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Design of a new medium-temperature Stirling engine for distributed cogeneration applications

Fabrizio Alberti[1,*], Luigi Crema[1]

[1] Fondazione Bruno Kessler, REET Unit, Via alla Cascata 56/C, 38123 Trento, Italy
* Corresponding author. Tel.: +39-0461-314941; fax: +39-0461-314930
E-mail address: alberti@fbk.eu.

Abstract

This paper presents and discusses the design and first prototype realization for a brand new generation of Stirling engines. This unit is realized within the DiGeSPo Project, in which it is coupled with a small-size parabolic trough concentration solar field. The engine is conceived for working with low-temperature heat sources (200-300°C), in order to match the typical temperatures for the solar field itself.

The first part presents the thermodynamic design, which is realized by using models and simulations, and give the specifications for each component, including pistons dimensions; the number, length, diameter for the heat exchangers tubes; regenerators porosity, length and diameter. Four independent and equally working spaces were chosen, as a compromise between the compactness of the overall system, limits imposed by the maximum charging pressure, and the target electrical power (3 kW). The parameters of the overall system were optimized during this phase with an iterative procedure, taking into account different concurrent constraints, such as the heat exchange requirements, mechanical friction power losses, and small dead spaces. The engine has been subsequently arranged in a double-acting mechanical configuration, in which the cylinders are opposed as in a boxer engine. This configuration gives the advantages of reducing leaking losses and can work with four pistons. The heat exchangers, which are the most crucial and complex components, have been realized by the Selective Laser Melting (SLM) manufacturing technique.

The specific scientific and technical details related to a low-temperature Stirling engine, and the solutions adopted, are discussed and presented trough the paper, and finals recommendations are provided.

Keywords: micro csp, stirling engine, low temperature, solar and conventional heat source

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1. Introduction

This paper presents and discusses the design and first prototype realization for a brand new generation of Stirling engines. This unit is realized within the Digespo Project [1], in which it is coupled with a small-size parabolic trough solar concentration field. The engine is conceived for working with low-temperature heat sources (200-300°C), in order to match the typical temperatures for the solar field.

The cogeneration unit will be integrated with high-vacuum solar tubes and a parabolic trough mirror, which together provide a heat source at medium temperature. Peak power from the solar field under realization is estimated to be 10 kW. The target efficiency for the cogeneration unit is 20 % in electrical conversion and 65 % in the thermal efficiency, which is recuperated as hot sanitary water for heating and domestic consumption.

The load profile for a solar energy application has a typical non-constant curve, which should be followed by engine. The heat power extracted by the engine should be adjustable, otherwise the fluid in the collectors is cooled/heated, and the source temperature is perturbed. The power can be reduced or increased by a factor of 2-3 by acting on the engine speed.

Due to technological constrain, both on materials and fluid, the maximum temperature for the source has been imposed to 300 °C. The cold sink is water for heating purpose, at temperature which can vary from 40 to 60 °C. The nominal mechanical output for the project is 3 kW. The engine is required to be adjustable to lower nominal power, down to 1 kW, in order to be scalable with the input source, which is a function of the solar field dimensions. The nominal output power for the engine can be increased or reduced by a factor of 10 by managing the pressure charge.

One of the first engine realized for comparable applications [2] has shown to be quite heavy, with a total weight of about 1000 kg. This can be problematic if the engine has to be installed in domestic roof. The lessons learned directed the new development to a high energy density engine, which can reduce total weight and cost for raw materials. The engine under development intends to take advantage of new manufacturing technology, the selective laser melting (SLM). It is widely recognized that heat exchanger are a critical part for any endothermic engine, and its design is even more crucial in Stirling engine, where dimensions has to be optimized and balanced between antagonist needs [3,4].

2. Methodology

2.1. Design of the thermodynamic cycle

The thermodynamic design of the cycle has used different tools in order to find the optimal parameters for main components of the engine, including pistons, the regenerator and heat exchanger. The starting point is Schmidt analysis and the Beale number [5], from which qualitative parameters can be extracted. From the beginning the Beale number make clear that an high density power, which means reducing the swept volume, can be achieved by increasing the charge pressure or the working frequency.

Increasing the working frequency has the disadvantages of increasing mechanical losses from friction, and leaves with the option of increasing instead the pressure. Other engine realized for medium temperature application show that the design of a low speed engine is the way to follow [2], in order to achieve good efficiency and reduce mechanical losses. A low speed engine can also operate more quietly and with less noise. From the beginning the results was the selection of a low speed high charged pressure concept, and the initials parameters derived from typical Beale number have been used to perform the iterative simulations.

The simulation tools used are the one developed by Urieli and Berchowitz [3], which performs a thermodynamic and heat transfer simulation for all the main components. Those method results in sinusoidal varying temperatures in the working spaces, temperature drop between the compression space and cooler, temperature drop between the heater and the expansion space, non-constant working fluid temperatures over the cycle, and pressure losses across the cooler, regenerator, and heater, as shown in Fig 1. The set of differential equation resolved numerically with Matlab is shown in Fig 2.

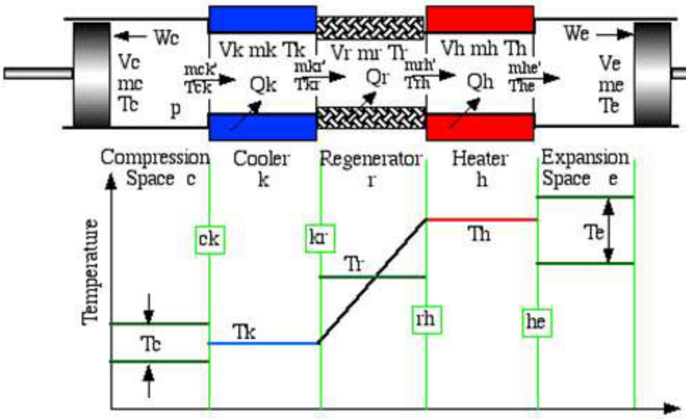

Fig 1: components and parameters model (Urieli and Berchowitz, 1984)

$p = M\,R / (Vc/Tc + Vk/Tk + Vr/Tr + Vh/Th + Ve/Te)$ $dp = \dfrac{-\gamma\,p\,(dVc/Tck + dVe/The)}{[Vc/Tck + \gamma(Vk/Tk + Vr/Tr + Vh/Th) + Ve/The]}$	Pressure
$mc = p\ Vc/(R\ Tc)$ $mk = p\ Vk/(R\ Tk)$ $mr = p\ Vr/(R\ Tr)$ $mh = p\ Vh/(R\ Th)$ $me = p\ Ve/(R\ Te)$	Masses
$dmc = (p\ dVc + Vc\ dp/\gamma)/(R\ Tck)$ $dme = (p\ dVe + Ve\ dp/\gamma)/(R\ The)$ $dmk = mk\ dp/p$ $dmr = mr\ dp/p$ $dmh = mh\ dp/p$	Mass Accumulations
$mck' = -dmc$ $mkr' = mck' - dmk$ $mhe' = dme$ $mrh' = mhe' + dmh$	Mass Flow
$dTc = Tc\ (dp/p + dVc/Vc - dmc/mc)$ $dTe = Te\ (dp/p + dVe/Ve - dme/me)$	Temperatures
$dQk = Vk\ dp\ cv/R - cp\ (Tck\ mck' - Tk\ mkr')$ $dQr = Vr\ dp\ cv/R - cp\ (Tk\ mkr' - Th\ mrh')$ $dQh = Vh\ dp\ cv/R - cp\ (Th\ mrh' - The\ mhe')$ $dWc = p\ dVc$ $dWe = p\ dVe$ $dW = dWc + dWe$ $W = Wc + We$	Energy

Fig 2: differential equation set which is resolved numerically to simulate the thermodynamic cycle [3]

2.2. Working fluid selection

At the initial stage the working fluid has been selected. The initial candidate were air, nitrogen and carbon dioxide. Other gases such as helium have been discharged for economic reasons. The potential use of carbon dioxide has been investigated at supercritical conditions. It was concluded that even if it has a greater density compared to air or nitrogen, there is no advantage when used in a Stirling cycle, since its performance are worsened for the following reasons:

- The lower compressibility of supercritical CO2 near critical point affects the pressure
- There is a lower entalphy (energy) change in supercritical CO2 compared with air
- The higher in-cylinder variations of supercritical CO2 density affects the extractable power during the expansion phase and the heat transfer is lower compared to air

Air and nitrogen share similar thermodynamic proprieties and have similar performance in a Stirling cycle. The use of nitrogen is preferred to air because it's an inert gas, reduces the risk of explosions and it's less likely to introduce humidity in the cylinders. Humidity should always be avoided because water drops condensing in the cold heat exchanger can damage the cylinders. Nitrogen has therefore been chosen as the working fluid during the simulations.

The engine designed here will be integrated with a solar collector field, which provide oil at 300 °C as the energy source for the heat exchanger. Compared to conventional engine, which extract energy from combustion gases, this design permits to reduce the irreversibility in the external part of the heat exchanger, thanks to the fact that liquids can reach higher heat transfer coefficients. The heat transfer bottleneck, located in the hot heat exchanger, is one of the main problematic which reduces thermodynamic efficiency in Stirling engines [4]. The use of a liquid as the hot source, in the form of diathermic oil, can be strategic to reduce the bottleneck and increase efficiency.

3. Design Results

The thermodynamic design of two different engine, following the requirement and constrains presented before, have been realized through the iterative use of simulations. The first engine designed is a single acting engine while the second one is a double acting engine with four opposite cylinders. The advantages and weakness for both option are presented and discussed.

3.1. Thermodynamic results for the Single acting Stirling engine

This engine is an Alpha type with two cylinders. The volume swept ratio is varied until the target power is achieved. Table 1 show the results from the iterative calculations and the optimal parameters for each component.

Table 1: parameters and dimension for the single acting stirling engine

Working Fluid	N2	
Charge pressure	160	bar
Rated speed	375	min-1
Expansion cylinder & piston		
Bore	82	mm

Stroke	68	mm
Swept Volume	360	cm3
Heater Exchanger		
Number of tubes	85	
Lenghth (exposed)	220	mm
Inner Diameter	2,5	mm
Regenerator		
Diameter	140	mm
Length	40	mm
Volume filling fact	0,3	
Gas Cooler		
Number of tubes	420	
Lenghth (exposed)	110	mm
Inner Diameter	1,5	mm
Compression cylinder & piston		
Bore	76	mm
Stroke	105	mm
Swept Volume	480	cm3

The expected performance for the engine are listed in the following Table 2.

Table 2: performance and efficiency for the single acting stirling engine

Indicated power	4.400,40	W
Cycle Work	704,10	J
Effective Power	3.474,10	W
Expansion work	18,21	kW
Heat to Working Fluid	17,84	kW
Heat out of Working Fluid	13,40	kW
Compression Work	13,81	kW
Cycle Carnot Factor	0,347	
Mechanical efficiency	0,347	
Overall efficiency	0,195	

The effective mechanical power for the engine is 3,4 kW at the charging pressure of 160 bar with an overall efficiency from thermal to mechanical of 19 %. The rated power can be reduced, if needed, by decreasing the charging pressure. The main disadvantage for such option is a very high charging pressure (160 bar) and need for a flywheel of 200 kg in order to maintain stable operating conditions.

3.2. Thermodynamic results for the Double acting Stirling engine

Starting from the results obtained with the single acting machine, the research has moved toward a different configuration, which could avoid the problems arising from the charge pressure and flywheel requirement. The configuration with four opposite cylinders, shown in Fig 3, has been found as a good alternative, since the pressure and mechanical forces are better balanced.

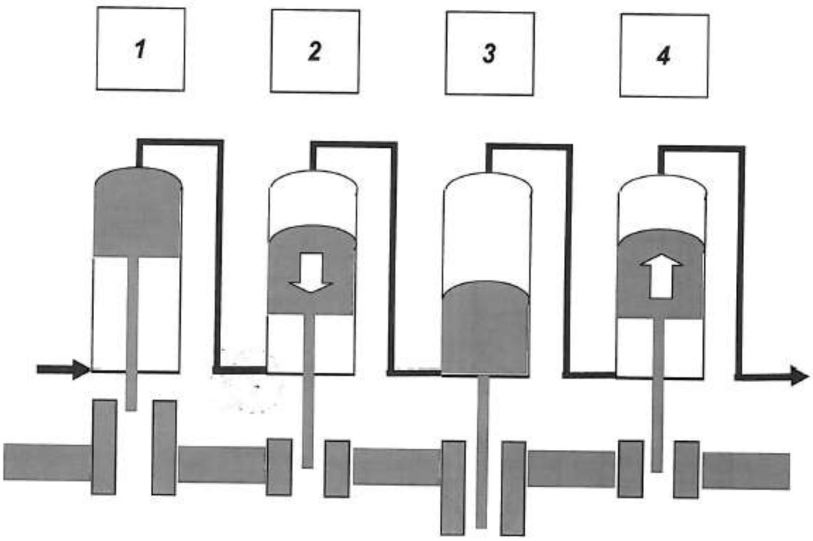

Fig 3: basic configuration for the double acting Stirling engine with four working spaces

The use of double acting pistons permits to reduce gas leakages, since only the rod is connected with the external environment. The total net work is shared between the four cylinders, which are thermodynamic delayed by a cinematic angle of 90°. This means that the compression and expansions phases are performed during each cycle in different moments by the four cylinder. The parameters of each components derived iterative optimization are show in Table 3.

Table 3: parameters dimension for the single acting stirling engine

Working Fluid	N2	
Charge pressure	135	bar
Rated speed	250	min-1
Expansion cylinder & piston		
Bore	68	mm
Stroke	40	mm

Swept Volume	145	cm3
Heater Exchanger		
Number of tubes	34	
Lenghth (exposed)	210	mm
Inner Diameter	2,5	mm
Regenerator		
Diameter	70	mm
Length	40	mm
Volume filling fact	0,3	
Gas Cooler		
Number of tubes	125	
Lenghth (exposed)	170	mm
Inner Diameter	1,5	mm
Compression cylinder & piston		
Bore	68	mm
Stroke	40	mm
Swept Volume	140	cm3

The length, number and diameters of the heat exchanger has been optimized in order to reduce the temperature drop from the sources. The oil heat exchanger is constituted by 34 tubes with a diameter of 2,5 mm, while the cooler has 125 tubes with a diameter of 1,5 mm. A efficiency of 19% and a mechanical power 3,3 kW can be achieved with a charging pressure of 135 bar (Table 4).

Table 4: performance and efficiency for the double acting stirling engine

Indicated power	3.801,60	W
Cycle Work	912,40	J
Friction losses	447,20	W
Effective Power	3.354,40	W
Expansion work	18,21	
Heat to Working Fluid	17,84	kW
Heat out of Working Fluid	13,40	kW
Compression Work	13,81	kW
Cycle Carnot Factor	0,359	
Mechanical efficiency	0,882	
Overall efficiency	0,190	

The pressure-volume diagram for the simulated cycle is presented in Fig 5, while the pressure evolution in the four working spaces during one cycle can be observed in Fig 6. The maximum pressure is found to be 175 bar, during the compression phase. This picture show the thermodynamic delay by a mechanical angle of 90° in the working spaces. Mechanical forces on pistons rod are calculated from pressures, by subtracting the opposite forces acting on each piston (Fig 4); their values over one cycle are show in Fig 7.

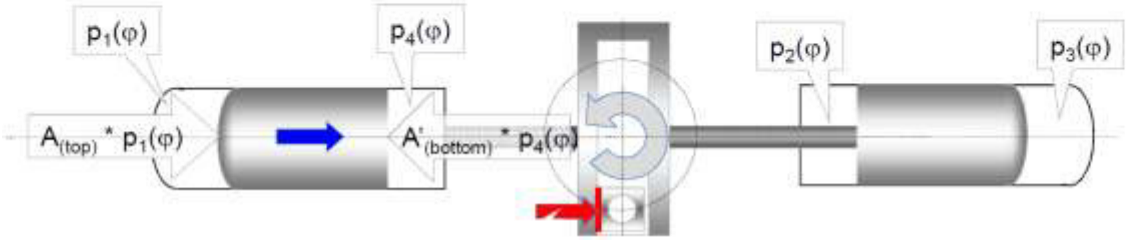

Fig 4: force acting on the piston rod and resulting force on the shaft

The forces acting on the crankshaft are decomposed by radial and tangential components, and net torque acting on the shaft is calculated (Fig 7). The torque has always a positive value, which means that the engine is always delivering a positive force; for this reason, the flywheel weight is less than 20 kg. This aspect represent a main advantage compared to the flywheel requirement for a comparable single acting engine. The results from the thermodynamic cycle are the inputs used during the mechanical design phase.

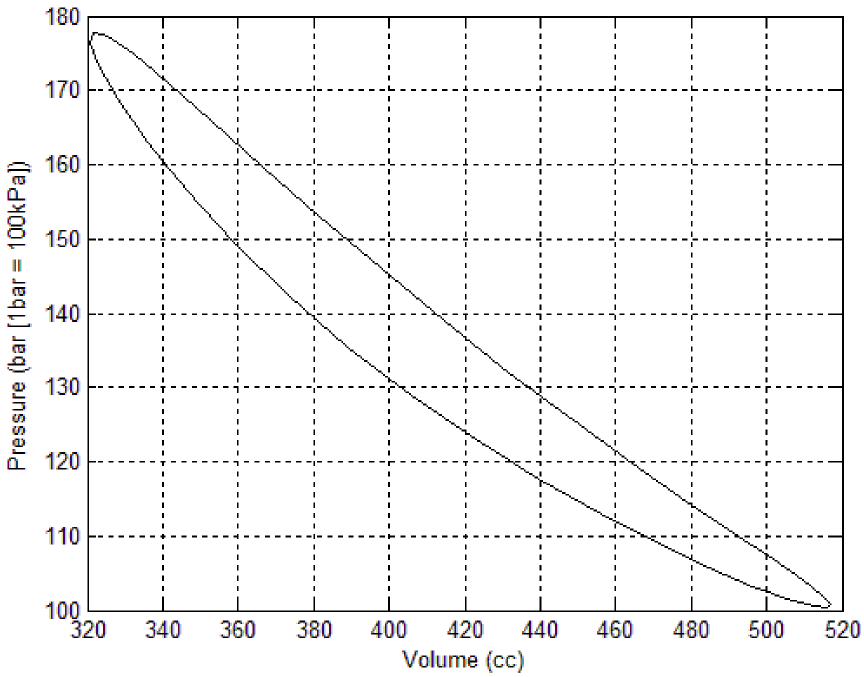

Fig 5: pressure-volume diagram for a single working space. The area represent the net work done by the fluid

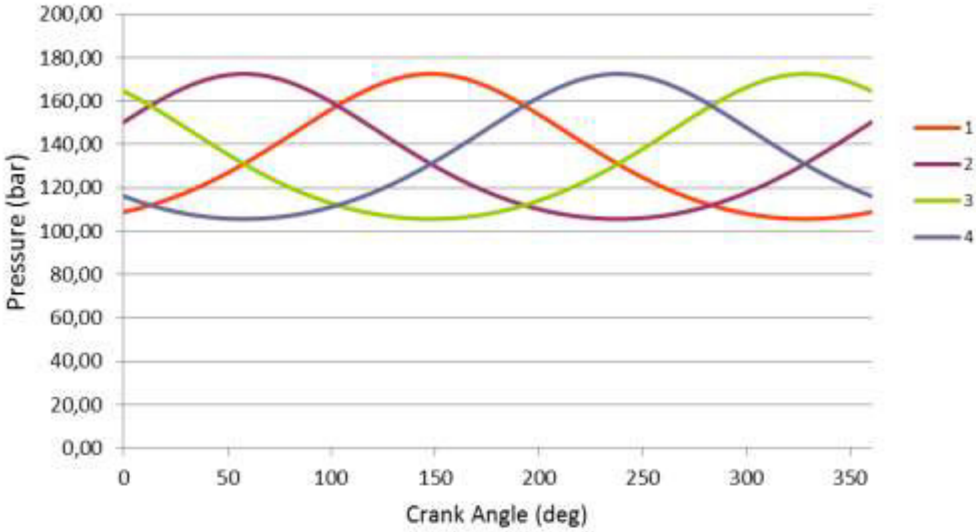

Fig 6: pressure versus crank angle in the four working spaces

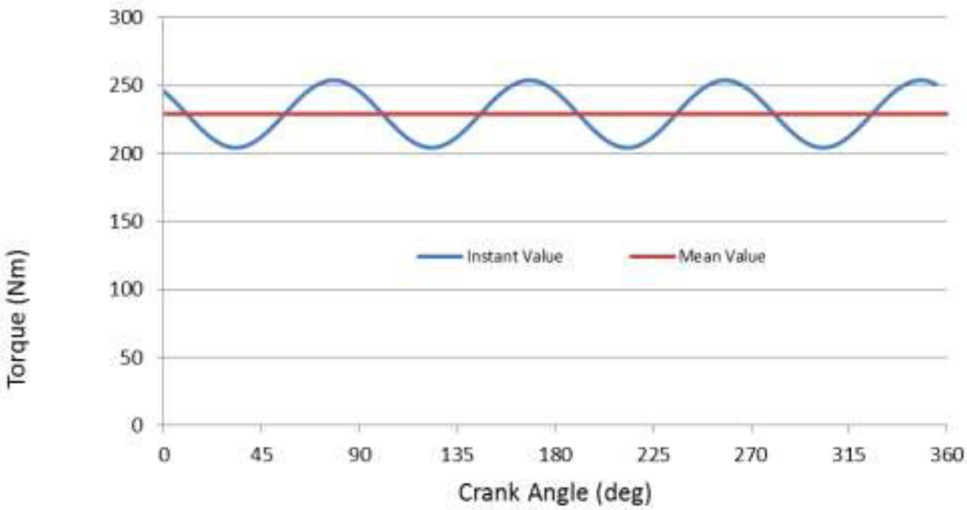

Fig 7: torque on the shaft

Fig 8 show the simulation results for the temperature in the working spaces. The heat transfer resistance from the sources to the heat exchangers is responsible for the difference between those two temperatures. The temperature in compression and expansions space varies accordingly to the model assumptions. The mean temperature for the regenerator is found to be an intermediate value between the hot and cold source. During the first part the thermal energy from the gas is passed to the regenerator matrix, while during the second part of the cycle the process is reversed. The net heat transfer to the regenerator over a cycle is zero.

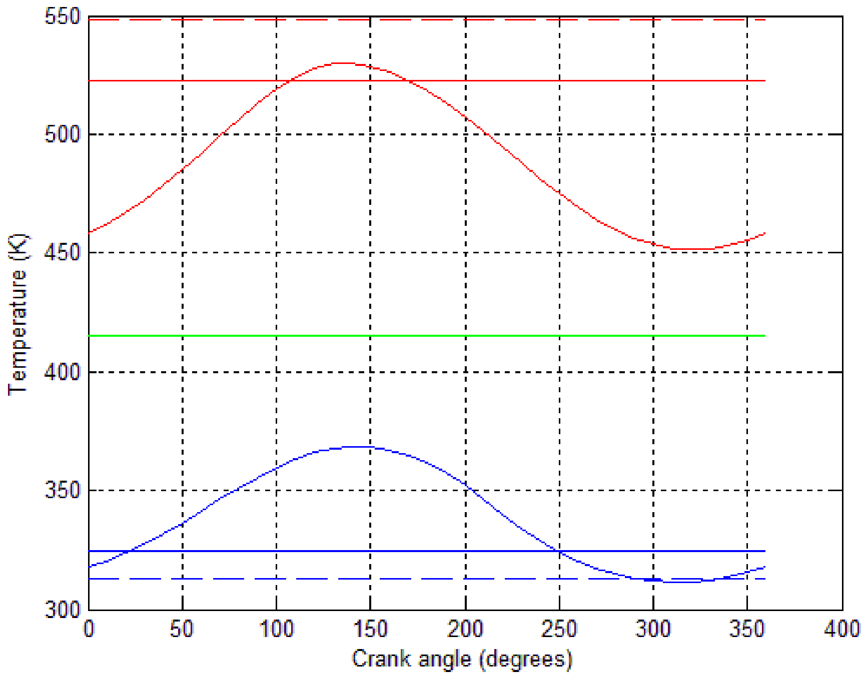

Fig 8: wall and gas temperature versus the crank angle (red dashed=hot source temperature, red constant=heater temperature, red variable=compression space temperature, green=mean regenerator temperature, blue variable=compression space temperature, blue constant= cooler temperature, blue dashed=cold source temperature)

4. Conclusions

The design procedure followed for the realization of a low-temperature Stirling engine is presented. Thermodynamic and heat transfer calculations are conducted in order to achieve and optimize the requirements fixed within the Digespo Project.

Results obtained during this phase (pressure, forces) will be used to design the mechanical components, and to realize a full-scale demonstrative prototype.

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