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Modelling and Simulation of a Three-stage Air Compressor Based on Dry Piston Technology

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Abstract: In order to develop an inventory cylinder-piston assembly used for compressing air, it is of a great interest to develop a simulations tool to characterize theoretically this compression system. Each of the three stages of the dry piston has been modeled, using COMSOL multi-physics software. The main difficulty is there to study the heat transfer and fluid dynamics in a system where the topology is changing, as it is a piston regularly in movement. COMSOL as Finite Element Method software has been chosen, well dedicated for such a study.

Keywords: Isothermal process, Heat transfer, Efficiency, Moving mesh.

1. Introduction

In the general frame of Compressed Air Energy System, the LEI Laboratory of EPFL has introduced the concept of dry piston. The main goal is to achieve an efficient energy storage system by the means of compressed air thanks to high efficiency compression and expansion processes. In this regard both processes should be as close as possible to isothermal processes. For achieving this goal, we have defined a new compression and expansion machine, based on three dry pistons, with increased area of heat transfer through fins inside cylinder piston assembly. The association of three pistons allows a compression process by stages: from 1 to 5.8bars, from 5.8 to 34bars, and from 34 bars up to 200bars. A test bench is actually under development in order to propose an experimental validation of such a compression process concept. In order to allow some reasonable simulation times with a “limited” power for the computation, each of the compression stage has been modeled independently. the core of this work was to develop tools for the modeling and the characterization of the three stages of a dry piston system. All the geometries presented here can from now on be optimized and easily modified to match the characteristics that will be finally implemented.

2. Generalities on the dry piston three stages modeling topologies

Each geometry of the stage of the dry piston system has been predefined. From the defined dimensions, we have directly drawn each stage directly with the CAD Comsol interface. Each stage has been considered separately from the two others. Moreover, the direct structural environment of each stage has not been considered. This last simplification is justified by the assumption that the temperature of the solid part of each stage does not vary significantly for few cycles compression/expansion, with no heat transfer between the solid part of the piston and their environment.

As each piston geometry has an evident symmetry along the vertical axe of the compression chamber, we have only drawn the half of each stage, in a 2D plane. This allows reducing the simulation constraints in time and in needed computational power. The geometry of each stage is proposed in Figure.1.

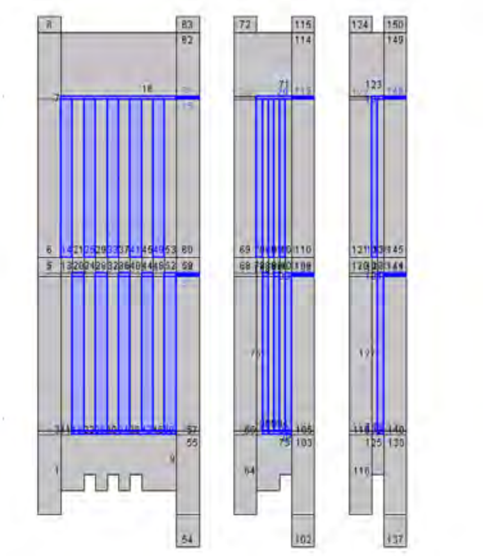


Figure 1. The three stages of the compressor.

For each of the stages, the blue domains represent volume of air to compress, while the gray surfaces represent the structure of the pistons.

We must note that:

- The vertical axe defined by x=0 defines the symmetry axe for each piston

- Each domain is divided in much more sub-domains that what is normally needed.

This is required for enabling the deformation of the structures with a correct mesh displacement in order to allow the expulsion of the compressed air, as well as the injection of air at the end of one cycle, two sets of openings have been considered at the right upper and right middle side of each stage, as shown in Figure. 2.

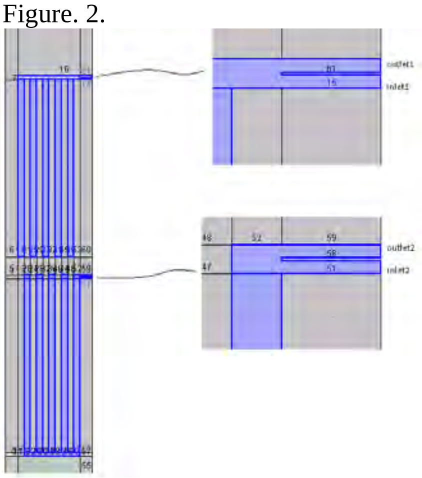


Figure 2. outlets and inlets.

In each set, the upper openings will be considered as the outlet of the compression chamber (expulsion of compressed air). The lower opening is an inlet, required for the admission of low pressure air.

2.1 Main parameters

Independently from the structure, various parameters must be defined. These parameters are generic for each stage.

The first set of parameters is required for modeling the displacement of the moving parts of the piston. These parameters are:

-The period of one cycle compression/expansion.

- The length of the piston travel.

- The radius of the wheel that allows the movement of the piston.

- The displacement of the piston is deduced from these parameters and the current time of the simulation. Its speed is obtained by the derivative of the displacement versus the time.

The second set of parameters defines different pressure and temperature specifications:

- T_init and p_init define the initial temperature and pressure.

- T_amb defines the ambient temperature around the piston.

- p_ref defines the reference pressure at which the air in the compression chamber will be expulsed during compression. For the admission of air during expansion, the reference pressure we consider is the initial pressure p_init.

The last set of parameters is related to the main characteristics of the materials considered for modeling the stages.

- Solid part: we have considered Aluminum. It is defined with its density, its thermal conductivity and its heat capacity.

- The air: it is also defined with its density, its thermal conductivity, its heat capacity and its ratio of specific heats for the heat transfer study. It also defined with its dynamic viscosity for the fluid dynamics. It is obvious that these parameters are affected by the temperature, and the pressure. This is taken into account in our model, at each simulation step of the calculation.

2.2 Considerations on the Finite Element modeling of the three stages Generalities

The core of the modeling is to study heat transfer and fluid dynamics processes for the three stages.

The main particularity of this study is that heat transfer and air movement are due to the movement of the piston. It means that the volume change of the compression chamber must necessarily be taken into account, to identify the pressure evolution, as well as the linked temperature increase/decrease.

This is the reason why, using a FEM software for this study, we have first to implement a “moving mesh” solver to compute the volume changes of the compression chamber.

The second solver to implement is a “Fluid dynamics” type solver. It must allow the correct computation of the fluid behavior in the compression chamber. It must enable to identify the pressure change of the fluid, and thus the heat generation impacts.

The last solver is logically a “heat transfer” type solver, to identify the temperature gradient of the structure.

One must keep in mind that if the “moving mesh” solver can be considered as a strict input, the “heat transfer” and the “fluid dynamics” effects are strongly linked, and influence them each other. It is then a strongly non-linear modeling approach.

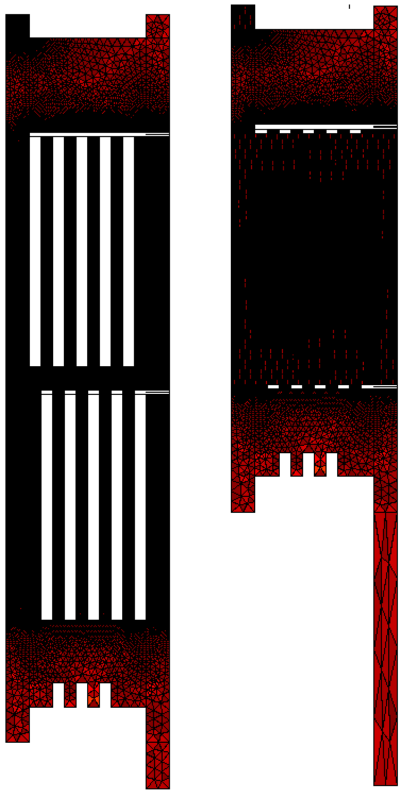
2.3 Moving meshes and meshes

The first step is to create the mesh needed by the FEM software from the drawn presented in Fig.1. This must be made following a few imperative requirements:

- To obtain the needed accuracy for the simulation results, it is needed to mesh the structure with significant fine meshes for the fluid dynamics part of the model. This is also an imperative condition for the correct convergence of the solver.
- As the structures are related to moving piston, the meshes must be re-defined at each simulation step for a moving topology. However, this can be time consuming. This is the reason why a second possibility is implemented. It consists in defining some given mesh, and in specifying the way the mesh will be deformed as a function of the piston displacement. For each domain, the mesh displacement is strictly defined. The definition of the mesh displacement must be strictly made, in order to give no degree of liberty for the solver to choose by itself the way the meshes could move. This is required to avoid any mesh inversion, which could lead to a decrease of the simulation accuracy.

An example of such a moving mesh generation process is given in Fig. 3 for the first compression stage, and for the full expansion and compression modes. For this illustration of the moving mesh we have implemented, we present only the mesh related to the aluminum part of the piston. This illustrates the reason why the structure we have implemented present more domain than normally needed: during the compression process, the mesh of some part are effectively compressed, while the meshes of other parts of the geometry are expanded.

This underlines also a main difficulty is the definition of the mesh size in each domain: the size variation of the meshes during a compression/expansion is continuously varying. we must find a trade-off that leads to homogeneity of the mesh distribution whatever the state of the piston is.



Full expansion Full compression

Figure 3. Mesh of the structures.

3. Use of COMSOL Multiphysics

3.3 Heat Transfer

Heat transfer inside the cylinder occurs through both convection and conduction in gas, convection from gas to solid and conduction inside solid. For this reason we have considered conjugate heat transfer to consider all this effects. The relations below can formulate the complicated heat transfer correlation inside the cylinder:

$$\rho\frac{\partial \mathbf{u}}{\partial t}+\rho\left(\mathbf{u}\cdot\nabla\right)\mathbf{u}=\nabla\cdot[-\nabla pI+\mu\left(\nabla\mathbf{u}+\left(\nabla\mathbf{u}\right)^{T}\right)-\frac{2}{3}\mu\left(\mathbf{u}\cdot\nabla\right)I]+F$$

$$\rho\frac{\partial \rho}{\partial t}+\nabla\cdot(\rho\mathbf{u})=0$$

$$\rho C_p\frac{\partial T}{\partial t}+\rho C_p\mathbf{u}.\nabla T=\nabla.(k\nabla T)+Q+Q_{vh}+W_p$$

As Mentioned previously, the main parameters needed for this solver are the density, the thermal conductivity, and the heat capacity of the air and the aluminum. The ratio of specific heats for the air only is also needed. We remind that the air parameters are function of the pressure and the temperature. It is taken into account at each simulation step.

Moreover, we must specify to the solver that parts of the geometry will move to take into account some coefficients directly linked to movement in the heat transfer theory. This is the reason why, in Fig. 4, some domains must be specified by some translational motion rules (part of the geometry which are the mobile part of the piston).

For the air, we must also specify the pressure work. In order to simplify the set of equations to solve, the low Mach number formulation has been chosen. This a priori hypothesis has been largely validated by the analysis of the simulation results, where the air speed has never been over 50m/s. for a maximum piston speed of 2m/s.

At least some boundary conditions must be defined. Considering again the three geometries defined in Fig. 1, Fig. 2 and Fig. 3:

- The left side of the geometries is defined by some symmetry conditions.

- The lower side of the geometries is defined by strict temperature conditions (Ambient temperature).

- The right and the upper side of the geometries are defined by cooling conditions

(Interface between aluminum and the non-simulated external air), where a per default Heat transfer coefficient h=50W/(m2.K) has been considered a the first guess.

But this last parameter must be adjusted more precisely. A general approach to modelling the heat transfer between the gas and the cylinder head walls, $\dot{E}_w$ is

$$\dot{E}_w = H_c A_c (T_w - T_c)$$

Where H_c, is the overall heat transfer coefficient, A_c, the cylinder control volume surface area, T_w, the surface area temperature and T_c, the instantaneous gas temperature.

The variation in area can be calculated from the piston motion. The wall temperature may be assumed a constant mean value during the cycle due to high heat capacity and conductivity of the metal structure. Regarding the heat transfer coefficient, one approach to use may be that given by Eichelberg (2)

$$H_c = 2.47\sqrt{P_c T_c}\sqrt[3]{V_{pm}} \qquad (1\text{-}1)$$

Where P_c, and T_c, are instantaneous cylinder gas pressure and temperature and V_{pm}, the mean piston velocity. This relation is correlated for diesel engines, but should also be useful for compressors.

After solving the model with first guess and extracting these parameters, h will be calculated from this formula.

Finally, two more boundary constraints have been defined regarding the inlet and outlet of each geometry (see Fig. 2):

- Inlet: the boundary is defined by a strict temperature condition (ambient temperature).

- Outlet: the boundary is defined by an open boundary condition, only valid for the heat transfer solver. Such a condition set the outlet temperature of the air to that we defined by the air from the chamber, affected by the surrounding aluminum.

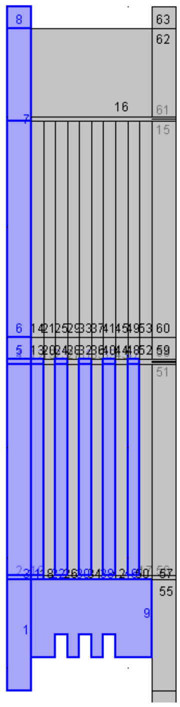


Figure 4. Translational motion for the first compression stage.

3.3 Fluid Dynamics

This solver is valid for the air only. The additional parameter needed is the dynamic viscosity of the air. This parameter is of course a function of the pressure and the temperature. It is taken into account in our simulations.

We takes also into account the effect due to the volume force effect: due to the non-homogeneity of the temperatures and pressure inside the air, some gradient of air density are observed, leading to additional and non homogenous forces on the air. Regarding the constant time of the compression/expansion process, the influence of this effect is negligible for the balance of temperature and on the movement of the air inside the piston. However, it is of a great importance to take this phenomenon into account, as it stabilizes the system and enable the solvers to find more easily accurate solutions.

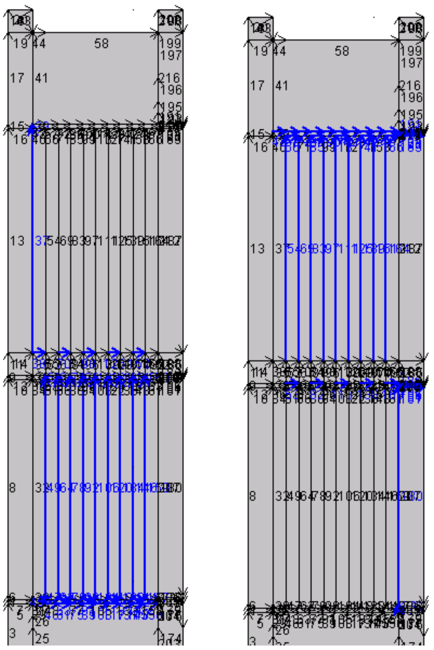
The boundary conditions have been set as follow:

- For all the boundaries except the inlet and the outlet, the condition is a wall with no slip condition.

- As seen in Fig. 5, two kind of wall are defined. A fix wall for the fix part of the piston, and a moving wall for the moving part of the piston. The speed of the moving wall is defined by the speed of the piston

- For the outlet (see Fig. 2): the boundary condition is a speed for the outrush air. This speed is 0 when the pressure is below the reference pressure. The speed is a function of the piston speed when the pressure is equal or higher than the reference pressure.

- For the inlet (see Fig. 2): the boundary condition is a speed for the inrush air. This speed is 0 when the pressure is higher than the initial pressure. The speed is a function of the piston speed when the pressure is equal or lower than the initial pressure.



Fix wall, no slip   Moving wall, no slip

Fig. 5 – Boundary conditions for the fluid dynamics solver - Second compression stage

4. Main results

Once the three compression stages model are implemented, calculations must be operated along 80’000 up to 12’0000 meshes. The average element quality is 0.98, for a mesh area near of 0.0132m2.

For the reason of limitation of pages we will just show the results for the first stage.

4.1 Temperature gradient during compression and expansion

First qualitative result is observed for the temperature gradient.

This temperature gradient is plotted in Fig. 6 in the middle of the last expansion travel.

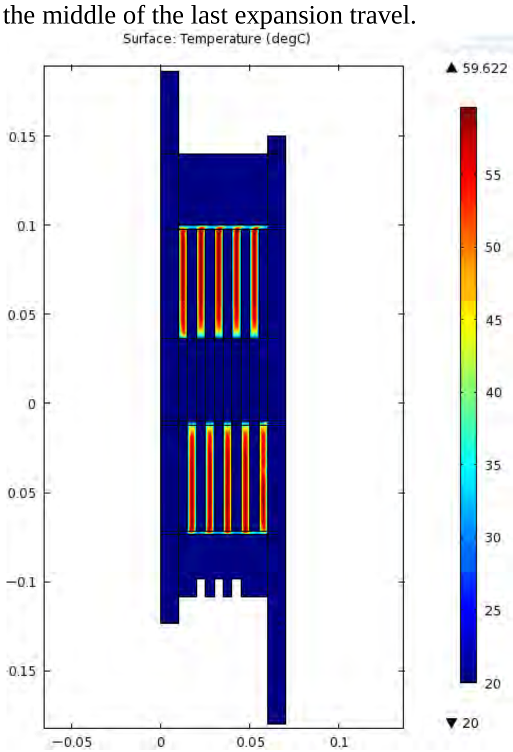


Fig. 6 – Temperature gradient – mid of last expansion travel

4.2 Global results: Speed, temperature and pressure

Some quantitative results are more interesting, as they enable to analyze directly the performances of the three stage dry piston.

At each simulation steps, various speeds, temperatures and pressure are calculated. The first results we present are related to the speed, compared to the speeds:

- Of the air expulsed during the compression travel.

- Of the air absorbed during the expansion travel.
These results are given in Fig. 7 for just the first stage.

The same way, the pressure profiles can be obtained. We present in Fig. 8 the pressure profiles of:
- The average pressure in the compression chamber.
- The pressure at the outlet during air expulsion.
- The pressure at the inlet during admission.

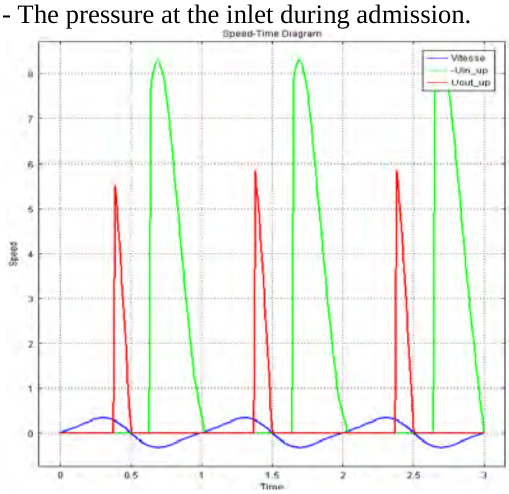


Fig. 7 – Speeds for the first stage

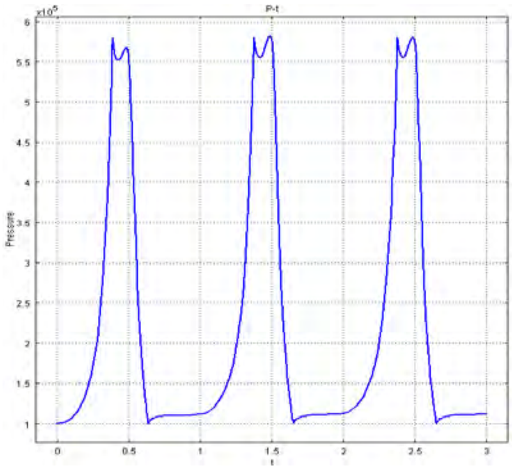


Fig. 8 – Pressure for the first stage

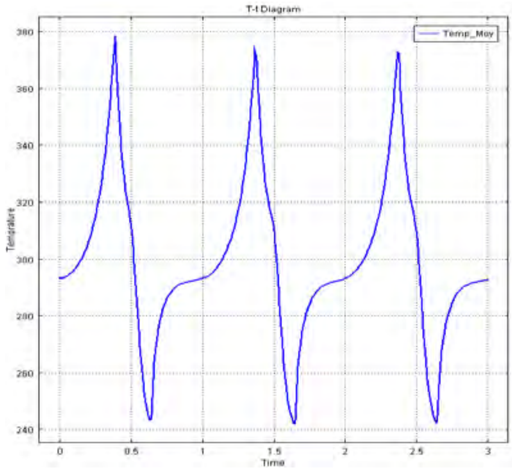


Fig. 9 – Temperature for the first stage

4.3 Power and energy

At least, knowing the pressure at the bottom of each piston, as well as the surface of the pistons, we can compute the power needed for moving the pistons of each stage.

By numerical integration of these power profiles, the energy (work) needed for the compression/expansion processes can be deduced.

The power and work profiles for the first stage along 3 cycles are presented in Fig. 10 and 11.

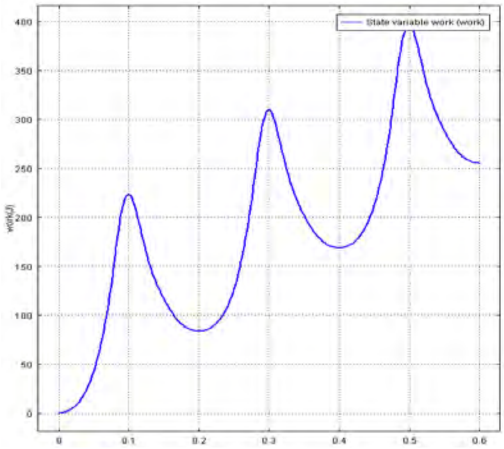


Fig. 10 – Work for the first stage

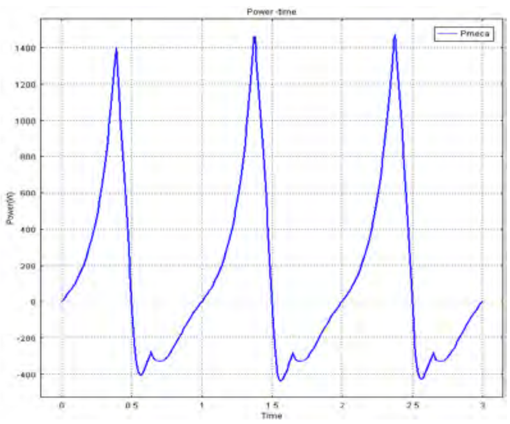


Fig. 11 – Power for the first stage

The power profiles show some positive half-wave, linked to the compression processes. The second half-wave is negative, during the expansion process, when the system restores part of the energy invested in it.
This last comment can also be made regarding the energy profiles.
Energy must be given to each piston during the compression process (the energy level is increasing). During the expansion process, each piston brings back part of this energy (the energy level is decreasing). But in all cases, the energy recovered during the expansion is lower than the energy invested for the compression:
- Because part of the energy invested has been transferred during the expulsion of a high temperature compressed air.
- Because part of the energy, transformed in heat during the compression, has been exchanged with the aluminum. It is not necessarily recovered during the expansion.
The analysis of the energy profiles are the profiles when have to focus now in order to characterize precisely the efficiency of the three stage dry piston. The last unknown to solve from now on is, as already mentioned, the deep study of the heat transfer from the air to aluminum.
Finally P-V diagram has been plotted in Fig.12 The area under this diagram shows us the work required by compression process.

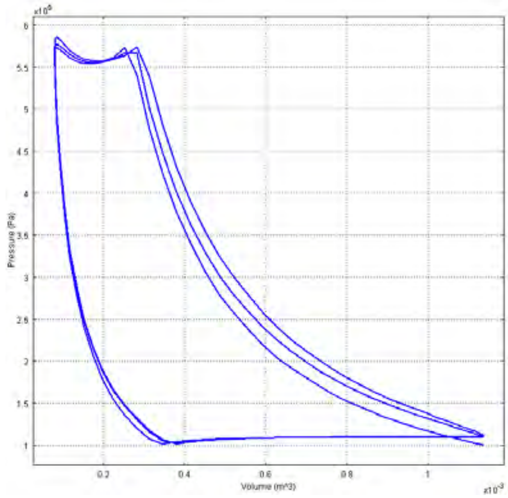


Fig. 12 – Pressure-volume diagram for the first stage

Based on these diagrams we are able to calculate the efficiency of the system that has been shown in table 1.

Table 1: compressor Performance indicators for each stage

variable	1st stage	2nd stage	3rd stage
Net work (kJ/kg)	166	168	157
Isothermal efficiency	0.88	0.87	0.94
Polytropic factor	1.164	1.168	1.06
Volumetric efficiency	0.75	0.65	0.41

7. Conclusions

The results will be validated by an experimental setup currently under construction. Current results show an efficient, close to isothermal process for compression expansion devices, thanks to increased heat transfer surface through added fins in the design.

8. References

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2. C. Eichelberg, Some new investigations of old combustion engine problems. *Engineering*, (1939)