Comparison of Classic and Finned Piston Reciprocating Linear Air Compressor Using COMSOL Multiphysics®

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Abstract: The intention of this paper is to provide a finite element model for a finned piston compressor based on a classic compressor model using COMSOL. An analytical lumped model for both of the systems is developed already (verified with an experimental test), but it is interesting to have a detailed study on distribution of parameters and their local behavior as well. The procedure of developing the model is described and results are compared. The results confirm very well that the temperature rise in finned piston is much less resulting in reduced work required. The results are also compatible with analytical model

Keywords: Compressor, Heat transfer, Moving mesh, Isothermal.

1. Introduction

Positive displacement machines like reciprocating compressors are one of the most important machines used throughout the industry. The current work is dedicated to develop a detailed CFD model for such machines. This work can provided a basis for the modeling of an engine as well. In general, a cycle of operation of a high speed positive displacement compressor (Figure 1) can be described as a number of complicated phenomena, interacting and taking place in a short period of time. One of the applications of such machines is compressed air energy storage (CAES). In this application, a compressor is used to compress the air during the off-pick electricity demand time, and store it in a reservoir, and during the pick time, the reverse mode (expander) can extract the stored energy to feed the grid. The closer the compression/expansion to isothermal process, the more efficient it is. So, the idea is to keep the air close to ambient temperature by increasing the heat transfer rate through increasing the heat transfer surface by introducing a "finned piston" (Figure 2).

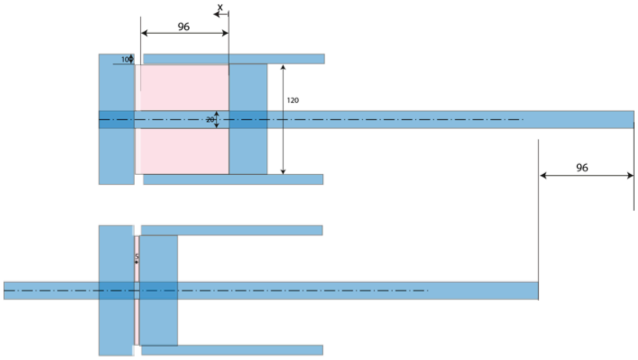

Figure 1. Classic piston positions at BDC and TDC.

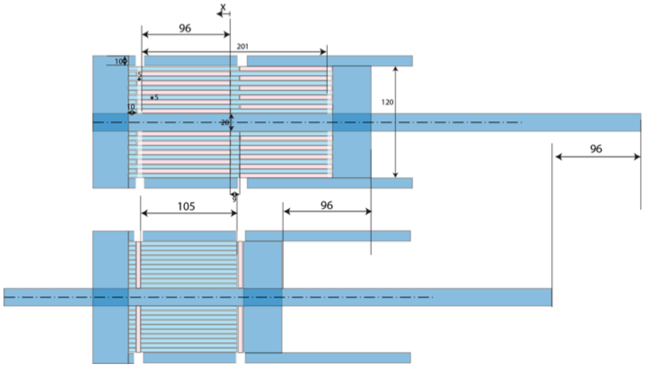

Figure 2. Finned piston positions at BDC and TDC.

2. Analytical model

The analytical model is base on the mass and energy balance together with ideal gas law and an equation relating energy to temperature. First this model is applied to the simple compressor and then developed to a finned one.

2.1 Classic piston

A reciprocating compressor is an open, non-uniform, unsteady state (transient) machine. The procedure of the model is briefly described here, but can be found in detail in [1].

2.1.1 Driver mechanism

An electrical motor provides the compressor with a constant velocity during compression and expansion through a ball-screw driver, that's why it is called a liner drive.

Eq. (1) shows the relationship between rotational speed of electrical motor ω, and the piston speed $\dot{x}_p$, as well as the relation between motor torque τ, and piston force F, recognizing the complicated function of θ as a transformer modulus, and calling it $m(\theta)$:

$$m(\theta) = \frac{\dot{x}_p}{\omega} = \frac{\tau}{F} \qquad (1)$$

Plus, the relation between the volume change and piston displacement can be deduced from Eq. (2).

$$\dot{V} = -A_p\dot{x}_p \qquad (2)$$

2.1.2 Cylinder head

The mass and energy balances for the cylinder head can be demonstrated using Eqs. (3) and (4).

$$\dot{m} = \dot{m}_i - \dot{m}_e \qquad (3)$$

$$\dot{U} = \dot{E}_i - \dot{E}_e - p\dot{V} + \dot{Q} \qquad (4)$$

Here U is the internal energy of the control volume,

$$U = mc_vT \qquad (5)$$

and it is assumed that the air obeys the gas law:

$$PV = mRT \qquad (6)$$

2.1.3 Valves

The valves can be assumed to be as a simple orifice with effective cross-sectional area, A_v, and isentropic flow. The mass flow rate, $\dot{m}_v$ through the valve as a function of valve area, upstream pressure P_u, temperature T_u, and downstream pressure, P_d can be calculated using Eq. (7).

$$\dot{m}_v = A_v p_u \sqrt{\frac{2k}{(k-1)RT_u}}\sqrt{(p_r)^{2/k} - (p_r)^{(k+1)/k}}$$

(7)

Where k is the ratio of specific heats and R is the ideal gas constant.

The area modulations A_i for the inlet and A_e for the exhaust are needed to be specified by a logic expression.

2.1.4 Heat transfer

The rate of heat transfer between the gas and the cylinder head wall is modeled by a general approach given in Eq. (8).

$$\dot{Q} = H_cA_c(T - T_w) \qquad (8)$$

Where H_c is the overall heat transfer coefficient, A_c the cylinder surface area, T_w the surface area temperature and T the instantaneous gas temperature. Regarding the heat transfer one approach to use may be that given by Eq. (7) developed by Adair et al. (1972).

$$H_c = 0.053\frac{k}{D}(Re)^{0.8}(Pr)^{0.8} \qquad (7)$$

Where Re and Pr numbers are based on flow prosperities, the flow speed and the cylinder diameter.

2.2 Finned Piston

The Finned piston is developed based on the same principals described for classic piston, but instead of one chamber it has numerous chambers that interact with each other from pneumatic as well as heat transfer aspect. Authors have used a detailed pneumatic- thermo-electric [2] and thermo-electric [3] to model the complicated finned piston, which is out of the scope of this paper.

3. Use of COMSOL Multiphysics®

The analytical model developed so far considers a uniform distribution of particles in cylinder, meaning one pressure one density and one temperature in the cylinder. In order to have a more detailed result on each area in the cylinder and study the flow and heat transfer we have developed a FEM model using COMSOL Multiphysics® and compared it to our analytic model.

There are some common parts in modeling both compressors, which are related to definition of the driver motion and properties of material and initial and ambient conditions. These definitions will be covered in section 3.1. Then classic and finned piston will be described from the geometry and moving mesh and solver point of view.

3.1 Global definitions

Independently from the structure, various parameters must be defined. The first set of parameters defines variables such as different pressure and temperature specifications:

- t_per defines the period of one cycle compression/expansion.
- 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.

3.2 Definitions

What is notable in defining the global definitions, is to define the linear drive displacement as a piecewise linear function. The velocity of the piston can be derived by the derivative of the displacement, which is basically constant during compression and expansion, with the exception of start and stop transition periods. The function for defining the displacement is shown in Table 1. Velocity is defined in the variables as the derivative of displacement. The change of displacement and velocity is shown in Figure 3.

Table 1: Definition of a piecewise function for displacement.

Start	End	Function
0	0.99	0.974*(0.1*t)
0.99	1.98	0.99

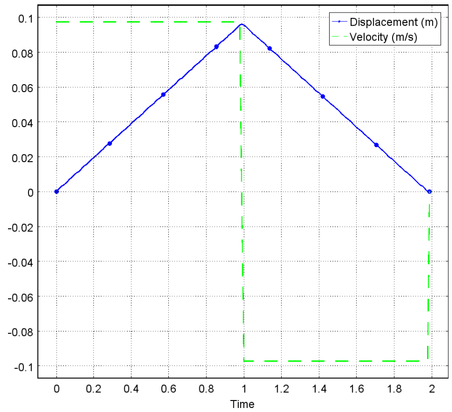

Figure 3. Displacement and velocity of the piston in one cycle.

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

- Solid part: Aluminum has been considered as the body of cylinder and the piston. The density, thermal conductivity and heat capacity is defined for these parts.

- 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 temperature. This is taken into account in our model, at each simulation step of the calculation.

4. Classic piston

The general approach for modeling a cylinder-piston assembly with inlet and outlet will be described in this section.

4.1 Geometry

The geometry of the classic piston has been drawn in COMSOL environment. From the defined dimensions, we have directly drawn each stage directly with the CAD COMSOL interface. Such a geometry with inlet/outlet valves is shown in Figure 4, where the blue part is air and grey part is the metallic cylinder piston assembly.

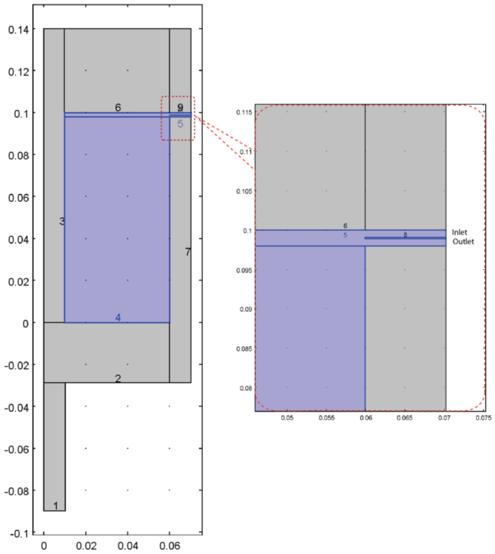

Figure 4. Geometry of a classic cylinder-piston assembly with axial symmetry (left) and zoom on inlet/outlet area (right).

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

4.2 Moving mesh and meshes

The geometry of the classic piston can be divided into three parts to define the moving mesh:

- Cylinder, which is stationary: Fixed mesh.
- Piston, that moves without deformation: Prescribed deformation, which is equal to displacement defined already.
- Air, which both moves and deforms using a prescribed deformation defined as: Displacement*(Y-A)/(-A), where A is the stroke of the piston.

Moving mesh specifies the way the mesh will be deformed as a function of the piston displacement. For each domain, the mesh displacement is 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. The moving mesh during compression of the classic piston is illustrated in Figure 5.

Once the classic piston model is implemented, calculations must be operated for 38'000 elements of meshes. The average element quality is 0.99, for a mesh area close to 0.014.

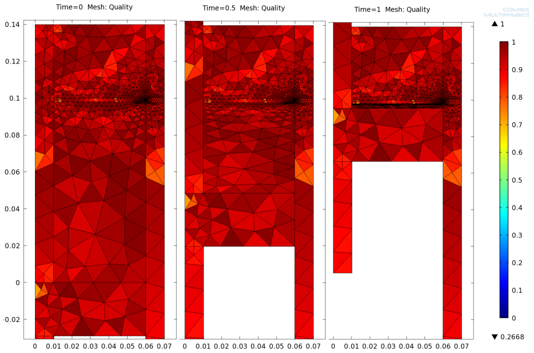

Figure 5. Mesh of the classic piston for full expansion mid-travel and full compression respectively.

4.3 Heat Transfer

Heat transfer inside the cylinder occurs through both convection and conduction in gas, convection from gas to solid and conduction inside solid. Besides, a part of the heat will be stored in the body of the metal. For this reason we have considered conjugate heat transfer to consider all this effects.

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, the parts of the geometry that will move should be specified to the solver. This is the reason why in Figure 6 some domains (part of the geometry which are the mobile part of the piston) must be specified by translational motion rules.

For the air, pressure work should be also specified.

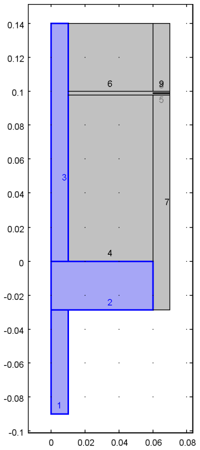

Figure 6. Translational motion for the piston.

The boundary conditions must be defined for heat transfer solver. Considering again the geometry defined in Figure 4.

- The left side of the geometry is defined to have symmetry conditions.
- The outer sides of the geometry is set to have convection cooling.
- Inlet: the boundary is defined by Laminar inflow with an entrance pressure of max(Pin,Pinit) and entrance length of 0.1.
- Outlet: the boundary is defined by laminar outflow with an exit pressure of min(Pout,P_ref) and exit length of 0.1.
- For all the boundaries except the inlet and the outlet, the condition is a wall with no slip condition.
- As seen in Figure 7, two kind of wall are defined. A fixed wall for the fixed 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

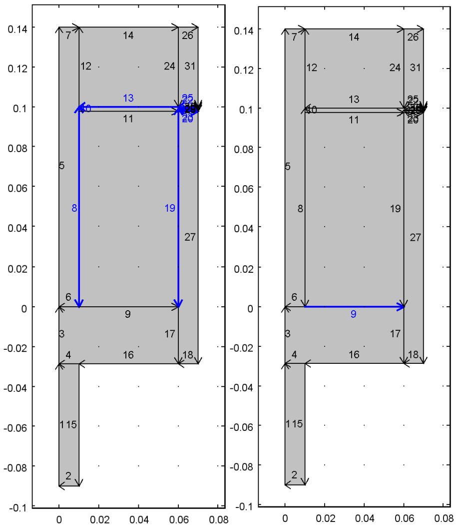

Figure 7. Boundary conditions with no slip condition (left) for fixed wall and moving wall (right) respectively.

4.4 Study

The solver uses a time-dependent compiler, and dependent are T, u (velocity) and p. The range of the study is equal to one complete cycle of compressor (0,1.98) with time step of 0.01 seconds.

5. Finned piston

The general approach is the same as described for a classic piston, so only the additional items or differences will be described in this section.

5.1 Geometry

The geometry of the finned piston with two inlets and outlets is sketched according to Figure 2, showing air and metallic areas respectively with blue and grey colors as shown in Figure 8. Figure 9 shows that the fines are not in contact with each other but a layer of air flow between them. This as will be seen later will enhance the heat transfer.

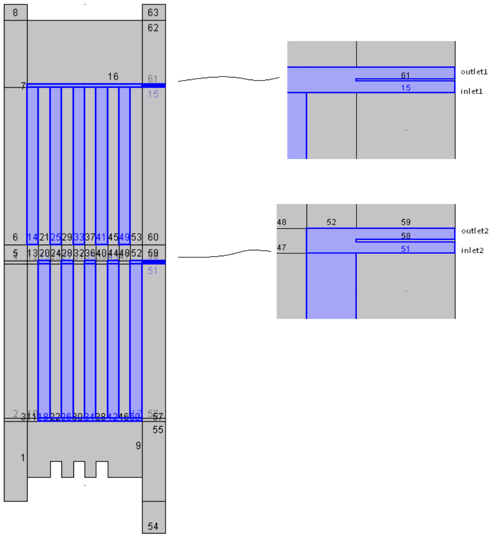

Figure 8. Geometry of finned piston with outlets and inlets.

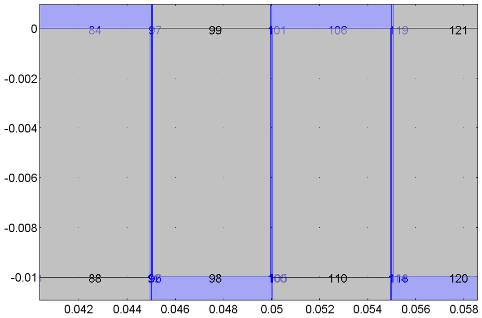

Figure 9 The air can flow Inter fin space.

5.2 Moving mesh and meshes

The principal of applying moving mesh is the same as described for classic piston. One can see the mesh distribution during compression and expansion in Figure 10.

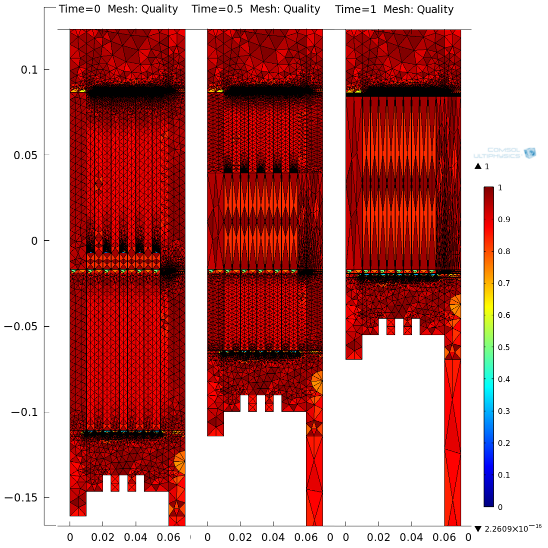

Figure 3. Mesh of the finned piston for full expansion mid-travel and full compression respectively.

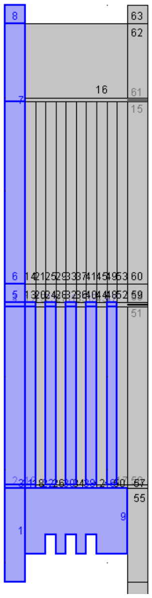

Figure 4. Translational motion for the finned piston.

Once the finned model is implemented, calculations must be operated along 80’000 up to 12’0000 meshes. The average element quality is 0.98, for a mesh area close to 0.0132.

5.3 Heat Transfer

The heat transfer occurs in each chamber of the finned piston very similar to what happens in a classic piston. However care must be taken in defining the translational motion (Figure 11) and also boundary conditions for walls (Figure 12).

The study is pretty much the same for finned piston as it was for simple piston.

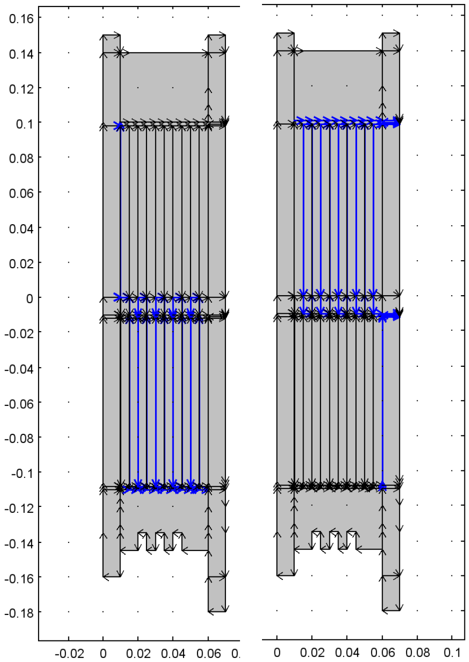

Figure 5 Boundary conditions with no slip condition for fixed wall and moving wall respectively.

6. Results

Tanks to the analytic simulation it is possible to verify the finite element results. The results are shown for one complete cycle of compressor operation.

Figure 13 shows the volume change from full expansion (TDC) to full compression (BDC).The comparison is based on the fact that both compressors should have the same dead volume and displacement volume.

The Pressure evolution is shown in Figure 14. As it can be seen, the pressure rise is less steep for finned piston since the process is closer to isothermal conditions. The analytic and COMSOL results match very well during compression but the deviation of results is a bit more during expansion.

Figure 15 shows the temperature change for both pistons. As expected temperature rise is much less for finned piston thanks to increased heat transfer area. The COMSOL model predicts less steep rise at the beginning, but steeper rise at the end of compression compared to the analytical model.

Figures 16 and 17 are related to the power and work comparison respectively. As it is obvious the work is the area under power curve.

The difference of the power (or work) between analytic and FEM result is 4.2% for classic and 1.7% for finned piston.

Figures 18 shows the mass change in one cycle. One can see that the mass remaining in the cylinder is more for the finned piston. This is no surprise because since the temperature rise is lower, the density will be higher at the end of compression, and with the same volume the mass of air remaining in the finned compressor will be higher as a result. This fact results in lower volumetric efficiency in a more isothermal compression, which is a draw back of the finned piston.

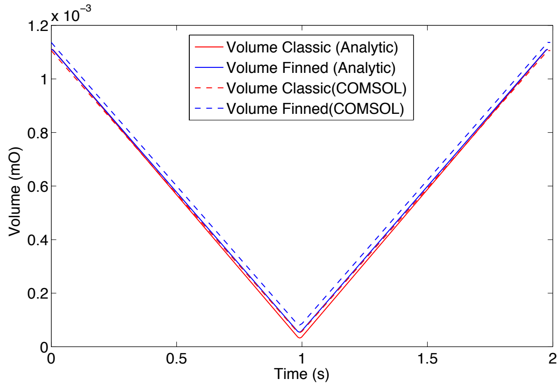

Figure 13. Volume change during in one cycle.

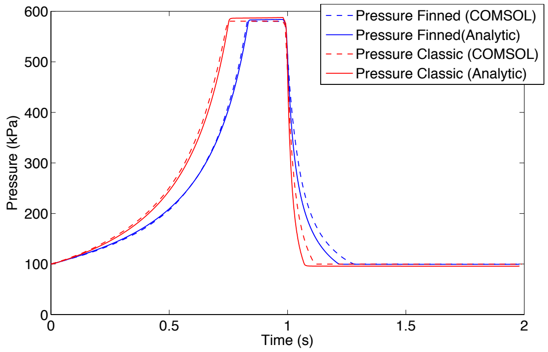

Figure 6. Pressure evolution in one cycle.

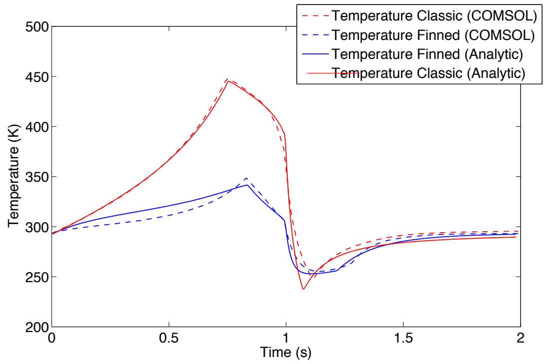

Figure 7. Temperature evolution in one cycle.

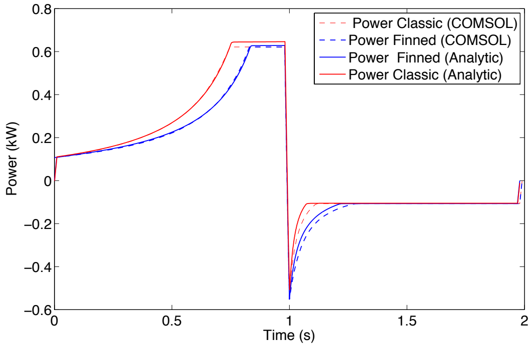

Figure 8. Power during one cycle.

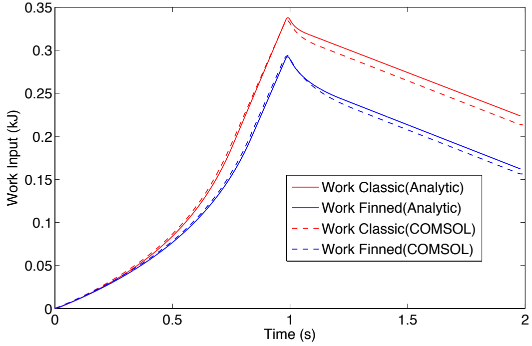

Figure 9. Work during in one cycle.

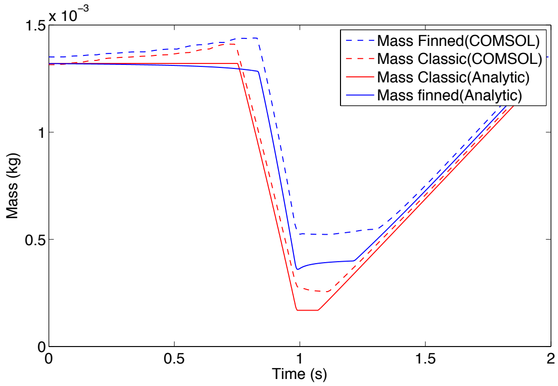

Figure 18. Mass change in the cylinder head in one cycle.

6.1 2D plots

First two-dimensional result is observed for the temperature gradient. This temperature gradient is plotted in Figure 19 for the middle of the compression process. One may notice that the maximum temperature at the same time is much less for the finned piston.

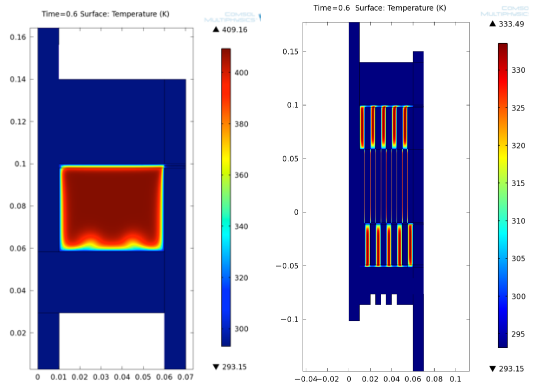

Figure 19. Temperature gradient – mid of last expansion travel

Figure 20 illustrates the velocity magnitude in color and velocity field in arrows. One can notice the flow in the very small Inter-Fins Space (IFS). This air circulation will enhance the heat transfer the more the compressor approaches the end of the compression.

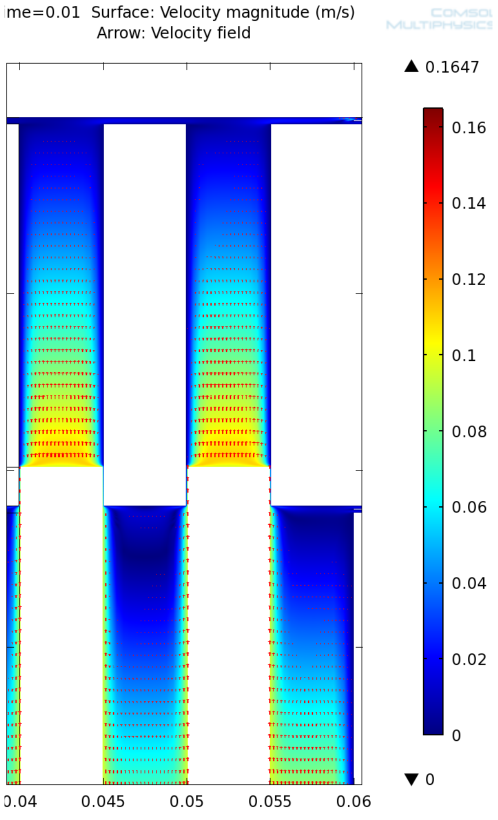

Figure 20. Velocity of air movement in the chambers and inter fin space.

8. Conclusions

COMSOL Multiphysics has proved to be a very powerful tool in unsteady state thermofluid problems with complicated geometry. The moving mesh is the key to solve such problems.

While analytic methods are mostly convenient to be used with a bulk method, COMSOL has the advantage of predicting details of parameters distribution in the system over the whole process time. Moreover the graphical representation of the parameters evolution makes the COMSOL advantage double.

However one should not forget without the without the insight that the analytical methods provide, FEM softwares can be a blackbox for the users and the user firs should have necessary knowledge about the physic of the problem.

The results of achieved using FEM method shows very good compatibility with analytic ones. The best approach is to use both analytic and FEM methods to have insight to the physics as well as accuracy and results for parameters distribution.

9. References

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