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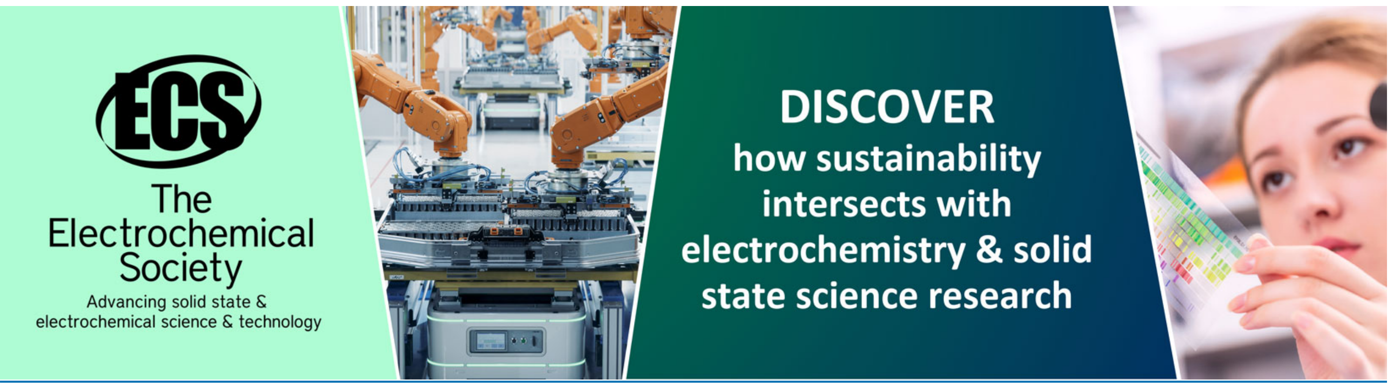

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3D Transient CFD Simulation of Scroll Compressors with the Tip Seal

Haiyang Gao, Hui Ding and Yu Jiang
Simerics, Inc.,
1750 112th AVE NE Ste A203
Bellevue, WA 98004, USA

E-mail: hg@simerics.com

Abstract. A new template simulation tool is developed for scroll compressors/expanders capable of modelling tip seal leakages. This scroll template generates a high quality 3D multi-block structured mesh from user-input stationary and orbiting scroll surfaces. The mesh movement is then automatically calculated to account for every position of the orbiting scroll, maintaining good grid quality and smooth movement throughout the whole revolution. A state-of-the-art efficient CFD solver is used to solve Navier-Stokes equations, capable of simulation with both real gas and ideal gas. A case study is presented for a generic scroll compressor with refrigerant R410A. The case was run with and without the tip seal volumes. Comparisons are made to show the impact of tip seals on the compressor performance.

1. Introduction

Scroll compressors are widely used in many industries, including refrigeration, air-conditioning and automobile (as superchargers). It is believed that scroll compressors have the advantages of high efficiency, lower noise and vibration levels. The tip seal is a common sealing mechanism placed on top of involute wraps, in order to reduce axial leakage and maximize the efficiency.
Computational Fluid Dynamics (CFD) simulations can be a useful tool in understanding the flow field and help to reduce various flow losses. However, CFD simulation of a scroll compressor has proven to be a challenging task. Because the fluid domain of a scroll compressor contains gas pockets with large deformation and complicated leakage paths of tip seal and radial gaps. There are studies to solve it via simplified mathematical models [9], Two-Dimensional (2D) simulations [8] as well as a few unsteady, 3D numerical simulations in recent years [2, 4, 6]. However, there are few numerical studies in which the tip seal leakage volumes are included.

For most commercial packages, simulation of the scroll compressor with the tip seal leakages requires lengthy set up and running time. It would also require the user to have extensive CFD knowledge and reasonable familiarity of the tools on hand.

Gao and Jiang (2014) [3] successfully developed a fully automated template tool for scroll compressor simulation, which greatly simplified the simulation process and significantly reduced the turn-around time of scroll compressor/expender simulations.

In the current work, the tool is further extended to include the leakage flow of tip seals. The template can generate high-quality structure mesh from user-input outlines of stationary and orbiting rotors. The mesh for the tip seal is also built simultaneously based on a series of input parameters. The

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mesh movement is then calculated to account for every position of the orbiting scroll, maintaining good grid quality and smooth movement through the scroll revolution.

2. Scroll Rotor Template Mesher

To ensure mesh quality throughout the movement of the orbiting rotor, especially for areas with small gaps, a continuous high quality multi-block structured mesh is generated for the scroll compressor.

As the first step the rotor CAD surfaces (Figure 1) are imported into the software and the stationary and orbiting rotors are identified. These surfaces can be easily output from any CAD package as STL files. The mesh generator is designed in such a way that the resulted mesh movements are independent of the initial relative positions of the stationary and orbiting scrolls. To define the rotating motion, some geometric parameters, such as the center of both rotors and the rotation axis vector, are also provided as inputs. The parameters that define the shape of the tip seal volumes are also input by the user, more details to follow in the next section.

Figure 1: A typical CAD surfaces input by user, stationary rotor in grey, orbiting rotor in red.

To start the meshing process, a 2D outline is extracted from the imported 3D surfaces imported. Then the template mesher will search for several key geometry points, with which the computational domain is divided into several meshing partitions.

Figure 2 shows that six partitions are created for the scroll rotor domain. Partitions 1 through 3 are the two branches of the spiral fluid volumes and the outer boundary area. These narrow, long fluid volumes are meshed by lines that are normal to either rotor outline, with evenly distributed nodes between the rotors.

The discharge area is further divided into Partitions 4-6. Partitions 4 and 5 are established in such a way that at most one “contact” point is allowed in each partition. For a continuous mesh, the stationary and orbiting scroll cannot interfere with each other, but instead relatively small gaps are maintained. The template mesher uses lines that are normal to either solid wall to mesh the area close to the “contact”, to ensure high level of orthogonality, while uniformly distributed nodes are used to draw mesh lines for the rest of the volume. Special treatments are also employed to preserve the possible sharp corners in these partitions.

In Partition 6, there could be as many as two “contact” points. For meshing purposes, the partition is further divided into two smaller ones. The dividing line is defined such that it splits the domain into two symmetric ones for symmetric rotors, and it has the smallest angle from normal direction on both rotor walls (shown as the center dash line in Figure 3). After the division, the same procedure as in Partitions 4 and 5 is applied to mesh each of the two smaller partitions.

Ecliptic smoothing is then applied to the 2D mesh to improve the orthogonality in highly distorted areas. After the 2D mesh is built, it is stretched in the third direction to form a 3D grid. The meshing process is repeated for each time step when the orbiting rotor moves to a new position.

The meshing algorithm described above has been successfully applied to various types of scroll rotors, such as various shapes at the discharge, symmetric and non-symmetric scrolls, etc.

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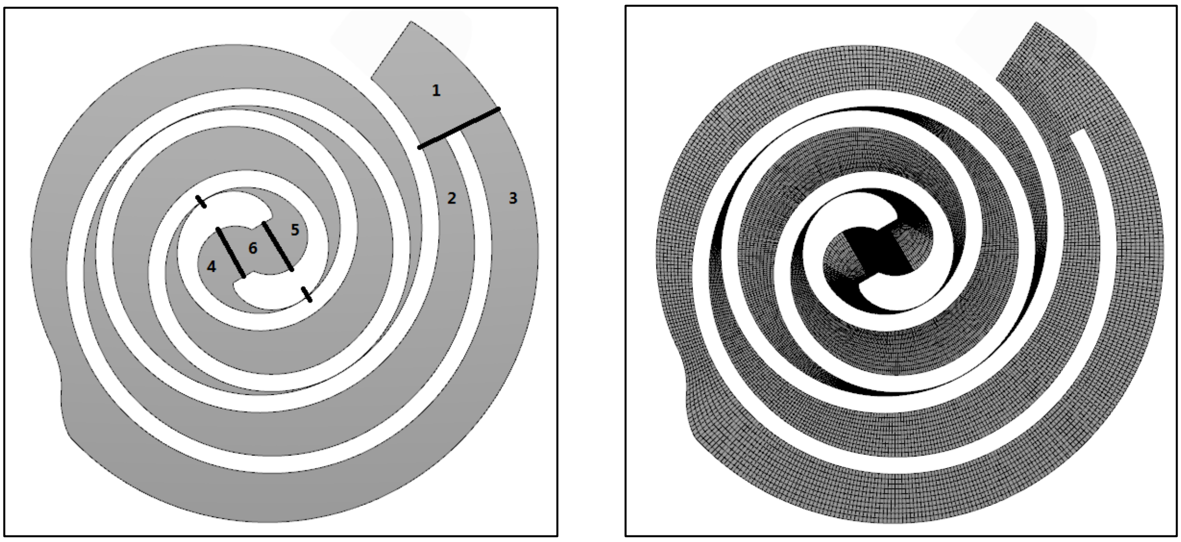


(a) (b)

Figure 2: Rotor domain partitions (a) and resulting mesh (b)

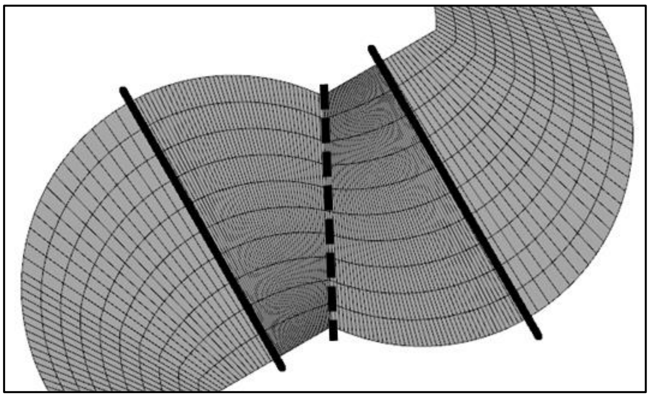


Figure 3: Further division of Partition 6.

3. Automated Meshing for the Tip Seal

A typical cross-section of the tip seal leakage path is shown in Figure 4. As demonstrated by Ancel et al. (2000) [1], the tip seal is pushed to the low pressure side of the groove. For a scroll compressor, we currently assume the tip seal is always attached to the outer side of the groove (Figure 4), considering the pressure is higher in the inner pocket most of the time. The resulting leakage path is the z-shaped fluid volume shown in Figure 5a. Once the user input all the five size parameters in Figure 4, the shape of the fluid domain is well defined. A multi-block structured mesh is then generated for the fluid volumes (Figure 5b).

From the top view of the scroll rotor in Figure 6, it can be seen that the tip seal leakage volume starts at the point where the width of the scroll becomes constant. The leakage volume ends when the spiral/involute shape stops. Both points are automatically detected by the template. Additional input is available for the user to control the starting and ending position of the leakage volume.

The scroll and the tip seal volumes generated in the last two sections are then connected via Mismatched Grid Interface (MGI) to form a single computational domain. Once the user input the necessary geometries and parameters, the whole procedure from the Sections 2 and 3 are fully automated.

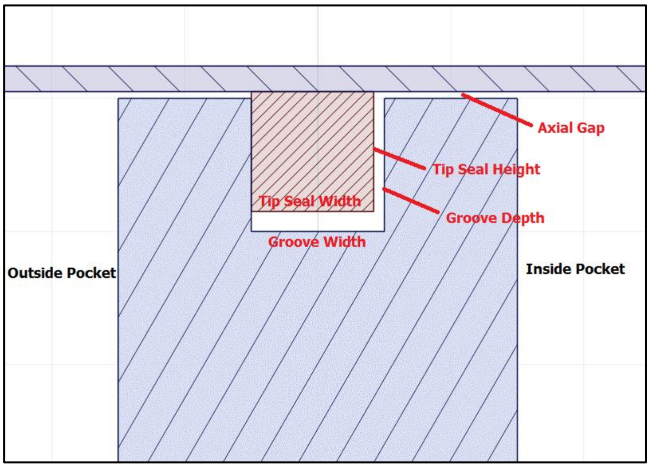
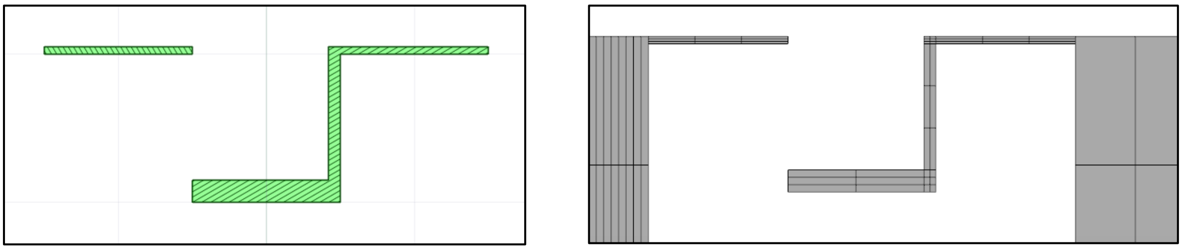
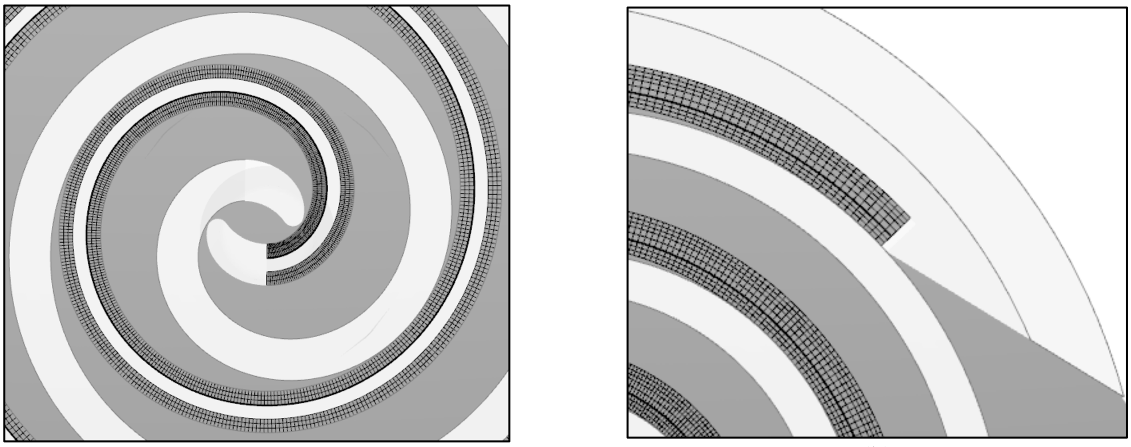


Figure 4: A typical cross section of tip seal leakage path.



(a) (b)

Figure 5: Cross sections of tip seal fluid volume (a) and the mesh generated (b).



(a) (b)

Figure 6: Top view of the compressor showing the start (a) and end (b) of tip seal volumes.

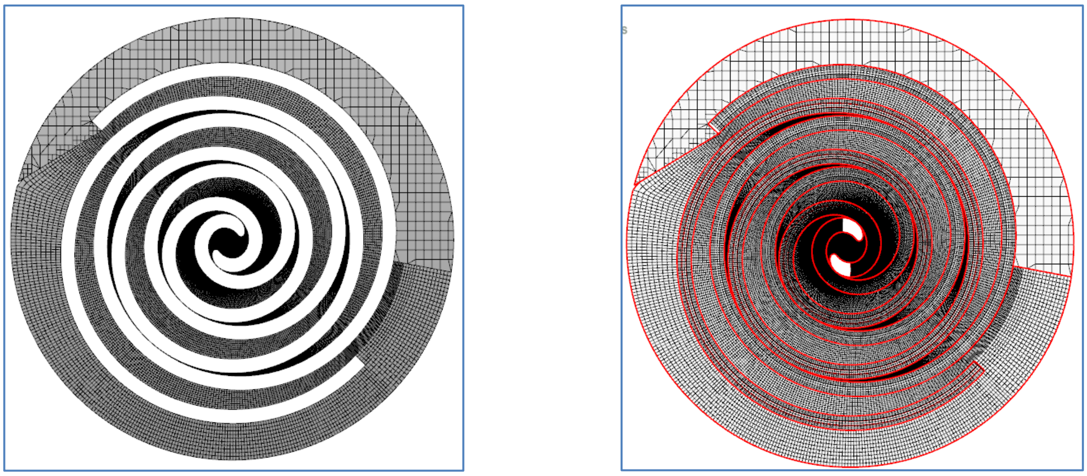
4. A Case Study

Next, the scroll template tool is put to the test for a generic scroll compressor. The compressor works with refrigerant R410A, with inlet pressure at 10 bar and the outlet at 34 bar. The inlet temperature is set at 298K. The compressor has a diameter of 90mm and height of 20mm. The gap between stationary and orbiting scrolls is set to 18 µm. For the tip seal, the groove width is set to 2 mm, groove depth 1 mm, tip seal width 1.94 mm, tip seal height 0.9 mm and axial gap 50 µm.

A commercial CFD package, PumpLinx, is used for the simulation. A proprietary pressure-based algorithm [5, 7] is employed for solving the Navier-Stokes equations. The properties of R410A are read from property tables generated by the NIST database.

Static pressure boundary conditions are applied at inlet and outlet. For the energy equation, the inlet temperature is fixed at 298K while the fully developed temperature boundary [7] is assumed at the discharge.

Two simulations at 3000RPM have been performed: one includes the tip seal leakage (Figure 7b) while the other doesn't (Figure 7a).



(a) (b)

Figure 7: Overview of computational domain: (a) without tip seal volumes (b) with tip seal volumes.

The meshes generated by the scroll template tool at four orbiting scroll positions are shown in Figure 8. It can be seen that for all positions, great mesh quality is maintained by the meshing algorithm.

Instantaneous pressure and temperature field is compared for both cases in Figure 9 and 10. From the pressure comparison, it can be observed that the tip seal volume results in substantial leakage between adjacent pockets, making an apparent difference in the pressure inside the pockets. From the temperature distribution, it is obvious the fluid inside the tip seal volume is heated up from the leakage flow, and this can lower the efficiency of the scroll compressor due to this extra energy loss.

Figure 11 and Figure 12 shows the comparison of discharge mass flow rate and temperature between two cases. Meaningful changes can be observed. Average mass flow rate is 12.1% lower when the tip seal leakage volumes are included.

The impact of the tip seal volumes on the compressor efficiency is also studied. The isentropic efficiency is defined as follows:

$$\eta = \frac{H_{2,isen} - H_1}{H_{2,act} - H_1}, \tag{1}$$

where H_1 is the enthalpy at the inlet, $H_{2,isen}$ is the enthalpy at the outlet from an isentropic process and $H_{2,act}$ is the actual enthalpy at the outlet. The leakage flow due to the tip seal volumes causes the heating of the fluid and thus lowers the efficiency. The average isentropic efficiency drops from 76.2% to 60.1% after including the tip seal volumes.

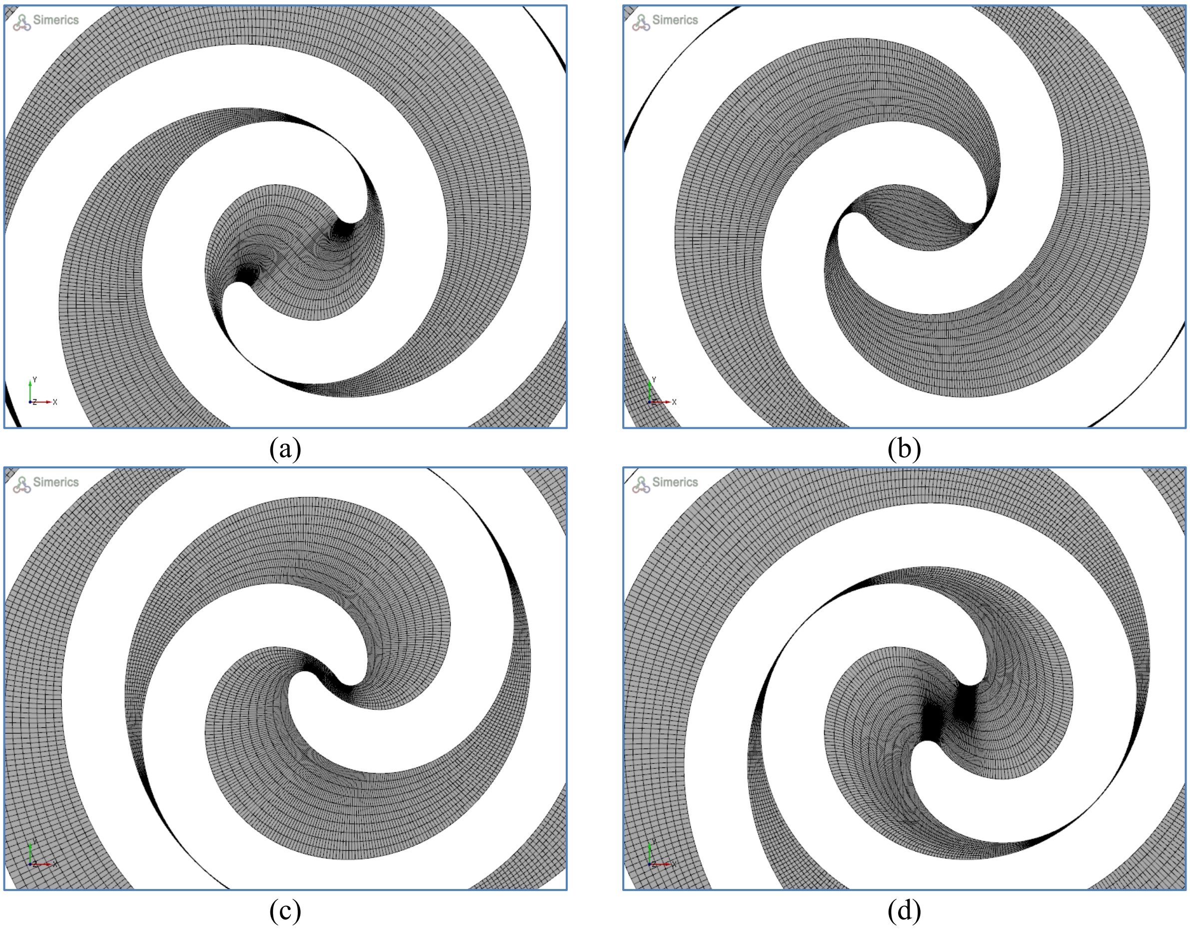


Figure 8: Smoothed mesh for scroll pocket volumes at four rotation positions: (a) 90, (b) 180, (c) 270 and (d) 360 degrees.

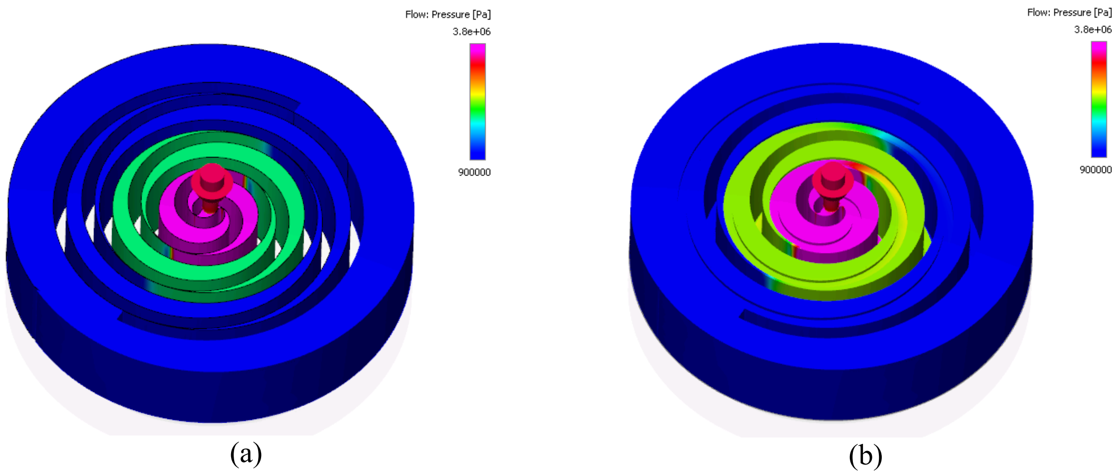
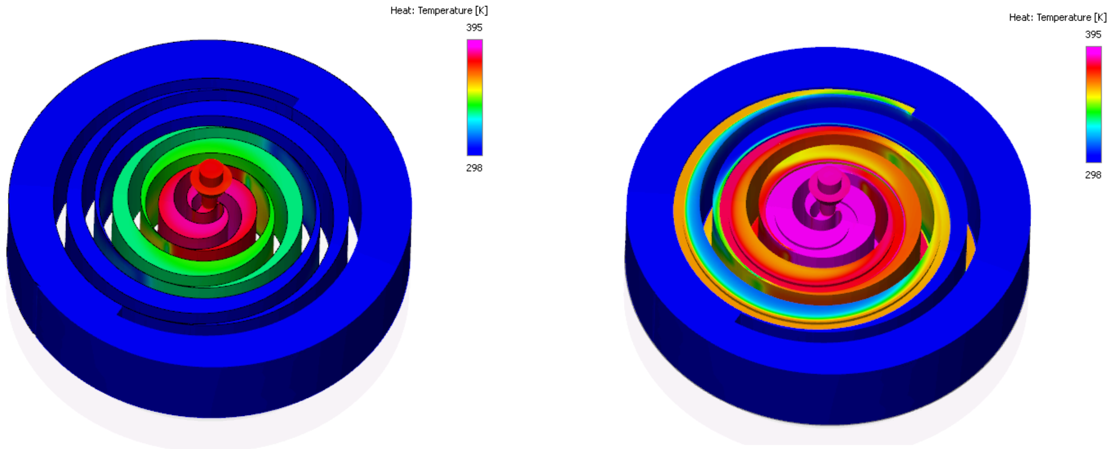


Figure 9: Instantaneous Pressure field at 180 degrees rotation position: (a) not including tip seal volumes (b) including tip seal volumes.



(a) (b)

Figure 10: Instantaneous Temperature field at 180 degrees rotation position (a) not including tip seal volumes (b) including tip seal volumes.

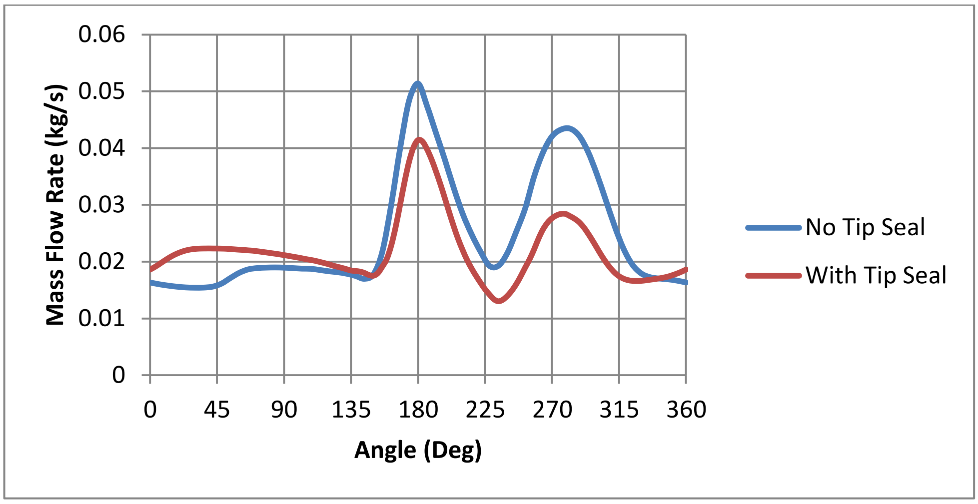


Figure 11: Mass flow rate at discharge during one scroll revolution.

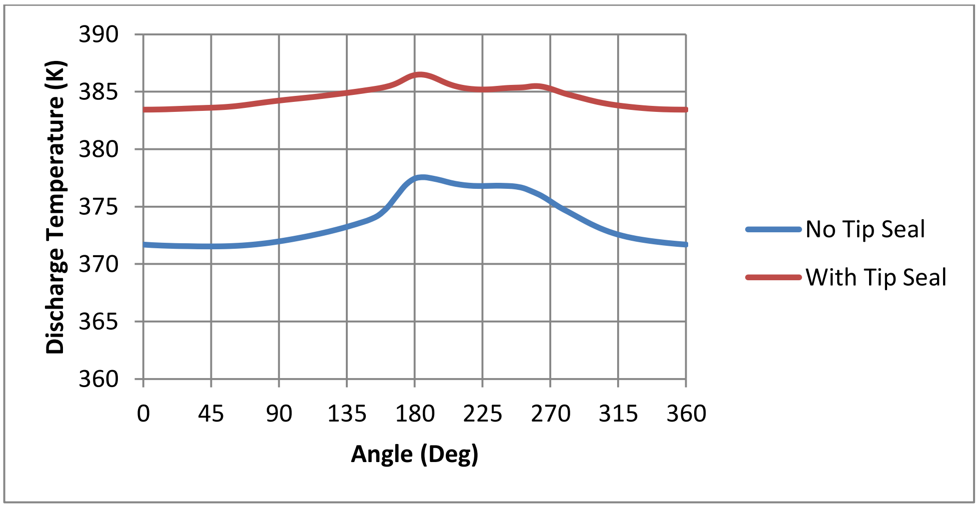


Figure 12: Temperature at discharge during one scroll revolution.

5. Conclusions

A new template tool that can streamline the simulation procedure of the scroll compressor and expander has been developed. The tool is also capable of simulating the tip seal leakage flow along with the scroll compressor. The new tool is tested on a generic scroll compressor, showing meaning impact by including the tip seal volumes.

With easy setup and a very efficient numerical solver, the unsteady flow field of scroll compressor along with the tip seal leakage path can be simulated in hours. The results of the simulation provide insightful flow information to guide the optimization and design of such systems, including pressure/velocity distribution, flow rate and temperature/efficiency of the compressor.

The ease of use and the short simulation turn-around time makes the proposed method an ideal candidate for simulations of scroll compressor systems.

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