POLITECNICO DI TORINO

Master's Degree in

AUTOMOTIVE ENGINEERING
(INGEGNERIA DELL'AUTOVEICOLO)

Master's Degree Thesis

The Barrel Internal Combustion Engine

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ABSTRACT

Traditional Internal Combustion Engines are characterized by the presence of the crank mechanism. This solution is able to convert the linear motion of the piston, pushed by the combustion pressure, into a rotational motion. The crank mechanism is still used nowadays in all the ICEs. One of the main aspects of the crank mechanism is the output torque that is generated and its dependence on the crank angle. It is important to have in mind that the work done by the piston during its downward movement towards the BDC is not fully converted into useful work at the crankshaft. This effect is due to the different angles and positions the components of the crank mechanism assume during the operation. Based on this concept, my thesis is focused on the analysis of an alternative mechanism to be implemented in an ICE. The innovative system considered is based on Mr. Bertotti's idea. Its system has the target to exploit in a larger amount the downward movement of the pistons and consequently obtaining an increase in the output torque from the engine. In practice, my job was to reproduce the mentioned engine mechanism in a 3D environment (SOLIDWORKS) and to simulate the working operation in order to verify the possible gain in the output torque of the system.

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Chapter 1

Traditional crank mechanism

The crank mechanism is a system made up by a crank that is coupled to a connecting rod. The function of this system is to convert the reciprocating linear motion of the piston into the rotational motion of the crankshaft. In practice, the crank mechanism is able to convert the thermal energy of the charge within the combustion chamber into mechanical work. This process occurs through the engine working cycle that corresponds to the periodical sequence of intake, compression, expansion and exhaust phase. If we consider a four-stroke (4-T) engine, each of the four phases of the engine cycle corresponds to one stroke of the piston. So, to complete the engine cycle, a 4-T engine requires four strokes. In particular, the intake and expansion strokes are downward movements of the piston while the compression and exhaust ones are upward movements. In general, the crank mechanism is studied from a kinematic perspective to determine displacements, velocities and accelerations, and from a dynamic point of view in order to evaluate forces and moments that characterise this mechanism.

1.1 Kinematics

In order to evaluate the kinematics of the crank mechanism, it is usually adopted a simplified scheme of the system in which the conrod and the crank are represented by two lines characterized by the main parameters of the real mechanism. The crankpin, the wrist pin (corresponding to the piston position in the cylinder) and the crankshaft rotational axis are instead indicated by three different points. This scheme is represented in the *figure 1.1*. In particular, the configuration taken into account is the centred layout characterized by the cylinder axis that intersects the crankshaft rotational axis and the wrist pin one.

Starting from the geometrical correlations characterizing the simplified scheme, it is possible to extract the relationship between the piston displacement x_p and the crank angle θ.

$$x_p = r\left[1 - \cos\theta + \frac{1}{\Lambda}\left(1 - \sqrt{1 - \Lambda^2 \sin\theta^2}\right)\right] \tag{1.1}$$

where $\Lambda = \frac{r}{l}$ is the elongation ratio, that is and index of the maximum angle reached by the conrod.

Once this relationship is extracted, it is possible to determine both the speed and acceleration characteristics of the piston motion.

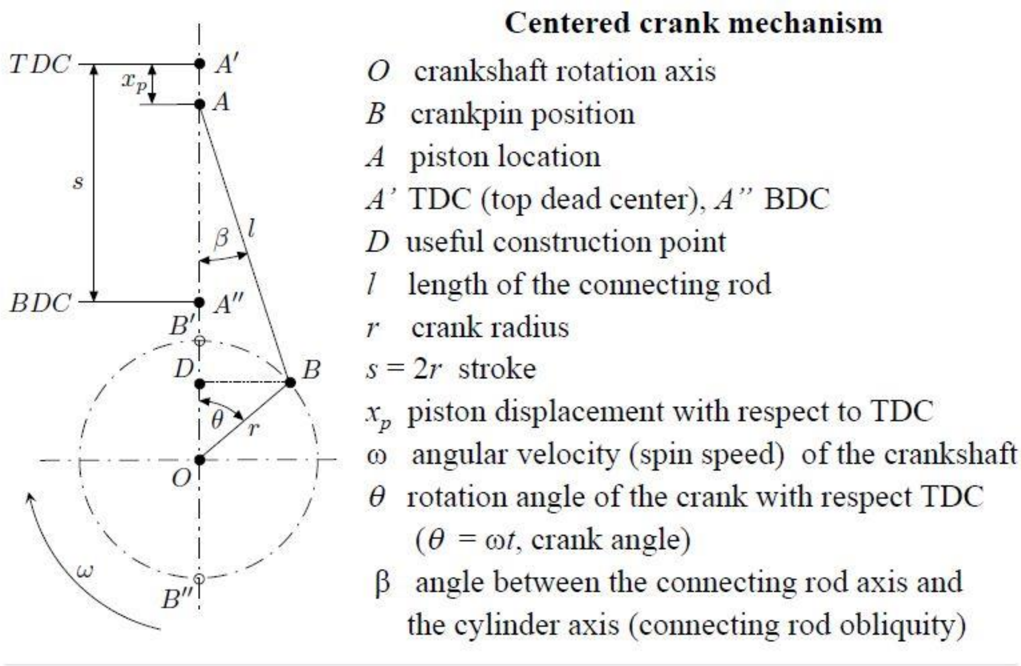

Figure 1.1. Scheme of the crank mechanism (centred layout)

1.2 Dynamics

For what concerns the forces acting on the crank mechanism, the starting point for the dynamic investigation of the system is the force F, acting on the piston along the cylinder axis. This force F is the summation of the instantaneous gas pressure due to the combustion of the charge inside the chamber and the instantaneous inertia force mainly due to the mass of the piston. As it is depicted in the *figure 1.2*, the force F can be split into two components that are F_n, the normal force acting in a direction that is perpendicular to the liner, and F_{cr} that is the contribution of the force that acts along the connecting rod. Considering the *figure 1.2*, the axial thrust force along the conrod F_{cr} has an arm d. Consequently, the torque resulting from this mechanism acting on a single crank is

$$M = F_{cr} d \tag{1.2}$$

It is then possible to express the torque M as the function of the crank angle θ and the force F, through the application of the geometrical relationship that characterize the crank mechanism.

$$M = Fr\left(\sin\theta + \frac{\Lambda \sin\theta \cos\theta}{\sqrt{1 - \Lambda^2 \sin^2\theta}}\right) \tag{1.3}$$

Neglecting the term $\Lambda^2 \sin^2\theta$ with respect the unit

$$M \approx Fr\left(\sin\theta + \frac{\Lambda}{2}\sin 2\theta\right) \tag{1.4}$$

The result is the torque acting on the single crank. It is clear that its value has a strong dependence on the crank angle and so, on the relative position between the connecting rod and the crank.

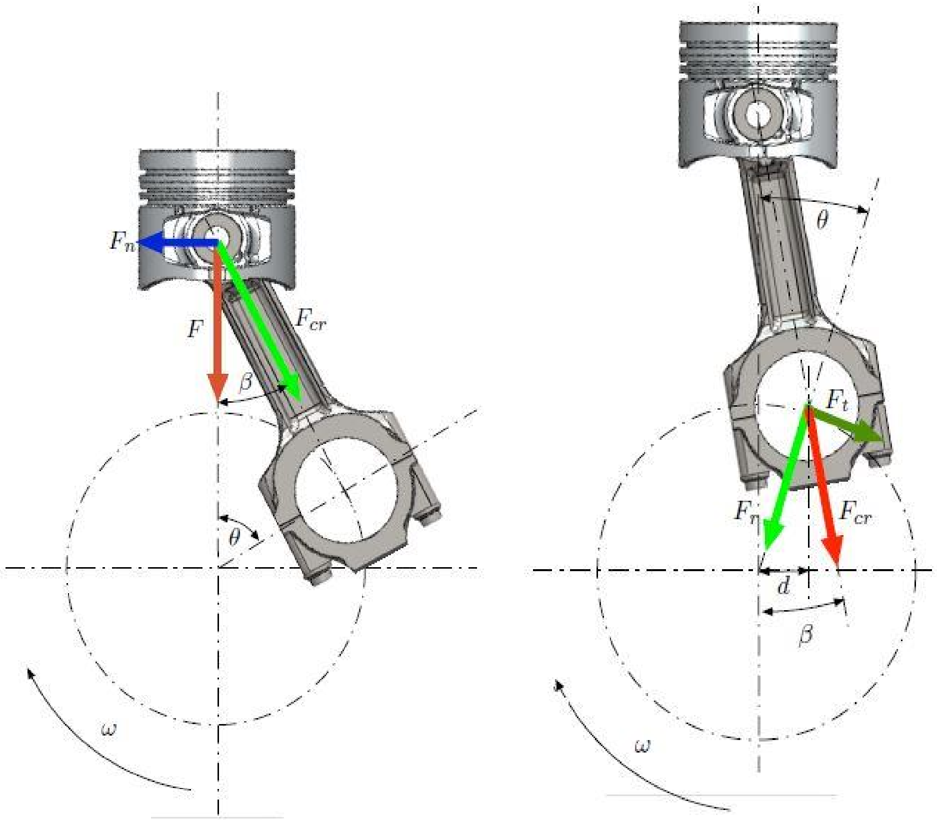

Figure 1.2. Force components in the crank mechanism

Chapter 2

Mr. Bertotti's project

Considering a traditional internal combustion engine, the explosion of the charge inside the combustion chamber, that occurs in the proximity of the Top Dead Centre (TDC), forces the piston towards the Bottom Dead Centre (BDC). The pressure peak of the combustion is converted into useful torque by the crank mechanism, as it is explained in the previous chapter. Mr. Bertotti's attention is focused on the fact that the explosion and the production of useful torque occurs during the first quarter (0° - 90°) of the full crank angle, while in the second quarter (90° - 180°) the output torque results to be negligible due to the position of the crank mechanism characterising this second part. So, taking into account this fact, Mr. Bertotti's idea is to recover the torque in the second quadrant.

"*Ci si prefigge di recuperare la coppia non utilizzata nel secondo quadrante*" [1]

In order to obtain these results, a barrel is thought to be put in-between two transversally aligned cylinders (*figure 2.1 blue rectangle*). This barrel is equipped with an appropriate cam that works as a guide for the roller mounted in a modified piston. In this way, the piston transfers to the barrel, the power received by the explosion occurring inside the chamber, making the barrel itself rotating. The rotation of the barrel, and the consequent torque, is then transferred to the output shaft of the engine through some conical gear couplings. In addition, the motion of the barrel is coupled to the rotational motion of the two crankshafts, one for each of the two pistons, also in this case through the use of spur gear mating (*figure 2.3*). This is done in order to transfer the motion and also to obtain a perfect phasing of the system so that every component is in the right position during the right time instant.

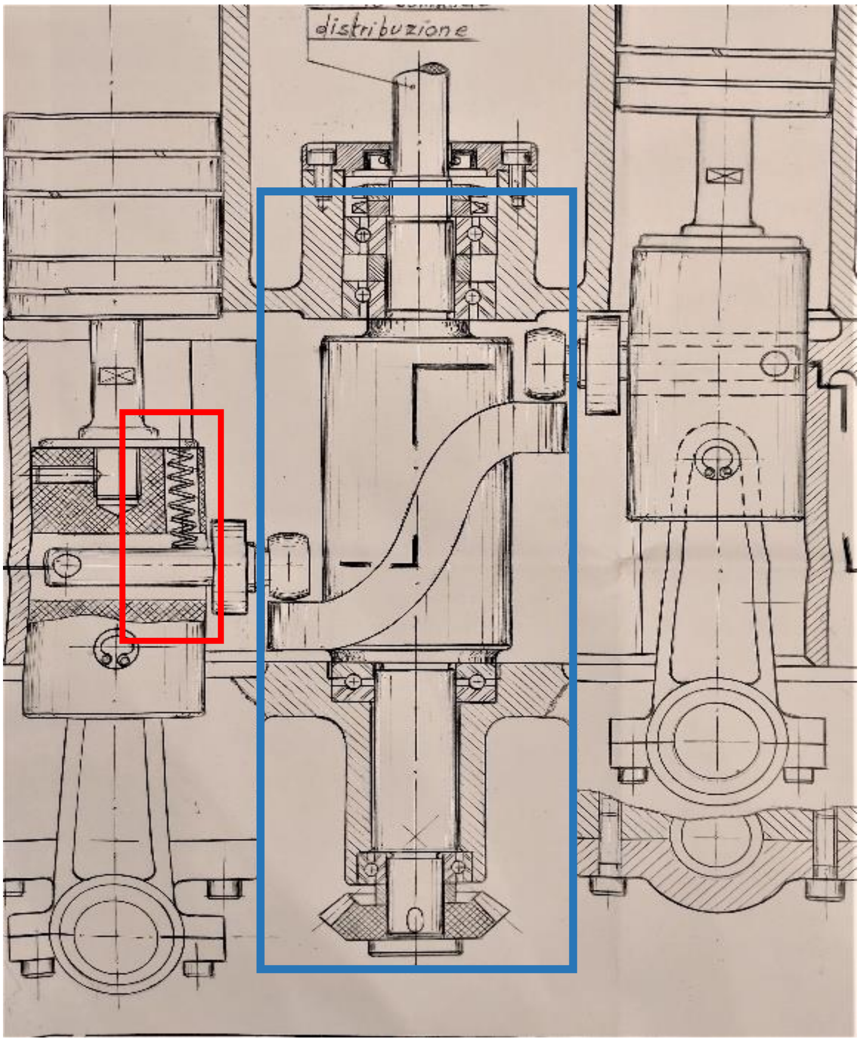

Figure 2.1. Scheme of the Barrel ICE (frontal section)

Considering in a more detailed way the system, and in particular taking into account the piston and the barrel, the profile of the cam is thought to be linear with respect to the semi-circumference of the cylinder. In practise, the profile to be enveloped around the surface of the barrel is an isosceles triangle (*figure 2.2*), with the base that corresponds to the length of the circumference of the cylinder, while the height is equal to the stroke value of the piston. This means that the piston is forced to follow the linear profile of the cam during its working operations. But this is in contrast with the displacement profile characteristic of the traditional crank mechanism seen in the previous chapter. In other words, there is a difference in the displacements coming from the cam profile and the traditional piston profile.

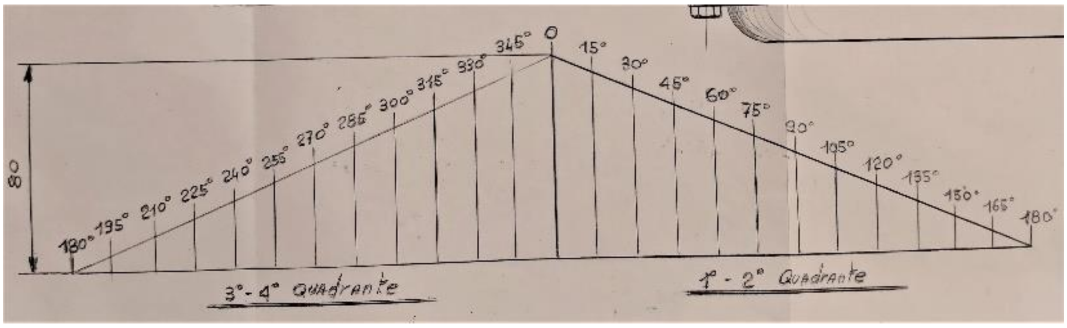

Figure 2.2. Planar development of the cam profile around the barrel

To overcome this issue, Mr. Bertotti’s solution is to couple the roller, inserted into the extension of the piston, with a spring represented in the red rectangle of *figure 2.1* in order to allow the roller itself to tilt and so keeping the contact with the cam while following the motion law of the crank mechanism. Another important aspect considered during the realization of the project is the external shape of the roller. The chosen shape is the barrel one. This solution is driven by the fact that the cam is realized in the axial direction of the cylinder, determining two different diameters with a different linear development. This characteristic of the cam obliges to avoid the use of a cylindrical roller because the result would be different from a pure rolling motion, that is the one desired.

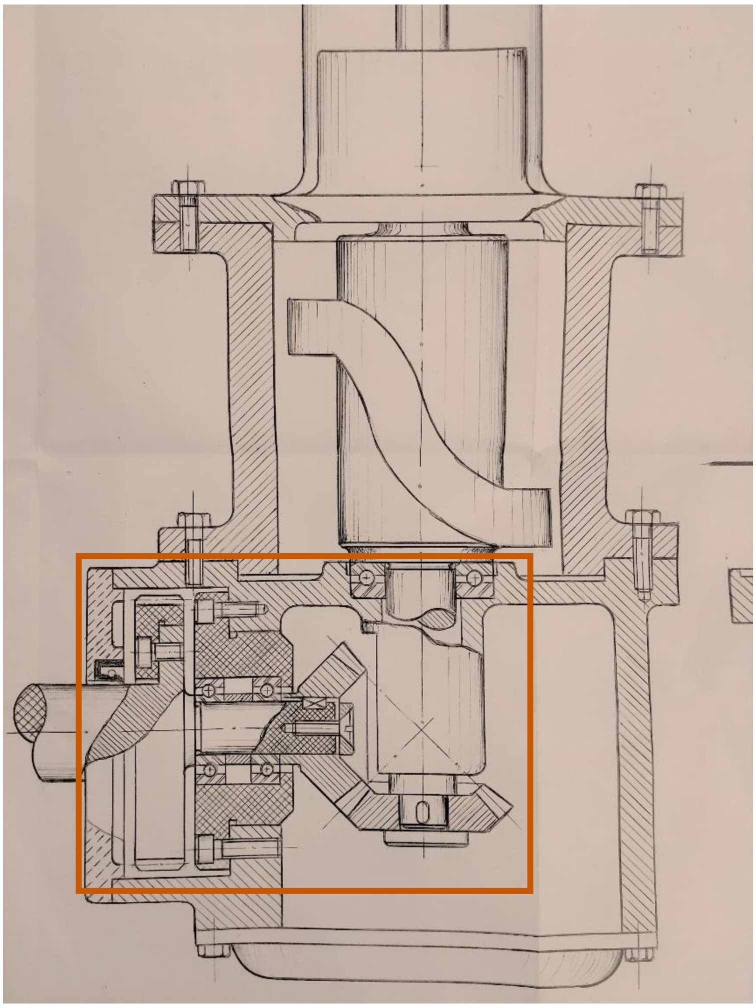

Figure 2.3. Scheme of both spur and conical gear mates

Chapter 3

Preliminary considerations

Starting from Mr. Bertotti's project, described in the chapter 2, the main assignment is to analyse the considered innovative mechanism, and to determine the possibility or not of an increase in the output torque with respect to a traditional two-cylinders ICE. The first step of the analysis is the development of the 3D model of the proposed unconventional engine. Before starting the build-up of the different components of the engine, a preliminary analysis of the project must be performed, with the consequent considerations that are driven by a deeper observation of the schemes and their understanding.

3.1 Main target of the mechanism

The first point to be clarified is the following: the main goal behind the idea of the "barrel cam ICE" is the recovery of some of the power that is not fully exploited by the crank mechanism during the second quarter of its operation. However, realistically talking, during the second quarter of the crank cycle the combustion of the charge is almost fully done. This means that the torque thought to be recovered in this part is practically negligible. Therefore, it is decided to analyse the possible increase in useful torque determined by the barrel cam device during the first quarter (0 - 90 C.A.°) of the crank cycle.

3.2 The barrel cam

Going deeper inside this preliminary analysis of the project, the attention is focused on the cylindrical element that it thought to act as a guide for the movement of the piston. The first aspect considered is that the cam profile is extruded positively/externally with respect to the cylindrical surface. So, in order to increase the level of manufacturability of the component, it is decided to extrude negatively/internally the guide for the roller to obtain a sort of deep groove around the surface of the barrel. The second aspect of the component under analysis is the actual profile of the groove. An important feature of the roller and cam mating is the difference between the path forced by the groove and the one resulting from the traditional crank mechanism. Since, according to the project, the cylindrical cam is actively involved in the transmission of the motion and power only during the downward stroke of the piston, the idea is to create an asymmetrical groove profile. One half (semi-circumference of the cylinder), corresponding to the upward stroke of the piston, has the same characteristic of the piston displacement in the crank mechanism, while the other half has a steeper profile. This allows to have in theory no difference between the piston motion driven by the crank mechanism and the one driven by the barrel cam during the motion of the piston from BDC toward the TDC.

3.3 The piston

Strongly related to the cylindrical cam analysis, it is the one related to the piston. It is clear indeed that the two main components characterizing the unconventionality of the engine taken into account are the barrel cam and the piston. The piston has the peculiarity of being composed by an upper part that corresponds to the piston head of a traditional piston and an extension composed by the assembly of some subcomponents such as the roller. One important aspect of the piston in the project is the elastic mating of the roller with respect the piston itself. This solution, as it is explained in the previous chapter, has the function to maintain the roller in contact with the cam profile even if the motion laws derived from the different parts of the mechanism are not coincident. The issue derived from the adoption of this expedient is that it will not transfer the energy, and therefore the torque, in the desired way. To achieve this task, it is necessary a rigid link between the roller and its housing in the piston. By the application of this new configuration, a consequent problem arises. The piston is found to be forced to follow the two different axial pathways, highlighted before. It is clear that this condition is impossible to be achieved unless there is the decoupling of the two motion laws. This decoupling is thought to be realised by converting the holes in which it is inserted the wrist pin into a slot. In this way the wrist pin is free to move along the length of the slot and consequently decoupling the motion of the piston driven by the barrel cam from the one driven by the crank mechanism. For what concerns the other components, there are no aspects to be further understood or modified.

Chapter 4

Development of the 3D model and simulation

The preliminary analysis of the project gives the basis for the successive step of the investigation process, that is the modelling of the engine in a 3D software environment. The software used to accomplish this task is SOLIDWORKS. Thanks to its large number of functions, it is possible to reproduce the considered assembly, with the proper simplifications, and to perform a first kinematic and dynamic simulation during the working operation. It is then possible to visualize in a graphic representation the significant results, such as forces, torques, velocities and displacements, that are useful for the verification of the advantages of this particular device.

4.1 Creation of the components

By doing several measurements on the concrete project papers and taking into account the modifications discussed in the previous chapter, I am able to reproduce in the Solidworks environment the several components to be later assembled.

4.1.1 Barrel cam

The cylindrical cam is realized starting from the simple revolved extrusion of the semi profile of the drawn component. The groove to be machined inside the lateral surface of the cylinder is realised later. Considering the cam profile proposed in the project, some modifications with respect to the original line must be done. Instead of a triangular profile, the line describing the motion of the roller must be the succession of tangent lines in order to avoid the detachment of the roller from the guide or an incorrect motion. Therefore, the vertices of the triangular profile must be smoothened but contemporarily maintaining the original height of the profile.

Taking into account the sum of the considerations of the previous chapter and the ones expressed above, the procedure for the construction of the guide consists of the preliminary build-up of the traditional piston displacement line as a reference, and the consequent construction of the actual profile of the cam. In order to have a correct profile to be enveloped around the cylinder, the sketch has length equal to the circumference of the base of the cylinder itself. As expressed in the Chapter 3, only one half of the entire profile corresponds to the *equation 1.1*. The other half is characterized by a steeper slope. This can be easily observed in *figure 4.1* where the blue line represents the piston displacement expressed in the Chapter 1, while the black one is the modified profile. The blue profile is represented in both the halves of the sketch for a better comparison between the lines. Once the profile is realized, it is used as guideline to cut the groove in the cylinder. The groove width is chosen equal to 10 mm, while its height is 15mm. An important additional comment to do is that the groove position is chosen in order to keep unchanged the relative positions between roller, piston and cam expressed in the drawings.

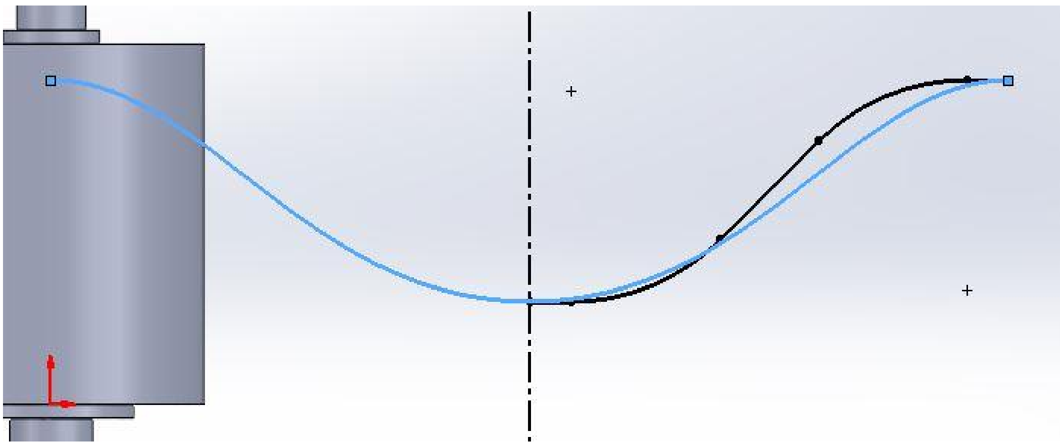

Figure 4.1. Planar development of the profile of the cam

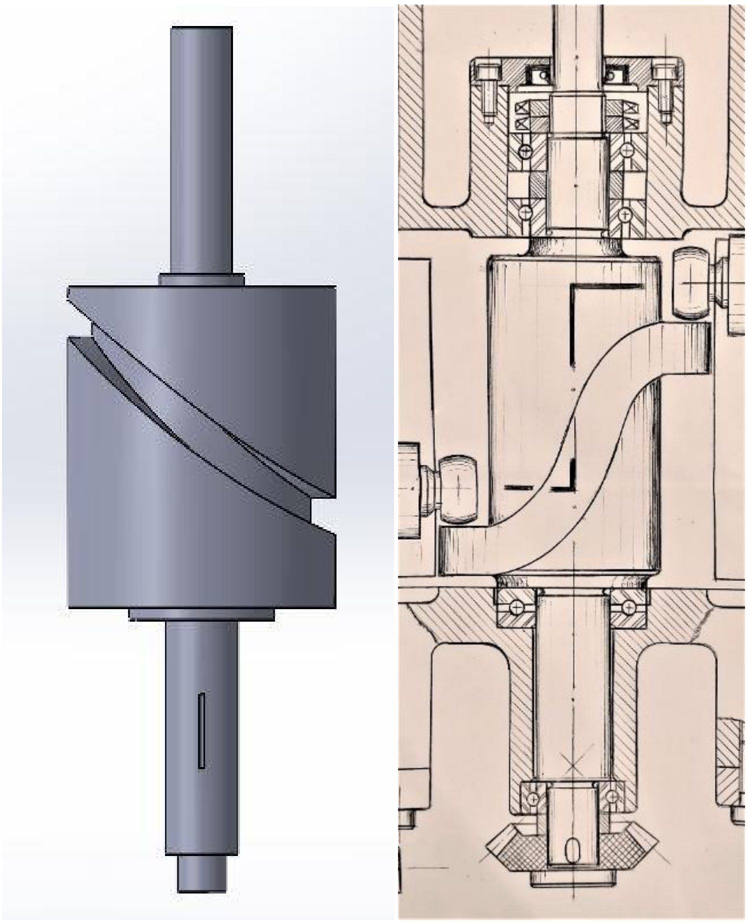

Figure 4.2. Barrel cam - 3D model and sketch

4.1.2 Piston and roller

The piston that is designed by Mr. Bertotti is significantly different from the traditional one due to the presence of the second cylinder that is connected to the one facing the combustion chamber and connected trough a simple rod. This second cylinder is particularly more complex than a traditional piston. It is in fact composed by many subcomponents such as the pin to fix the rod that connects the two cylinders, the roller with its corresponding pin and the spring for the elastic connection between roller and housing in the piston. During the creation of the 3D model, many simplifications are adopted. First of all, the piston is made up as a single solid body, so considering the component as a single part and not as the assembly of several components. The roller itself is reproduced as a cylindrical extrusion coming out from the second cylinder, with a length equal to the one that goes out from its housing in the piston. The second important difference between the designed Piston and the modelled one is the shape of the housing for the wrist pin. Instead of being a simple circular hole, it is modelled as a slot with the width equal to the diameter. In this way the wrist pin is free to move along the length of the slot, with a resulting decoupling of the crank mechanism with respect the piston motion as it is highlighted in the Chapter 3. The slot chosen length is at least as large as the maximum axial distance between the blue and black profile in the *figure 4.1*. As it can be seen in the *figure 4.1*, the black line crosses the blue line determining a portion in which the former is above the latter and a portion in which the opposite condition occurs. As a result, the slot is realised taking as reference the original hole and its position, and then an increase in the slot length is applied along the axial direction of the piston for both the upper and lower side of the hole.

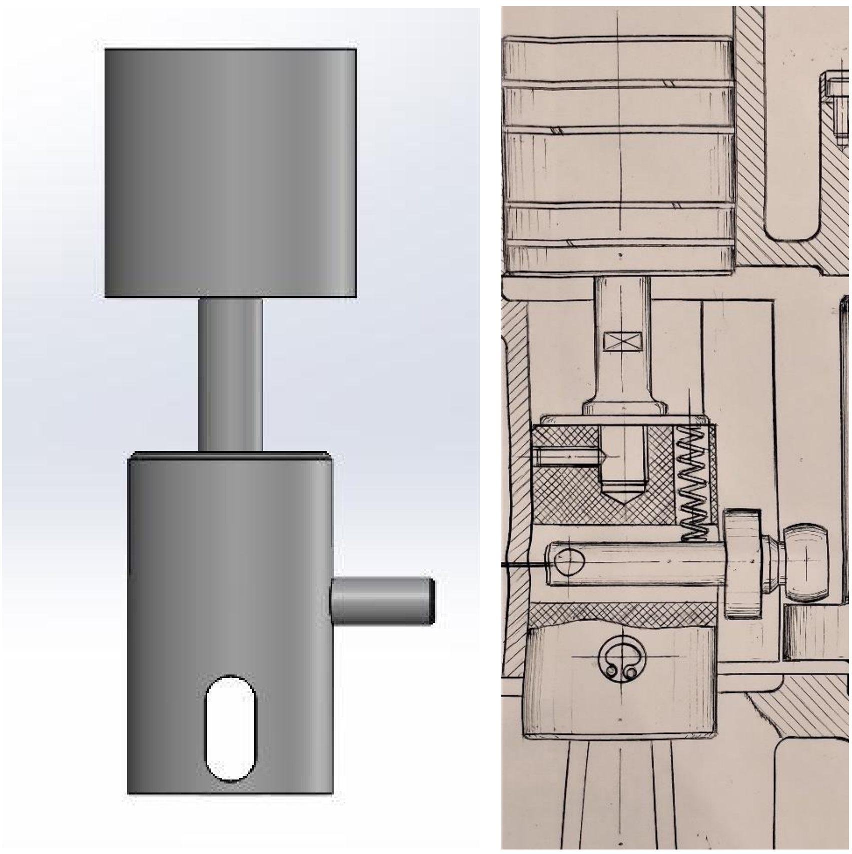

Figure 4.3. Piston - 3D model and sketch

4.1.3 Connecting rod and wrist pin

Due to the intrinsic simplicity of the wrist pin and the connecting rod, these two components are not so difficult to be modelled in a 3D environment. In particular, the wrist pin is realised as a simple cylinder with a length equal to the maximum diameter of the lower cylinder of the piston to facilitate the assembly within Solidworks. For what concerns the connecting rod, instead of reproducing the one represented in the drawings of Mr. Bertotti, it is chosen to model a more traditional conrod. So, the basic structure is made up by the two eyes, the big and the small one, connected by the stem. The distance between the centres of the two eyes is exactly equal to the corresponding value in the project. In this way it is achieved also a simplification for what concerns the assembly of the modelled engine.

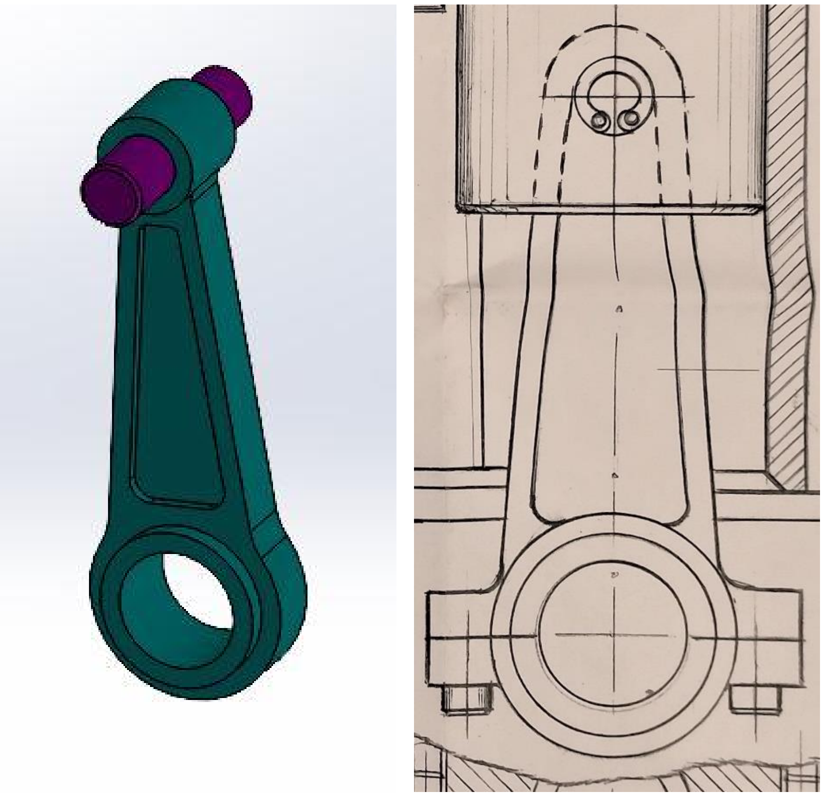

Figure 4.4. Conrod and pin - 3D model and sketch

4.1.4 Crankshafts, output shaft and conical gear

Starting from the description of the project done in the Chapter 2, it is explained that the different parts of the engine, in particular the two crankshafts, the output shaft and the barrel cam are joined through the adoption of gears. This solution, as it is expressed, allow the transmission of motion and the maintenance of the correct phasing among the several moving parts. The mating adopted between the three rotating shafts is achieved by mean of three spur gear wheels characterized by the same diameter. In addition, due to the need to have the shafts rotating at the same angular speed, the gear ratio chosen is equal to 1. So, considering the creation of the model, the two crankshafts and the output shaft are respectively built-up as single solid bodies with their corresponding gearwheel. The gears themselves are modelled as simple discs. In order to simplify the model and to reduce the computational effort of the computer. The simulation of the gear coupling will be possible thanks to the specific mate present in Solidworks. For what concerns the shafts modelling, the two crankshafts are simply reproduced taking as reference the drawings of the project and maintaining the most important lengths unchanged, such as the crank radius. To simplify the assembly in the Solidworks environment, the distance between the crank web corresponding to the length of the crank pin is chosen equal to the thickness of the big eye of the conrod. Also the output shaft of the engine is a simple reproduction of the one drawn in the sketch.

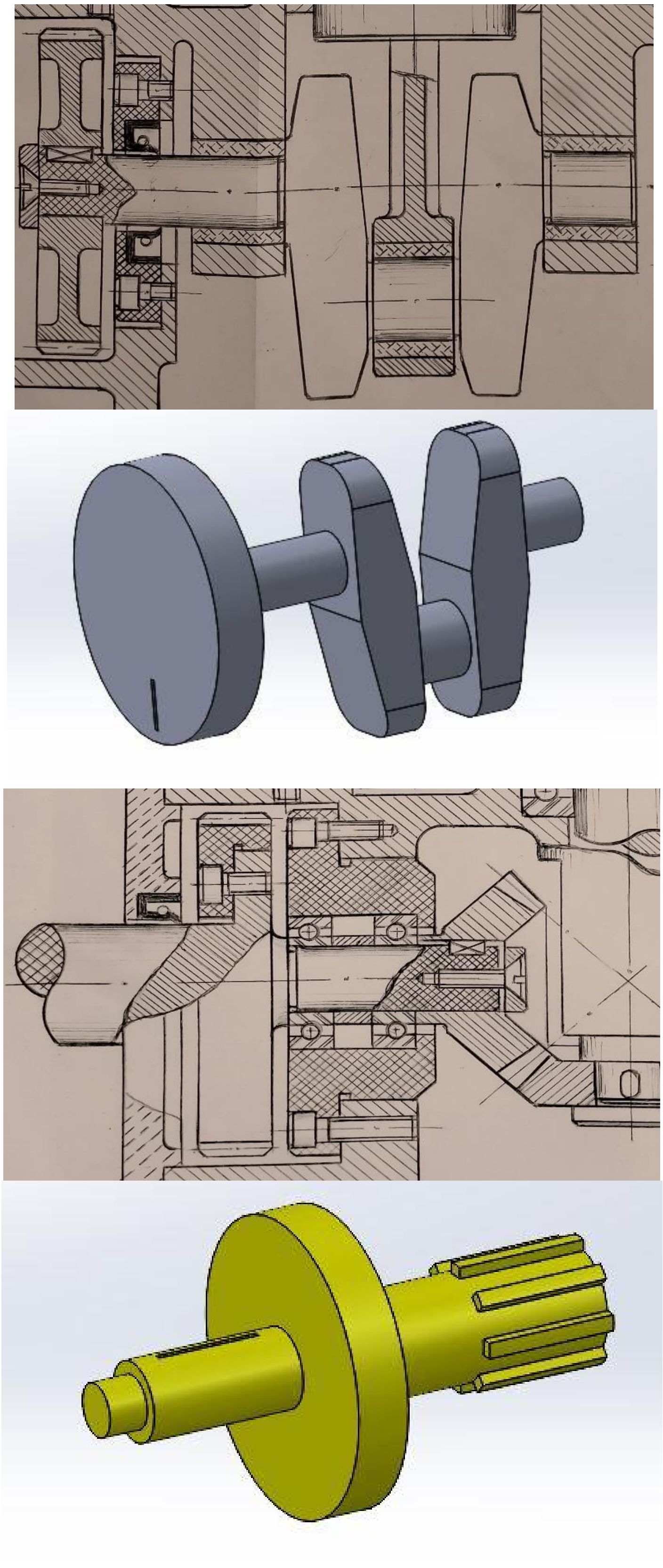

Figure 4.6. Output shaft - 3D model and sketch

Considering again the gear link that characterize the whole mechanism, the coupling that is designed between the barrel cam and the output shaft consists of two conical gears with gear ratio equal to 1. In order to model these components, it is created a single conical gear that can be matched with both the output shaft and the cylindrical cam. In this way, by duplicating this component it is possible to reproduce the conical gears mating that can be seen in the project. An important aspect to be underlined is that in the assembled engine model this part is not inserted due to the fact that during the simulation the gear mating between the two conical gear is not considered. To overcome this issue, a gear mate is applied between the bar of the barrel cam and the output shaft maintaining the gear ratio equal to 1. As a consequence, the conical gears are not part of the final assembly because are not necessary and a smaller number of components is preferred in order to further reduce the complexity of the model and consequently the computational times.

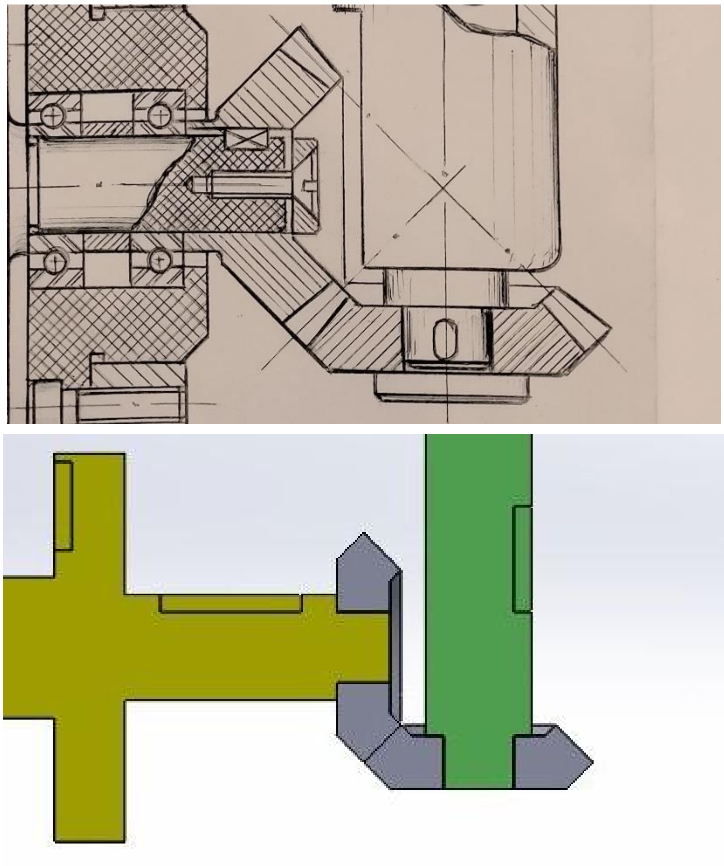

Figure 4.7. Conical gear mate - 3D model and sketch

4.1.5 Crankcase

Once the components of the mechanism are completed, it is necessary to create the crankcase because the Solidworks software needs, in order to assemble all the several parts, a fixed element that acts as a reference for the moving ones. So, the main idea behind the modelling of the crankcase is to have a set of reference surfaces and volumes that make possible the correct positioning of all the components. The main volumes and distances adopted as starting point for the build-up of the crankcase are the ones corresponding to the two pistons and the Cylindrical cam. Once the interaxial distance between the mentioned elements is measured, it is possible to evaluate a sort of overall width of the engine block. The following step is measurement of the distances between the pistons and the crankshaft, and the barrel cam and the output shaft, in order to create the housing holes for the three rotating shafts of the system. Now that all the distances are evaluated, it is possible to model the entire structure of the block. After the creation of the crankcase is done, one half of the component is cut out for a better visualisation of the system and the mechanism that is inserted inside it.

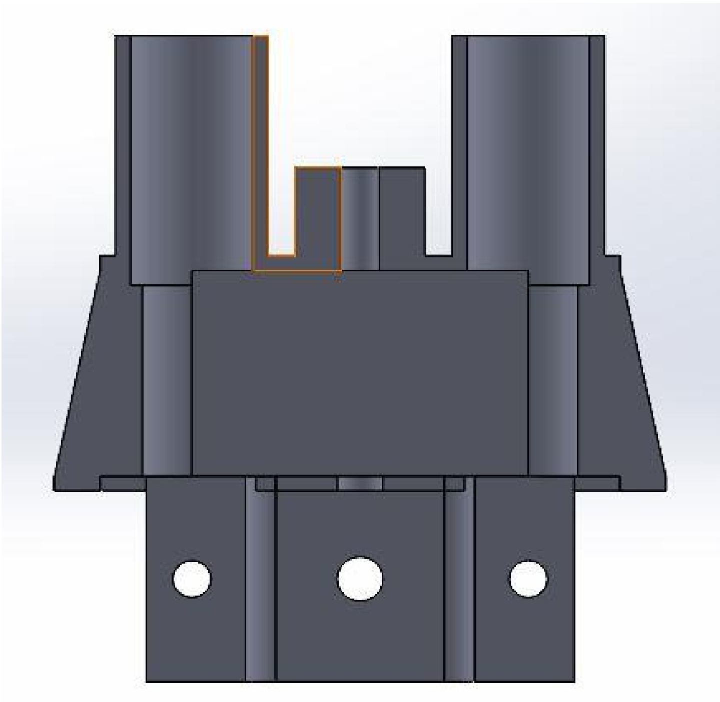

Figure 4.8. Crankcase 3D model

4.2 Assembly of the engine

The following phase in the creation of the 3D model of the engine under analysis is the assembly of all the components presented above. The starting point is the crankcase that is fixed in the assembly environment of Solidworks. After that all the components are inserted into the environment, it is now the time to connect all of them. Thanks to the large variety of mate typologies present in the used software, it is not so difficult to assemble a major part of them. It is in fact possible to use approximately only concentricity and coincidence constraints to couple almost every part of the assembly. However, there are some needed relative constraints that require a little more attention such as the connection between the wrist pin and the piston and between the roller and the barrel cam groove. Taking into account the former coupling, the use of the proper mate for pin and slot present in Solidworks is not possible to be adopted because it does not work for what concerns the simulation environment of the software. So, the solution thought for this issue is to limit the relative position of the pin with respect the two edges of the slot in order to obtain a slot mate with an indirect constraint. The pin is forced to move in a length space equal to the one that characterizes the slot in the piston. This is possible thanks to the Solidworks mate concerning the limit distance between two objects.The other significant point concerning the assembly of the mechanism is the roller-barrel cam coupling.

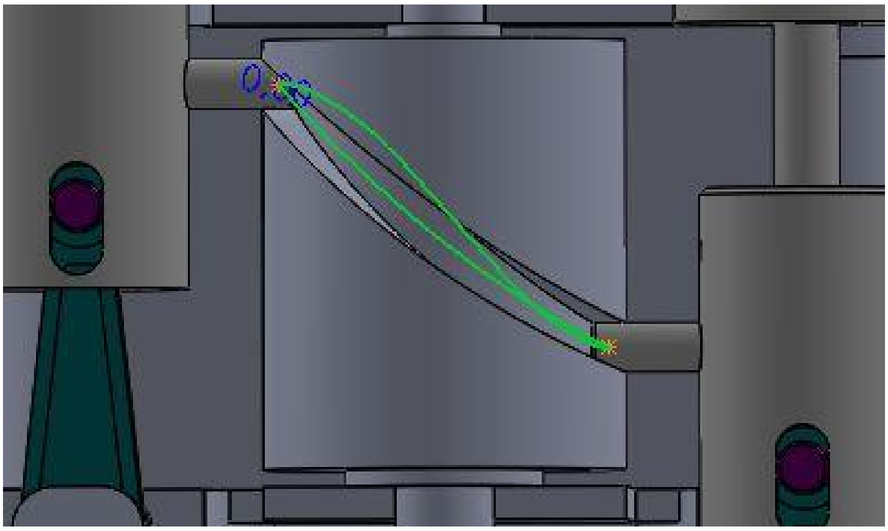

Figure 4.9. Roller-cam mate (3D model)

The basic concept under this kind of mate is that the roller must be forced by the groove cut inside the cylinder to follow the cam profile. The starting idea to achieve the roller-cam mating is to constrain the roller to be tangent to the surfaces of the groove. Unfortunately, also in this case, the adoption of this type of mate is not possible because during the simulation of the working system the tangent constraint is not considered valid. So, in order to simulate the considered joining, it is thought to create a profile within the groove that corresponds to the one that the centre of the roller would follow in case of the hypothesised initial condition. Then a point is created along the roller axis at a distance from the outer face that is equal to half the depth of the groove. Once these preliminary steps are done, the final solution is to use the "path mate" that is present among the several mates available on Solidworks. In practice, the mentioned mate forces a point of the desired part to follow the specified path. In this way it is possible to obtain the desired relative motion between the roller and the barrel cam (*figure 4.9*). At this stage, the assembly of all the components and parts can be considered finished and ready for the next phase regarding the simulation during the working operation of the engine.

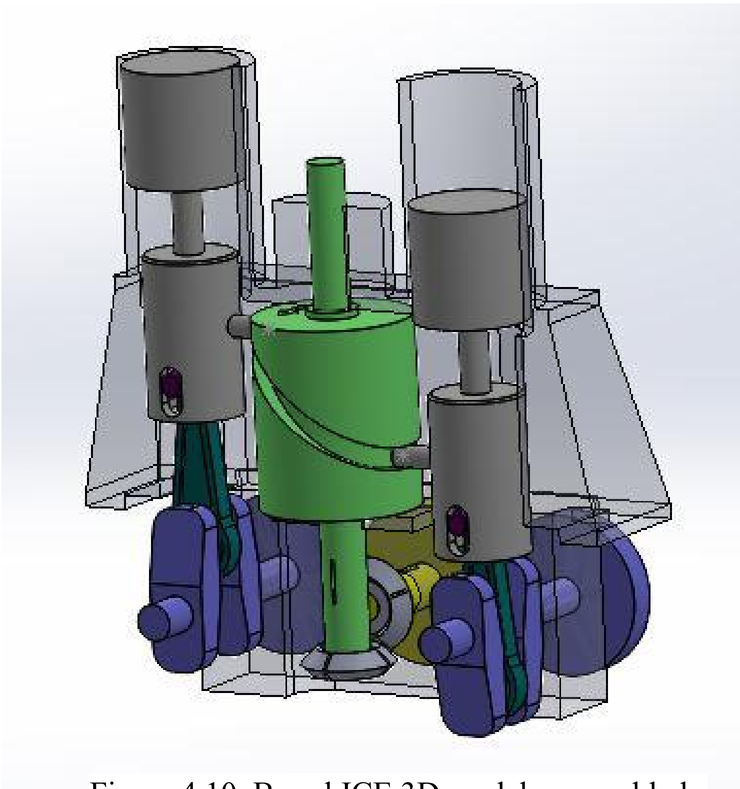

Figure 4.10. Barrel ICE 3D model - assembled

4.3 Conventional 2-cylinder ICE model

Before starting the simulation of the unconventional engine under analysis, it is necessary to interpose an additional step useful for the following phases of the investigation process. In fact, once the model of the "*Barrel ICE*" is completed, it is unavoidable to create the model of a traditional engine in order to be able to perform the comparison between the two mechanisms. Since the analysed machine is composed by two cylinders, the traditional one must have the same number of cylinders in order to achieve a direct comparison of the results. The model which this chapter is focused on is a very simplistic version of an automotive ICE. It is composed by two pistons, characterized by the same bore diameter of the unconventional one, two wrist pins and conrods, that are the same ones explained in the paragraph 4.1.3, and one crankshaft. The latter component is realized by simply mirroring the crankshaft of the paragraph 4.1.4 with respect its end, and by eliminating the disc at the opposite edge. The elongation ratio adopted in this case is equal to the investigated engine one, so that the comparison is more and more accurate and significant. For what concerns the assembly of this engine, to position all the components in the correct way, it is created a simple structure that as the task of reference for all the other parts. The construction of the system in this case is particularly simpler and intuitive. At this point, all the 3D models necessary for the simulation, and the consequent extraction and comparison of the results are completed.

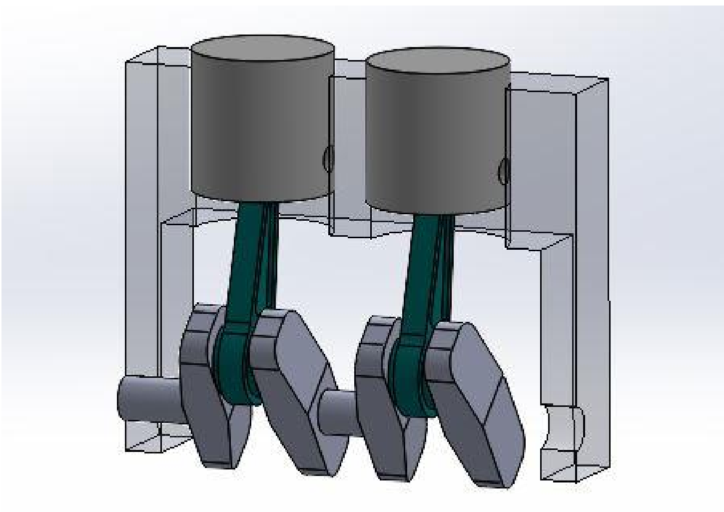

Figure 4.11. Conventional ICE - 3D model

4.4 Simulation

In this chapter it is presented all the adjustments and the parameters important for the correct execution of the simulation. It is important to specify that in this phase the main task is to evaluate the output torque and power under idealized conditions. This means that the simulation is characterized by the absence of the gravity force, the mass of each component and the friction between components that are in contact. Therefore, this first step has as main goal to evaluate the possible gain in power considering the simplest and theoretical conditions.

4.4.1 Input forces

The main goal of the simulation is to analyse the system during the working operation and extract the mean value of the output torque in order to determine the possible gain. The main input to be given to the model to work properly is the pressure acting on the pistons. The input used in this case are a set of data of the in-cylinder pressure as function of the crank angle that was given during the course of "Powertrain components design". These data are represented in the corresponding graph of the *figure 4.12***.** The pressure values cover the whole working cycle of a thermal engine. In this case, taking into account that the engine considered is a 4-T ICE, the dataset is extended for 720 C.A.°. The target is to manipulate these data with the Excel software and create the proper function to be given as input to the model. Due to the fact that in Solidworks it is required a force and not a pressure, the first step is to convert the pressure values into force values. This is simply done by multiplying every value by the surface area of the piston and expressing the result in Newtons. Then it is necessary to correctly phase the force peak with the motion of the piston so as to have the peak when the piston is near the TDC, as it happens in a conventional engine. To do that it is important to correctly specify the initial position of all the components of the system. In this case, the initial configuration corresponds to the condition in which the first piston (the left one) is at TDC while the second one it at the BDC as it is shown in *figure 4.10*.

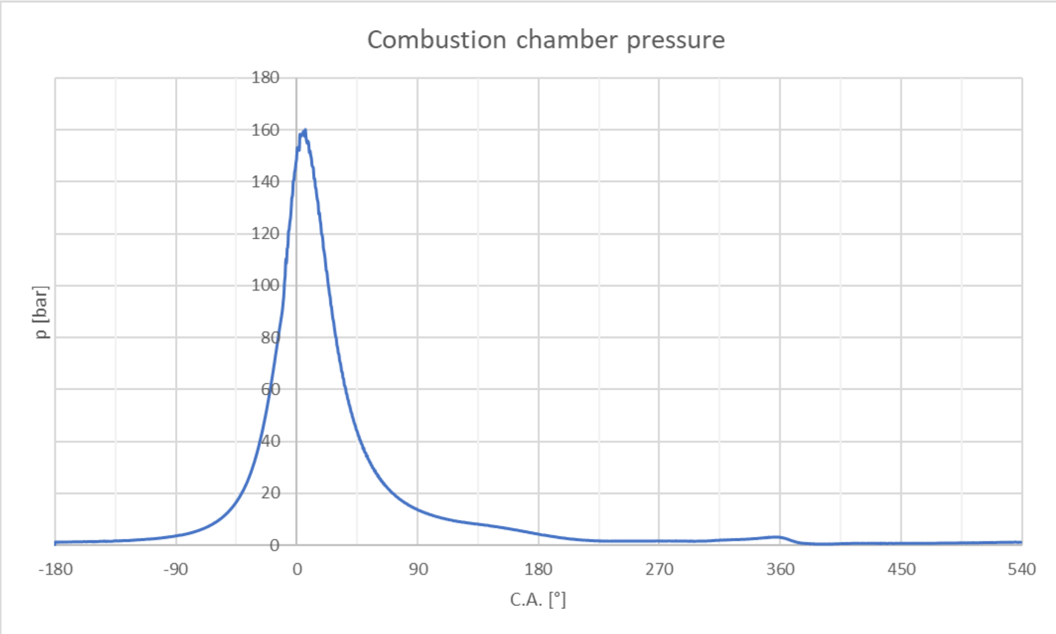

Figure 4.12. Pressure trace inside an ICE combustion chamber

Once this step is done, it is possible to copy and paste the function obtained just to have at least three peaks of pressure in the simulation. At this point, one more important step must be done. It is in fact necessary to convert the current function, that is actually expressed with respect to the crank angle, into a function of time due to the fact that Solidworks uses the time as the independent variable during the motion analysis. The simplest way to perform this conversion is to hypothesise a constant rotational speed of the engine during the simulation. In this way it is possible to obtain the force as function of time by simply dividing the angles by the desired angular velocity. The value chosen for the constant speed is equal to 2000 rpm. This value corresponds to the speed of an engine during normal operations. Therefore, what is obtained is a force function expressed with respect the time. Once the force function is completed, it is necessary to create a second function since the two pistons are characterized by an alternative motion, unlike a conventional 2-cylinders ICE in which the pistons move in parallel. Therefore, the second function that has to be applied on the right piston, must be shifted by 180 C.A.°. Once the first function is obtained, the process to derive the second one is much simpler and faster. The final functions can be clearly seen in *figure 4.13*, where it is evident that the temporal extension of the force, and consequently of the simulation, is very short. This is caused by the high value of rotational speed applied to the model.

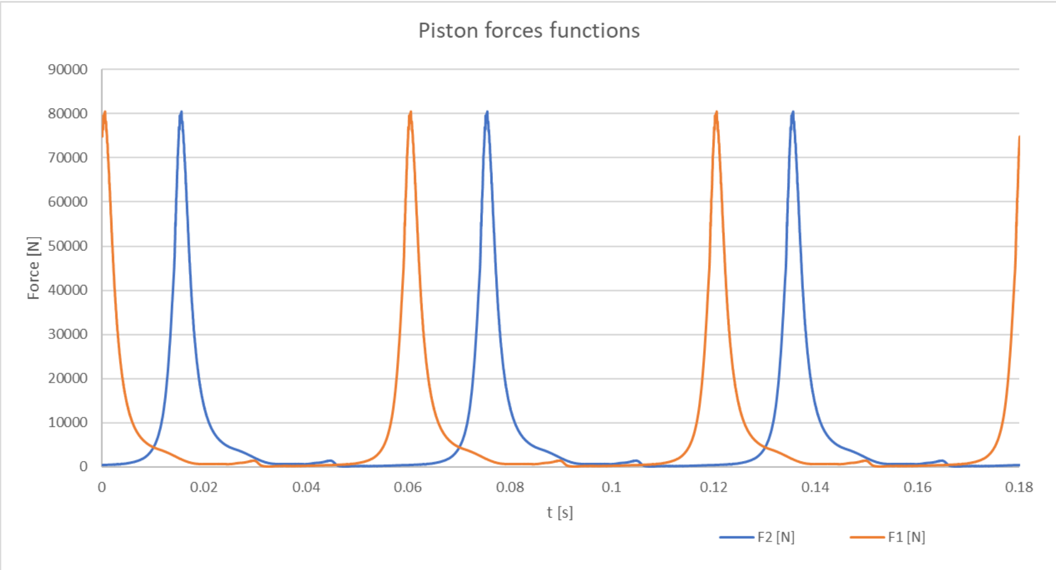

Figure 4.13. Force functions applied to the pistons during the simulation

4.4.2 Simulation parameters and items

Before the start of the simulation, it is necessary to set some parameters of the simulation and ad to the model some elements. First of all, it is needed to set the temporal duration of the simulation. The total time is decided equal to 0.18 seconds. This value corresponds to the time length of the function of the forces discussed in the paragraph 4.4.1. The second important value to be appropriately set is the number of frames per second. This parameter expresses the total number of frames that are captured during the simulation time previously set. Since the mechanism to be studied is thought to rotate at constant speed equal to 2000 rpm, it is necessary to set a very large number of fps. In this way it is possible to get the proper amount of data in order not to lose some significant phenomena characterized by too short time length. Therefore, taking into account this important aspect, the frame parameter is set to 4000 fps. Connected to the aspect of the rotational speed, another important point to focus on is the necessity to force the engine to work at the desired constant speed. The corresponding solution is simply to add a simulation element that operates as an ideal rotational motor. One of the options that can be chosen for the working operation of the rotative motor element is to set a constant rotational speed. It is applied to the output shaft and set to rotate in the desired direction. This expedient is important for two main aspects. It assures the engine to rotate at the decided constant velocity. This means that the forces are correctly phased with respect to the motion of the pistons, and consequently, of the whole mechanism. The other advantage is that the output torque and power that is produced by the engine can be simply evaluated from the torque/power value required by the motor element to rotate at constant speed. Once that the simulation environment and the model are properly set, it is possible to proceed with the actual simulation.

4.4.3 Conventional ICE model simulation

For what concerns the simple model of the 2-cylinders ICE, the only difference with respect to the unconventional one is in the functions of the forces to be applied at the piston top faces. It is in fact necessary to modify one of the two functions described in the paragraph 4.4.1 in order to have the peaks of the forces shifted by an angle of 360 C.A.°. This characteristic is determined by the fact that the angular phase shift between the two cranks of a 2-cylinder ICE is equal to 360°. Consequently, at each crankshaft revolution corresponds the expansion stroke of one piston in an alternating succession. Once the force function is modified, it is applied to the corresponding piston upper surface. The other parameters discussed in the previous paragraph are kept unchanged.

Chapter 5

Initial results

After that the simulation is completed, it is now possible to extract the significant results that allow the comparison between a traditional ICE and the analysed one. The main parameter that is under investigation is the output torque. It is extracted through the evaluation of the torque necessary to maintain the engine at the same speed. Task that is performed by the motor item described in the Chapter 4.

5.1 Conventional 2-cylinder ICE results

It is preferred to start with the presentation of the results regarding the conventional ICE whose model is presented in the Chapter 4, because it represents the reference for the comparison to be done with respect the studied innovative system. Before the actual verification of the simulation results, the theoretical characteristic of the output torque of a 2-cylinder ICE is computed in an Excel file and represented as function of time in the *figure 5.1*. The torque 1 and torque 2 pictured in the *figure 5.1* correspond to the torques that act on the corresponding single crank. The values of these two plots are computed through the application of the equation 1.4 presented in the Chapter 1. The superposition of the two curves produces the theoretical output torque coming from the modelled conventional ICE and represented by the grey dashed line in the *figure 5.1*. Once this preliminary step is completed, the torque extrapolated from the simulation is extracted and displayed graphically as function of time. The result is shown in the *figure 5.2*. At first sight, it is possible to notice the strong similarity between the computed graph and the simulated one. The graph coming out from the simulation is in fact well approximated by the theoretical one computed before. This first observation is useful in order to understand if the procedure used to extract the output torque from the model is correct. Comparing the *figure 5.1* and *figure 5.2* is possible to assess that the procedure adopted is the right one.

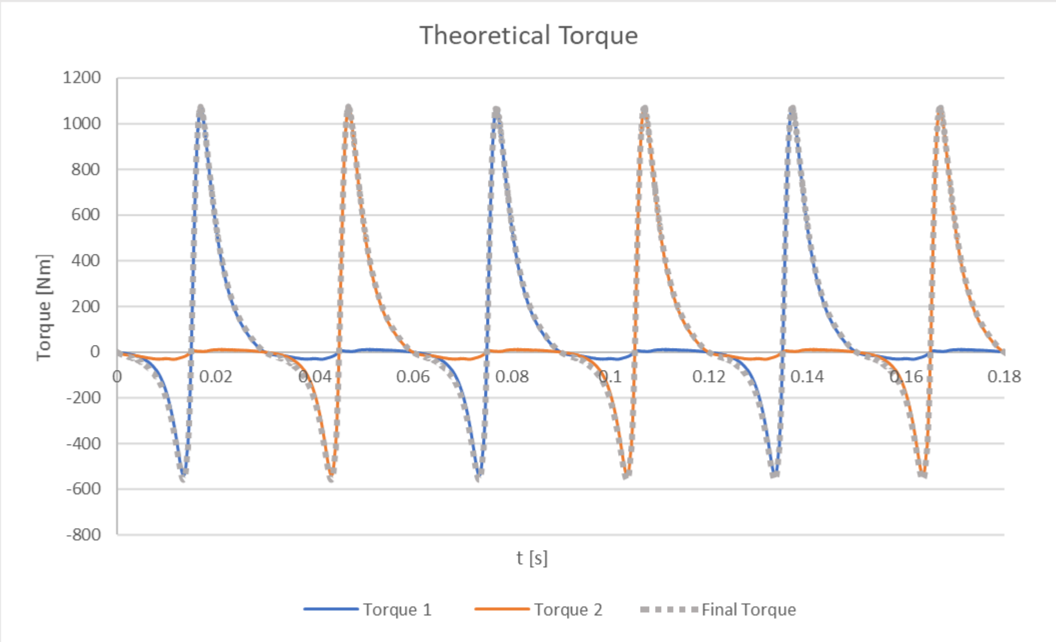

Figure 5.1. Theoretical output torque of a 2-cylinder ICE

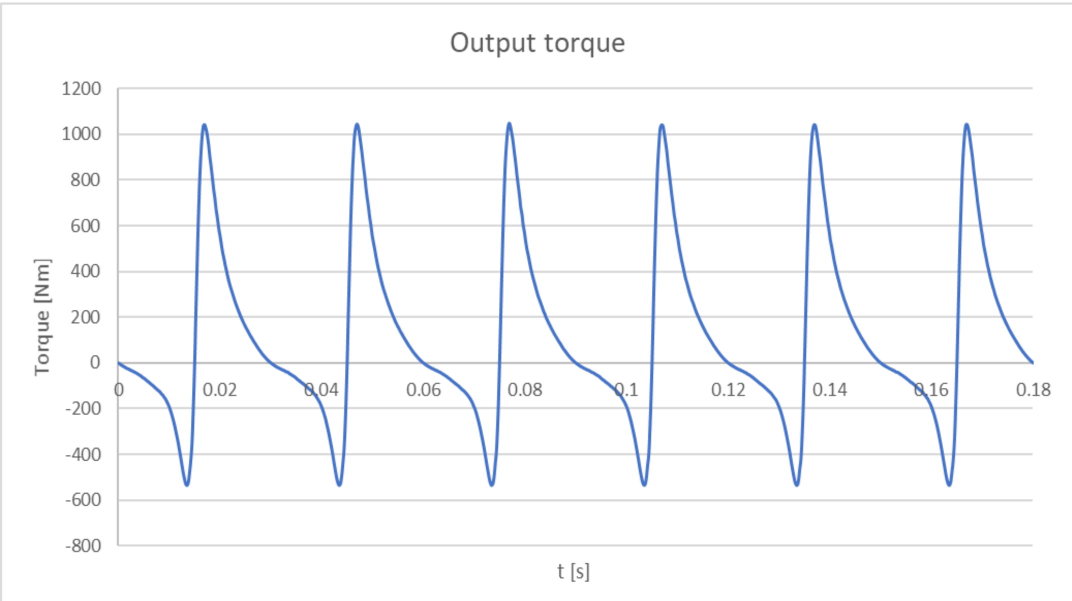

Figure 5.2. Output torque of a 2-cylinder ICE

The second aspect relevant for this study is the actual value of the output torque and its mean value. The periodic curve represented in *figure 5.2* is characterized by a peak of torque about 1000 Nm and a negative peak of about -550 Nm. By extrapolating the data set produced by Solidworks into an Excel sheet, it is possible to evaluate the mean value of the output torque. The average torque is the main parameter adopted for the comparison between the two models.

Table 5.1. Conventional 2-cylinder ICE results

Mean Torque [Nm]	105.8
Mean Power [kW]	22.16

Due to the imposed condition of constant rotational speed, it is also possible to easily evaluate the mean power by multiplying the mean torque value by the velocity. It is important to underline that the mean value of torque coming from the theoretical computation is in line with the results in the *table 5.1*.

Table 5.2. Conventional 2-cylinder ICE theoretical results

Mean theoretical Torque [Nm]	105.7
Mean theoretical Power [kW]	22.14

5.2 Barrel ICE results

The output torque generated by the unconventional engine taken into account is depicted in the *figure 5.3*. The engine torque corresponds to the blue line. In the same graph shown in the *figure 5.3*, the curves corresponding to the piston forces discussed in the Chapter 4 are depicted. In this way it is possible to see the effect of the peak pressure on the output torque of the engine.

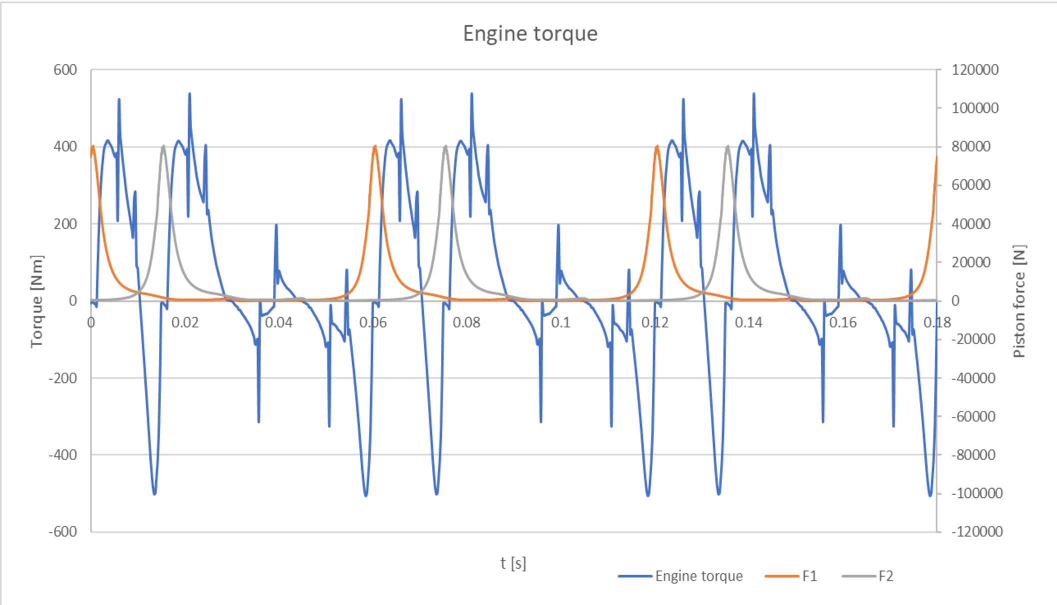

Figure 5.3. Barrel ICE output torque and piston forces

In fact, the positive peaks of the blue line are subsequent to the forces peaks that are applied on the pistons. Therefore, the piston roller correctly transfers the force towards the barrel cam and consequently to the output shaft. An important aspect to be highlighted is the value of the torque peaks. It is in fact visible that the maximum value reached is about 500 Nm. Almost the same absolute value is reached by the blue curve in the negative part of the chart. The engine torque plot is in fact characterized by negative peaks whose values are comparable to the maximum peaks. The negative parts of the torque chart represent the necessary torque to drive the system when the combustion phase is already finished. This similarity between the negative and the positive torque peaks has an important effect on the mean value of the engine torque. Performing the same procedure explained in the paragraph 5.1, it is extrapolated the average torque and the average power produced by the analysed engine.

Table 5.3. Barrel ICE results

Mean Torque [Nm]	38.86
Mean Power [kW]	8.139

The results are shown in the *table 5.3*. By comparing the obtained value with the ones resulting from the conventional ICE and represented in the *table 5.1*, the first conclusion is that the unconventional engine produces a significantly lower torque level. It is in fact clearly visible that the peak torque in the *figure 5.2*, is importantly higher than the peak torque shown in the *figure 5.3*. The positive peaks play a fundamental role in the final comparison between the two models because the negative peaks are characterized by similar values. Therefore, the mechanism of the barrel cam probably can not exploit in a good way the force applied on the piston. In order to have a better understanding of the mechanism behaviour, a deeper investigation is performed. The reaction force between the roller and the cam is extrapolated. The component of the force taken into account is the one tangential to the cam profile. Once the data are available, they are multiplied by the mean radius of the cylinder, considering the outer diameter and the inner one corresponding to the inner face of the groove. Proceeding in this way it is possible to evaluate the contribution of the cam and roller coupling during the working operation of the engine. Since, what is obtained from this computation is a sort of torque, it is subsequently expressed graphically on the same graph of the engine torque. What comes out is depicted in the *figure 5.4*. The reaction force considered takes into account only one of the two roller-cam mates and so the curve obtained is only a part of the final engine torque plot. The main aspect to observe is that the negative torque peaks are determined by the reaction force between the roller and the cam. This means that during the operation of the engine, the cam is always engaged and transfer torque in each working phase. The behaviour of the mechanism differs from the starting idea. Theoretically, the barrel cam should be exploited only during the downward motion of the piston when pushed by the combustion pressure. Instead, during the downward stroke, the roller should disengage the cam and allow the piston to be guided by the classic crank mechanism. This means that some modifications have to be done to the actual model in order to achieve the desired behaviour.

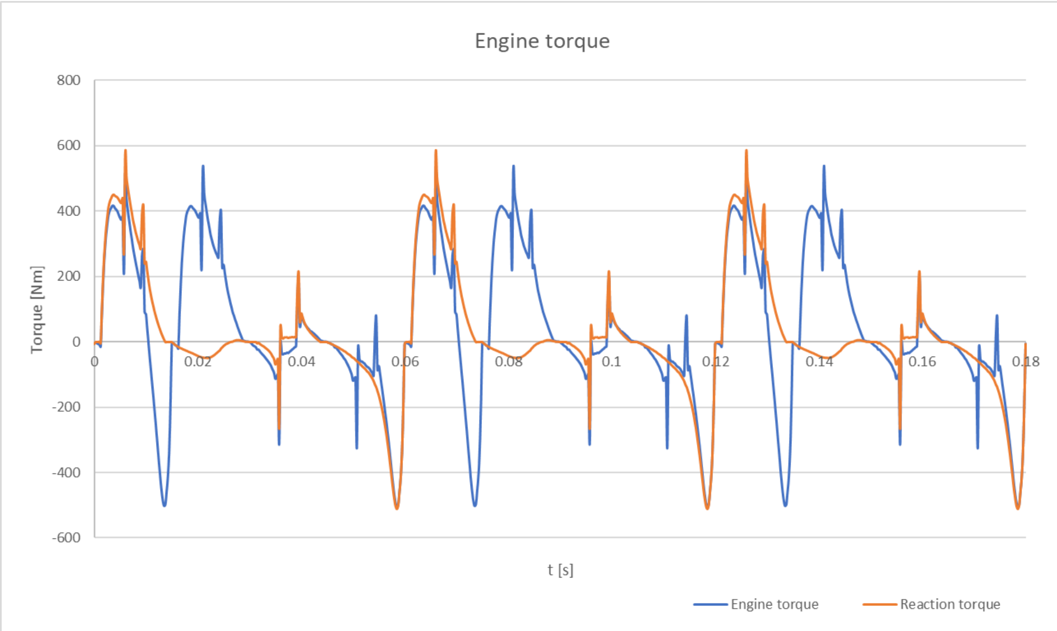

Figure 5.4. Barrel ICE output torque and roller-cam reaction torque

Chapter 6

Intermediate results

6.1 Intermediate version of the model

The results coming out from the simulation of the model gives an overview of the functioning behaviour of the actual system. In particular, by the observation of the *figure 5.4*, it is clear that the roller-cam mechanism, that actually is the innovative element added to a conventional ICE, instead of being engaged only during the expansion stroke, it transfers torque during the whole cycle. This aspect differs from the starting idea. In theory, during the upward strokes the system should work as a conventional engine and therefore, the piston should be pushed up by the crank mechanism exploiting the inertia of the system. Since, this condition does not occur during the working operation of the model, it is necessary to perform some modifications in order to obtain the desired behaviour and exchange of forces.

6.1.1 Barrel cam modifications

The main goal is to avoid the engaging of the barrel during the upward strokes. In order to achieve this result, the basic idea is to extrude externally from the cylinder only one half of the original cam profile. Instead of having a groove that is cut around the barrel, it is built-up an external guide as represented in the *figure 6.1*. The extruded profile corresponds to the one designed for the downward stroke of the piston, as described in the Chapter 4. Therefore, the other half of the profile that corresponds to the piston displacement driven by a conventional crank mechanism (equation 1.1) is completely avoided. By applying this modification, it is avoided the contact between the roller and the cam when it is not desired. The fact that in this new version the profile is extruded outside instead of being cut into the barrel is determined by the need to disengage the roller from the cam when it is required. Another aspect to be taken into account is the cam profile itself. It is in fact necessary that the actual profile must be made in such a way that its linear development stays always above or at least at the same level with respect the profile expressed by *equation 1.1*. The *figure 6.1*. better explains the just mentioned concept.

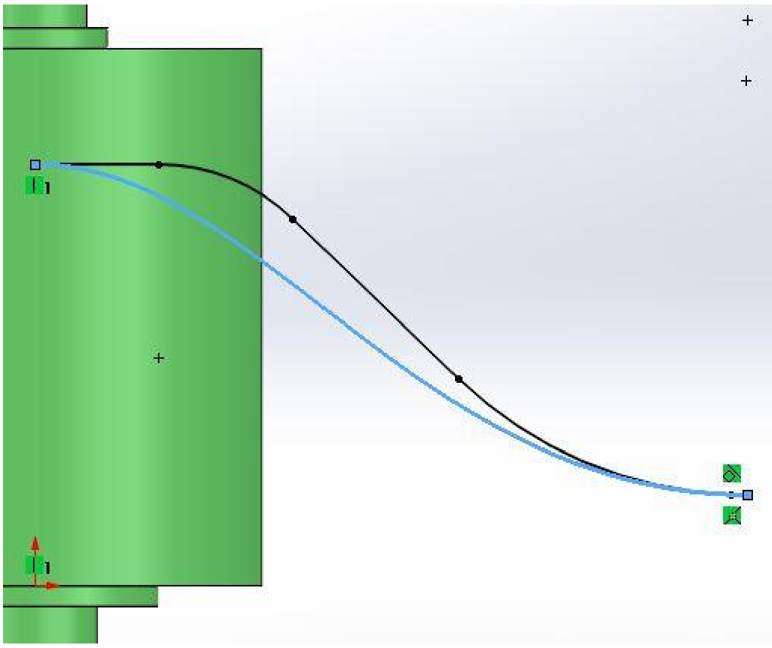

Figure 6.1. Barrel cam profile - new version

As it is clear, the black line represents the modified actual cam profile while the blue one represents the piston displacement characteristic expressed by equation 1.1. The combination of this last change and the ones that are applied to the piston and presented in the next chapter allow to have the piston driven only by the crank mechanism during the upward strokes of the engine. The resulting model is represented in *figure 6.2*.

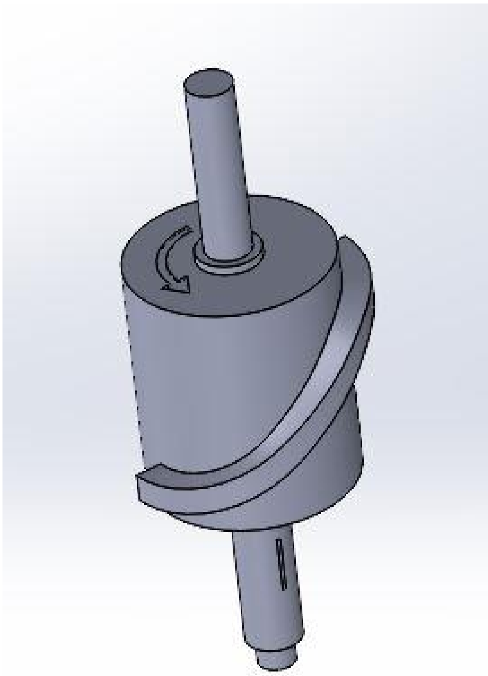

Figure 6.2. Modified barrel cam

6.1.2 Piston modifications

This first change in the model drives the necessity to also modify the piston. In particular, what must be changed is the position of the slot in which it is inserted the wrist pin. Theoretically speaking, the upward motion of the piston should be guided from the crank mechanism. To achieve this behaviour, it is needed the pin to be pushed against the upper side of the slot in order to transfer the force coming from the crankshaft trough the connecting rod. So, the position of the slot must be shifted downward to have the pin positioned at the upper edge of the slot when the piston is at its BDC and TDC. These two configurations are in fact the points in which the motion law of the wrist pin and of the piston, guided by the cam profile, coincide. In the previous version, the slot is positioned in such a way that when the piston is at TDC (or BDC) the pin is placed almost in the middle of the slot. Once the changes are applied, the new version of the model is obtained, and it is represented in the *figure 6.3*. By the observation of the image, it is possible to notice that the left piston, that is moving upward, is pushed by the crank mechanism and the wrist pin is forced against the upper side of the slot (red circle in the *figure 6.3*). While the right piston, that is moving downward, follows the profile of the cam and it is decoupled by the motion of the crank mechanism (green circle in the *figure 6.3*).

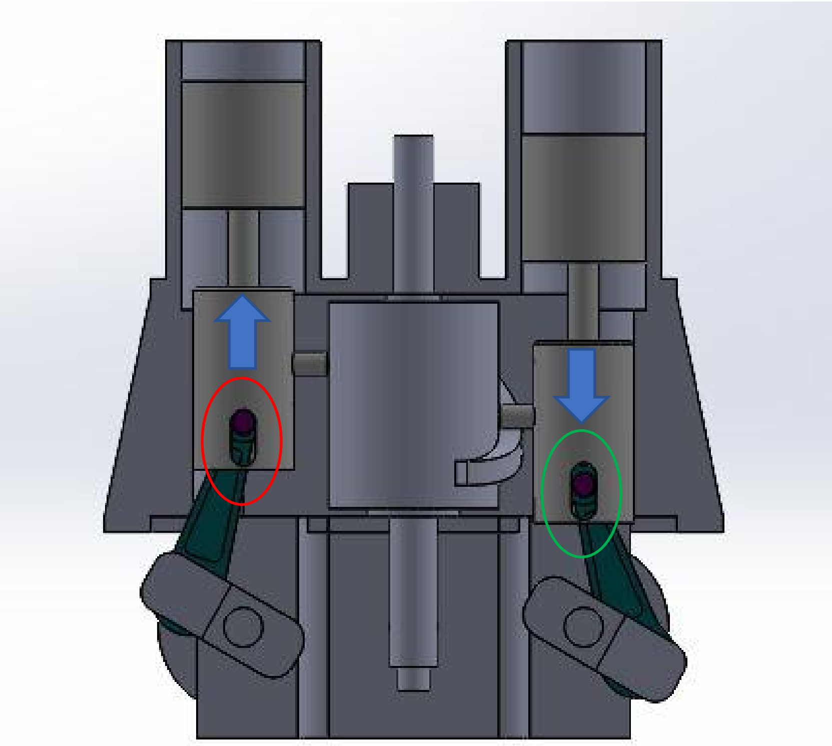

Figure 6.3. Modified barrel ICE - working condition

6.1.3 Assembly and simulation modifications

Before proceeding with the simulation, it is necessary to highlight an important aspect. For what concerns the assembly of the modified model, it is not possible to use the mates available in Solidworks for the coupling between the roller and the cam profile. This is because the roller is forced to follow the guide only during one half of the barrel circumference while it detaches from the cam during the other half. Therefore, the solution applied to solve this issue is to simulate the contact between the roller and the cam. Thanks to this expedient it is possible to simulate both the engagement and disengagement between the roller and the barrel cam. The contact to be simulated is ideal. This means that there is no friction between the elements considered and the impact is characterized by a coefficient of restitution equal to 0. This means that the impact is perfectly inelastic. At this point it is possible to perform a new simulation in order to extract the output torque from this new version of the engine.

6.2 Modified barrel-ICE results

The resulting output torque generated by the new version of the module is represented in the *figure 6.4*. As it can be seen, the engine torque curve, the blue one, is characterized by a trend like the one observed in the *figure 5.3*. Also in this case, to better understand the force transfer between the components, it is represented also the reaction force between one roller and the cam. The corresponding curve is the orange one in the *figure 6.4*. The main aspect of the reaction torque plot is that there is no more the negative peaks as in the *figure 5.4*. This means that the barrel cam and the roller are actually disengaged during the upward stroke of the piston and consequently there is no exchange of forces within these phases. Therefore, it is possible to assess that the main goal of keeping the barrel engaged specifically during the expansion stroke is achieved.

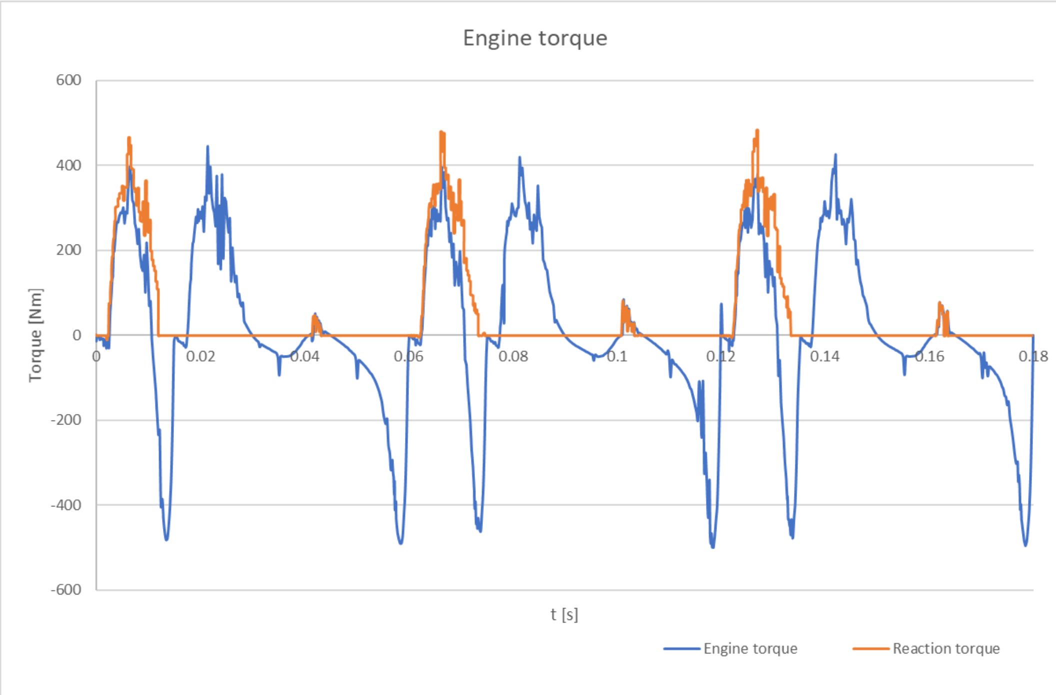

Figure 6.4. Modified barrel-ICE output torque and roller-cam reaction torque

However, even if the barrel is not acting during the upward strokes, the graph shown in the *figure 6.4* highlights that the negative peaks are still present in the chart. If a deeper observation of the resulting plot is performed, it is possible to see that the negative parts of the engine torque curve are very similar to the ones shown in the *figure 5.2* representing the output torque of the conventional ICE model. This aspect can be easily explained by the fact that during the upward strokes of the pistons, these latter are pushed by the crank mechanism and therefore, consume the same amount of energy as in the conventional ICE. Once that this point is clarified, the main aspect to focus on is the very low level reached by the engine torque during the active phases. By the computation of the average torque and power coming out from the model the following results are obtained.

Table 6.1. Modified barrel-ICE results

Mean Torque [Nm]	38.11
Mean Power [kW]	7.982

As it can be seen from *table 6.1*, the mean torque value is not so different from the one shown in the *table 5.3*, representing the previous version of the model. From these results, it is possible to assess that the roller-cam mechanism is inefficient if adopted during the expansion stroke of the piston. However, additional investigations can be performed in order to try to exploit this mechanism in an alternative way.

Chapter 7

Final results

7.1 Additional modifications of the model

The results derived from the engine version discussed in the Chapter 6 indicates that the roller-barrel cam mechanism is not able to transfer the forces and energy with a sufficiently high efficiency.

7.1.1 Barrel cam under a new perspective

By the comparison between the *figure 5.2* and the *figure 6.3*, it is clear the significant difference in the maximum useful torque generated by the 2-cylinder ICE and the Barrel ICE. This means that, considering an equal force applied to the pistons in the two mechanism under comparison, the crank mechanism is able to convert the axial force into useful torque with a stronger effectiveness. Once that this concept is clarified, the alternative idea is to use the barrel cam in the opposite way to exploit its capacity to convert a high force value into a not so high torque. The basic principle under this solution is that the coupling between the roller and the cam can be seen as a screw. The screw is based on the principle of the inclined plane. The aspect under interest for what concerns the engine model is the capability of the screw to convert a low level of torque into a higher level of axial force. Starting from this principle, it is possible to reconfigure the system in order to exploit the barrel cam during the upward strokes of the pistons. In this way, theoretically speaking, the piston themselves can be pushed towards the TDC consuming a lower torque from the system and determining a higher level of the mean torque generated by the engine.

7.1.2 Practical modifications of the model

In practice, the changes to be applied to the actual model are simple. The main aspect to be changed is that during the expansion stroke of the piston pushed by the pressure, the wrist pin must stay in contact with the upper side of the slot in order to allow the force transfer between the piston and the crank mechanism. To achieve this result, the slot is shifted downward. In this way the pin is in the desired position when the piston is at TDC. The second important aspect to be changed is the starting position of the barrel. It is in fact necessary that the extruded profile engages the contact with the roller during the upward stroke of the piston. In this way, the piston itself is pushed toward the TDC by the cam and not by the conrod.

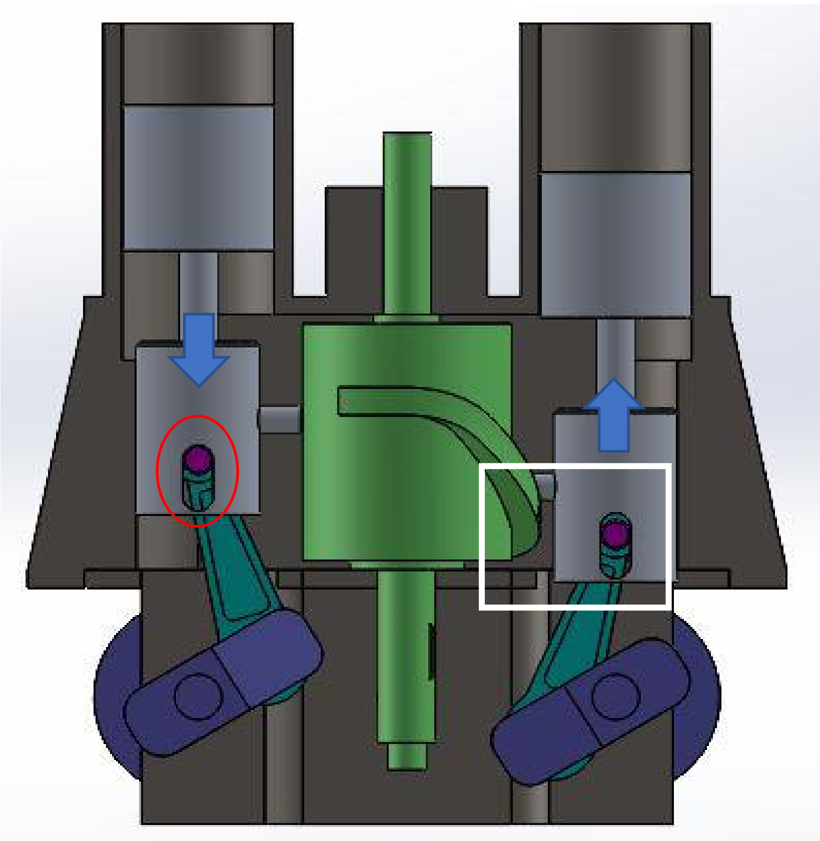

Figure 7.1. Barrel-ICE final version - working condition

The representation of this concept is shown in *figure 7.1*. It is possible to see that during the expansion stroke the wrist pin is pushed against the upper wall of the slot, transmitting the piston force to the crankshaft (red circle in the *figure 7.1*). Instead, during the upward strokes the piston lean on the barrel cam through the roller and it is pushed up by the rotational movement of the barrel itself. Since the profile of the cam is steeper with respect the characteristic of the piston displacement expressed by the equation 1.1, during the upward strokes the piston and the wrist pin are decoupled (white rectangle in the *figure 7.1*). Once that these corrections are applied to the model, it is possible to begin with the proper simulation. For what concerns the simulation environment, all the conditions and all the parameters are kept unchanged with respect the simulation described in the Chapter 6.

7.2 Barrel-ICE final version results

As done for the previous versions of the model, the output torque generated by the engine is extracted from the simulation. The main results are plotted in the *figure 7.2*. The blue line represents the engine torque that is actually the main indicator that is investigated. The first aspect to focus on is the significant difference in the maximum torque generated by this version of the engine compared to the one presented in the Chapter 6. The torque peaks reach values about 1000 Nm instead of about 400 Nm. The second aspect to be highlighted is the similarity between the shape of the peaks in the *figure 7.2* and the ones characterizing the conventional ICE torque chart shown in the *figure 5.2*. This resemblance can be easily explained by the fact that during the expansion strokes of the pistons, they are coupled to the crankshaft by means of the crank mechanism therefore the system is working as a normal ICE. However, there is also a big difference between the conventional and the actual barrel ICE. The latter is characterized by significantly lower values of negative torques, considering the absolute value without the sign. It is visible that the negative parts of the curve have the maximum values around -200 Nm. The spikes at the beginning of the active phases reaching values of torque around -400 Nm are probably due to the instantaneous passage from the roller-cam coupling to the piston-wrist pin one, and so are not so significant.

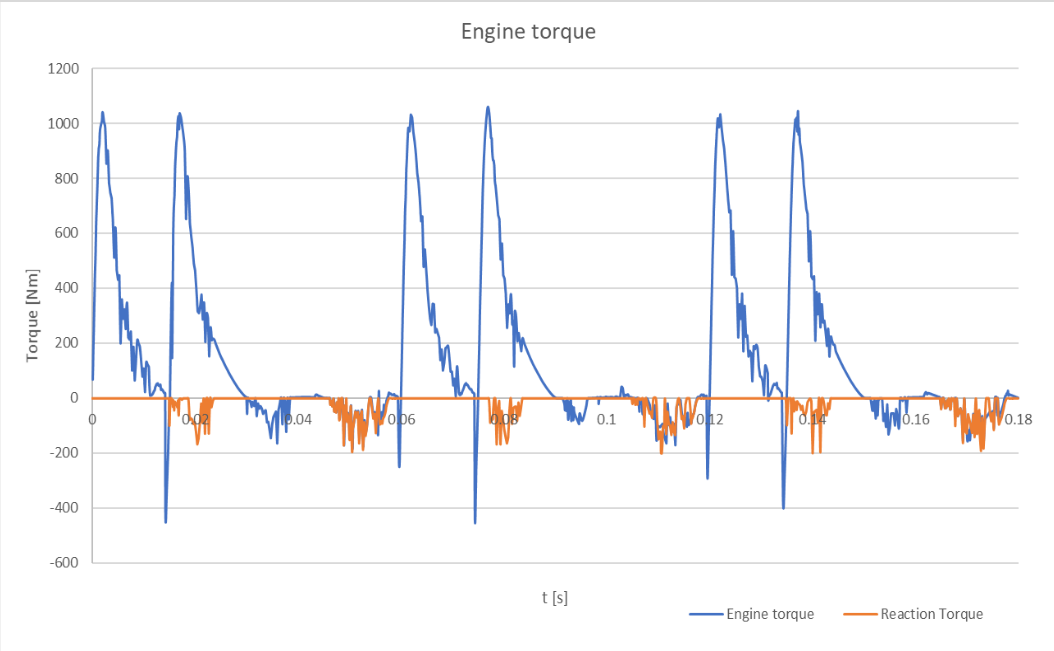

Figure 7.2. Barrel-ICE final version - output torque

Similarly to what is done in Chapter 5 and Chapter 6, the reaction force between roller and cam is extrapolated and represented graphically in the same plot of the *figure 7.2*. This procedure is mainly done to confirm the actual functioning of the mechanism that is theorized in the paragraph 7.1.1. The segments of the reaction torque curve that are not coinciding with the negative parts of the engine torque curve are cancelled out by the bigger peaks occurring in the same instant. Once the data are graphically represented, the next step is the computation of the average torque and power that characterises this configuration.

Table 7.1. Barrel-ICE final version - results

Mean Torque [Nm]	167.6
Mean Power [kW]	35.10

It is clear that there is an important difference in the mean value of the output torque with respect the previous version investigated. In addition, from the comparison between the values in *table 7.1* and the ones present in *table 5.1*, it can be assessed that this last version of the model is able to reach a significantly higher average torque. Therefore, the exploitation of the barrel-cam during the compression and exhaust phases can bring, in theory, a significant improvement in the power of the engine.

7.3 Evaluation of the losses

Before drawing the conclusion, an additional step it is required in order to have a deeper and more precise evaluation of the system under investigation. It is necessary to determine the effects of the friction between the elements in contact and the mass of the components of the mechanism.

7.3.1 Friction and mass simulation

The additional features of the model described above produces a more realistic simulation of the working engine and results. The first step is the assignment of the material to each component. In particular, the pistons are chosen to be made in aluminium (1060 Alloy) while the other elements are made in a generic steel (A286 Iron Base Superalloy) whose characteristics are described in the *table 7.2* and *table 7.3*. In this way, the mass of each single component is taken into account by the software while performing the simulation.

Table 7.2. A286 Iron Base Superalloy characteristics

Elastic Modulus	69000	N/mm^2
Poisson's Ratio	0.33	N/A
Shear Modulus	27000	N/mm^2
Mass Density	2700	kg/m^3
Tensile Strength	68.9356	N/mm^2
Compressive Strength		N/mm^2
Yield Strength	27.5742	N/mm^2
Thermal Expansion Coefficient	2.40E-05	/K
Thermal Conductivity	200	W/(m·K)
Specific Heat	900	J/(kg·K)
Material Damping Ratio		N/A

Table 7.3. 1060 Alloy characteristics

Elastic Modulus	201000	N/mm^2
Poisson's Ratio	0.31	N/A
Shear Modulus	77000	N/mm^2
Mass Density	7920	kg/m^3
Tensile Strength	620	N/mm^2
Compressive Strength		N/mm^2
Yield Strength	275	N/mm^2
Thermal Expansion Coefficient	1.65E-05	/K
Thermal Conductivity	15.1	W/(m·K)
Specific Heat	420	J/(kg·K)
Material Damping Ratio		N/A

Then, it is needed to simulate the contact among every component that actually touch the other ones. The characteristics of the contact applied to the model are summarized in the *table 7.4*.

Table 7.4. Contact features

Material	Part 1	Steel (greasy)	
	Part 2	Steel (greasy)	
Friction	Dynamic Friction Velocity	10.16	mm/s
	Dynamic Friction Coefficient	0.05	
Static friction	Static Friction Velocity	0.10	mm/s
	Static Friction Coefficient	0.08	
Elastic Properties	Impact		
	Stiffness	100000.00	N/mm
	Exponent	1.5	
	Max. Damping	49.91566312	N/(mm/s)
	Penetration	0.10	mm

Once all the additional features of the model are correctly set, it is possible to proceed with the simulation and the analysis of the results.

7.3.2 Conclusive results

The results extracted from the simulation are represented graphically in the *figure 7.3*. The image considered shows the engine torque curve. It is important to clarify that, due to the necessity to simulate the different contacts among the components, the original plot of the engine torque is characterized by several values that are out of the order of magnitude of the possible torque. These isolated points are probably due to numerical errors or corresponds to some instants during which the system is blocked. The graph shown in the *figure 7.3* is the result of the skimming performed to the set of data extrapolated from the simulation. Observing the *figure 7.3* it is possible to notice that the resulting chart is similar to the one shown in the *figure 7.2*. Therefore, it can be assessed that the basic functioning of the mechanism is remained unchanged. By an accurate observation it is noticeable that the maximum torque values and the negative peaks are slightly lower to the ones depicted in the *figure 7.2*. This little change in the graph can be related to the effect of the friction and of the mass applied to the model.

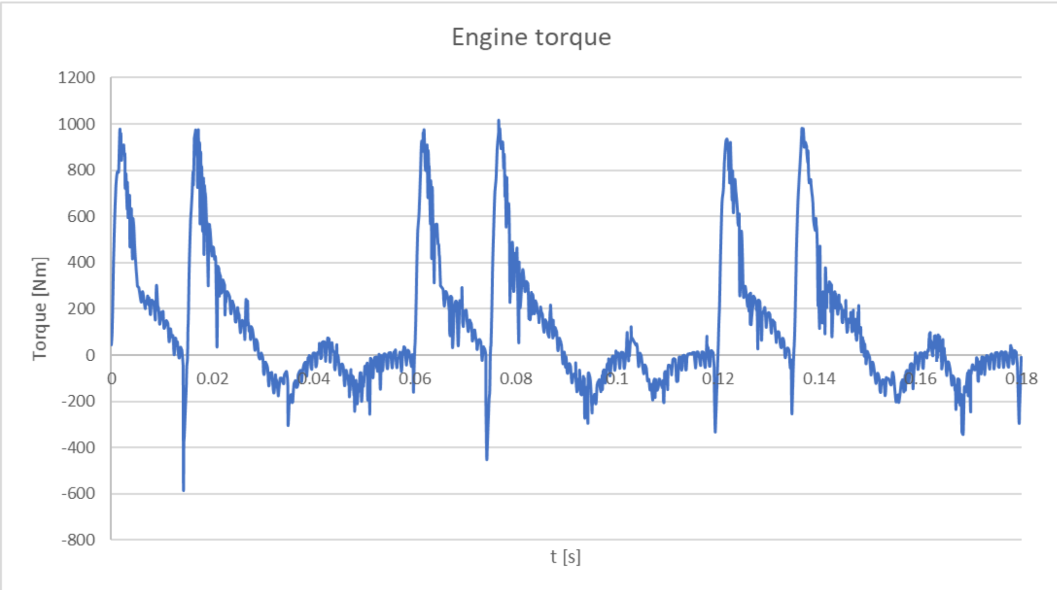

Figure 7.3. Barrel-ICE final version - output torque considering masses and frictions

In order to have a numerical evidence of the change in the graphs, the average value of the output torque is computed.

Table 7.5. Barrel-ICE final results

Mean Torque [Nm]	116.8
Mean Power [kW]	24.47
Loss with respect ideal condition	30.30%

The computed mean torque value assesses that even considering the friction within, and the mass of the components, the actual version of the engine is capable to produce a higher power with respect the conventional one. In particular, the value of torque in the *table 7.5* is higher than the one in *table 5.1* that corresponds to an ideal condition. In the *table 7.5* it is also present the computed loss determined by the friction and the mass compared to the ideal configuration of the model. After this final test, the investigation about the barrel ICE can be considered completed.

Chapter 8

Conclusions

At the end of the investigation, it is possible to assess that from a theoretical point of view, the barrel ICE can determine some advantages for what concerns the useful torque and power generated. However, it is important to specify that the initial idea about the adoption of a barrel in between the two pistons is significantly different from the final configuration of the engine and its working behaviour. The starting principle was to exploit the barrel in order to transform the energy freed during the combustion into mechanical energy with a higher level of efficiency. By means of the simulations performed and the analysis of the related results, a better understanding of the mechanism has been achieved. Thank to this higher knowledge level, it has been possible to modify the model and to reach a sort of final version of the engine characterized by the advantages presented in the Chapter 7. An important point to highlight is that the results obtained are only a small portion of the entire process of investigation that should be done to actually prove the feasibility or not of this innovative mechanism. However, the Mr. Bertotti's project has given me the opportunity to approach something similar to the investigation work performed by an engineer and consequently, to better understand my future targets and goals.

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