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Jerk-limited Profiles for a Grooved Cam to Operate a Stirling Engine Displacer Piston

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Abstract— This paper discusses the importance of groove cam and follower mechanisms in emerging applications like driving the displacer piston of a Stirling engine used to recover wasted heat and transform it into electricity. While conventional cam profiles aim to minimize acceleration and jerk for cycloid and harmonic motion, these designs fall short of providing a continuous jerk in the transition from rise, dwell, and return. This paper utilizes B-spline parametric curves to achieve cam profiles with continuous jerk. Continuous jerk is beneficial in this application because it limits the vibrations, sliding between cam and follower, and wear between the groove surfaces when the follower jumps throughout the cycle. Cam profile optimization is further investigated by altering the displacer piston stroke length and follower pressure angle and observing efficiency fluctuations. Manufacturing considerations including dynamic balancing, temperature, and material wear are discussed as these can affect the engine's performance.

Keywords: Groove cam and follower, Stirling engine, Jerk profile

I. NOMENCLATURE

h	Base Circle displacement (mm)
β	Interval of Rotation (deg)
θ	Angle (deg)
L	Lift (mm)
F	Force of interference (N)
ω	Radial speed (RPM)
R	Distance from cam center to follower center (mm)
X_C	Cam Profile X-axis Coordinate (mm)
Y_C	Cam Profile Y-axis Coordinate (mm)
γ_C	Cam profile angle
r_b	Base circle radius
r_C	The radius of the base circle and follower (mm)
r_r	Follower radius (mm)
S	Offset (mm)
φ	Pressure Angle (deg)
P_i	B-spline control points
$N_{i,p}(u)$	Bernstein basis polynomials

II. INTRODUCTION

Cam and follower mechanisms are a robust way to convert rotational to translational motion [1], and they are continually being applied to new applications with unique demands. In cam design, there are three main types of cams: disk, cylindrical, and groove cams. Disk cams are the most prominently used type of cams, often found in internal combustion engine camshafts to operate the opening and closing of a valve [2]. Cylindrical cams control the follower within the groove of a cylinder, where the follower contacts the profile perpendicular to the surface of the cylinder [3]. Groove cams fully control the follower to trace the cam profile parallelly and are relatively unexplored but essential for situations requiring precise and irregular motion control, such as controlling the motion of pistons in external heat engines, such as Stirling engines.

Stirling engines have the highest theoretical efficiency for the conversion of heat to power and they utilize external heat sources, thus offering a way to produce power from a wide range of sustainable options. Natural heat sources or waste heat from industrial processes can be used to run these engines, making other production systems greener while reducing greenhouse gas (GHG) emissions. The application of groove cams in Stirling engines is of particular interest since these engines require precise displacer control to operate close to the actual Stirling Cycle, which can be facilitated with groove cams because they provide complete control of the motion throughout the cycle. A previous study of Stirling engines found that the efficiency was reduced by as much as 30% from the maximum attainable Carnot efficiency when sinusoidal motion was utilized [4]. A groove cam profile offers control of the displacer stroke length and timing, with complete cycle control, thus enabling a Stirling engine to closely follow the Stirling cycle and improve the engine performance.

In the context of groove cams design, Nguyen and Sevcik [5] proposed an analytical method to design a groove cam with a ball follower, focusing on geometric constraints and pressure angle. In another study, Xuan et al. [6] optimized grooved cam profiles using non-uniform rational B-spline (NURBS) curves and employing an improved optimization algorithm to reduce fatigue wear and improve performance. In 2022, Hsu et al. [7] presented a novel grooved cam mechanism using three roller linked together to distribute force on the contact surface. In their design, the three connected roller follower is in simultaneous contact with the cam leading to less contact stress. However, none of the preceding works focused on designing a cam mechanism to move the displacer of an engine and dictating the motion based on custom profiles.

In this paper, various motion profiles for cams including uniform velocity, simple harmonic, cycloidal, polynomial, and B-spline are investigated to ensure that the requirements are met for controlling displacer motion in Stirling engines. This study is particularly focused on investigating the jerk behavior, which is a critical issue for use in the engine, since high-speed and long-term applications require jerk continuity without abrupt transitions and vibration.

III. Working principles of Stirling engines

A Stirling engine produces work using the mechanical motion created by the expansion and compression of air under different temperatures. See Fig. 1 for the components on the Stirling Engine. A temperature gradient is required to enable the thermal energy to be converted into work. This temperature gradient is created by placing a heat source and an opposing cold source on the chamber exterior, and the motion of the displacer is used to move the working fluid into either the cold or hot chambers. A power piston is used to expand the working fluid when it is hot and compress it when it is cold, to generate a net work output and rotate the shaft, which also rotates the cam groove resulting in the displacer motion. To optimize the efficiency of the Stirling engine, the displacer piston and cam follower mechanism are key components that directly affect the thermodynamic principles of the Stirling cycle by controlling the displacer motion. Therefore, the integration of a groove cam, which would provide complete control of the displacer stroke with the follower is ideal. The geometrical profile details for this particular cam type are presented in the next section.

IV. Geometric Profile of a Groove CAM

Positive motion cam mechanisms are a specific category of cams where the motion of the follower is completely controlled by the cam profile for the entirety of a cycle [8]. Groove cams, a subcategory of positive motion cams, do not require an external force, including gravity, to ensure contact between the follower and the cam is maintained. Fig. 1 shows how a groove cam can be integrated into a Stirling engine system. In this application, constant contact is necessary along the profile to prevent catching or sliding, which happens when the follower jumps between the inner and outer edges of the grooved profile. Catching or sliding hinders the kinematic motion of the piston stroke and interferes with the proper operation of the Stirling engine.

Designing a cam capable of precisely generating the desired motion requires a thorough knowledge of cam geometric parameters, shown in Fig. 2. There are five main elements of geometric profiles in a groove cam: cam profile, base circle, pitch curve, trace point, and lift. The cam profile is the surface of the cam where the follower travels, in this application it is the groove of the cam. The pitch curve is the average path completed by the follower's center or trace point. The base circle has the smallest circumference tangent to the cam profile, aligned with the center of the camshaft [8]. The lift is the farthest distance the follower will travel from the base circle, which dictates the length of the displacer piston stroke. One of the key considerations when designing cam profiles is the pressure angle. The pressure angle is the angle between normal to the motion of the trace point and the pitch [8]. A higher-pressure angle will increase the applied force along the horizontal axis of the follower and increase friction and wear on the mechanism. To reduce the pressure angle, the diameter of the base circle or roller can be increased and the motion or offset of the follower can be adjusted.

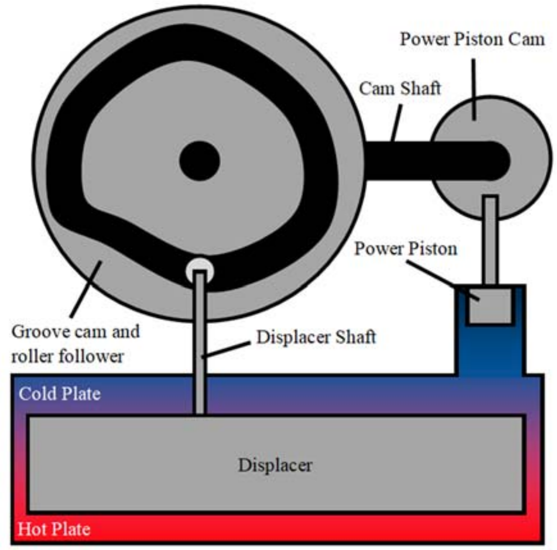

Fig. 1: Diagram of the main components in a Stirling engine.

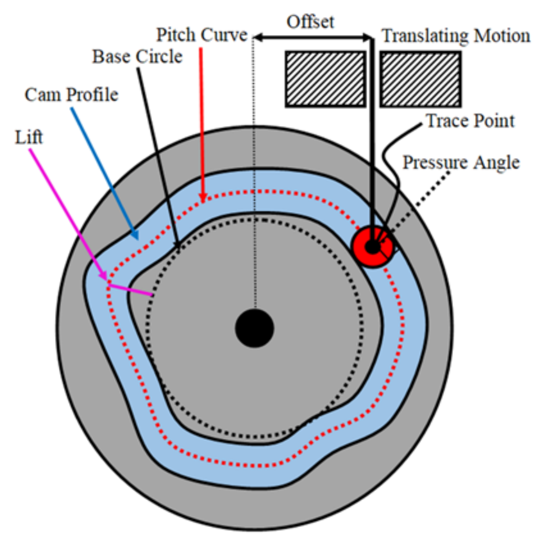

Fig. 2: Groove cam and roller follower labelled diagram.

Proper selection of the type of follower also plays an important role in successful operation of a cam and follower mechanism, and this is also crucial for Stirling engine applications. There are many types of followers including: knife edge, flat faced, spherical-faced and roller followers. The roller follower was selected for this study due to the benefits of reduced friction, high-speed capability, reduced noise, and complex profile design compatibility. The roller follower allows for a smooth transition between inner and outer edges while adding compatibility with the smoother machined surfaces. The contact surface between the cam and follower can further be reduced with a crown. A crowned roller follower has a beveled surface so only one point is in contact with the profile instead of the entire exterior face.

V. Cam and Follower Motion

Cam profiles are typically designed for three phases: rise, dwell, and return to match the motion of follower as it moves to and from the cam center. The transition between each phase should be as smooth as possible to prevent infinite jerk, when the follower suddenly jumps along the profile, disrupting the path, and causing potential failures. This is especially important for the control of displacer piston motion to reduce wear over long term continuous operation and enhance the performance of the engines. This section discusses how different types of motion profiles will affect the cam follower mechanism performance.

A. Jerk Profiles

A jerk profile can be created by projecting the slope of the acceleration kinematic profile to determine when jumps or vibrations might occur [9]. In cases where the acceleration profile is not continuous, jerk tends to have an infinite value which induces unwanted vibrations through the motion. Vibration induces wear, fatigue, generates noise, and can impact the overall pathing of the cam profile with repeated occurrence. Reducing vibration during the operation of the Stirling engine is essential because the motion directly affects the displacer piston's stroke. Any offset will correspondingly change the duration of each stage in the Stirling cycle and reduce the overall efficiency of the kinetic motion. In order to avoid infinite jerk, the transition from each phase of follower motion, dictated by the cam profile, should be smoother meaning that the acceleration profile must be gradually increased. The value of this gradual increase is defined by jerk values. Figure. 3 provides a comparison between a jerk limited and acceleration limited motion profile.

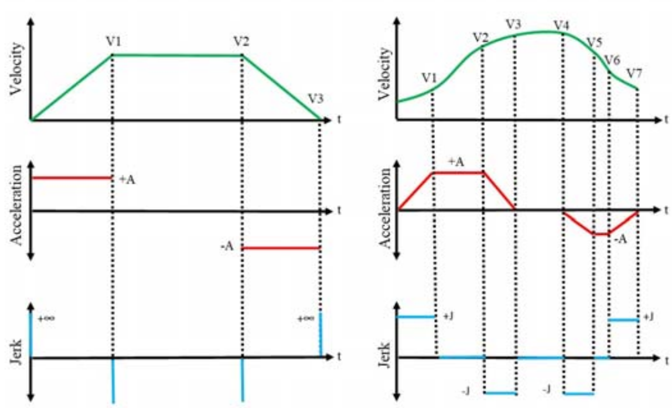

Fig. 3: Comparison of acceleration limited and jerk limited motion.

B. General Types of Follower Motion

Among the conventional types of follower motions, uniform velocity motion has smooth operation due to constant speed, timing, and simplistic synchronization. Although this provides predictable motion, it requires precise control over the profile as the system has infinite jerk and is more prone to vibrations or sliding from rapid velocity changes. Another common cam profile is simple harmonic motion in which the rising pattern presents a sine or cosine pattern and a mirrored acceleration and deceleration for the return phase [10]. Simple harmonic motion reduces the induced wear between the cam and follower in comparison to uniform motion because of its smoother pathing. When imposing certain constraints on the acceleration or jerk profile is desired, more complex profiles need to be devised. Such complex profiles cannot be achieved by uniform and harmonic motion. Adjusted or complex profiles require custom cam designs and consequently become more expensive due to their manufacturability difficulties. Although uniform velocity and simple harmonic motion's profile could facilitate the process of cam design for application, when dealing with custom motion equations in high speed, such as the requirements for displacer motion control in Stirling engines, they cannot provide a desirable response. When utilizing cams in high speed applications, the designed profile should at least provide a limited jerk profile to minimize wear or jamming on the cam profile. Cycloidal and 3-4-5 polynomial profiles, are two of the common profiles used for their ability to limit the jerk value along the motion. The following sections will discuss these profiles and compare them for the integration in Stirling engines.

C. Cycloidal Motion

Cycloidal motion is one of the most widely used profiles which provides a finite jerk value at the transition points between each phase of motion [11]. The mathematical expressions of kinematic profiles for cycloid motion are presented in Eqs. (1-4). In these equations, h is the amount of displacement in one motion, θ is the angular position of the follower during a movement, and β is the angular span in which the motion occurs. Fig. 4 provides the displacement, velocity acceleration, and jerk profile of a cycloid motion for a rise in a 90-degree span angle.

Displacement $$y = h\left(\frac{\theta}{\beta} - \frac{1}{2\pi}\sin\frac{2\pi\theta}{\beta}\right) \tag{1}$$

Velocity $$y' = \frac{h}{\beta}\left(1 - \cos\frac{2\pi\theta}{\beta}\right) \tag{2}$$

Acceleration $$y'' = \frac{2h\pi}{\beta^2}\sin\frac{2\pi\theta}{\beta} \tag{3}$$

Jerk $$y''' = \frac{4h\pi^2}{\beta^3}\cos\frac{2\pi\theta}{\beta} \tag{4}$$

Based on Fig. 4, the plot shows that the jerk has a finite value at the beginning and end of the motion. This limited value is one of the main advantages of cycloidal in comparison to the constant velocity and simple harmonic profiles. Although cycloidal motion provides finite jerk, when transitioning between phases, the value of the jerk is not continuous which leads to a sudden change on the jerk profile.

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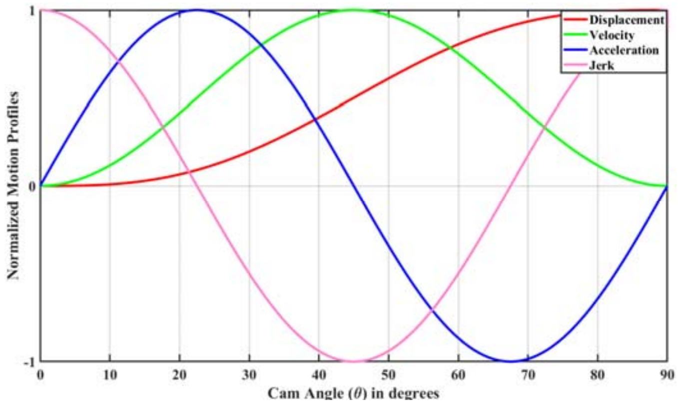

Fig. 4: Cycloidal motion: displacement, velocity, acceleration, and jerk profiles.

D. 3-4-5 Polynomial Motion

A cam profile with 3-4-5 polynomial motion also offers finite jerk. In general, the motion created by this profile follows a similar trend to the cycloid one with one main difference. In the case of 3-4-5 polynomial, start and end phase of each motion has a more gradual increase and decrease which provides a smoother motion along the path [11]. The smoother behavior of the 3-4-5 polynomial at transition points leads to smoother motion resulting in lower wear and damage. Implementing the 3-4-5 polynomials could be done by following Eqs. 5 - 8 where h, θ, and β are defined similar to the cycloid motion. Kinematic profiles of a 3-4-5 spline are presented in Fig. 5.

Displacement $$y = 10h\left(\frac{\theta}{\beta}\right)^3 - 15h\left(\frac{\theta}{\beta}\right)^4 + 6h\left(\frac{\theta}{\beta}\right)^5 \quad (5)$$

Velocity $$y' = \frac{30h}{\beta}\left[\left(\frac{\theta}{\beta}\right)^2 - 2\left(\frac{\theta}{\beta}\right)^3 + \left(\frac{\theta}{\beta}\right)^4\right] \quad (6)$$

Acceleration $$y'' = \frac{60h}{\beta^2}\left[\left(\frac{\theta}{\beta}\right) - 3\left(\frac{\theta}{\beta}\right)^2 + 2\left(\frac{\theta}{\beta}\right)^3\right] \quad (7)$$

Jerk $$y''' = \frac{60h}{\beta^3}\left[1 - 6\left(\frac{\theta}{\beta}\right) + 6\left(\frac{\theta}{\beta}\right)^2\right] \quad (8)$$

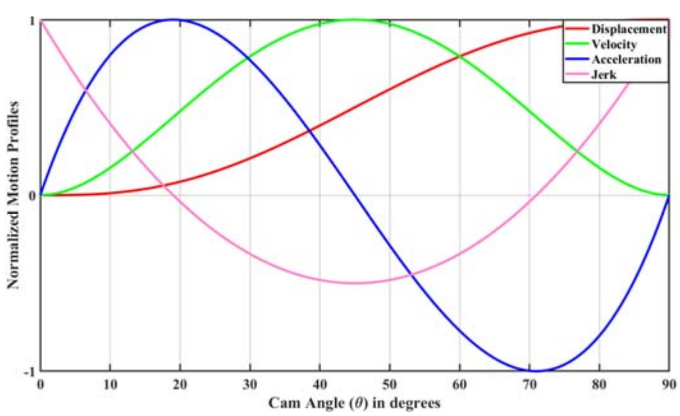

Fig. 5: 3-4-5 Polynomial motion: displacement, velocity, acceleration, and jerk profiles.

For both cycloidal and 3-4-5 polynomial, acceleration at the junction points between phase transitions (e.g., rise to dwell or dwell to fall) exhibit only C^0 continuity. This means that only the values of the acceleration are equivalent, and their respective derivatives are not equal. The C^0 continuity results in a sudden change in the derivative of acceleration (jerk) at the transition points. Although the jerk remains finite, it still undergoes an abrupt change at transition points. Conventional cam profiles do not inherently control kinematic limits, causing higher acceleration and jerk with increased lift. The sudden change in higher jerk can result in wear and damage to the cam at the transition points. To address this issue, a method is proposed to ensure continuity in the jerk profile throughout the motion.

E. B-spline Motion

This approach facilitates smoother transitions between motion segments, reducing abrupt kinematic variations. To achieve this, the follower displacement is defined using a B-spline polynomial, which ensures higher continuity throughout the motion [12]. In general, a B-spline curve of degree p with $n+1$ control points and $m+1$ knots is defined by Eq. (9).

$$C(u) = \sum_{i=0}^{n} N_{i,p}(u)\, P_i \qquad 0 \le u \le 1 \qquad (9)$$

In Eq. (9), P_i represents the control points, and $N_{i,p}(u)$ denotes the Bernstein basis polynomials. To ensure the curve passes through the initial and final control points, the knot vector is constructed with $p+1$ repeated knots at both the beginning and the end, as shown in Eq. (10).

$$U = \{\underbrace{0,\dots,0}_{p+1}, u_{p+1}, \dots, u_{m-p-1}, \underbrace{1,\dots,1}_{p+1}\} \qquad (10)$$

In order to use Eq. 9 for creation of the cam profile in the angular domain, it can be rewritten as Eq. (11).

$$C\left(\frac{\theta}{\beta}\right) = \sum_{i=0}^{n} N_{i,p}\left(\frac{\theta}{\beta}\right) P_i \qquad 0 \le \theta \le \beta \qquad (11)$$

Figure. 6 shows the kinematic profiles for the B-Spline motion. In this profile the jerk values at the beginning and end of the motion are equal to zero which makes the jerk value continuous at the transition points.

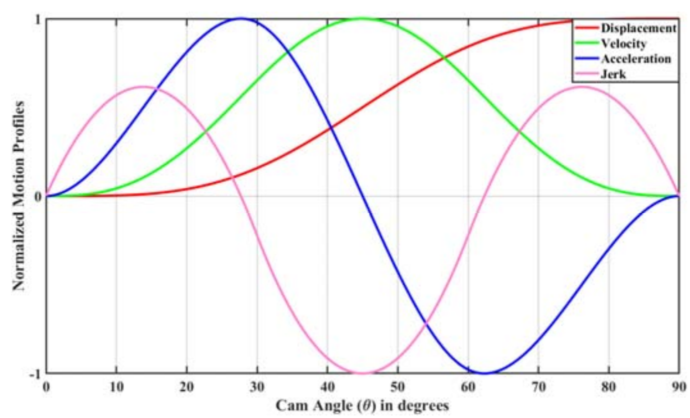

Fig. 6: B-spline motion: displacement, velocity, acceleration, and jerk profiles

MATLAB was used to develop motion profiles, generating a degree-6 B-spline curve with the spmak function, which requires control points and a knot vector as inputs. To ensure jerk continuity at junction points, control points which define the initial and final position of the motion were repeated based on the curve's degree. For example, in a rise motion where the follower moves from 0 to 10 mm, the control points are set to 0 and 10, each repeated six times. The knot vector, another input for spmak, is defined according to Eq. (10). Velocity, acceleration, and jerk profiles are obtained by differentiating the displacement profile, which is computed in MATLAB using the fnder function.

VI. CAM PROFILE GENERATION

The phases of the cam profile in the application of a Stirling engine are namely rise, dwell, fall, and dwell again. This ensures the motion of the follower, and relative displacer piston motion, matches the Stirling cycle. To generate the cam contour profile, the displacement of the follower is determined at every interval of the cam to create XYZ coordinates for the curve. See Table 1 for the input parameters used to generate the profile in Fig. 7.

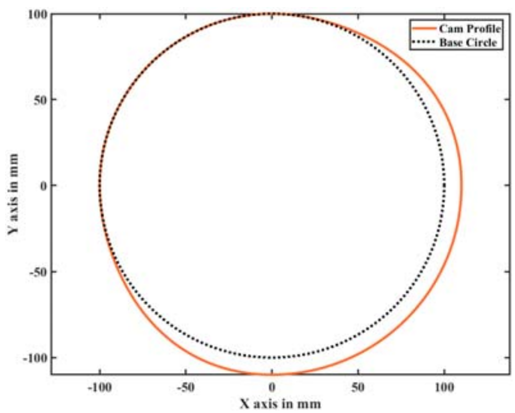

Fig.1: MATLAB Cam Profile Contour

Other input parameters including the base circle radius, follower radius, and offset were considered to generate the profile. Refer to Eq.(11-17) for the formula used to generate coordinates for a curve with these parameters [8].

Table 1: Cam Profile Generation Inputs

Base Circle Radius	100 mm
Follower Radius	10 mm
Offset	0 mm
Rotational Speed	600 RPM (62.83 rad/s)
Follower Motion	Translating Roller
Lift (Piston Stroke)	10 mm

$$R = R_0 + f(\theta) \tag{11}$$

$$X_C = r_C \cos \gamma_C \tag{12}$$

$$Y_C = r_C \sin \gamma_C \tag{13}$$

$$R_0 = [(r_b + r_r)^2 - S^2]^{1/2} \tag{14}$$

$$r_C = [(R - r_r \cos \varphi)^2 + (S + r_r \sin \varphi)^2]^{1/2} \tag{15}$$

$$\gamma_C = \frac{\pi}{2} - \tan^{-1}\left(\frac{S + r_r \sin \varphi}{R - r_r \cos \varphi}\right) \tag{16}$$

$$\varphi = \tan^{-1}\left(\frac{f'(\theta) - S}{R}\right) \tag{17}$$

VII. CAM PROFILE MANUFACTURING CONSIDERATIONS

This section discusses the effect of pressure angle, static and dynamic balancing and wear on the performance of the cam motion, and the potential effects on the Stirling engine performance. These factors must be considered before manufacturing the cams to reduce the effects of vibrations, sliding, jamming, or frequent maintenance.

A. Pressure Angle

The pressure angle of a cam is recommended to be below 30° for effective use, otherwise, effects such as sliding, jamming, wear, and catching are more likely to occur [8, 11]. The pressure angle of the sample cam with a 10 mm stroke was calculated to be roughly 12°, the base circle diameter could be reduced as there was room to increase the pressure angle. While this would promote the option to reduce the overall stock size and save material, it would remove flywheel functionality of the cam. A flywheel is commonly used to store kinetic energy and minimize disruption to the Stirling engine due to fluctuations in the heat source. Therefore, the increase in pressure angle was applied by reducing the size of the base circle from 100 mm to 50.00mm on the same stock billet so the weight of the component is unchanged, ensuring the system remains uniform. See Figure 8 for the STL files of the 50mm and 100mm cams.

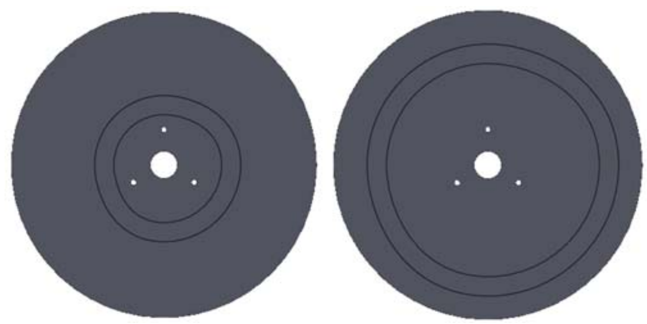

Fig. 8: 50mm Base Circle Cam (left) and 100mm Base Circle (right)

Other potential root causes of vibration in the system also include machining errors, assembly misalignment, and wear within the cam profile or follower. These interferences were considered in the simulation of the cam motion in ANSYS to ensure the cams were statically and dynamically balanced by checking the center of motion and center of mass overlapped at the origin. A study on wear detection for a disc-type prototype was conducted using a digitized reference CAD model to highlight deviations of selected points using a color map [13].

B. Balancing the Cam

Imbalance can induce vibrations, and lead to sliding or wear at certain profile points. Small cavities or mirrored pathing on either side of the cam is recommended to avoid these scenarios. See Fig. 9 for the ANSYS results of this simulation on a cam with separate motion paths on either side of the cam. In this case, the cam did not dynamically balance, it deviated from this point by a negligible amount of 0.968N. To determine the static balance, the center of mass should align with the center of the base circle, which was the case for this model.

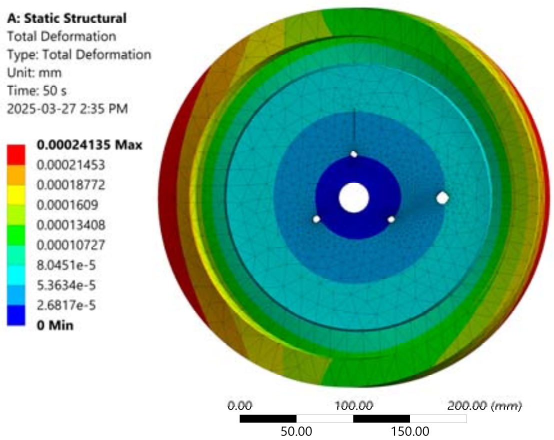

Fig. 9: Total Deformation (non-normalized scale)

C. Material Wear

The material of the cam and follower must be considered to determine where the wear will occur as these components interact. The harder material type will induce the most wear. Considering the custom profile of the cams and the standardized roller follower, the tougher material should be the cam. For this experiment, the material used by the cam is aluminum and the follower is comprised of hot hardened steel. Aluminum was selected due to its quick machining time; however, it results in high wear along the cam profile over time. Manufacturing errors induced by machining the cam groove profile with alternative materials should also be considered as imperfections can disrupt the system.

VIII. Conclusions

The grooved cam and roller follower integration to control the motion of the displacer piston stroke is very important in the efficiency performance of a Stirling engine. Proper cam profile design is crucial to reduce potential errors like vibrations, jamming, catching, and sliding with consideration of jerking. Standardized motions including uniform velocity and simple harmonic motions have discontinuous infinite points of jerk, which cause disruptions to the Stirling cycle. For custom cam profiles, other motion options such as cycloidal, 3-4-5 polynomial, and B-spline profiles are preferred for their consideration of jerk. However, cycloidal and 3-4-5 polynomial motion profiles only provide limited jerk whilst B-spline offers continuous jerk, making it the ideal method for implementation in this application.

Furthermore, the cam performance is dependent on the pressure angle, balancing, and wear. These considerations are outside of the motion profile but still affect the overall cam follower mechanism performance. The pressure angle will affect the friction between the groove cam and roller follower. The balancing will prevent wobbling or misalignment when in use and wear will affect the precision of the cam profile over time. All of these considerations will affect the overall performance of the Stirling engine.

This study provides information for generating a groove cam profile that provides precise control of the follower for the application of controlling a displacer piston, effectively improving the Stirling engine’s efficiency performance. This action will help to combat climate change by mitigating GHG emissions with the integration of innovative solutions like Stirling engines to reduce our dependence on fossil-fueled energy generation methods. These engines can utilize wasted heat in industrial applications as a clean energy resource with zero emissions. The proposed jerk-limited cam profiles offer practical benefits for industries. By improving displacer control and reducing wear, the design enhances engine efficiency and reliability. This makes the approach suitable for compact, low-maintenance power generation systems.

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