Investigation of Pressure Pulsation and Vibration of the Internally Geared Screw Compressor

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Abstract. The Internally Geared Screw Compressor (IGSM) is a relatively novel and promising type of positive displacement compressor concept that takes inspiration from classic gerotor pumps. The IGSM uses two rotors rotating in the same direction about parallel axes while maintaining continuous contact between both rotors which isolates several working chambers. By using ported end plate, the IGSM can determine the timing of the fluid entering and exiting the working chamber. There are several key advantages to the IGSM over traditional twin-screw compressors including reduction of the rotor to casting leakage, elimination of the blow hole area, reduction in sliding velocity at the point of contact, and a uniform circumferential rotor temperature profile. Although there is some research regarding the geometric profiling and chamber modeling of this type of machine, there is no research currently available on the pressure pulsation and vibration characteristics of the IGSM. The purpose of this paper is to provide an initial look into the pressure pulsation and vibration characteristics of the IGSM and how they differ from that of a traditional twin-screw machine. By explicating the key differences between the IGSM and the industry standard twin-screw compressor, this paper aims to better understand an important yet underdeveloped area of IGSM design.

1 Introduction

The history of the screw compressor is a relatively old technology that can be dated back to 1878. Since its inception, there have been substantial improvement made to the design and operation of screw compressors including increased efficiency, lower noise and vibrations, as well as improvements to many other compressor properties. However, research into screw compressors usually adheres to either twin screw machines, or offshoots of twin screw machines. For example, Akei et al. proposed the use of a mini screw to achieve high flow so that it can use the low GWP refrigerant DR-12[1]. Little research have been conducted regarding new styles of rotary compressor that defy the conventional twin screw setup.

The Internally Geared Screw Machine (IGSM) is a novel compressor that takes inspiration from the gerotor pump. Similar to the gerotor pump, the IGSM has 2 meshing internal rotors that has continuous contact with each other to create separate working chambers. To achieve the desired compression, a stationary ported end plate is used so that the fluid can only leave once the desired volume ratio is achieved. Figure 1 shows a 3D model of the IGSM.

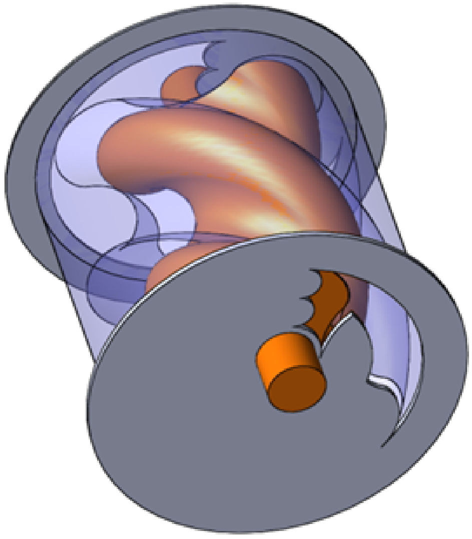


Figure 1: Model of the Internally Geared Screw Machine (IGSM)

Since this is a novel compressor that is still undergoing development, there is limited research available for this topic. The IGSM compressor design was proposed by Read et al.[2] where the author detailed the geometric analysis of the machine as well as several operational characteristics. In [3], the authors detailed a chamber modeling approach to achieve performance calculations, and in [4], the authors used CFD to investigate the performance of an oil injected model. Amongst the work currently being pursued, one area that has yet to be explored is the pressure pulsations and how that affect the vibration and structural integrity of the system.

The study of pressure pulsations and vibration for screw compressors is paramount in the successful application of such a machine to industrial applications. There are many benefits to having a compressor that have low vibration characteristics. For example, pressure pulsation causes cyclical loading which can cause fatigue stress, which in turn wears down the life of a mechanical component such as the rotors or the housing causing the need for the component to be prematurely replaced before its life cycle end. Another practical reason is that high vibrations may cause the overall structure to degrade and cause problems in the attached piping and sensor systems. The purpose of this paper is to preform an initial investigation of the pressure pulsation and vibration characteristics of the IGSM and compare these results to the traditional twin screw compressor.

2 Methodology

It is well established that the dominant source of fluid-induced vibrations for a screw machine is caused by the discharge pressure pulsations. This type of vibration analysis is a complicated process as it is in essence a fluid-solid interaction (FSI) problem. Therefore, purely mathematical models are either inaccurate or fails to provide the necessary range to perform such analysis. In order to accurately model this process, it is more advantageous to use a high fidelity numerical model that can accurately display the pressure profile. For this study, the method used to evaluate the system vibrations is based on a 1-way FSI analysis. A fully coupled FSI system is likely unnecessary as the structural vibrations will have minimal effect on the fluid. For this process, the discharge pressure pulsations are treated as forcing function acting upon the discharge port, which in turn causes structural deformations and vibrations.

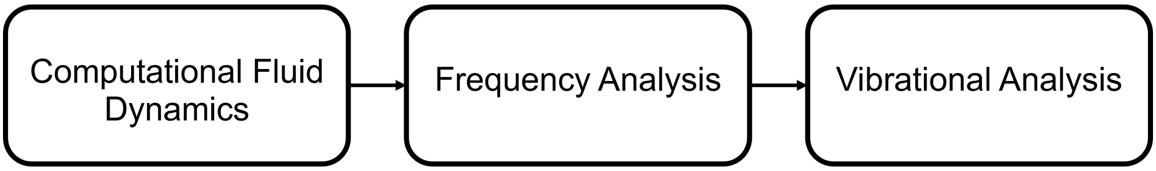


Figure 2: Procedure of vibration analysis procedure

Figure 2 shows the analysis procedure used to evaluate the surface deformation and velocity of the compressor. There are 3 main steps to this process: CFD of the compressor, frequency analysis of the pulsating discharge flow, and vibration analysis of the compressor system. Further analysis can be extended to look at the associated acoustic pressure field due to the system vibration, but that aspect is omitted from this study. The purpose of the CFD is to generate the pulsating pressure distribution onto the compressor discharge port. After obtaining the discharge pressure pulsation from the CFD, these results are then transformed to the frequency domain via spacial spectrum analysis. Frequency domain is generally what is used for vibration analysis as frequency data will provide characteristics such as resonant frequencies and peak amplitudes. Finally, the frequency domain results are then used as a forcing function to calculate the surface velocity of the compressor via FEA analysis. Note that the resultant vibrations are dependent on the shape and mass of the structure. In reality, compressors in HVAC applications are mounted on a chiller system, thus the fixed support used in the harmonic response module is not actually fixed as it will mechanically interact with the chiller components.

3 Numerical Simulation

As mentioned previously, the fundamental fluid-solid interaction that encircles pressure pulsation induced vibrations is complicated in nature and thus require full fidelity CFD simulations to fully capture the flow characteristics. The CFD simulation was performed using ANSYS CFX, and the vibration analysis was performed using ANSYS harmonic response. To simulate the rotor dynamics, SCORG was used to generate the moving mesh. The statics meshes used for the other fluid domains were generated using ANSYS meshing.

Three cases were considered for this study. For each of the 3 cases, the suction pressure was set at 1 bar, the suction temperature was set at 25 C, and the shaft speed was set at 12000 RPM. For the discharge pressure, a ramping function was used to linearly vary the pressure from 1 bar to 2 - 4 bar for the 1st rotation and kept constant at 2 - 4 bar for the remainder of the simulation.

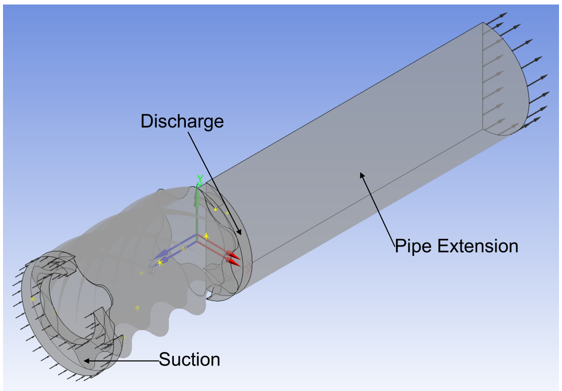
Table 1: SCORG parameters for generating rotor mesh

	IGSM	**Twin-Screw**
Circumferential Division	120	40
Radial Division	10	6
Angular Division	72	40
Interlobe Division	N/A	40

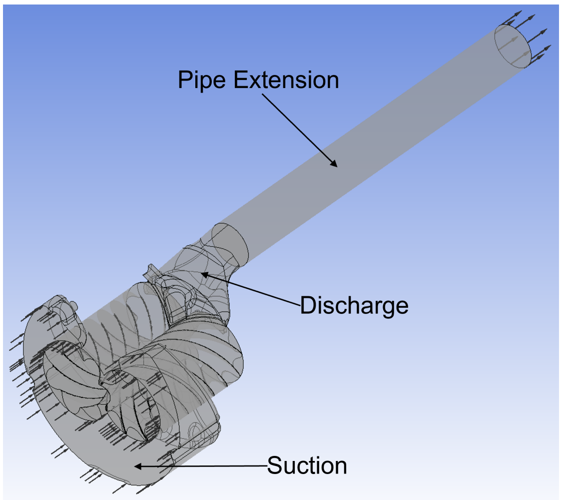
Table 2: Mesh statistics for CFD simulation

	IGSM	**Twin-Screw**
Suction Elements	30960	1478046
Rotor Elements	1677600	1643520
Discharge Elements	341214	404741
Pipe Extension Elements	1423200	113685

Table 1 shows the parameters used to generate the rotor mesh and table 2 shows the mesh statistics of each of the different fluid domains. Note that the interlobe parameter that defines the mesh between male and female rotors does not exist for the IGSM. Looking at the mesh statistics, it is evident that the suction elements of the IGSM is much lower than every other fluid domain. This is due to the fluid volume of the suction port being significantly smaller than either the rotor and pipe extension. Although the volume of the discharge port is even smaller than the suction port, the discharge port is purposely refined since that is the surface area of interest.



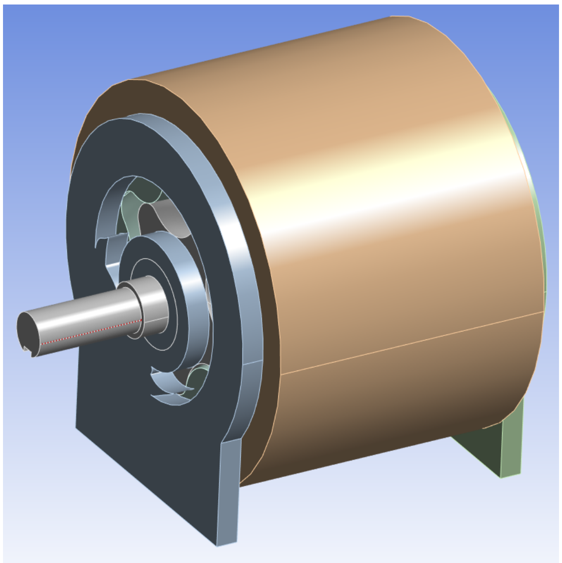
(a) IGSM Model



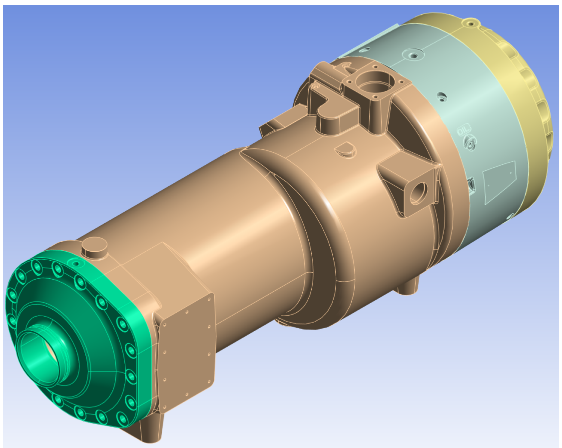
(b) Twin-Screw Model

Figure 3: Models used for CFD simulations of IGSM and Twin Screw compressors.

The fluid models that were used for the CFD simulations are shown in figure 3. Figures 3a and b shows the IGSM model and twin screw model respectively. The rotor configuration used for the IGSM is 5-6 for the inner and outer rotors respectively, and the rotor configuration for the twin screw is 7-8 for the male and female rotors respectively. The pipe extension domain is placed artificially to allow for better convergence of the fluid downstream of the discharge port.



(a) IGSM Model



(b) Twin Screw Model

Figure 4: Models used for mechanical simulations of IGSM and Twin Screw compressors.

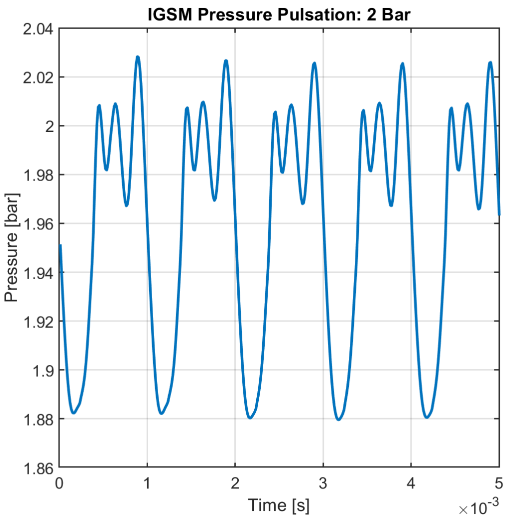
To complete the vibration analysis of both compressors, a solid model is needed for the FEA analysis. Figures 4a and b shows the solid models for the IGSM and Twin Screw respectively. In terms of scale, it should be noted that the IGSM model is significantly smaller than that of the twin screw model. This is because the IGSM is a currently ongoing project and not all the parts of the IGSM model has been fully realized whereas the twin screw model is fully designed and operational. Further analysis will be conducted in the future once the IGSM has a more complete design.

4 Results and Discussion

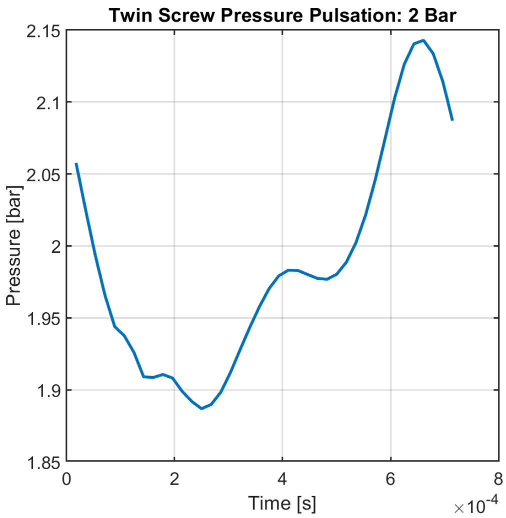
As previously discussed, CFD simulations were performed for both the IGSM and twin screw compressors at 2 - 4 bar discharge pressure. For the purposes of this study, the effects of oil was not considered, but it will be explored in a future study. Each result for the IGSM is compared against an analogous twin screw compressor to explore the specific differences between the novel IGSM and the traditional twin screw compressor. Three sets of results will be discussed in this section: section 4.1 will explore the discharge pressure pulsation that is applied to the discharge port, section 4.2 will consider the frequency response of the system, and section 4.3 will look at the surface velocity of the system.

4.1 Discharge Pressure Pulsation

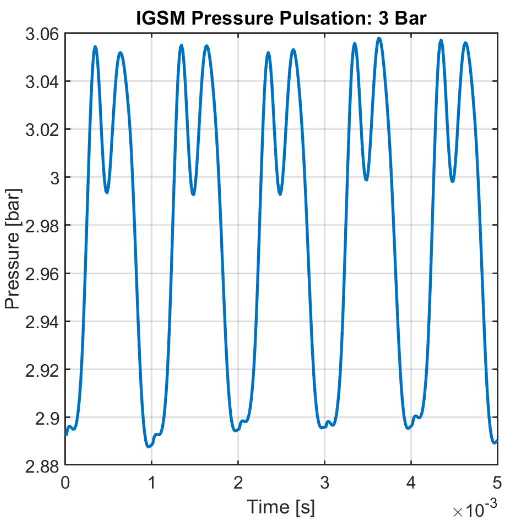
The purpose of this section is to analyze the pressure pulsations that is applied to the discharge port. Figure 5 shows the mean pressure pulsations that is acting on the discharge port for a discharge pressure of 2, 3, and 4 bar with a suction pressure of 1 bar. The left hand side of figure 5 show the IGSM pressure pulsations and the right hand side of figure 5 shows the twin screw pressure pulsations.



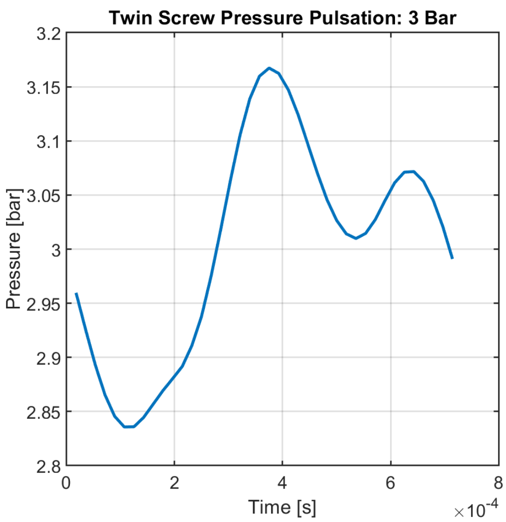
(a) IGSM: $P_{out} = 2$ bar



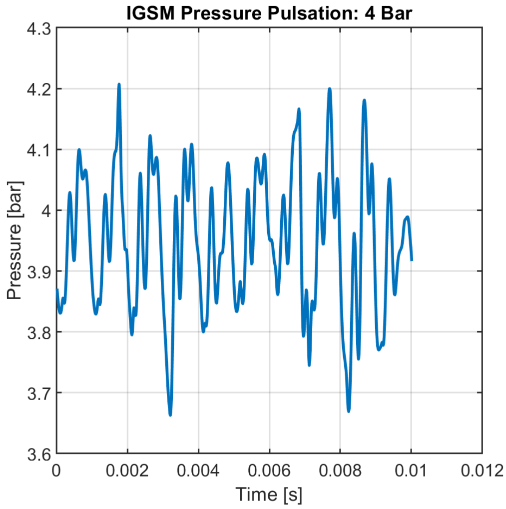
(b) Twin Screw: $P_{out} = 2$ bar



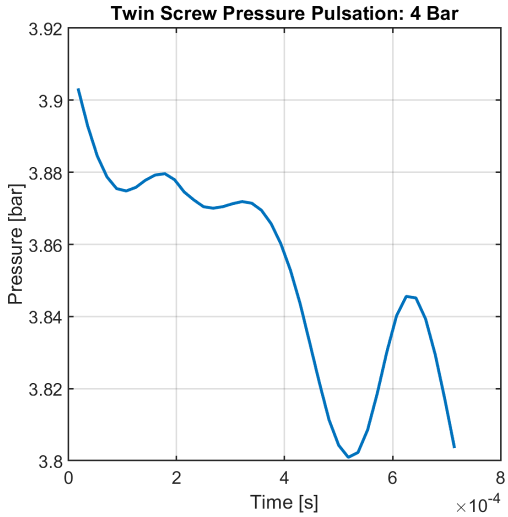
(c) IGSM: $P_{out} = 3$ bar



(d) Twin Screw: $P_{out} = 3$ bar



(e) IGSM: P_{out} = 4 bar



(f) Twin Screw: P_{out} = 4 bar

Figure 5: Discharge pressure pulsations for IGSM (left) and Twin Screw (right) at 2, 3 and 4 bar outlet pressure

One key property of the IGSM pressure pulsations as compared to the twin screw compressor is the existence of the micro oscillations that occurs at each cycle. These micro oscillations occur for both that 2 bar and 3 bar case, but does not exist for the 4 bar case. The reason that the oscillations no longer occur is due to the additional noise that is present in the 4 bar case. Since no oil is considered in this study, the sealing effect that oil provides also no longer exist. At a pressure ratio of 4, the inter-lobe leakage becomes too much to gather acceptable results. This effect can be seen in both the IGSM and twin screw cases, as although there is a clear oscillating presence, a distinct pattern no longer exist. This behaviour is especially present for the 2 bar case where 3 of these micro oscillations occur. This behaviour also seems to be damped down as the pressure ratio increases as the number of micro oscillations decreased at an outlet pressure of 3 bar as compared to the 2 bar case.

Table 3: RMS pressure of the pressure pulsations for the IGSM and twin screw compressors

Discharge Pressure	**IGSM**	**Twin-Screw**
2 bar	1.9603 bar	1.9891 bar
3 bar	2.9769 bar	3.0039 bar
4 bar	3.9502 bar	3.8530 bar

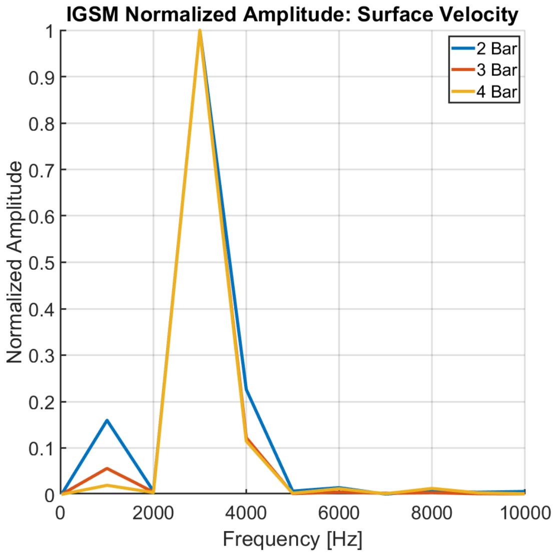
Table 4: Peak-to-peak pressure of the pressure pulsations for the IGSM and twin screw compressors

Discharge Pressure	**IGSM**	**Twin-Screw**
2 bar	0.1487 bar	0.2560 bar
3 bar	0.1701 bar	0.3286 bar
4 bar	0.5446 bar	0.1032 bar

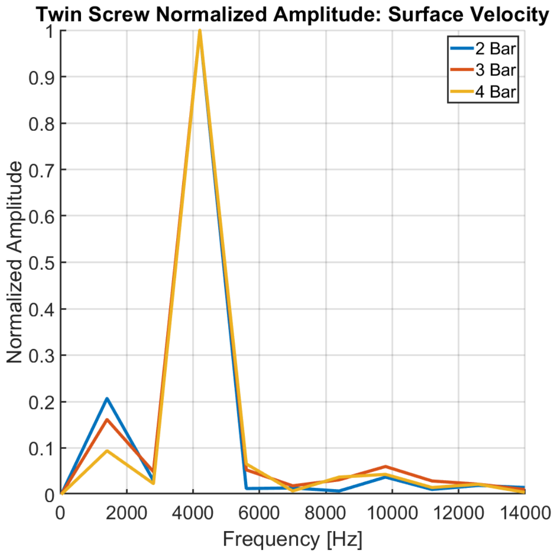
Table 3 shows the RMS pressures for each of the 3 cases for both the IGSM and twin screw cases. From this result, it is evident that the RMS pressure of the traditional twin-screw is closer to the set discharge pressure as compared to the IGSM. This is because the cyclical over and under compression is equally prevalent in the twin screw, i.e, there is just as much over compression as there is under compression, whereas the IGSM exhibit much more under compression as it does over compression. Therefore, the RMS value for the IGSM is skewed towards a lower value. Another difference between the pressure pulsations for the IGSM and the twin screw is the peak-to-peak pressure value. Although the pressure pulsations for the IGSM skews towards under compression, the peak-to-peak pressure decreases significantly as compared to the twin screw compressor (as shown in table 4).

4.2 Frequency Response

The purpose of this section is to look at the frequency response of the IGSM and twin screw in response to the surface velocity of the structure. The main purpose of frequency response is to regard the behaviour of the structure while it is under harmonic loading, which allows one to ascertain the dominant frequency of the system.



(a) IGSM frequency response due to surface velocity



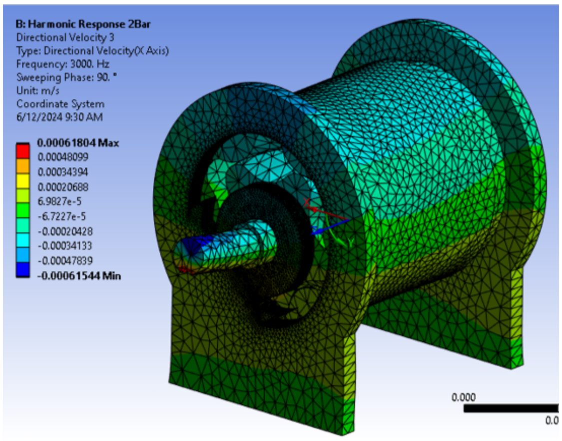
(b) Twin Screw frequency response due to surface velocity

Figure 6: Frequency response due to surface velocity for IGSM and Twin Screw at the first 10 harmonics

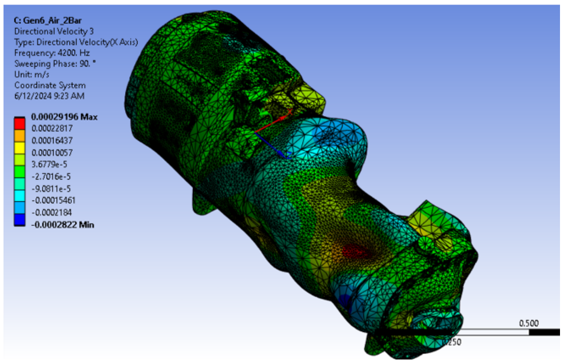
Figure 6 shows the frequency response of the first 10 harmonics for both the IGSM and twin screw compressor due to surface velocity. Since both compressors are running at 200 Hz and the male lobe count for the IGSM and twin screw compressor are 5 and 7 respectively, the fundamental frequency is 1000 Hz and 1400 Hz respectively. Looking at the result of figure 6, it can be seen that the dominant frequency reside at the 3rd harmonic and the 2nd dominant frequency is the 1st harmonic for both the IGSM and twin screw compressors. It should also be noted that there is very little influence at the 2nd harmonic as well as every harmonic past the 3rd harmonic. As for the 1st and 4th harmonic, both systems show that the same trend that increasing the pressure ratio decreases the influence of the 1st harmonic. In regards to the harmonic response of the system, it can be said that the IGSM behaves very similar to the twin screw compressor.

4.3 Surface Velocity

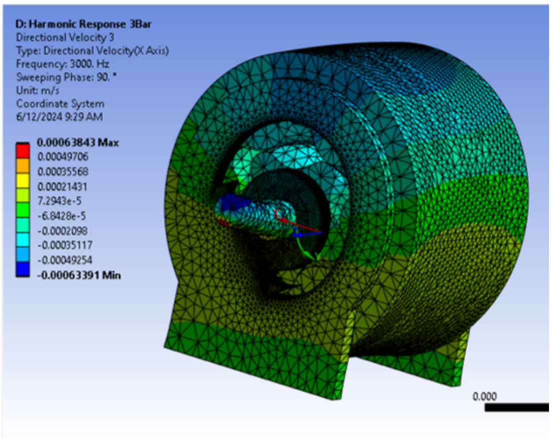
The purpose of this section is to examine the surface velocity of both the IGSM and twin screw compressor in response to the pulsating discharge pressure. Normally, one would extend this analysis to the connecting pipe as well to examine weakness in the piping design, but since a fully actualized port design is not yet available for the IGSM, this analysis is omitted for this study. It should be mentioned that the surface velocity of a structure is heavily dependent on both the discharge port design and the structure of the compressor. Since the IGSM here is still in the initial phase of design while the twin screw is a fully designed compressor, it is therefore difficult to make a direct comparison between the two compressors. However, what can be examined is to show that the magnitude of the surface velocity is near to that of the twin screw so that there is confidence a full design will be similar enough to the twin screw and no key design features need to be implemented.



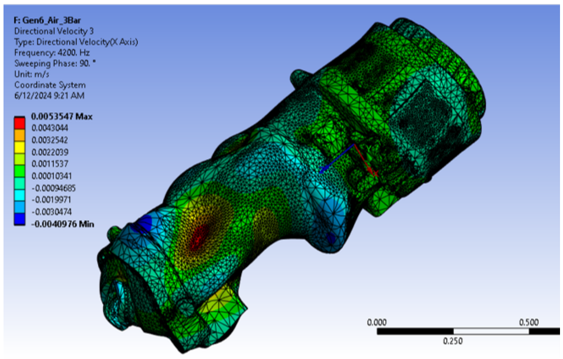
(a) IGSM: $P_{out} = 2$ bar



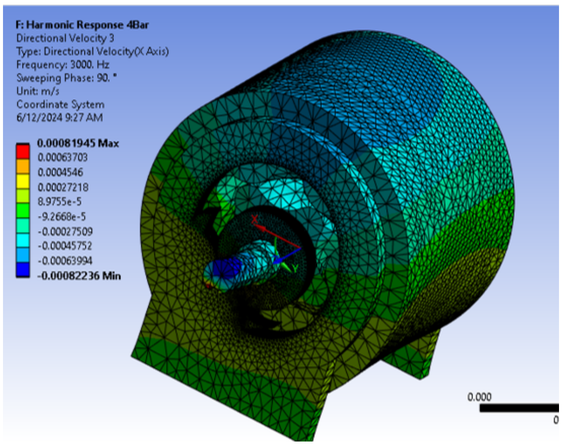
(b) Twin Screw: $P_{out} = 2$ bar



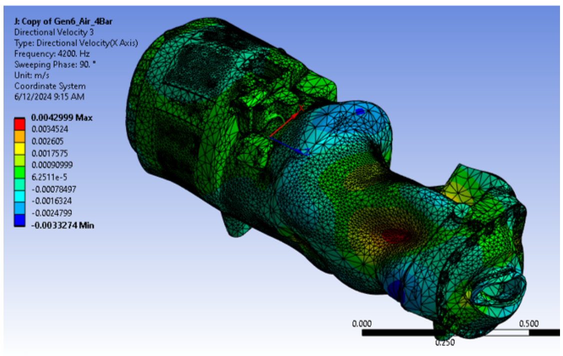
(c) IGSM: $P_{out} = 3$ bar



(d) Twin Screw: $P_{out} = 3$ bar



(e) IGSM: $P_{out} = 4$ bar



(f) Twin Screw: $P_{out} = 4$ bar

Figure 7: Surface velocity for IGSM and twin screw compressor at 2, 3 and 4 bar outlet pressure.

Figure 7 show the directional surface velocity of the IGSM and twin screw at the 3rd harmonic. As previously shown, the 3rd harmonic is the dominant frequency and the direction of velocity is shown is radial as that direction is best indicator of surface vibration of the system. One interesting feature about the IGSM is that there seems to be a region of low surface velocity around the top of the compressor that travels down in regions increasing in surface velocity as if reaches the fixed support at the bottom. This effect may be due to the symmetrical nature of the under designed structure as this does not occur in the twin screw machine.

Table 5: IGSM mean surface velocity for 2, 3, and 4 outlet pressure at the first 5 harmonics (unit: m/s).

	F_1	F_2	F_3	F_4	F_5
$P_{out} = 2$ bar	1.26E-06	3.04E-07	2.51E-05	1.40E-05	3.56E-07
$P_{out} = 3$ bar	1.04E-06	1.66E-07	4.15E-05	2.27E-06	3.84E-08
$P_{out} = 4$ bar	3.05E-06	5.11E-08	5.40E-05	1.36E-05	2.35E-07

Table 6: Twin Screw mean surface velocity for 2, 3, and 4 outlet pressure at the first 5 harmonics (unit: m/s).

	F_1	F_2	F_3	F_4	F_5
$P_{out} = 2$ bar	1.68E-05	2.86E-06	6.11E-06	4.90E-06	5.76E-07
$P_{out} = 3$ bar	2.47E-05	2.68E-06	1.47E-04	1.05E-06	1.45E-06
$P_{out} = 4$ bar	1.16E-05	6.30E-07	1.14E-04	8.41E-06	1.84E-07

Table 5 and 6 shows the mean surface velocity at the first 5 harmonics for the IGSM and twin screw compressor respectively. Recall that it was shown the largest amplitude shown in figure 6 occurs at the 3rd harmonic for both the IGSM and twin screw compressor and that result is confirmed here as the largest velocity magnitudes fall at that harmonic. Another characteristics that can be seen is that the magnitude of the surface velocity is shown to be in the neighborhood of the twin screw across discharge pressures. Although not a perfect trend, it can also be seen that on average that the surface velocity is lower for the IGSM. This result is to be expected as a fully designed discharge port will have more area affected by the pulsating flow thus resulting the more force applied to the structure.

5 Future Work

A study of the pressure pulsation and vibration characteristics were conducted for the novel IGSM. There are several key takeaways regarding this study including the existence of the micro oscillations that occur for the IGSM but not for the twin screw. There are also several similarities in that both machines is dominated by the 3rd harmonic of the system and have similar surface velocity values. As mentioned previously, a full port and case design has not been actualized yet, so one aspect of the future work is to reconfirm the results for a fully designed IGSM. Other future work include better understanding the micro oscillations that occur in the IGSM pressure pulsations and the effects that oil injection will play in the vibrations of the system.

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