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Design of a vane compressor to get a supercharged piston engine.

End of Degree Project

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Design of a vane compressor to get a supercharged piston engine

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ACKNOWLEDGMENTS

Almost four years have passed since I stepped foot in the University for the first time. It was the beginning of a different stage, although also complicated, which I took with great ambition. Now, four years later, I am a different person, I am not the same person who entered the university for the first time. I have grown both academically and personally. And all this I have not achieved alone. Along this path there have been countless people who have made this possible from start to finish.

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RESUMO

No documento mostrado abaixo, foi realizado um estudo sobre o efeito da sobrealimentação de um motor Honda GX-50 em relação ao seu estado original.

Para isso, em primeiro lugar, foi escrita uma breve compilação de alguns dos marcos mais importantes na evolução do mundo motor, desde seu primeiro precedente, como o motor a vapor, até hoje.

Posteriormente, foram escritos os fundamentos teóricos sobre motores a pistão e compressores volumétricos, diferenciando seus diferentes tipos e aprofundando no compressor de palhetas, visto que é o que será utilizado para o estudo da sobrealimentação, onde analisa a função que desenvolve por colocando-se antes do carburador, bem como suas vantagens e desvantagens.

Após isso, é adicionada uma peça de projeto do compressor, incluindo suas medidas, bem como sua montagem no motor.

Por fim, são elaboradas algumas conclusões que incluem uma análise teórica dos possíveis resultados e também dos problemas que surgiram durante o desenvolvimento do projeto, bem como a justificação do cumprimento ou não dos objetivos propostos, o que leva a propostas de projetos.

PALAVRAS-CHAVE

Motor;pistão;compressor;palhetas;sobrecarga;4tempos;2tempos

ABSTRACT

In the document shown below, a study has been carried out on the effect of supercharging a Honda GX-50 engine with respect to its original state.

To do this, first, a brief compilation of some of the most important milestones in the evolution of the motor world has been written, from its first precedent, such as the steam engine, until today.

Subsequently, the theoretical foundations about piston engines and volumetric compressors have been written, differentiating their different types and delving into the vane compressor, since it is the one that will be used for the study of supercharging, where It analyzes the function that it develops by placing itself before the carburetor, as well as its advantages and disadvantages.

After this, a compressor design part is added, including its measurements, as well as its assembly to the engine.

Finally, some conclusions are drawn up which include a theoretical analysis of the possible results and the problems that have arisen during the development of the project, as well as the justification of compliance or not with the proposed objectives, which leads to proposals for future projects.

KEYWORDS

Engine;piston;compressor;vane;supercharging;4stroke;2stroke

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1. INTRODUCTION

1.1. MOTIVATION

When the different project topics were shown to me, I was interested in those that required design, manufacture, and construction, due to at the end of the project I would get something tangible made for me, which is always satisfactory. On the other hand, the choice of a vane compressor for a 50 cc motorcycle seemed interesting to me since it is an element which is applied to something as familiar as a motorcycle in order to improve its performance by applying knowledge acquired throughout all four years of college.

1.2. OBJECTIVE OF THE INVESTIGATION

General:

Design a small vane compressor.

Specifics:

Review the forms of supercharging of a piston engine.

Understand the operation of vane compressors and piston motors.

Design the components of the compressor.

Redirect a dissertation to present a discussion.

Redirect a scientific article with a summary of the description of the state of the art, project procedure and discussion of results.

1.3. METODOLOGY AND ESTRUCTURE

In the first place, a search of the evolution of engines throughout history is carried out, as well as the theoretical foundations related to it, to understand its operation and be able to carry out its design, manufacture and construction. Subsequently, the laboratory will be accessed with the design carried out to proceed with its manufacture, construction and coupling to the engine to obtain the required data.

2. STATE OF THE ART

2.1. HISTORICAL CONTEXT

It is considered an engine any device capable of transforming any type of energy into mechanical energy. There are electric engines, which transform electrical energy in mechanical; chemicals engines, which obtain mechanical energy through chemical reactions; etc. In this case, volumetric compressors belong to the group of thermal engines, which transform the thermal energy of a compressible fluid into mechanical energy (figure 1).

Firstly, we are going to talk about the precedents for the first internal combustion engines. The most obvious precedent of this type of engines are steam engines (XVIII century) because these could produce mechanical energy from unnatural energies, these are those that do not come from the energies existing in nature as they are hydraulic or wind. The main feature that these machines share with an alternative internal combustion engine (AICE) is that both are a delimited volumetric chamber with a mobile wall, which modifies the volume of the fluid that is inside.

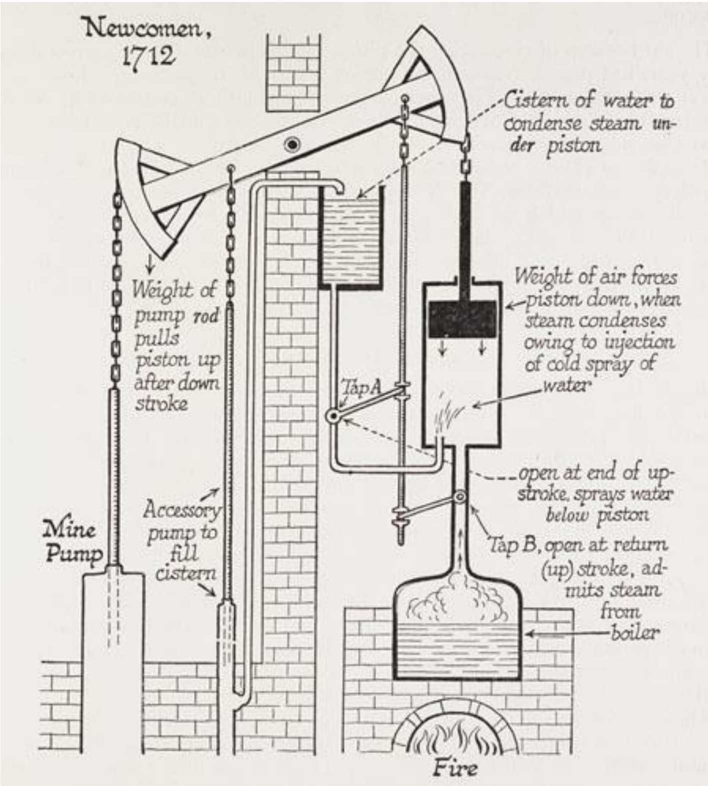

Figure 1. First steam machine scheme. [8]

For more than two hundred years steam engines were the only thermal machines, during which they had important applications in the propulsion of means of transport, both in the maritime and fluvial fields and in the railway. However, during these years the nature of the processes that allowed the obtaining of mechanical energy was unknown, until the nineteenth century, when thermodynamics began to develop.

The first alternative internal combustion engine arrived in 1860 by the Belgian Étienne Lenoir, which he called Dilated Air Engine with the combustion of gases by means of electricity (figure 2). It was a one-cylinder engine and a 2T cycle, whose fluid, without prior compression, was formed by coal gas and air. This meant poor engine performance, in addition to its small expansion ratio. Despite this, these engines reached between 6 and 20 hp.

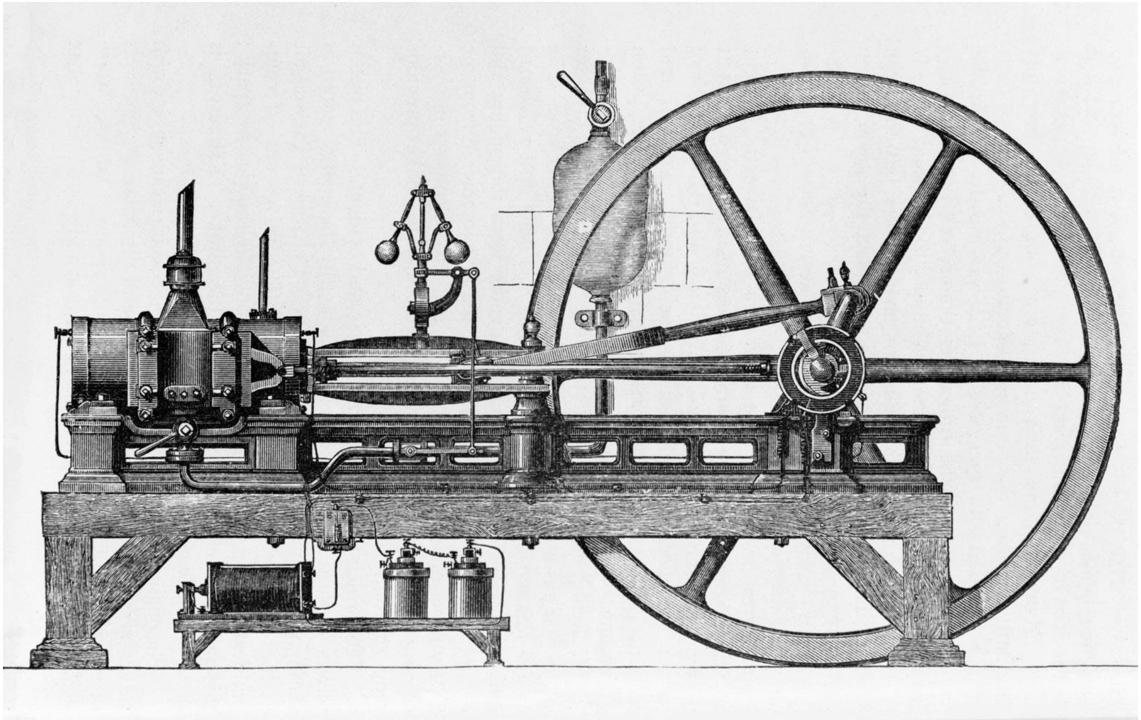

Figure 2. Drawing of one-cylinder engine 2 strokes of Lenoir. [9]

Two years later, in 1862, the idea of compressing the fluid previously of the combustion was introduced for first time. This milestone is attributed to the Frenchman Alphonse Eugène Beau de Rochas, who filed a patent for four chapters, the second of which was called Mixed Steam Gas Engine. Device with pre-compression. This engine had a 4T cycle, where the subsequent step to the aspiration of the fluid was a compression of this, all prior to combustion (figure 3).

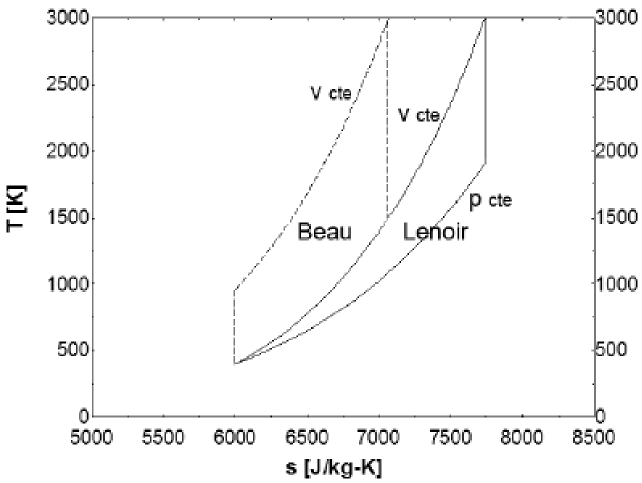

Figure 3. Comparation between Lenoir and Beau engines. [2]

In 1866 the German Nikolaus August Otto built an engine that operated by gas in 4T as well. A little later, together with Eugen Langen, they also made a version of this 2T. Although this milestone was later than that of Rochas, Otto paid economically to this to keep the fame of the invention, and to this day this cycle of operation is recognized as Otto cycle (figure 4).

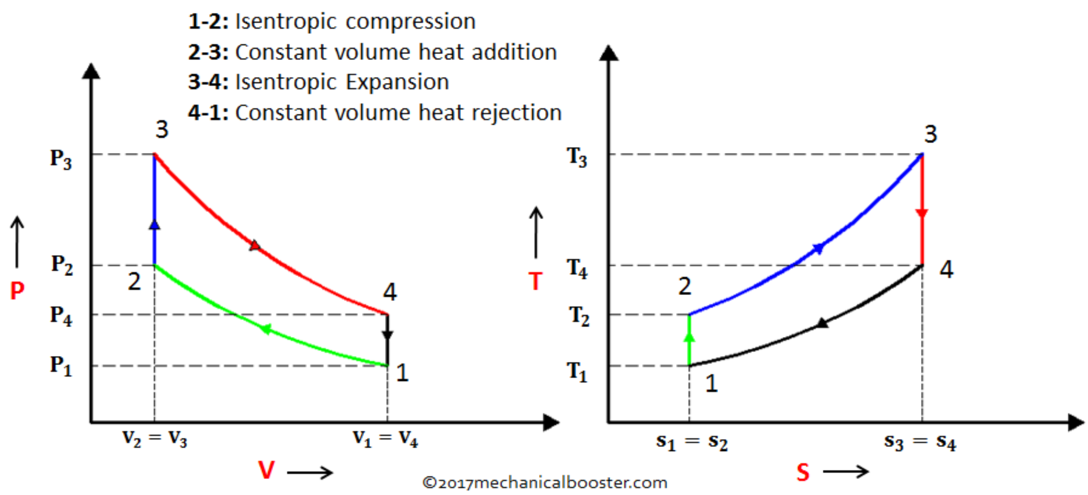

Figure 4. P-V and T-S Diagram of Otto Cycle [10]

The first design of a compressor to be applied to an MCIA engine was by Gottlieb Daimler in 1885. It is at this time that the concept of supercharging an engine was born.

This year (1885), Karl Benz invented the carburetor, obtaining his patent in 1886, the year in which he also filed the patent for the first car equipped with an MCIA

Parisian Rudolf Diesel vigorously pursued the idea of getting as close as possible to a Carnot cycle. He wanted to keep the temperature constant in both expansion and compression. To do this, he introduced the idea of injecting the fuel simultaneously with its combustion, compensating for the heating of the compression with the cooling of the expansion. Its cycle, therefore, consisted of an isothermal compression followed by an isentropic compression, and an isothermal expansion followed by another isentropic expansion (figure 5).

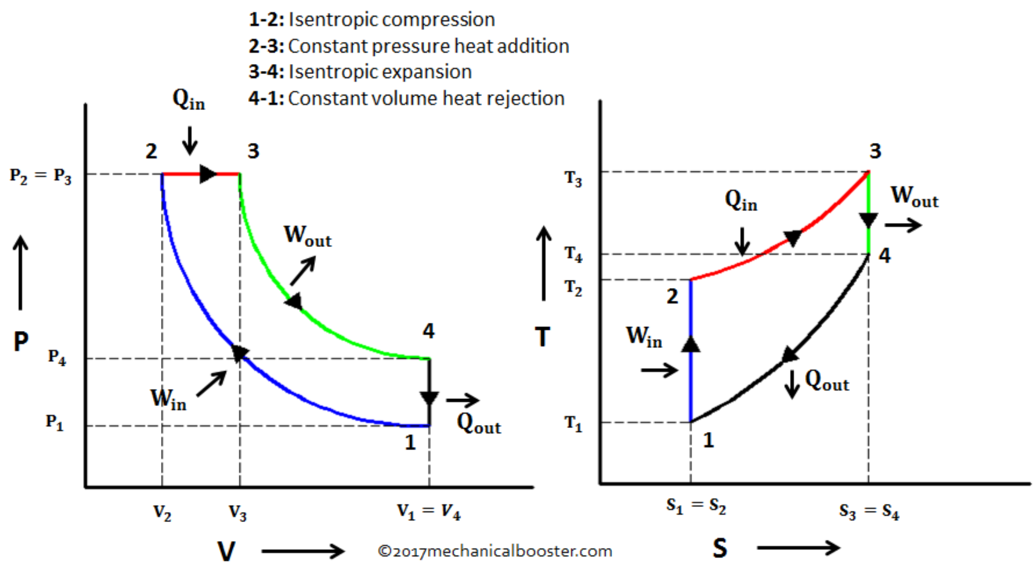

Figure 5. P-V and T-S Diagram of Diesel Cycle [10]

It was in 1892 when he applied for the patent of "Procedure to produce engine work by the combustion of fuels.

However, he did not define the process between the last expansion and the first compression. Later, in some variable of his patent, he defined the previously indefinite step as a cooling of constant volume.

During the XX century advances and improvements in this type of engines were abundant.

MCIA began to introduce the concept of supercharging in real engines, the use of which was justified in 1885 by Gottlieb Daimler.

In 1922 the first engine with a combustion pre-chamber was manufactured by Mercedes-Benz.

In 1925, Jonas Hesselman introduced the first gasoline engine with direct injection.

Over several years, these concepts were improved for their applications in different vehicles depending on their needs, such as in trucks or trains.

In 1961 comes the next revolutionary milestone in the world of MCIA, electronic injection, patented by Bendix. This meant the replacement of carburetors in the automotive market. The development of electronic injection continues to this day.

Until now, the development of the engines had focused solely on an improvement in their performance. However, in the 70s it began to give importance to fuel consumption, trying to reduce this while maintaining the efficiency of the engines.

In addition, in this decade environmental policies began to emerge, which meant the obligation to guide the improvement of engines in a reduction of CO and HC emissions. This gave rise to catalysts, which allowed a complete oxidation of CO and HC prior to their escape process.

From this moment, MCIA engines have been improved in terms of performance and reduction of consumption, as well as the introduction of hybrid systems to reduce pollution on the planet, an objective that already had its origin in the 70s and remains until today.

3. THEORICAL FOUNDATIONS

3.1. PISTON ENGINES

3.1.1. Components and concepts

A piston engine is that engine that, as its name says, uses the movement of pistons for the transmission of energy. To understand its operation, it is necessary to know the main elements that compose it (figure 6):

-Cylinder: is the cavity through which the piston moves.

-Crankshaft: it is in charge of transmitting the reciprocating movement of the piston in a rotary movement to supply it with a useful torque.

-Connecting rod: it is the intermediary between the piston and the crankshaft in the transformation of the movement.

-Piston: is the mobile element that moves along the cylinder in an alternative rectilinear movement.

-Cylinder head: it is the part that seals the cylinder at the top so that the volume is closed. Here are the valves.

-Valves: they are responsible for allowing the entry and exit of gases in the combustion chamber.

-Combustion chamber: it is the space between the cylinder, the cylinder head and the piston, where the combustion of the gases introduced into it takes place.

-Spark plug: it is the element that causes the spark in the combustion chamber (Only in provoked ignition engines)

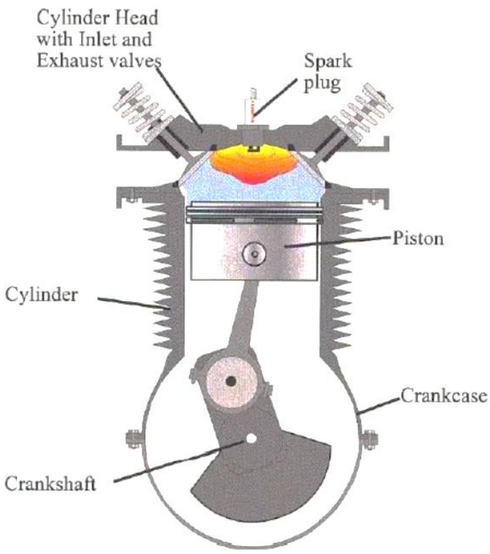

Figure 6. Drawing of a piston engine [1]

It is also necessary to describe some concepts of this type of motor (figure 7):

-Top dead center (TDC): is the position in which the piston is, considering positive displacement the ascending one, in its maximum displacement, that is, in the upper part of the cylinder.

-Bottom dead center (BDC): it is the position opposite to PMS, that is, in its minimum displacement, or what is the same, the position of the piston in which the cylinder has the maximum volume.

-Piston stroke (S): It is the path that the piston makes from PMS to PMI and vice versa.

-Compression ratio (rc): is the ratio of the volume when the cylinder is at bottom dead center to the volume when it is at top dead center. This value is significant to know the power of the motor. Its value is, for aviation, around 8.5/1 [xxx].

-Piston diameter (D)

-Stroke-bore ratio (S/D)

-Piston section (Ap)

-Unit displacement (Vd)

-Number of engine cylinders (z)

-Total displacement (Vt): is the product of the unit displacement by the number of pistons

-Combustion chamber volume (Vc): is the volume that remains when the piston is at TDC

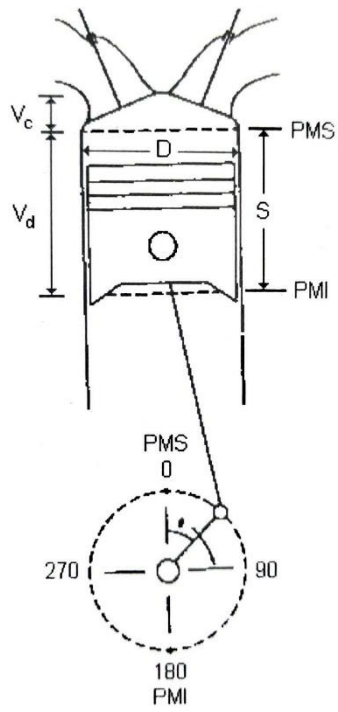

Figure 7. Piston engine with some keywords [1]

3.1.2. Classification

Alternative internal combustion engines can be classified into different types depending on what is being analyzed for said classification.

Depending on the type of ignition of the air-fuel mixture, these can be classified as spark ignition engine or compression ignition engine.

On the one hand, the MEPs have the formation of the mixture in the carburetor; as its name says, its ignition is caused by a spark, caused by the spark plug; its compression ratio is between 8-12 and it can work between 6000-18000 rpm. To regulate the load, it is done quantitatively, that is, it is regulated with more or less mixture, since the air and fuel are previously mixed in the combustion chamber. Its main fuel is gasoline. As for the dosage, this is usually stoichiometric (f~1).

On the other hand, the MEC have the formation of the mixture in the combustion chamber by injecting fuel into it. Its ignition is by self-ignition. The compression ratio is between 13-18, and its working range is between 4500-6000 rpm. In this case, the regulation of the load is qualitative, that is, it is regulated by injecting more or less fuel. The fuels used are self-igniting and lubricants, and the most common is diesel. In reference to the dosage, this is less than unity, so it is a poor mixture.

They can also be classified by the way they perform the duty cycle. In this classification we have the previously named 2-stroke and 4-stroke engines, which will be explained in detail later.

On the other hand, according to the type of cooling, they can be classified into air-cooled motors and water-cooled motors.

Regarding the intake pressure, here we find the type of engine that we will use for the study. There are naturally aspirated engines, in which the gas enters at ambient pressure, and supercharged engines, in which a pre-compressor is placed to increase the inlet pressure to the combustion chamber. The latter type of motor will be discussed in detail later.

Finally, they can be classified according to the number of cylinders and their arrangement. In reference to the arrangement, we find motors in line [Figure 8], in V [Figure 9], opposed [Figure 10], and radial [Figure 11].

Figure 8. Cyclinders in line [12]

Figure 9. Cylinders in V [13]

Figure 10. Oposed cylinders. [14]

Figure 11. Radial cylinders [15]

3.1.3. How it works?

The basis of operation of these engines is similar in both 2T and 4T engines. For 4-strokes, it starts with the piston at the top of the cylinder. The intake valve opens, and gas enters the combustion chamber so that the piston travels down on the cylinder. Once the piston has reached its bottom point and the gas has occupied the volume that it has left free. Next, the piston ascends compressing the mixture until it returns to its upper point. At this moment, either by self-ignition or by a spark from a spark plug, the mixture ignites, causing the gases to expand, causing the piston to move towards the bottom of the cylinder. Finally, the exhaust valve opens, and the piston rises again, thus expelling the gases that are inside the combustion chamber. In total, 4 piston strokes have been made, two ascending and two descending.

On the other hand, for 2T engines, the first and third steps, and the second and fourth steps are carried out simultaneously, that is, the first downward displacement of the piston into which the mixture is introduced is caused by the expansion caused by the combustion of gases, and in the compression of these, they are expelled from the combustion chamber.

The movement of these ascending and descending strokes carried out by the piston constantly activates the connecting rod-crank mechanism, which, as has been said before, transforms the linear movement into a rotary movement.

3.1.4. Advantages and disadvantages

First of all, it should be noted that one of the main advantages of these motors is that they can be adapted to a wide range of powers.

On the other hand, the amount of fuels with which these engines work is also high, and can be both gaseous, liquid or even solid. However, the most common are liquids due to their high calorific value, which provides great autonomy with a reduced volume of fuel.

In addition, these motors can be built in different sizes and materials, adapting to the needs that are required and the budget or available resources.

However, these motors also have disadvantages. The first of these is the emission of polluting gases such as CO_2 after the combustion process, being harmful to the environment. Hand in hand with this inconvenience, there is also dependence on oil, a limited and non-renewable energy resource, as well as a pollutant, as has already been said.

On the other hand, despite having a wide range of powers at which to work, there is a maximum power at which they can work, so that on large scales this type of engine cannot compete against others such as gas turbines.

3.1.5. Fields of application

Next, the main fields of application of 4T, 2T, MEP and MEC motors will be shown, with their different combinations, in order of motor size:

MEP and 2T:

-Mopeds

-Small tools

-Modeling

MEP and 4T:

-Cars and motorcycles

-Recreation boats

-Small planes

MEC and 4T:

-Cars

-Industrial vehicles

-Farm Equipment

-Industrial Stationary Engines

-Small-medium sized boats

-Public works machinery

-Railways

MEC and 2T:

-Public works machinery

-Railways

-Large boats

-Large stationary engines

3.1.6. Supercharged engine

The concept of supercharging consists of taking advantage of mechanical means to increase the pressure of the gases entering the combustion chamber so that the amount that enters is greater than the standard case, generally achieving greater engine power and efficiency. This phenomenon is mainly carried out with the existence of a compressor prior to said combustion chamber.

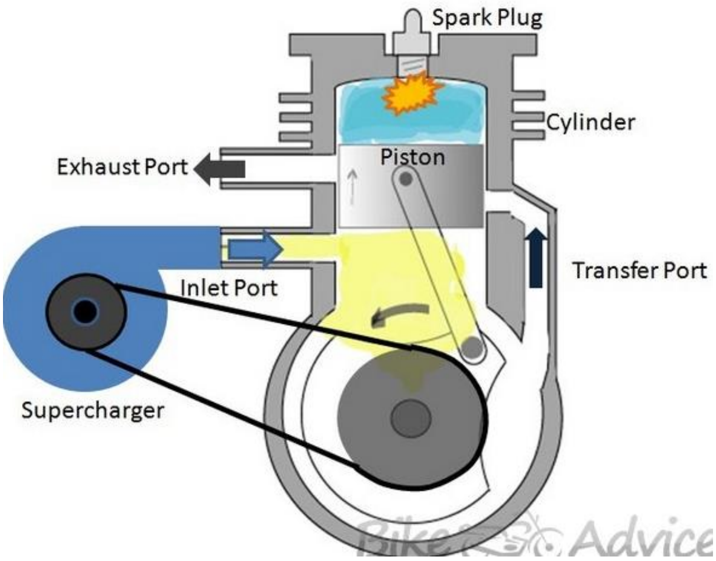

Figure 12. Two Stroke Engine with Supercharger [16]

Supercharging, as mentioned in previous sections, has existed since the beginning of engine mechanics. However, it is not a very common practice in automotive engines, so its greatest development and evolution took place in the aeronautical sector, since with the increase in flight altitude, the pistons used in engines of aircraft lost performance with decreasing air pressure and density, so this mechanism was a solid solution to this problem.

In this way, with the supercharging of an engine, an increase in power is achieved without the need to increase its displacement. This happens because both a diesel engine and a gasoline engine can only see increased power without increasing displacement if, by increasing the injected fuel, it has enough air to carry out complete combustion. The solution to this problem, as has already been mentioned, is an increase in air pressure, so that the same amount of air will occupy less volume or, what is the same, in the same volume more air can enter.

The main advantage of supercharged engines is an increase in engine efficiency and power, as initially stated.

However, it also shows disadvantages. Because the flow injected into the engine is forced, the displacement of the piston during the intake stroke consumes part of the available energy. For this reason, from a certain level of supercharging, the advantages mentioned above may be null or even reduce their performance with respect to the standard case (without supercharging).

In addition, another disadvantage occurs in gasoline engines, since as the air-fuel mixture is produced prior to its injection into the combustion chamber, the high temperatures and pressures at which the air flow comes out of the compressor can cause self-ignition. . That is why for gasoline engines its use is limited to competition vehicles, since the fuels used are high octane (the higher the octane, the higher the autoignition temperature)

3.1.7. 4 STROKE WORKING CYCLE

A 4-stroke engine is one, as previously mentioned, that completes a work cycle in two revolutions of the crankshaft, that is, four strokes of the piston (figure 13).

First run (intake): the intake valves are open and the exhaust valves are closed. The piston is initially at top dead center (TDC) and moves to bottom dead center (BDC) and the intake valve closes. The depression created causes the entry of gases through the intake duct.

Second stroke (compression): both valves are closed and the piston moves from bottom dead center to top dead center, thus achieving compression of the gas that had previously entered the cylinder and combustion of the gas occurs, either by compression or by spark, depending on the engine.

Third stroke (expansion): with both valves closed, there is a displacement of the cylinder from the top dead center to the bottom dead center due to the increase in pressure in the cylinder after the combustion of gases occurs. It is in this career that the only one where you get a job.

Fourth stroke (exhaust): the exhaust valve opens and the piston moves from bottom dead center to top dead center, thus expelling the burnt gases. After this, this valve closes, and with the opening of the intake valve, a new work cycle begins.

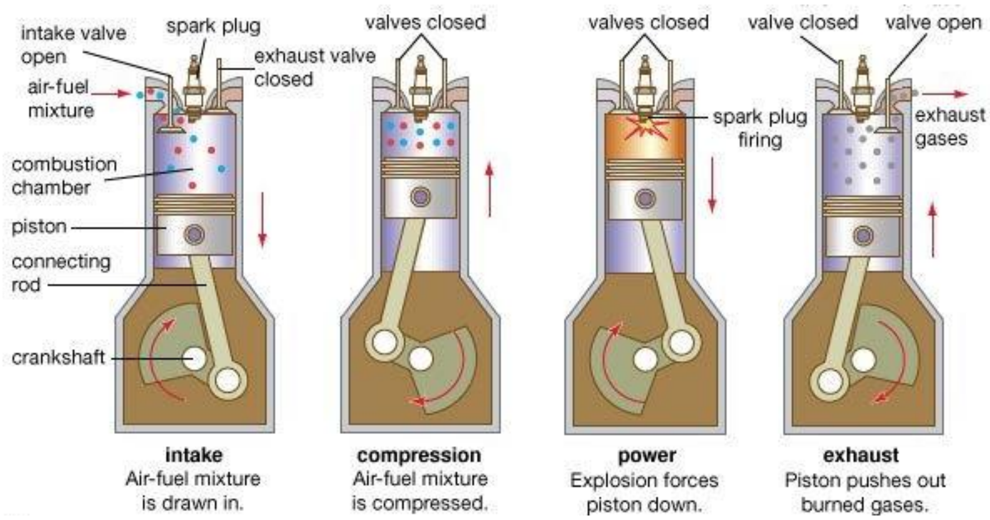

Figure 13. Four Stroke working cycle. [7]

3.1.8. 2 STROKE WORKING CYCLE

Unlike the previous case, it is recalled that in this type of engine the work cycle is completed in a single turn of the crankshaft or, being the same, in two strokes of the piston (figure 14).

In the ascending stroke of the piston, a sweep is produced first, that is, when the piston is at bottom dead center, the gases enter the cylinder through a lateral duct as shown in the image (figure 14). In the same upward displacement, once the gas has been introduced, their compression begins to take place. It is at the top dead center, in the same way as in a 4-stroke engine, where combustion takes place.

On the downstroke, there is an expansion caused by the increase in pressure in the cylinder due to combustion. In this same displacement, the exhaust valve opens on the opposite side, so that, while the piston continues the downward stroke to the bottom dead center, the burned gases are expelled by said valve.

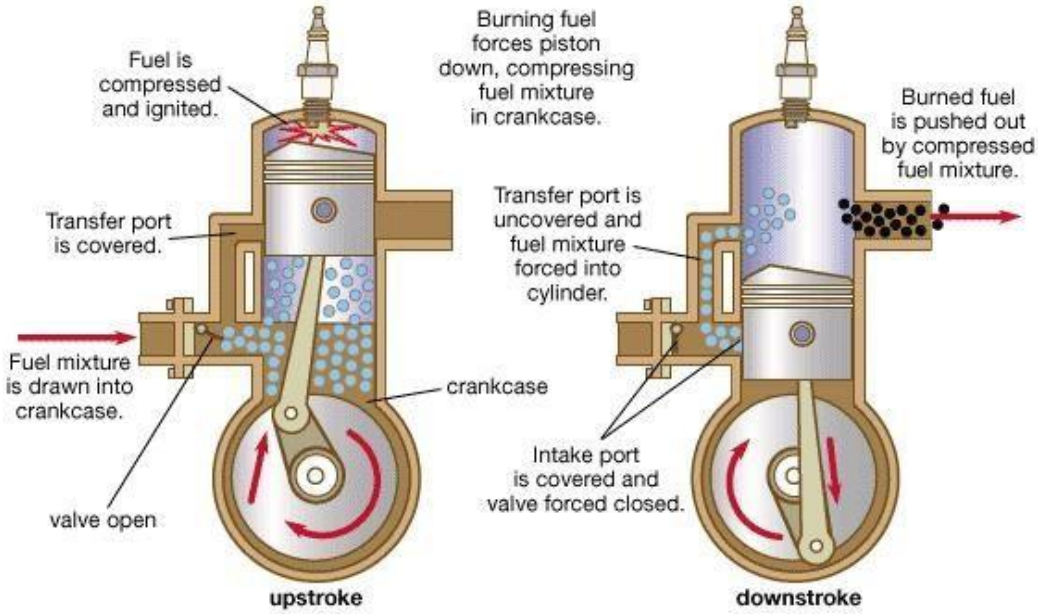

Figure 14. Two Stroke Working Cycle. [7]

3.2. VOLUMETRIC COMPONENTS

A compressor is a device which, through different mechanisms, is capable of increasing the pressure of a flow by a certain percentage. Its applications cover numerous fields. One of them is aerospace propulsion, where one or more of these can be found before the combustion chamber.

As mentioned, a compressor can use different mechanisms to perform its function. That is why, based on this, compressors can be classified into different types (figure 15). In the next section, each of them will be explained in detail.

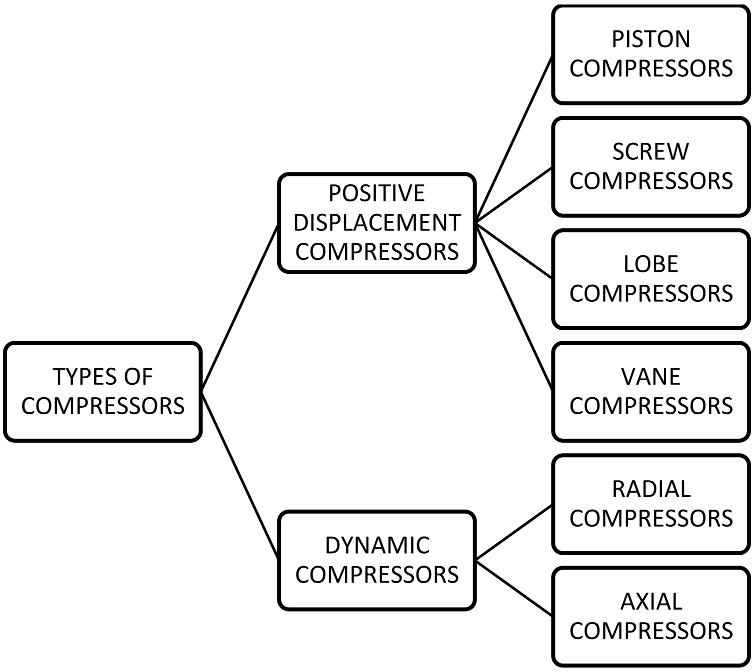

Figure 15. Scheme of diferent types of compressors

3.2.1. TYPES OF COMPRESSORS

POSITIVE DISPLACEMENT COMPRESSORS

They are those whose operation is based on reducing the volume of the chamber where the flow is enclosed, so that its pressure increases until it reaches the desired value to later release it into the system. As seen in the previous diagram, there are several types depending on the way to reduce its volume.

-Piston Compressors (figure 16):

Piston compressors have already been explained above. Remember that they are those in which the air is introduced into a cylinder, and through the upward and downward movement of a piston to compress the air and expel it through an outlet valve.

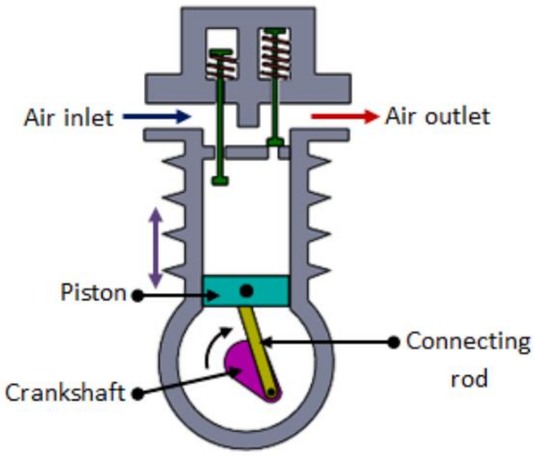

Figure 16. Piston compressor. [17]

-Screw Compressors (figure 17):

Screw compressors are those whose operation is based on the rotary movement of two screws, one male and one female, so that the air moves through the chambers that are created between them, thus being a fixed volume. The air moves linearly through these from the injection zone to the outlet, the pressure zone.

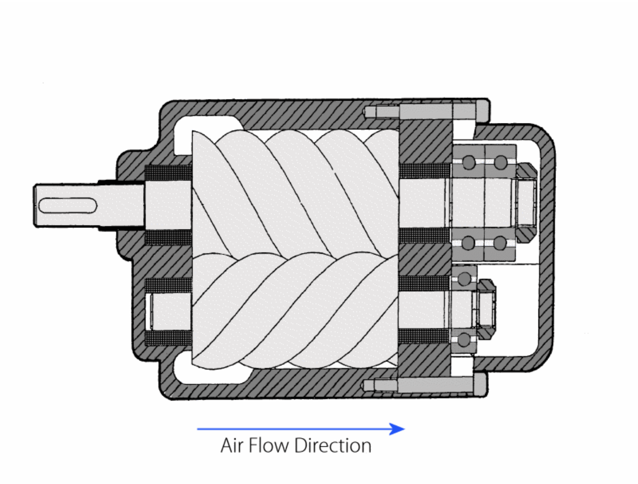

Figure 17. Screw Compressor. [18]

Lobe Compressors (figure 18):

Lobe compressors are those whose operation is based on the rotation in the opposite direction to the advance of the air and in a synchronized way of two lobes so that two chambers are formed (one on each side) through which the air passes. In this case the lobes do not compress the air, they only displace it. The pressure increase is generated due to the back pressure existing at the outlet of the mechanism. This makes them low pressure compressors.

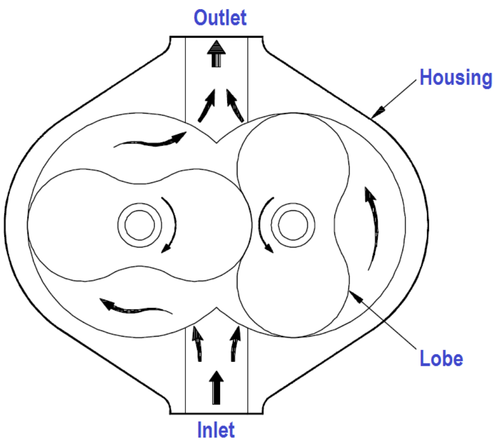

Figure 18. Lobe Compressor. [19]

-Vane Compressors (figure 19):

Finally, vane compressors, which will be analyzed in more detail later, are those whose operation is based on the rotation of an eccentric rotor inside a stator, with blades that go from the center of the rotor to the stator and of length variable, so that they drag the in small control volumes that are formed between the blades whose volume decreases with rotation due to said eccentricity.

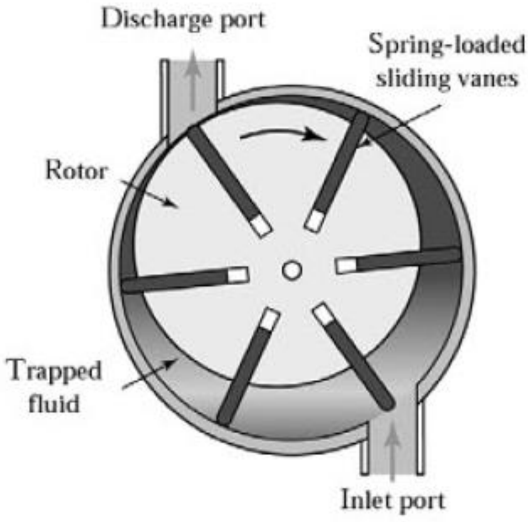

Figure 19. Vane Compressor. [20]

DYNAMIC COMPRESSORS

They are those whose operation is based on molecular acceleration. Air is sucked in through the inlet and accelerated to high speed. After that, it is discharged into a diffuser and all the kinetic energy is converted into static pressure to be released into the system as a compressed flow. The main difference with positive displacement compressors is that dynamic compressors do not require any element that confines the air in a specific volume, but instead it is a continuous flow.

-Radial Compressor (figure 20):

A radial compressor is made up of a rotor and a volume chamber around it. Through the center of the rotor the air flow is sucked in, and due to its rotation, the centrifugal force caused by the rotation of the rotor blades expels the flow outwards, towards the volume chamber. In this chamber, when the flow contacts the walls, the kinetic energy it has is transformed into static pressure. Following the conduit of this volume chamber, the flow exits by the diffuser to the system under the desired conditions. Outlet pressure varies with rotor revolutions.

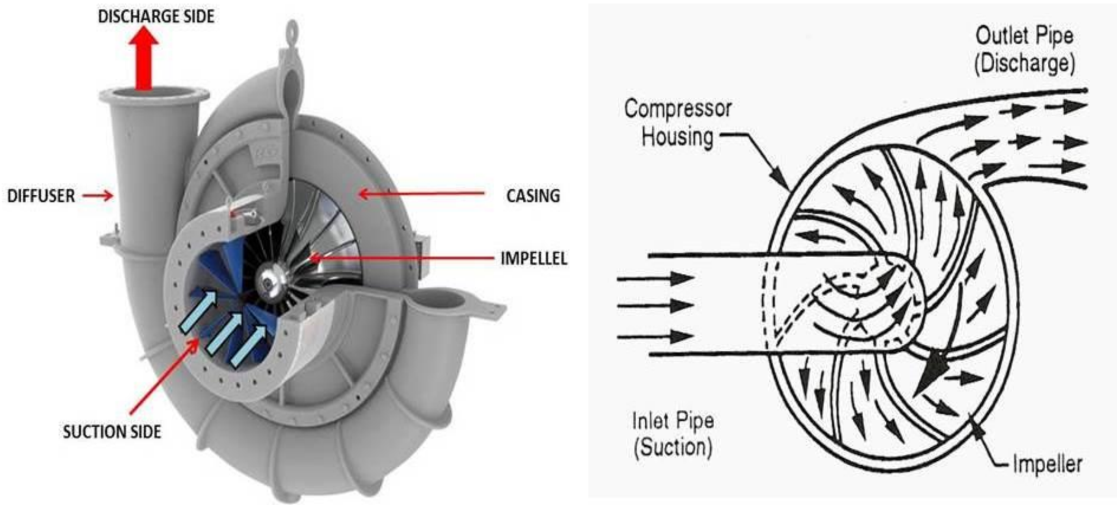

Figure 20. Radial Compressor. [21]

-Axial Compressor (figure 21):

An axial compressor is mainly formed by the achievement of two elements: a rotor and a stator. When the gas passes through the rotor, it is accelerated, and when it passes through the stator, it is decelerated, converting the kinetic energy into static pressure. As many rotor-stator as necessary are inserted to obtain the desired pressure at the same operating revolutions.

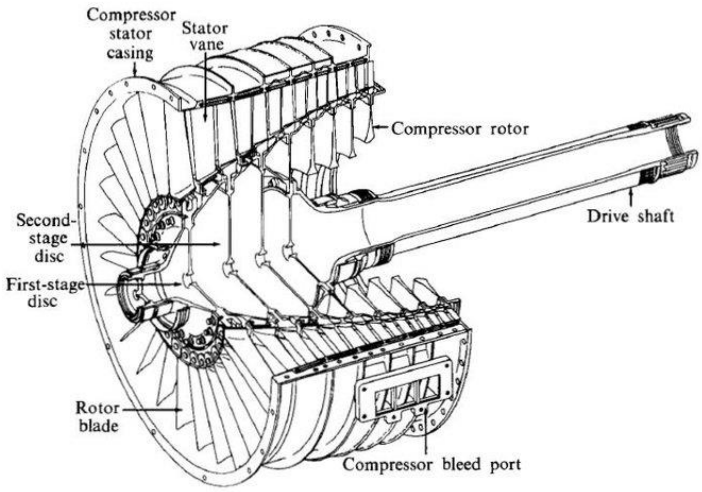

Figure 21. Axial Compressor.[22]

3.2.2. VANE COMPRESSORS

How it has been said before, volumetric vane compressors are a type of rotary compressors that achieve gas compression through the variation of the control volume caused by the eccentricity between the moving part, the rotor, and the fixed part, the stator, both of circular geometry. The blades are responsible for sectioning the 360º of a circle to achieve different control volumes within the same compressor so that the gas is enclosed in a specific space, which varies in volume with the rotation of the blades that drag it. Therefore, it can be said that the control volume of this type of compressor is limited by the rotor, the stator, and two consecutive blades.

In addition, for this to be possible, the vanes must have a different length depending on the position they occupy at any given time, that is, they must have the ability to retract and unfold. For this, floating paddles are used. These blades supported on a layer of oil to lubricate, reach the inner part of the stator (outermost position of the chamber between rotor and stator) thanks to the centrifugal force caused in the rotation.

The compression of the blades also causes an increase in temperature, so pressurized oil is injected into the different chambers. This oil is eliminated in the discharge area by means of an oil separator.

Both the inlet and discharge sections are located along the 360º of the mechanism depending on the classification of the compressor according to the number of blades:

-Monocellular compressor: contains a single blade. The intake section is located as close as possible to the contact point between rotor and stator.

-Bicellular compressor: contains two blades. The intake section is located approximately 90º with respect to the discharge section in the positive direction of rotation.

-Multicellular compressor: contains four or more blades. The intake section is located approximately 180º with respect to the discharge section.

The increase in the number of blades supposes a small decrease in the total volume that is displaced, since these occupy a volume which has to be subtracted from the theoretical total volume. In addition, it is not physically possible to increase the number of vanes infinitely, since for this same volume a number of vanes will be reached that covers the entire surface of the rotor, making it impossible to add more. However, increasing the number of blades to a certain extent favors greater air compression, since the difference in volume of each chamber between intake and discharge will be greater.

To start thinking about the design of a volumetric vane compressor, it is first necessary to know the geometric variables that exist in it (figure 22 and 23):

-Radius of the stator (Re)

-Radius of the rotor (Rr)

-Eccentricity (e) (distance between the center of the rotor and the stator)

-Number of pallets (N)

-Thickness of the blades (Nt)

-Length of the compressor (L) (depth of the compressor if it is viewed in such a way that both the rotor and the stator are two circles)

-Intake area (A1)

-Discharge area (A2)

-Volume of intake tank (V1)

- Volume of discharge tank (V2)

-Angle between blades (B) (obtained as 2PI/N, where N is the number of blades)

-Angular velocity (w)

-Initial intake angle (angIn1) (angle formed by the initial point of the admission with the center of the rotor)

-Final intake angle (angIn2) (angle between the end point of the inlet and the center of the rotor)

-Initial discharge angle (angOut1) (angle formed by the initial point of the exhaust with the center of the rotor)

-Final discharge angle (angOut2) (angle between the end point of the exhaust and the center of the rotor)

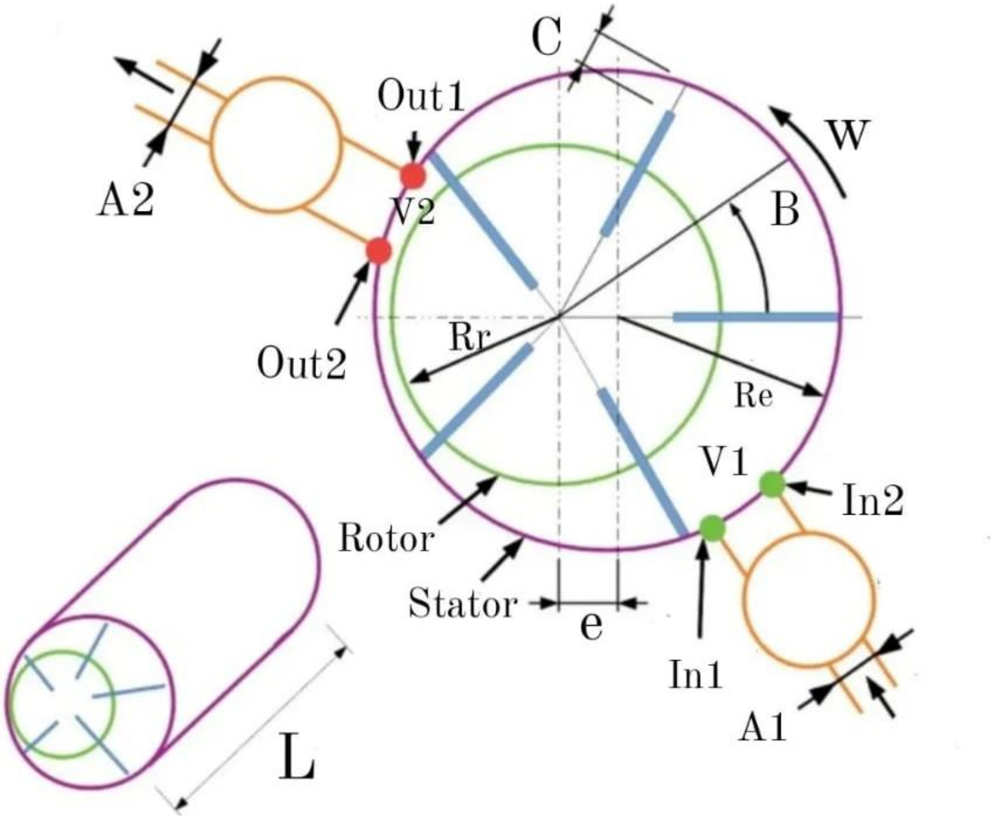

Figure 22. Parameters of a vane compressor. [4]

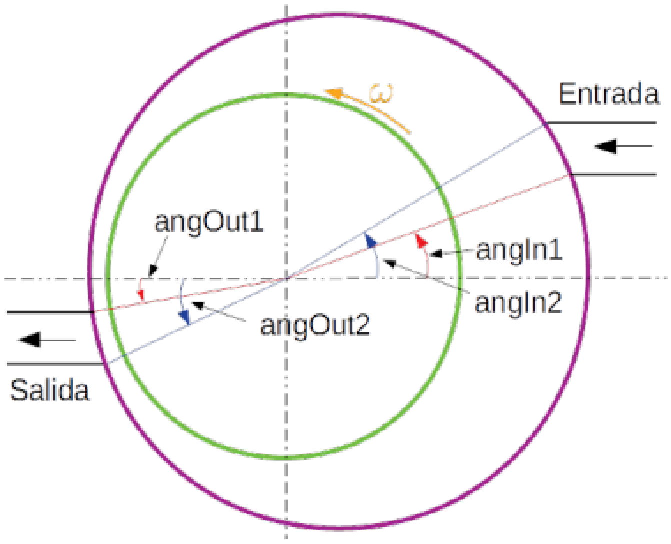

Figure 23. Parameters of a vane compressor. Angles. [4]

After knowing the geometric parameters of a volumetric vane compressor, some nuances must be made about them, since there are restrictions which do not allow us to attribute arbitrary values to all of them:

The rotor must not collide with the stator, so the stator radius must be greater than the sum of the rotor radius and the eccentricity.

The number of pallets must not exceed the maximum allowed, as previously explained.

Regarding the defined entry and exit angles, the initial entry angle must be less than the final entry angle, and the initial exit angle must be less than the final exit angle.

Considering angle 0 the point where the distance from the rotor to the stator is maximum, both input angles must be less than both output angles (in the case of the compressor). In case of using the mechanism as an expander, the previous inequality would be the other way around.

The difference between the initial entry angle and the final exit angle, and the difference between the final entry angle and the initial exit angle must not exceed the angle between the blades, since in this case it would be loading and unloading air simultaneously.

Lastly, as the blades slide outward by centrifugal force and enter again through contact with the wall, they must have a length greater than the maximum distance between rotor and stator (Re-Rr-e) so that it does not come off. of the rotor and less than the minimum distance between the center of the rotor and the stator (Re-e) so that when it passes through this section it can continue rotating and the mechanism is not blocked.

Advantages and disadvantages

As advantages of this type of compressor, we can find first of all the high efficiency that they provide due to the fact that their moving parts are scarce (only the blades, although the rotor is rotating, do not suffer displacement). In addition, the noise levels they generate are reduced. Lastly, their small size and lightness mean that they can be coupled to systems without increasing their weight, which is why in the automotive sector they increase the power of vehicles without excessively modifying their weight.

As its main disadvantage, it is the opposite of the last-mentioned advantage, since its small size means that the air flow it compresses is reduced and it cannot be used for large machines. Also, other disadvantage is that the separation between cells is not fully closed due to the contact between vanes and stator is not perfect, so there is a small gas flow from one cell to other ones. Finally, due to the friction of the blades with the stator at their ends, a good lubrication system is needed so that said friction does not produce losses in the system or considerably increase the temperature.

4. DESIGN

4.1. DIMENSIONING

In the first place, to obtain the dimensions of the compressor, it is necessary to know the material that is available. For the stator part, there is an aluminum cylinder 60 [mm] high and 50 [mm] in diameter.

For the rotor, a 25 [mm] diameter cylinder is also made of the same material. On the contrary, for the pallets the available material is variable and a great variety of these can be applied, so in the dimensioning part this is not relevant.

Once the limitations are known, it is necessary to calculate some dimensions on which to calculate some others in order to obtain a displaced volume of 50 [cc] for each revolution in the combustion chamber (piston in this case), since it is the displacement of the engine HONDA-GX50. In addition, it is also necessary to know that this motor works at 2500 [rpm] (see the engine properties table (Table 6)) and we need to obtain an increase of around 20% of the initial pressure. As losses are estimated, we will try to obtain a greater increase in pressure so that the real pressure obtained is as close to 20% as possible, always being on the side of safety, but without exceeding it, since, as has been seen previously an excessive increase in pressure is counterproductive.

To first obtain the length of the compressor (L), we start by calculating the displaced volume in the compressor with the radius of the rotor and stator, simplifying and taking as a difference between said constant radius in each section, despite knowing that this It is not like this. However, the displaced volume is the same so it is valid for this calculation. Therefore, starting from an internal stator radius of 20 [mm] and a rotor radius of 12.5 [mm], the way to obtain L is:

$$L = \frac{50000}{\pi \cdot (r_1^2 - (r_2 + e)^2)}$$

Equation 1

We have two incognits, the length of the compressor, L, and the excentricity, e. So, we need to obtain other expression with these two incognits.

On the other hand, to obtain the eccentricity between the rotor and the stator, we assume the hypothesis that the temperature of the fluid remains constant (despite suffering an increase due to friction with the ends) and only the pressure and volume are variable. In this way, it must be fulfilled that:

$$\frac{P_2}{P_1} = \frac{V_1}{V_2}$$

Equation 2

As mentioned above, we want to increase the pressure by 20%, but due to the losses, we suppose to increase it by 25%, so the first part of the equality is 1.25. In this way, the expressions of the initial and final volume are put in function of both radio and of the eccentricity and the latter is solved. It should be noted that for this calculation it will be assumed that in the initial volume the difference between radio is constant throughout the section between blades and equal to the maximum and the final volume is the volume of the combustion chamber, because is the volume we want to get:

$$1.25 = \frac{(r_1^2 - (r_2 - e)^2) \cdot L \cdot \pi}{50000}$$

Equation 3

Thus, a length of 72 [mm] is finally obtained. As we have seen that there was a certain limitation due to the materials, this length is not possible. In addition, in the final construction a small component volume is sought. The way to reduce this length is by increasing the transmission ratio, that is, the speed ratio between the engine and the compressor. It is estimated to obtain a length of around 20 [mm]. To obtain a transmission ratio with an integer, a length of 18 [mm] is established, so the transmission ratio established between the compressor and the motor is 4:1. Solving for eccentricity, an eccentricity of 0.88 (~ 0.9) [mm] is obtained.

In this calculation of both the eccentricity and the length of the compressor and its transmission ratio, an angle between inlet and outlet of 180º has been assumed, as previously indicated that it was recommended for multicellular compressors.

To obtain the inlet and outlet areas, it is only necessary to have a diameter less than the length of the compressor and for the angles between one end and the other to be less than the angles between the blades. The latter refers to the fact that the diameter of the outlet section must not be greater than the chord between two blades, which is calculated as follows:

$$k = 2r_1 \cdot sin\frac{B}{2} = 23\ [mm]$$

Equation 4

As this length is greater than L, the maximum diameter of the inlet and outlet section is delimited by L. Then a diameter of 14 [mm] is established.

Below (table 1) is a table with the final values of the dimensions in question proposed in section 3.2.2:

Table 1. Final dimensions values of the compressor

Re	20 [mm]
Rr	12.5 [mm]
e	0.9 [mm]
N	5
Nt	1[mm]
L	18 [mm]
B	72 [º]
w	10000 [rpm]
A1	154 [mm^2]
A2	154 [mm^2]
Compression ratio	4:1

Finally, it is showed some draws of the rotor (figure 24), of the stator (figure 25), of the vanes (figure 26) and of the final compressor (figure 27):

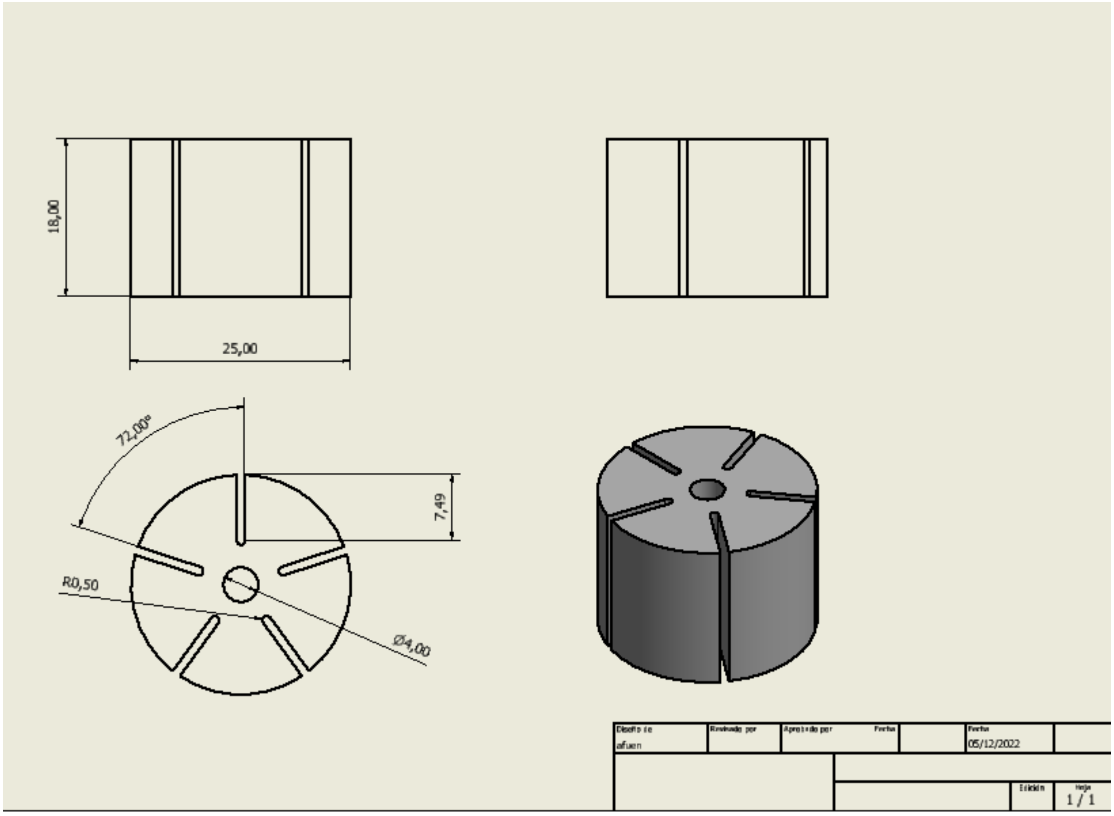

Figure 24. Draw of the rotor. Designed by Inventor

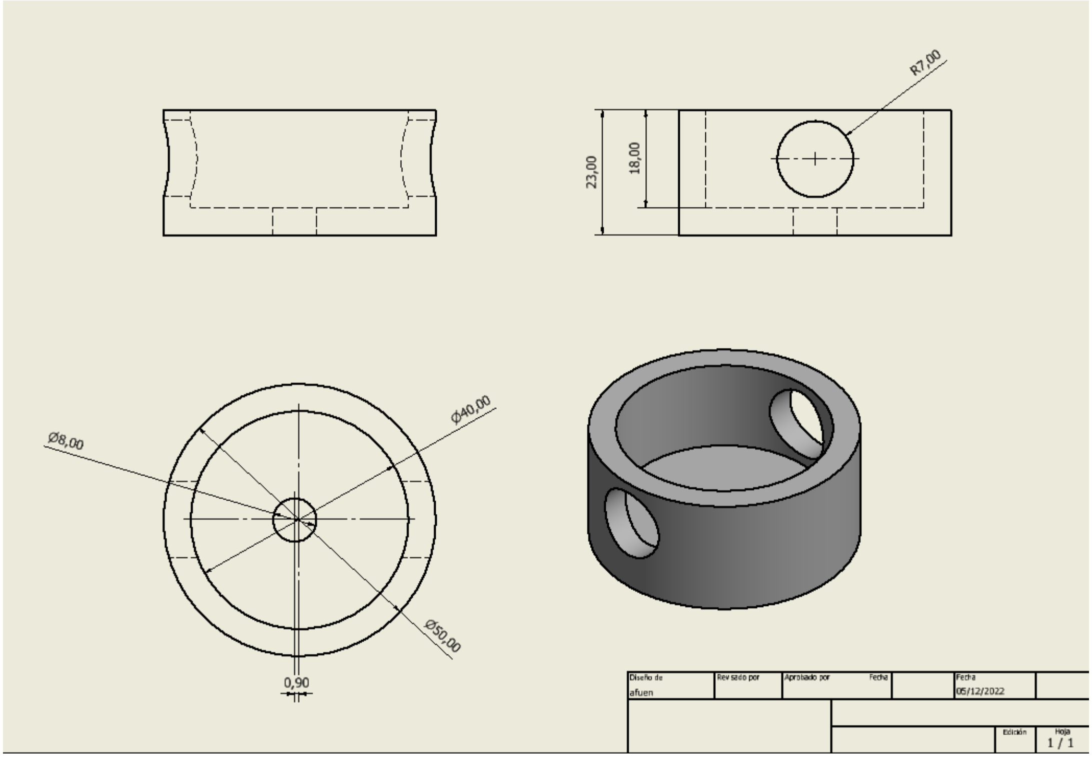

Figure 25. Draw of the stator. Designed by Inventor

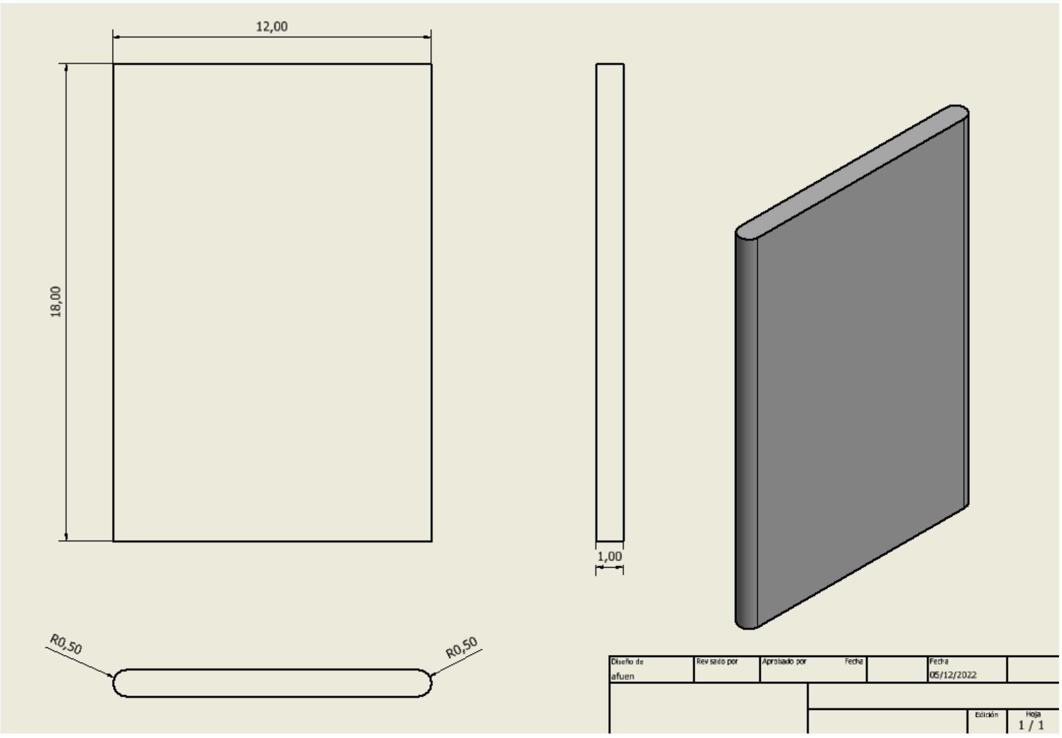

Figure 26. Draw of the vanes. Designed by Inventor

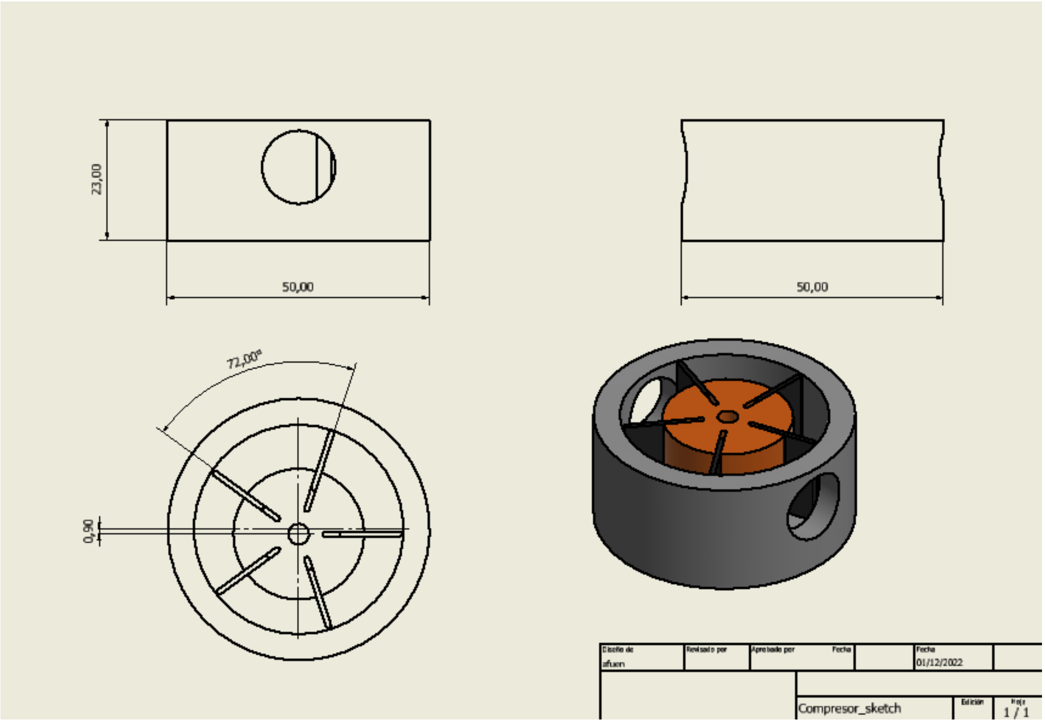

Figure 27. Draw of the final compressor. Designed by Inventor

4.2. ASSEMBLY TO THE ENGINE

Once the measurements of the compressor have been obtained, it is necessary to think about how it will be attached to the engine's carburetor. To do this, a sequence of tubes will be used (figure 24), whose inlet and outlet diameters are those corresponding to the compressor outlet diameter and the carburetor inlet diameter, respectively. In this way, the flow coming out of the compressor is directed directly towards the carburetor so that it mixes with the fuel and enters the combustion chamber of the engine.

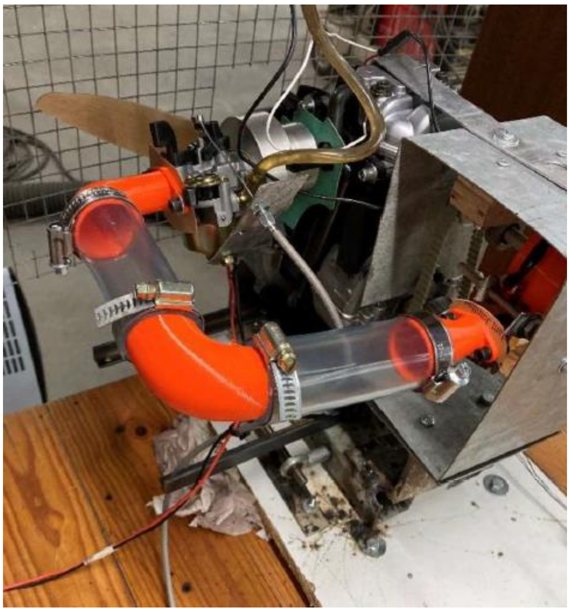

Figure 28. Union pipes. [24]

5. CONCLUSIONS, FUTURE PROJECTS, AND DEVELOPMENTAL PROBLEMS

5.1 CONCLUSIONS

As has been seen previously, despite not having been able to observe the practical case, supercharged engines allow the performance of the engine to be increased without the need to increase the volume of the combustion chamber.

On the other hand, it has been seen how there are different types of compressors that can be previously adapted to the carburetor of an engine to achieve said supercharging. However, the volumetric vane compressor is the simplest mechanically, so to study the difference between a supercharged engine and a normal one it is the most appropriate with the available resources, since the others require a very high precision in their manufacture and a very complex modeling of its geometry, as has been seen in the explanation of each one of them.

The proposed objectives have not been fully met. The part of manufacturing and building and testing the engine has not been carried out due to problems that have arisen in the laboratory. Regarding the understanding of the operation of a volumetric compressor and its usefulness, it has been able to more than meet this objective. Its design has been carried out digitally but it has not been carried out, so this objective has been achieved, but no concrete conclusion can be drawn about it.

In the dimensioning of the compressor, the calculations have been carried out in a simplified way since the objective of the project was not to build an optimal compressor but to build one to be able to test with it later, for which it is possible to obtain approximate dimensions, despite the fact that later the results obtained were slightly different from those expected.

In reference to the practical case, as has been commented, it has not been possible to proceed with its study, so it has not been possible to obtain a conclusion about how it would be possible to improve the performance of the compressor, either by varying the geometry or by perfecting the manufacturing. of the parts and their assembly. From what can be deduced from being knowledgeable about the process of design, manufacture and assembly of the same, as well as its operation, it is possible that the main problems that made the performance of the compressor not as expected were the assembly of the different parts. of the compressor, especially the blades, since as it is a system of floating blades inserted in the rotor, which move outwards by centrifugal force and inwards by contact with the stator, these could not perform such movements as It is expected due to the friction force existing with the components with which it makes contact, so that the different closed volumes that should be created between the blades were not really closed and the gas passed from one to another, losing its compression capacity.

5.2. FUTURE PROJECTS

As in all projects that are carried out, once they have been completed, new questions or new objectives to be achieved in the future arising from the development of the first project always arise.

In this case, the main objective to be achieved in future projects is to carry out the manufacture, construction and testing of the designed compressor. This objective should have been part of this project, but it has not been possible to achieve it, as has already been mentioned and will be explained in detail below.

On the other hand, in future projects I am very curious to go a little further than comparing an engine with and without supercharging. I would like to analyze what would be the optimal compressor parameters starting from a fixed stator diameter and varying both the number of blades, the rotor diameter, the eccentricity, the compressor revolutions... However, for this type of experiment it is not possible to carry out a trial and error experiment manufacturing each model with different dimensions, but it should be carried out with CFD programs until obtaining the optimal model and then manufacturing said model to test it together with the motor and compare the real case with the theoretical one.

5.3. DEVELOPMENTAL PROBLEMS

During the development of the document, it has already been commented on several occasions that problems have arisen in the months of work that have prevented, mainly, carrying out one of the main objectives of the project: the manufacture, construction and testing of the compressor. This is due to the fact that in the months that it was time to attend the workshop there was a workload since there were more students and the tutor had to divide the time he dedicated to each one, which added to the fact that this time was indicative and not exact, meant that some days the working time was minimal.

On the other hand, in the last days of work, using the machines to make the piece, one of these broke down and that part of the manufacturing could not be carried out. After finding his solution and being able to complete the piece, the tutor fell ill and was unable to attend the laboratory during the last days of our stay at the destination.

As for minor problems, the dimensions established in the initially proposed design could not be applied because the available material did not adapt to them and a modification had to be made so that the compressor measurements would adapt to the available material.

6. APPENDIX

HONDA GXH50 [23]

Engine Specifications	
Model	GXH50
Type	4-stroke, OHV, single cylinder
Displacement	49 cm^3 (2.99 cu-in)
Max. horsepower	Net: 2.1 HP (1.6 kW) at 7,000 rpm Gross: 2.5 PS (1.8 kW) at 7,000 rpm
Max. torque (crank PTO)	Net: 2.7 N·m (0.27 kg·m, 2.0 ft·lb) at 4,500 rpm Gross: 3.04 N·m (0.31 kg·m, 2.25 ft·lb) at 4,500 rpm
Carburetor	Float type
Cooling system	Forced-air
Ignition system	Transistorized magneto ignition
Lubricating system	Splash
Starting system	Recoil starter
Stoping system	Ignition primary circuit ground
Fuel used	Unleaded gasoline (octane number 86 or higher)
Fuel tank capacity	1.2 L (0.32 US gal, 0.26 Imp gal)
Fuel consumption	240 g/HPh
PTO shaft rotation	Counterclockwise (from PTO shaft side)

Table 2. Engine Specifications

Cylinder block		
Compression ratio:		8.0:1
Bore x Stroke		41.8 mm X 36 mm (1.65 x 1.42 in)
Sleeve internal diameter		41.800-41.815 mm (1.6445-1.6463 in)
Piston skirt outer diameter		41.770-41.790 mm (1.6445-1.6453 in)
Ring side clearance	Top/second	0.015-0.050 mm (0.0006-0.0020 in)
Ring end gap	Top	0.15-0.30 mm (0.0059-0.0118 in)
	Second	0.15-0.30 mm (0.0059-0.0118 in)
Ring width	Top	0.77-0.79 mm (0.030-0.031 in)
	Second	0.97-0.99 mm (0.038-0.039 in)
Connecting rod small end ID		10.006-10.007 mm (0.3939-0.3944 in)
Connecting rod big end ID		15.000-15.011 mm (0.5906-0.5910 in)
Crankpin outer diameter		14.973-14.984 mm (0.5895-0.5899 in)

Table 3. Cylinder Block Specifications

Cylinder Head		
Warpage (limit)		0.10 mm (0.004 in)
Valve Arrangement:		OHV
Valves:		2
Stem outer diameter (standard):	INTAKE	3.970-3.985 mm (0.1563-0.1569 in)
	EXHAUST	3.935-3.950 mm (0.1549-0.1555 in)
Spring free length		23.7.0 mm (0.93 in)
Cam height		27.972 mm (1.1013 in)

Table 4. Cylinder Head Specifications

Tightening torque specs	
Spark plug	12 Nm; 1.2 kg·m; 8.6 ft·lb
Connecting rod bolt	6 Nm; 0.6 kg·m; 4.3 ft·lb
Crankcase cover bolt	7.5 Nm; 0.75 kg·m; 5.4 ft·lb
Oil case bolt	7.5 Nm; 0.75 kg·m; 5.4 ft·lb
Flywheel nut	22 Nm; 2.2 kg·m; 16 ft·lb
Igniton coil bolt	6 Nm; 0.6 kg·m; 5 ft·lb
Carburetor drain screw	1.2 Nm; 0.12 kg·m; 0.9 ft·lb
Carburetor drain bolt	4.5 Nm; 0.45 kg·m; 3.3 ft·lb

Table 5. Tightening torque Specifications

Engine	
Maximum speed	7,800 ± 150 rpm
Idle speed	2,500 ± 200 rpm
Cylinder compression	4.3 kg/cm^2 (61 psi) at 1000 rpm

Table 6. Engine Working Values

Valve clearance	
Intake valve	0.06-0.10 mm (0.0024-0.0039 in)
Exhaust valve	0.09-0.13 mm (0.0035-0.0051 in)

Table 7. Intake and Exhaust Values

Oil system	
Oil type	Honda 4-stroke or an equivalent (SE or SF)
Recommended oil	SAE 10W-30
Oil capacity	0.25 l (0.26 US. qt, 0.22 Imp. qt.)

Table 8. Oil System Specifications

Ignition system	
Ignition timing	30° B.T.D.C.
Spark plug	**NGK:** CR5HSB **DENSO:** U16FSR-UB
Spark plug gap	0.6-0.7 mm (0.024-0.028 in)
Spark plug tightening torque	12 Nm; 1.2 kg·m; 8.6 ft·lb
Primary ignition coil resistance	0.98-1.20 kΩ
Secondary ignition coil resistance	8-10 kΩ
Ignition coil air gap	0.3-0.5 mm (0.012-0.020 in)

Table 9. Ignition System Specifications

Carburetor	
Main jet	**#55**
Float height	12 mm (0.47 in)
Pilot screw opening	1-1/8 turns out

Table 10. Carburator Specifications

7. LITERATURE

[1] Motores de Combustión Interna Alternativos (MCIA) (1ºParte) (ingenieromarino.com)

[2] Motores de combustión interna alternativos. F Payri y J.M.Desantes. Universitat Politècnica de València.

[3] CICLO OTTO (combustión) | Qué es, tipos, tiempos y funcionamiento (como-funciona.co)

[4] Modelo paramétrico de compresor volumétrico. Trabajo Fin de Grado en Ingeniería Aeroespacial. Carlos Rodríguez Andrade. Universidad de Sevilla.

[5] Compresor de Paletas | PDF | Presión | Eje (scribd.com)

[6] Motores sobrealimentados - EcuRed

[7] El ciclo Otto: motores de dos y cuatro tiempos

[8] https://railroad.lindahall.org/engine500

[9] Etienne Lenoir MCIA ilustration

[10] https://isaacscienceblog.files.wordpress.com

[11] https://th.bing.com/ImgDet

[12] https://fotos.perfil.com/motoresde6cilindrosenlinea

[13] https://img.autosblogmexico.com/jpg

[14] https://2.bp.blogspot.com/jpg

[15] https://www.horizonhobby.com/dw/image

[16] https://bikeadvice.in/Supercharged-Two-Stroke-Engine

[17] https://www.mechdiploma.com

[18] https://th.bing.com/Screw-Compressor

[19] https://i.pinimg.com

[20] https://th.bing.com/ImgRaw

[21] https://th.bing.com/CENTRIFUGAL-COMPRESOR

[22] https://www.researchgate.net/axial-compressor

[23] Honda GXH50 (Gross: 2.5 PS; Net: 2.1 HP) small engine: review and specs (engine-specs.net)

[24] Implementação e ensaio de um compressor centrífugo mecânico de pequenas dimensões. Dissertação para obtenção do Grau de Mestre em Engenharia Aeronáutica. Édi Élson Pires Monteiro. Universidad de Beira Interior.