaust valve at the top of the cylinder is opened and the exhaust exits the cylinder and the piston moves up. This cycle encompasses two crankshaft rotations. The camshaft is a rod which controls the timing of the induction and exhaust valves in the engine. It is connected to the crankshaft by a set of speed reducing gears or chains, allowing it to turn at half the rate of the crankshaft (Khurmi and Gupta, 2008). Four-Stroke vs. Two-Stroke Engine. Two-stroke and four-stroke engines are the most common forms of gasoline engines available for conversion to compressed air. A two-stroke engine completes the combustion cycle in two piston strokes of the piston.

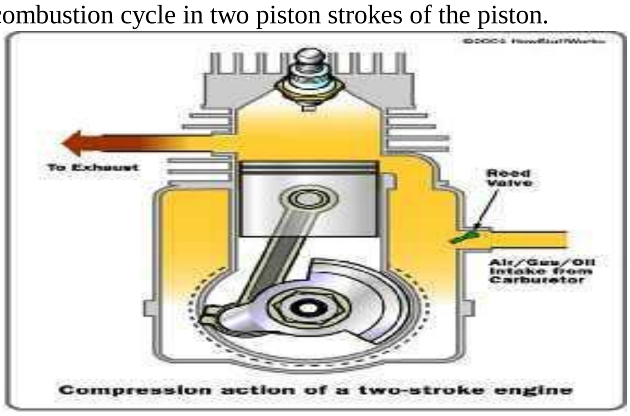


Fig. 2.2. Two stroke internal combustion engine

The primary advantage of selecting a two-stroke engine is that the engine does not have to be retimed, unlike a four-stroke engine; there are no valves to be operated in a set sequence. A two-stroke engine has fewer moving parts, since it lacks a camshaft and an oil system. In a two-stroke engine, the oil is mixed in with the fuel to lubricate the system. This means that two-stroke engines do not last long, since the parts of the system wear a lot faster from a lack of a dedicated lubrication system (Gordon, 1996). Another disadvantage is that the intake and exhaust ports must be flipped or sealed up. If flipped, there is a significant loss of efficiency during the compression stroke. If the ports are sealed up, a new exhaust port would need to be designed and implemented, whether it is a slam valve or solenoid. The exhaust port can also be located near the bottom of the piston's travel with the sacrifice of some percent of the overall efficiency (Benson, 2013).

A. Compressed Air Engines

In order to convert a petrol engine to a compressed-air engine, it is crucial to understand how a compressed-air engine works in relation to its petrol powered counterpart. The important parts of a compressed-air engine are the compressed air tank, regulator, delivery system, camshaft, valves, and piston cylinder. These parts are either unique to a compressed-air engine or operate differently when powered by compressed air (Rolfe et al., 2016).

The compressed-air tank provides the driving force of the engine. It is equivalent to the fuel tank for a petrol engine, as it is the source of the energy used to move the piston. Unlike petrol's energy content, which is a constant value per unit volume, the amount of energy available from the compressed air is completely dependent on the pressure at which it is stored. Unfortunately, as flow leaves the tank the pressure within the tank drops. This causes an inconsistent flow into the engine, thus a regulator is crucial to the compressed-air engine. In order to stabilize the compressed-air engine, a regulator is used. A regulator allows for a controlled pressure of the flow into the engine. The ideal regulator would give a constant pressure and flow rate from the high pressure tank.

IV. MATERIALS AND METHOD

The compressed air vehicle consists of several components (engine, frame, steering, braking system, the compressed air storage cylinder and others). Air is stored in the cylinder below the designed pressure and supplied adequately through transmission mechanism (e.g. pipe and gate valve) to drive the modified engine. Therefore, proper modification of the selected engine must be looked into, coupled with the chassis and load to be carried.

Here the most important parts of the design are the engine modification (consisting of camshaft, valve mechanism, piston, cylinder, crankshaft), air storage cylinder, frame structure, shafts, bearings, chain drive followed by material selection to encourage less weight of the vehicle.

Table 3.1 Engine Cylinder Design

Design Considerations.

i. Type of load and stress cause by the load (kinematics of machine inclusive)

ii. Availability of materials

iii. Cost of acquisition and construction

iv. Assembling of parts and systems

v. Workshop facility

vi. Safety of operation

vii. Conventional economical features

S/No.	Initial	Formula/calculation	Result
1	D=52.4	Thickness of the cylinder wall : $t = 0.045D + 0.6 =$ 0.045(52.4) + 0.6 = 2.95 = 3mm	t = 3mm
2	D= 52.4mm P= 8bar Do= 56mm	Longitudinal stress: $\sigma = \frac{force}{area} = \frac{\frac{\pi}{4}D^2 \times p}{\frac{\pi}{4}(D_o^2 - D_i^2)}$ $\frac{52.4^2 \times 8}{(56^2 - 52.4^2)}$	$\sigma = 56.29 N/mm^2$
3	D= 52.4 $l = 1$	Circumferential stress $\sigma = \frac{f}{A} = \frac{(D \times l \times p)}{(2t \times l)} = \frac{D \times p}{2t}$ $\frac{52.4 \times 8}{2 \times 4}$	$52.4 N/mm^2$
4	N=50	Number of working strokes per sec (n) = N/2 = 120/2	60 r.p.s
5	D= 52.4	$A = \frac{\pi D^2}{4} = \frac{\pi 52.4^2}{4}$	$A = 2156.5\ mm^2$
6		Length of the cylinder (L) L=1.15×l=1.15 ×49.5	L=56.925mm
7	Cylinder head C= 0.1 P= 8bar	Thickness of cylinder head : $t_h = D\sqrt{\frac{C.p}{\sigma_c}} = 52.4\sqrt{\frac{0.1 \times 8}{35}}$	t_h =7.922mm

8	Pm= 6 L= 49.6 A= 2156.5	Power produced in the cylinder that is indicated power $I.P = \frac{P_M \times L \times A \times n}{60} = \frac{6 \times 49.5 \times 2156.5 \times 50}{60}$	$I.P$ =533.733J
		Number of bolts or studs to be used: n_s 0.01D + 4 = 0.01(52.4) + 4 0.02D + 4 = 0.02(52.4) + 4	n_s =4.524, 5.048 Say 4 bolts

Table 3.2 Piston Design

S/N.	Initial	Formula/calculation	Result
1	D= 52.4 $\delta_t = 60$ P= 8bar	Piston head or crown $t_h = \sqrt{\frac{3pD^2}{16\delta_t}}$ $= \sqrt{\frac{3 \times 8 \times 52.4^2}{16 \times 60}}$	=8.29mm
2	$p_w = 0.042$ $\sigma_t = 95$	Piston rings $t_1 = \sqrt{\frac{3p_w}{\sigma_t}}$ $t_1 = \sqrt{\frac{3 \times 0.042}{95}}$ • Minimum axial thickness • $t_s = \frac{D}{10n_R}$ • Width of the top land $b_1 = t_H \ to\ 1.2t_H$ • Width of other rings loads $b_2 = 0.75t_2 \ to\ t_2$ • Gap between rings $G_1 = 3.5t_1 \ to\ 4t_1$ • Gap when the rings is in cylinder =0.002D to 0.004D	$t_1 = 1.9mm$
3		piston barrel $t_3 = 0.03D + t_1 + 4.9$	$t_3 = 8.372mm$
4	D= 52.4mm $p = 8$	piston skirt : maximum gas load on piston $P = p \times \frac{\pi \times D^2}{4}$ $8 \times \pi \times \frac{52.4^2}{4}$ Maximum side thrust on the cylinder $R = \frac{P}{10}$ Length of piston skirt (l) $l = 0.75 \times D$ $= 0.75 \times 52.4$ Length of piston (L) $L = 1.35D = 1.35 \times 52.4$	P=17252N $R = 1725N$ $l = 39.4mm$ $L = 70.74mm$
5	$d_0 = 12mm$	Piston-pin Outside diameter of the piston pin = $d_0 = 12mm$ $L_1 = 0.45D = 0.45 \times 52.4 = 23.58mm$ Bearing pressure (bronze material). $p_{B1} = 25N/mm^2$ Length between the supports of the piston pin. $L_2 = \frac{L_1 + D}{2} = \frac{23.58 + 52.4}{2} = 3.799 = 4mm$ Maximum bending moment at the center of the pin $M = \frac{P.D}{8} = \frac{17252 \times 52.4}{8} = 113 \times 10^3 N = 113KNmm$ Internal diameter $d_1 = 0.6d_0 = 0.6 \times 12 = 7.2mm$ Modulus of Rigidity $Z = \frac{\pi}{32}\left[\frac{d_0^4 - d_i^4}{d_0}\right] = \frac{\pi}{32}\left[\frac{12^4 - 7.2^4}{12}\right] = 147.66mm^4$ Allowable bending stress	$l_1 = 23.58mm$ $L_2 = 4mm$ $M = 113KNmm$

		$M = Z \times \sigma_b \sigma_b = \frac{M}{Z} = \frac{110 \times 10^3}{147.66} = 765.27N/mm^2$	

Table 3.3 Connecting Rod

S/N	INITIAL	FORMULA/CALCULATIONS	RESULTS
1	D= 52.4mm Stroke = 49.6mm	dimension of I-section of connecting rod $F_c = F_l = \frac{\pi D^2 \times P}{4} = \frac{\pi \times 52.5^2 \times 8}{4}$ =17318N F_c=force on connecting rod F_l=force on piston w_b = buckling load $= F_c \times F.S = 17318 \times 6 =$ 103,908N From $k_{xx} = 1.78 + thickness\ of\ the\ I\ section\ (t) =$ $1.78 + 7$mm=8.78mm $A = (4t \times t) + 3t \times t = 11t^2$ $A = 11 \times 7^2 = 539mm^2$ Length of crank r = $r = \frac{stroke\ of\ piston}{2} = \frac{49.5}{2} = 24.75mm$ Length of connecting rod L= 145mm Width of the section B = 4t = 4 × 7 = 28mm Depth or height of the section H= 5t =5×7=35mm Depth near the big end $= H_1 = 1.2H = 1.2 \times 3.5 = 4.2m$ Depth near the small end $= H_2 = 0.85H = 0.85 \times 3.5 = 2.98m$	F_c=17318N $A = 539mm^2$ $r = 24.75mm$

Table 3.3 Valve Mechanism

S/N	INITIAL	FORMULA/CALCULATIONS	RESULTS
`1	Size of port: inlet diameter = 20mmoutlet diameter = 22mm	$ap = \frac{av}{v_p}$	
2	Thickness of valve: k(constant) = 0.42 for steel and 0.54 for cast steel d_p= σ_b = Permissible bending stress 50-60 Mpa for carbon 100-120 Mpa for alloy steel, p =	$t = k.d_p\sqrt{\frac{p}{\sigma_b}} == 0 - 54. \times 22\sqrt{\frac{33}{50}}$	t=9mm
3	Valve stem diameter ds α =30	$ds = \frac{dp}{8} + 6.35mm\ to\ \frac{dp}{8} + 11mm$ Max. lift $h = \frac{d_p}{4\cos\alpha} = \frac{22}{4\cos 30} = 6.35mm$	ds=9.1 h= 6.35mm
4	Valve spring:	$w = w_1 + w_2$, $w_2 = 6.35 \times 10 =$ 63.5mm, $f_s = \frac{\pi \times 22^2}{4} \times 0.02 - 1.96 = 5.64N$, W = 5.64 + 63.5	W = 69.1

5	Outside diameter of the piston pin = $d_0 = 12mm$ D= 52.4 Bearing pressure (bronze material). $p_{B1} = 25N/mm^2$	Piston pin $L_1 = 0.45D = 0.45 \times 52.4 = 23.58mm$ Length between the supports of the piston pin. $L_2 = \frac{L_1 + D}{2} = \frac{23.58 + 52.4}{2} = 3.799 = 4mm$ Maximum bending moment at the center of the pin $M = \frac{P.D}{8} = \frac{17252 \times 52.4}{8} = 113 \times 10^3 N = 113KNmm$ Internal diameter $d_1 = 0.6d_0 = 0.6 \times 12 = 7.2mm$ $Z = \frac{\pi}{32}\left[\frac{d_0^4 - d_i^4}{d_0}\right] = \frac{\pi}{32}\left[\frac{12^4 - 7.2^4}{12}\right] = 147.66mm^2$ Allowable bending stress $M = Z \times \sigma_b \sigma_b = \frac{M}{Z} = \frac{110 \times 10^3}{147.66} = 765.27N/mm^2$	L_1 =23.58mm $L_2 = 4mm$ M=113KN mm Z=147.66mm $\sigma_b = 746.27N$ $L_2 = 3.799$

Table 3.4 Cam Specification

Inlet cam	Inlet cam	Exhaust cam
Base diameter	20mm	20mm
Lift	4mm	6.35mm
Open/close angle	355°- 55°,175°-235°	35°-185° ,265°-5°
Thickness	10mm	10mm

Table 3.4 Design of Camshaft

S/N	INITIAL	FORMULA/CALCULATION	RESULT
		Diameter of camshaft D = 0.16 × cylinder bore + 12.7mm = 0.16×52.4 +12.7	
	cylinder bore= 52.4mm	Width of camshaft W=0.09 ×cylinder bore + 6mm	W=10.716mm

Table 3.5 Design of Crankshaft

S/N	INITIAL	FORMULA/CALCULATIOS	RESULTS
1	D= 52.4mm $F_p = 17318$ W =20N	design of the crankshaft when the crankshaft is at the dead Centre $F_p = \frac{\pi D^2}{4} P$ Assuming the distance (b) between the bearings 1and 2 is equal to the piston diameter. b = 2D = 2×52.4 = 104.8mm $b_1 = b_2 = \frac{b}{2} = 52.4mm$ Due to the piston gas load, there will be two horizontal reactions H_1 and H_2 at bearings 1 and 2 respectively such that $H_1 = \frac{F_p \times b_1}{b} = \frac{17318 \times 52.4}{104.4} = 8659N$ $H_2 = \frac{F_p \times b_2}{b} = \frac{17318 \times 52.4}{104.4} = 8659N$ W= what is W =20N $V_2 = \frac{W}{2} = \frac{20}{2} = 10N$ $V_3 = \frac{W}{2} = \frac{20}{2} = 10N$	$F_p = 17318N$ B =104.8mm b_1=52.2mm $V_2 = V_3 = 10N$

2	Choosing $\sigma_b = 50N/mm^2$ d_c= 4.3mm	Design of crank pins $d_c =$ Bending moment at the Centre of the crank pin $M_1 = H_1 \times B_2 = 8659 \times 52.4 = 454KNmm$ $M_l = \frac{\pi (d_i)^3}{32}$ $\sigma_b = \frac{\pi(d_c)^3}{32} \times 60 = 5.85(d_c)^3 \times 10^3$ $(d_c)^3 = 77.61$ Length of crank pin $l_c = \frac{F_p}{d_c \times P_s} = \frac{17318}{4.3 \times 10} = 402mm$	$l_c = 402mm$
3	d_c= 4.3mm	Design of left hand crank web t=0.65d_c + 6.35 t=285.85mm With of crank web w = 1.125dc + 12.7 = 496.45mm Maximum bending moment B.M on crankshaft $M = H_1 = \left[b_2 - \frac{l_c}{2} - \frac{t}{2}\right] =$ $M = H_1 = [52.4 - \frac{402}{2} - \frac{285.85}{2} = 2524kNmm$ Section modulus $Z = \frac{1}{6} \times wt^2 = \frac{1}{6} \times 285.85^2 \times 496.45 = 6761 \times 10^3\,mm^3$ Therefore bending stress $\sigma_b = \frac{M}{Z} = \frac{2524}{6761 \times 10^3} = 0.4 \times 10^{-4} kN/mm^2$ Direct compressive stress on crank web $\sigma_c = \frac{H_1}{w.t} = \frac{8659}{285.85 \times 496.45} = 0.061kN/mm^2$ Total stress of the crank web = 61 + 0.043 = 61.043 N/mm^2 iii. Design of right hand crank web From the balancing port of view, the dimensions of the right hand crank web (i.e. thickness and width) are made equal to the dimensions of the left hand crank web iv. Shaft under the flywheel Diameter of shaft under flywheel = 18mm Length of shaft = 205mm.	t=285.85mm $\sigma_{c=0.061kN/mm^2}$
	$l = 26.5$ $r = 4.5$	Design of crankshaft when the crank is at the angle of maximum twisting moment Notes: - when the crank has turned 35 from the top dead Centre the pressure on the piston is 1n/mm2 and the torque on the crank is maximum (P^1= 1 N/mm^2) - ratio of connecting rod length to the crank radius is $\frac{l}{r} = \frac{26.5}{4.5} = 5.7$ $F_p = \frac{D^2 p^1}{4} = \frac{\pi}{4} \times 52.4^2 \times 1 = 2156.5N$ To find the thrust on the connecting rod, angle of inclination of the connecting rod with the line of stroke needed to be gotten (i.e. angle Φ)	$F_p = 2156.5N$

	$H_{R1} = 802.5$ b_2=52.4 τ = 35 N/mm^2	design of crank pin From B.M at Centre of the crankpin $M_c = H_{R1} \times b_2 = 802.5 \times 52.4 = 42051Nmm = 42.051k$ $T_c = H_{T1} \times r = 707.1 \times 4.5 = 3181.95Nmm = 3.81kNmm$ Therefore equivalent twisting moment on the crank pin Taking τ = 35 N/mm^2 or 35Mpa $42\ 000 = \frac{\pi(d_c)^3}{16}\tau = \frac{(d_c)^3\pi \times 35}{16}$ $d_c = 8.5mm$ Taking the mean of d_c $d_c = \frac{8.5 + 4.3}{2} = 6.4mm$	$d_c = 8.5mm$ $T_c = 3.81kNmm$
	C= width of flywheel = 60mm = t	design of shaft under flywheel Allowing space for gearing and clearance by taking c = 100mm $c_1 = c_2 = \frac{C}{2} = \frac{100}{2} = c$	Diameter of shaft under flywheel = 18mm

Table 3.70: Shaft design

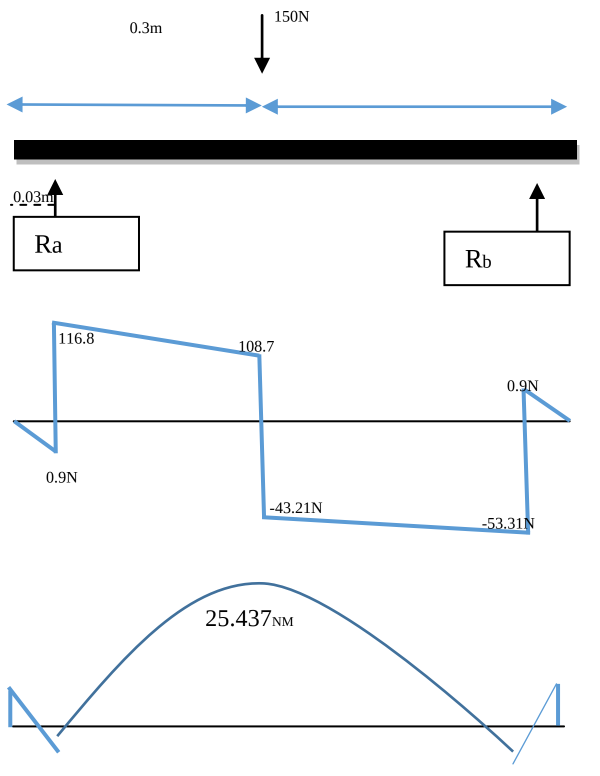
S/N	INITIAL	FORMULA/CALCULATIOS	RESULTS
1	Torque = ? Power = 100w N = 150rpm	Torque = $\frac{P \times 60}{2\pi N}$	Torque = 6.4
2	Diameter of shaft d =? Torque T = 6.4 Shear stress $\tau =$	d = $\sqrt[3]{\frac{16T}{\pi\tau}} = \sqrt[3]{\frac{16 \times 6.4}{\pi \times 500}} = 0.04m =$	D = 4cm = 40 mm
3	Length of shaft	As specified in the design of frame	L= 0.8m
4	Shear force and bending moment Ra= Rb Dynamic loading N= 200N		Maximum shear force= 116.89 Maximum bending moment = 25.437Nm Maximum deflection = 3.83/EI

Table 3.8.1: chain drive design

S/N	INITIAL	FORMULA/CALCULATIOS	RESULTS
1	No. of teeth of bigger sprocket T_2 $T_{1=12}$ $\frac{N_1}{N_2} = 3$	$T_2 = T_1 \times \frac{N_1}{N_2}$ $T_2 = k_s 15 \times \frac{N_1}{N_2}$	$T_2 = 36$ Due to availability and cost a sprocket with 46 number of teeth was selected
2	Service factor k_1 k_2 k_3	$k_s = k_1 \times k_2 \times k_3$ $k_s = k_1 \times k_2 \times k_3$	$k_s = 1.875$
3	Design Power Rated power Service factor k_s= 1.875	Design power = Rated power × 1.875 100 × 1.875	Design Power = 187.5 0.1875kw

From catalogue (Indian standard- IS: 2403-1991) the following specification is selected

Chain type: roller with link type P =100wD =4mmW_B= 2890w

Table 3.8.2: Chain drive design (continued)

S/N	INITIAL	FORMULA/CALCULATIOS	RESULTS
4	Pitch Diameter of smaller sprocket d_1 Pitch p = 1.2	$d_1 = pcosec\{\frac{180}{T_1}\}$ $d_1 = 1.2cosec\{\frac{180}{16}\}$	$d_1 = $6cm
5	Pitch Diameter of smaller sprocket d_2 Pitch p =1.5	$d_2 = pcosec\{\frac{180}{T_2}\}$ $d_2 = 1.5cosec\{\frac{180}{36}\}$	$d_2 = 17.2$
6	Line Velocity of smaller sprocket V_l	$V_l = \frac{\pi \times d_1 \times N_1}{60}$ $V_l = \frac{\pi \times 6 \times 100}{60}$	$V_l = $31m/m
7	Load on chain W $rated\ power =$ $pitch\ line\ velocity =$	W = $\frac{rated\ power}{pitch\ line\ velocity}$ $\frac{100}{2.9}$	W = 34N
8	Factor of safety $W_B =$ $W =$	F.S = $\frac{W_B}{W}$ F.S = $\frac{2890}{100}$	F.S = 28.9

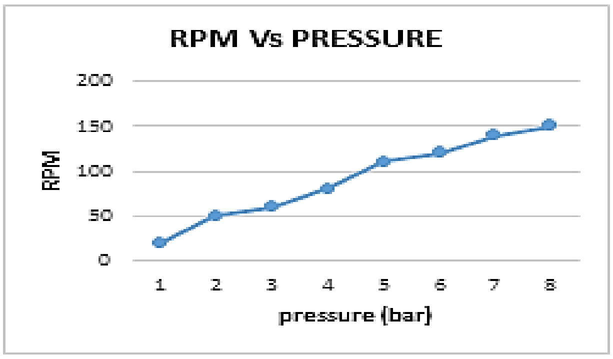
Compare the value x from standard table and take the appropriate value.

Table 3.8.3: Chain Drive Design (continued)

S/N	INITIAL	FORMULA/CALCULATIOS	RESULTS
9	Center distance between sprocket C Pitch p =	$C = y \times p$ $C = 38 \times 1.2$	$C = $45.6cm
10	Early sag correct distance	Correct distance = $[C - x]$ $[45.6 - 5]$	Correct distance = 40.6cm

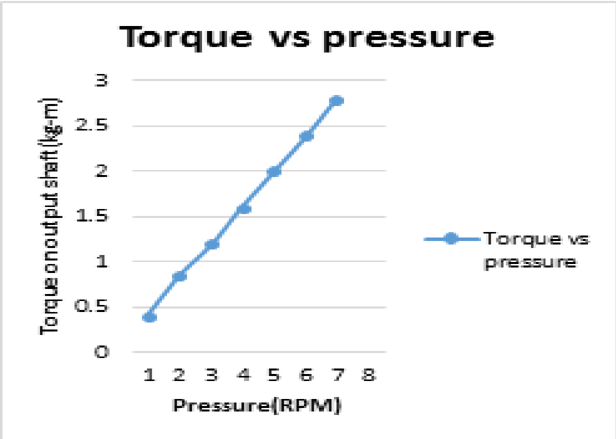
V.RESULTS

The test carried out on the project work is of two stages Engine Test: where RPM, Engine torque, power on output shaft, working pressure etc. were determined and the relationships between the parameters were also determined. The second stage test is the Road Test.

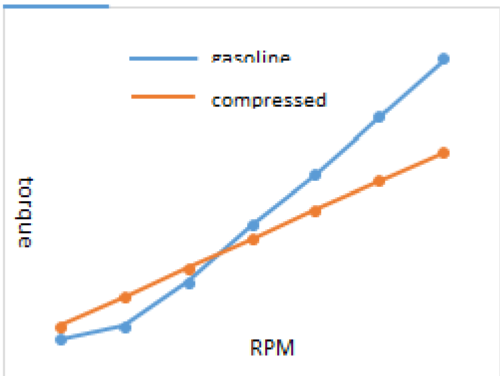


Graph: 4.1: Graph for the relationship between the RPM and the input pressure.

Graph showing the relationship between the pressure supplied and the torque on the output shaft.



Graph: 4.2: Graph of pressure in the cylinder against torque on the output shaft.



Graph 4.3: Graph of torque versus RPM for gasoline Engine and compressed air Engine.

The project engine was successfully converted completely to run on compressed air. The modified engine was tested in thermo- lab.II (AbubakarTafawaBalewa University Bauchi) twice in order to determine real power output. Air for the tests was taken from compressor at 8bar. An initial run showed a maximum output of 0.1 horsepower at 120 RPM. A second run with larger compressor and larger diameter tubing shows a maximum output of 0.2 horsepower at 150 RPM. When mounted on the vehicle, the engine ran to the distance specified in the test sheet. Although the engine ran smoothly, these results show that there was a little disparity between the theoretical engine cycle and the real engine cycle. Due to this mechanical issue, realistic theoretical data from the thermodynamic models were used for power and efficiency calculations.

Using the first model of the engine, the maximum power the engine could produce was calculated. This assumed that the air pressure was applied to the piston for the entire power stroke and that no air was compressed in the return stoke. This model also assumed that pressure was applied fully at TDC and the pressure returned to ambient as soon as the piston reaches BDC. A bore of 49.2mm and a stroke of 52.4mm were used in the model, giving a piston area of 76mm squire. Using 8bar of air pressure and an engine RPM of 150RPM the work calculated per engine cycle is 85.01 Nm. This equates to a power of 168 Nm/s. This value was obtained from code A.2.mat-lab.

The second thermodynamic model calculated the work in four separate steps, as explored in chapter 3. For this model the intake valve was assumed to be open from TDC until 90 degrees past TDC and the exhaust valve was assumed to be open from BDC until 90 degrees after BDC. The displacement of the engine was assumed to be 305 cubic centimeters (cc), RPM is 150, air supply pressure was 8bar and ambient pressure was 1013.25N⁄m^2 . From this model the four values for work representing the four steps were:

1W2 = 0.2864 Nm
2W3 = 0.3742 Nm
3W4 = -0.0362 Nm
4W1 = -0.1536 Nm

This gave a network of 0.4708 Nm per cycle. These values were calculated using code A.3.

The model used to calculate theoretical air consumption in the engine was based on the volumetric flow rate through the engine at ambient pressure and temperature. Then Using the RPM of the engine with the amount of air the engine uses per cycle a volumetric flow rate of air through the engine was obtained. This flow rate was then used with the volume of air stored in the tanks to give an estimated runtime for the engine. It was assumed that large air cylinder/tanks which can store air at 300bar and assumed an effective volume of the cylinder over 1000 cc and an effective pressure in the cylinder of 8bar. This model then predicted a runtime of over 10 minutes. This value was calculated from code A.4.

Engine efficiency was explored in chapter 3. The primary idea was that engine efficiency is equal to the amount of energy applied to the crankshaft divided by the amount of energy used to compress the stored air. With a tank storing air at 300bar, regulated down to an intake pressure of 8bar, engine efficiency was calculated to be approximately 15%. This was calculated using equations from chapter 3. Losses were determined to be from heat loss in the storage tank, pressure loss in the tubing and regulator, friction in the engine itself, and incomplete expansion of expended air. Total efficiency was the engine efficiency multiplied by the efficiency of the production of source electricity to run the compressor.

The total efficiency, of the entire theoretical compressed air system is approximately 10-12%.

VI. DISCUSSION OF RESULTS

The buildup of air pressure in the cylinder is what creates power in the engine. Before modification, the four-stroke petrol engine used a spark from a plug to ignite the mixture of fuel and air for burning of fuel to power the engine. In order to convert this to run off of compressed air, the engine was converted into a two-stroke engine by modifying the cam, modifying the inlet valve on the engine block to allow for the entrance of the pressurized air, grinding of the valve and setting of the timing chain. From the test result obtained, it was found that the engine run as required, calculated from the theoretical engine result obtained an engine output of 0.2 horse power at 150 RPM.

It was also found from the theoretical Results that engine RPM increase with increase in the inlet Pressure as shown in Graph:4.1 this indicate that there is direct proportionality relationship between the pressure and the RPM. Graph: 4.2 shows that torque increase with the increase in pressure and RPM this is because the more the available pressure in the cylinder to perform the work the more the RPM hence it gives high torque with increase in pressure and RPM to some permissible limits. The pressure to push down the piston has some certain limit were any additional pressure on it will lead lesser efficiency of the Engine.

Furthermore, from the Graph: 4.3 torque for compressed air engine dominates the torque of gasoline engine at low speeds (RPM), this is because the pressure available at startup for the compressed air engine higher than that of gasoline engines. But at higher speed the gasoline engine have more torque when compared with the compressed air Engine. This is because at startup of the of gasoline engines mostly there is incomplete combustion of the fuel, which results in lesser efficiency. And also from the graph obtained using the theoretical result, increase in the pressure entering into the engine increases the speed of the engine in revolution per min RPM and corresponding increase in the amount of torque, but as the load on the Engine increases and the RPM gets higher, the Gasoline engine provides more torque than the engine which uses compressed air as fuel. Thus for low speed applications, the compressed air engine is suitable and can work efficiently.

From the result obtained on the comparative cost analysis carried on 15 lifan 110 motorcycle running with light petroleum fuel shows that the cost of production, maintenance and operational is high when compared to the compressed air vehicles. The cost objects of production, maintenance and operations are on the increase for motorcycles running on gasoline. From Graph: 4.2 the entire cost object increases from 2012-2015. Therefore, compressed air vehicles have lesser life cycle cost LCC when compared with the vehicles running on gasoline.

VII. RECOMMENDATION

Compressesed air technology have many area for improvement. the same applied to these project which design and development is based on these technology. these project is far from been optimised. Due to many problems encountered shortage of time, unavailability of funding, and lack of research materials and sure if the project is to be continued in future there will be an inprovements. The following points are recommended for future improvement. Consideration should be made for tank with high storage capacity and can supply air at a constant pressure without necessary using more than one cylinder

Consideration should be made for a heater which can heat the air to increase the efficiency of the output Inprovement in the design to futher reduce the weight of the transmission by using lighter, strong and durable material or for overall reduction in the space and volume of the vehicle Design should be made for air of high flow rate The exhaust air can be channeled to perform some work, by installing a dyno and small electric generator on the exhaust. This can generate electric power for some auxiliary use.

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