REDOCTIOM OF DISCRETE BLADE RATE TONES VIA SLOWLY ROTATING STATOR

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Abstract

Noise reduction is important in both commercial and military turbomachines. This thesis investigates the reduction of discrete tones produced by an upstream stator interacting with a rotor in a single stage by slowly rotating the stator. This method was investigated both computationally and experimentally for a co-rotating and counter-rotating stator rotating at some small fraction of the rotor rpm. Theoretical calculations were conducted in MATLAB using the Unsteady Forces and Moments (UFAM) script which predicts the unsteady forces and moments on a rotor due to an unsteady upstream stator gust by means of the Sears function and 2D strip theory. UFAM calculates the unsteady forces acting on the rotor due to a stationary upstream stator; modifications to the code were made in order to calculate the forces acting on the rotor due to a rotating stator. This was done by re-calculating the relative velocity into the rear rotor using velocity triangles. These calculations predict a co-rotating stator to decrease blade rate tones and for a counter-rotating stator to increase blade rate tones, depending upon the stator rpm. Typically, a stator rpm greater than 10% of the rotor rpm would lead to increases in the blade rate tones regardless of the direction of rotation. An experiment was designed in order to validate these results. The experiment consisted of a ducted single turbomachinery stage with an upstream cruciform. A microphone downstream of the end of the duct measured the noise produced for different co-rotating and counter-rotating stator configurations. Measurements were made in an anechoic chamber to reduce background noise effects. The experimental results displayed the same trend as predicted by the theory; validating that slowly co-rotating an upstream stator reduces blade rate tones.
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List of Symbols

\( \alpha \) Absolute flow angle
\( \beta \) Relative flow angle
\( \lambda \) Source wavelength
\( \eta \) Skew of the rotor blades
\( \Omega \) Angular velocity of the rotor
\( c_{\text{rotor}} \) Rotor chord length
\( c_v \) Fourier coefficient normal to the rotor blade
\( C \) Axial velocity
\( D \) Diameter of the duct
\( d_f \) Fraunhofer Distance
\( J \) Advance ratio
\( k \) Reduced frequency
\( l_e \) Entry length
\( L \) Unsteady lift
\( n \) Inflow harmonic
\( r \) radius
\( r_{\text{ref}} \) Reference radius
\( Re \) Reynolds number
\( S(k) \) Sears’ Function
\( U \) Shaft rate
\( V_m \) Meridional Velocity
\( V_{\text{ref}} \) Reference velocity
\( V_\theta \) Absolute velocity in the tangential direction
\( W \) Relative velocity
\( _{RR} \) Subscript for rear rotor
\( _{FR} \) Subscript for front rotor
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Introduction

In the quest for quieter turbomachines, both for commercial and military benefits, the investigation of new methods to reduce the radiated noise for the ducted system is at the forefront of research.

One contribution to the radiated noise of turbomachines is the periodic, unsteady forces produced on a rotor by its interaction with an upstream stator gust [1]. Therefore, manipulation of the wakes generated from the upstream stator is key in reducing turbomachine noise. The behavior of these tones is significantly influenced by the in-flow conditions created by these wakes, which alter the relative velocity and effective angle of attack of the fluid entering the rotor, and hence the transient pressure distribution spanning the pressure and suction sides of its blades. A time-varying lift force arises from these disturbances in the pressure field and fluctuates at the blade passage frequency [2]. As sound is simply the manifestation of fluctuations in the pressure, the lifting force and the noise level are related parameters; in fact, they are directly proportional to one another. Therefore, any decrease in the lifting force will result in a decrease in the radiated noise.

When seeking to reduce the noise level via rotor-stator interaction, the two aerodynamic parameters to be investigated are amplitude reduction of wake shedding from the upstream blade row and the interaction of these wakes with the downstream blade row. Peters and Spakovszky (2012) investigated both of these, but for an open contra-rotating propfan [1]. Comparing their configurations to an A320/B737 short to medium range twin-engine aircraft, they found increasing rotor-rotor axial spacing to allow for increased decay of viscous wakes and tip-vortices shed by the front rotor decreased interaction noises. In addition, decreasing the rear rotor diameter relative to the front rotor eliminates the tip-vortex interaction of the front rotor with the rear rotor. This also yielded a decrease in interaction noise level, given the number of blades of the rear rotor was increased to compensate for increased loading. However, Peters and Spakovsky did not investigate the effect of varying the front rotor rpm on the rotor-rotor interaction tonal noise level. This thesis serves to investigate this for a ducted fan. It is hypothesized that by altering the incoming flow angle and velocity relative to the downstream rotor, the amplitude of the unsteady interaction noise will be reduced. In other words, typically a stator or an inlet guide vane (IGV) would remain stationary within a duct to guide the flow into the rotating rotor (hence the name). This new method studies whether a certain amount of swirl, imparted to the flow by a slowly rotating stator, will reduce the unsteady rotor lift force magnitude, and thus, the radiated noise level.

Previous attempts to reduce the noise profile in ducted turbomachinery via stator/IGV alteration focused on clocking or rotatable IGVs. This is where the circumferential position of the blades relative to the rotor are varied with each trial but remain constant for a given duration of testing. Blaszczak (2006) investigated the noise reduction and efficiency improvement of a two-stage low-pressure turbine by altering the clocking position of the first stator relative to the second stator. He found that lower overall acoustic levels emitted were correlated with higher levels of efficiency, yet that in certain cases, decreasing the noise signature can also result in lower levels of efficiency, and as such, the optimum clocking position is a tradeoff between maximum noise level reduction and maximum increase in efficiency [3]. A similar conclusion was reached by Kovalev et al. (2006). They deduced that by optimizing the mutual circumferential position between the two stators in a stator-rotor-stator interaction of both a compressor and turbine in a turbomachine, the
tonal noise could be decreased [4]. These two experiments, as well as several others investigating stator clocking effects, found a reduction in overall acoustic level and discrete tones, respectively, by varying the circumferential position of one stator in comparison to one downstream of the consecutive rotor. The method tested in this thesis treats the IGV/stator as though it were a slowly moving rotor, so the circumferential blade position is varied continually throughout a given duration of testing. Additionally, this thesis focuses on a single turbomachine stage.

In another experiment, Nouri et al. investigated the effect of relative rotation rate and relative axial spacing of a ducted counter-rotating fan system on the overall performance efficiency. The relative rotation rate is measured in terms of a rotation ratio, where the front rotor was rotated at a constant 2000 rpm and the rear rotor had a rotation ratio range of 0 to 1.2, with the lowest ratio, besides 0, being 0.5, meaning the rear rotor is being rotated at half the rate of the front rotor. For these two lower rotation rates, the counter-rotating system is highly inefficient, with a rear rotor rotation rate of zero yielding a maximum efficiency of less than 35% [5]. While Nouri et al. mention the importance of the compromise between high aerodynamic performance and a low acoustic signature, only the variable parameter’s effect on efficiency are addressed. This thesis, in contrast, investigates the effect a slowly co-rotating “stator” has on acoustics, specifically the blade rate tones impinging on a downstream rotor. The velocity triangles produced by Nouri et al. to model the flow between counter-rotating fans were manipulated to yield the relative inflow velocity to the rotor for both the co-rotating and counter-rotating cases, which was implemented into the Unsteady Forces and Moments code to predict these discrete tones.

This experiment is unique in that it studies the effect a stator, rotating at some low fraction of the rotor shaft rate, has on the tones produced by the rotor. In order to be directly applicable to turbomachine engines, this experiment would need to be scaled up in terms of funding and component sizing and materials to more closely model an actual engine. For the scope of this thesis, however, the small-scale experimental results indicate that such an apparatus could be implemented to reduce noise generation.
Theoretical Modeling

Unsteady Forces and Moments (UFAM)

To accurately model the flow in a turbomachine engine, its unsteadiness, which is characterized by axial, tangential, and radial velocities along the span of the rotor’s leading edge, must be taken into account. Although several techniques to calculate unsteady forces and moments on a rotor have been established, a method by William Zierke, which expands upon the work conducted by Kemp and Sears, is particularly useful. His method takes into account the flow interaction due to adjacent blades and the effect of camber and steady angle of attack on the inflow profile [2]. To calculate the lift along the entire span of the rotor blade, Zierke employs two-dimensional strip theory, in which the characteristics of the inflow are analyzed for each infinitesimally-thin, two-dimensional airfoil segment and summed over the entirety of the span. The non-uniform inflow, resulting from the swirl imparted by the upstream stator, is characterized using Fourier decomposition and affects the relative velocity into the rotor [2]. While the assumptions made when employing 2-D strip theory can restrict the application of Zierke’s method, for the purpose of predicting a change in blade rate tones and not absolute values, this technique will suffice.

The Unsteady Forces and Moments (UFAM) script, written by Peter D. Lysak, employs Zierke’s technique to computationally calculate the forces and moments impinging on the rotor blades as a function of span, due to the wakes shed from an upstream stator [6]. For the scope of this experiment, only the forces are analyzed. The code used for this thesis is a simplified model of Lysak’s script written by Margalit Goldschmidt, the flowchart of which is shown in Figure 1. The inputs to the code are shown in pink. Since this project is investigating changes in the blade rate tones, only the inflow velocity profile is significant, as the other input parameters remain constant. However, the rotor geometry, conical flow angle, and distance between the rotor and stator are nonetheless specified to model the future experimental setup. The blue boxes in Figure 1 represent the intermediate steps taken by the code to calculate the unsteady lift which is output in the form of visuals, as represented by the boxes in green.
Although the conical flow angle for the experimental duct is zero, this angle was set to an arbitrary value in UFAM to ensure the code would run; as this is a proof of concept experiment, the only parameters which have an outcome on the change in blade rate tones are the stator and rotor rpm relative to one another.

The unsteady lift is computed using

\[ \frac{L_n}{\rho V_{ref}^2} = \frac{\pi c_{rotor}}{r_{ref}} \frac{W_r}{V_{ref}} c_v S(k_n)e^{i\eta} \]  

(1)

where \( \eta \) is the leading edge skew angle of the rotor blades, \( S \) is the modified Sear’s function (modified because the reference point for gust penetration was shifted to the leading edge from the mid-chord. This is discussed in further detail in Ref. 2), \( k \) is the reduced frequency, \( n \) is the harmonic number, and \( W_r \) is the relative velocity. The reduced frequency can be thought of as the number of times an airfoil oscillates as an unsteady gust travels across its chord length. It is clear from this equation that the unsteady lift on a rotor blade is a function of the relative velocity into the rotor. The relative velocity for a stationary stator’s inflow into the rotor is defined as:

\[ W_r = \sqrt{V_m^2 + (V_\theta - r\Omega)^2} \]  

(2)

where \( V_m \) is the meridional velocity, \( V_\theta \) is the absolute velocity in the tangential direction, and \( \Omega \) is the angular velocity of the rotor. In UFAM, the velocities are non-dimensionalized by some reference velocity \( (V_{ref}) \), which in this case, is the maximum axial velocity of 2.74 m/s (8.99 ft/s). Equation 2 rewritten in its non-dimensionalized form is
\[
\frac{W_r}{V_{\text{ref}}} = \sqrt{\left(\frac{V_m}{V_{\text{ref}}}\right)^2 + \left(\frac{V_\theta}{V_{\text{ref}}} - \frac{\pi}{J}\right)^2}
\]  

(3)

where \( J \) is the advance ratio. The advance ratio is defined as:

\[
J = \frac{V_{\text{ref}}}{\Omega D}
\]  

(4)

where \( \Omega \) is the shaft rate and \( D \) is the reference diameter, which in this case is twice the radius of the stator blade or 5.4 cm (2.13 in).

UFAM is written to analyze a standard stator-rotor stage configuration [2]. In order for UFAM to be applicable to the case in this experiment, the inlet velocity to the rotor must be altered to simulate the flow induced by the wakes and motion of a rotating stator. In the code developed by Lysak, the relative inflow velocity to the stator is a function of the meridional velocity and rotor rotational speed, but since the stator is now rotating, the relative velocity into the rotor is found using a different method (to be outlined in the following section). The relative velocity into the rotor can be calculated in terms of known parameters, such as the stator and rotor rpm/rotational velocity, the axial flow velocity, and the stator exit angle.

Relative Velocity

Stationary Stator

The relative velocity entering the rotor for the standard stationary stator is given by Lysak in Equation 2. This equation in the context of the parameters used in this thesis is given in Equation 3, which is derived from Figure 2:

\[
W_{1RR} = \sqrt{(V_m)^2 + (V_\theta - U_{RR})^2}
\]  

(5)

where \( W_{1RR} \) is the relative velocity entering the rotor, \( V_m \) is the meridional velocity, \( V_\theta \) is the tangential velocity, and \( U_{RR} \) is the rotational speed of the rotor.

In its non-dimensional form, Equation 5 becomes

\[
\frac{W_{1RR}}{V_{\text{ref}}} = \sqrt{\left(\frac{V_m}{V_{\text{ref}}}\right)^2 + \left(\frac{V_\theta}{V_{\text{ref}}} - \frac{\pi}{J}\right)^2}.
\]  

(6)
The relative velocity is obtained using the same method for the counter- and co-rotating stator.

**Counter-Rotating Stator**

Nouri et al conducted an experiment to investigate the effect two consecutive counter-rotating rotors would have on the overall efficiency, in comparison to a stationary stator-rotor system. The velocity triangles for the front and rear rotor of their system are displayed in Figure 3.
These velocity triangles were used to model the flow of a stator counter-rotating with a downstream rotor; the relative velocity entering into the downstream rotor can be determined. The angle of the absolute flow exiting the stator is assumed to be the same as the angle of the absolute flow entering the rotor, $\alpha_{2_{FR}} = \alpha_{1_{RR}}$. Under this simplification, the velocity triangles between the rotor and stator from Figure 3 can be superimposed, and a velocity triangle for the relative flow between the two components is established in Figure 4.
Using trigonometry, an expression for the relative velocity into the rotor can be obtained:

$$W_{1RR} = \sqrt{(V_m)^2 + (U_{FR} + V_m \tan \beta_{2FR} - U_{RR})^2} \tag{7}$$

where $\beta_{2FR}$ is the stator exit angle, $U_{RR}$ is the rotor rotational velocity, and $U_{FR}$ is the stator rotational velocity.

In its non-dimensional form, Equation 7 becomes

$$\frac{W_{1RR}}{V_{ref}} = \sqrt{\left(\frac{V_m}{V_{ref}}\right)^2 + \left(\frac{r \pi}{J_{FR}} + \frac{V_{\theta}}{V_{ref}} - \frac{r \pi}{J}\right)^2} \tag{8}$$

where $J_{FR}$ is the advance ratio using the shaft rate ($\Omega$) of the stator/front rotor.

Nouri et al. conducted an experiment with a stage of counter-rotating rotors, in which an increase in efficiency was observed but the effect of this configuration on the noise level/blade rate tones was not studied. The case of the counter-rotating stator was included in the theoretical prediction of tones in UFAM to determine the merits of utilizing this set-up to reduce tonal noise. Both in theoretical predictions and in preliminary experimental results, the counter-rotating stator-rotor configuration produced increases in the blade rate tones.

**Co-Rotating Stator**

This experiment primarily investigated the effect of the co-rotating stator configuration. Using velocity triangles, an expression for the relative velocity entering the rotor was derived by reversing the direction of the stator rotation, represented in Figure 5 and Figure 6 as $U_{FR}$. The same method for finding the relative velocity for the counter-rotating case is used to find the relative velocity for the co-rotating stator set-up. The velocity triangles for the stator and rotor are shown in Figure 5.
For a co-rotating stator, the relative velocity into the stator is

\[
\overrightarrow{W_{1RR}} = \sqrt{(V_m)^2 + (-U_{FR} + V_m \tan \beta_{2FR} - U_{RR})^2}
\]  

(9)
In its non-dimensional form, Equation 9 becomes

$$\frac{W_{1RR}}{V_{ref}} = \sqrt{\left(\frac{V_m}{V_{ref}}\right)^2 + \left(-r \frac{\pi}{J_{fr}} + \frac{V_\theta}{V_{ref}} - r \frac{\pi}{J}\right)^2} \quad (10)$$

The discrete tones produced by the rotor occur at its blade passage frequency, which is simply the rotor’s number of revolutions per second multiplied by the number of its blades. When an upstream or downstream stator is introduced, the effective number of blades becomes the least common multiple of the rotor and stator blade count. It is this number that is multiplied by the rotor revolutions per second to yield the effective blade passage frequency, and hence the frequency at which the tones will occur. The stator was designed to have the same number of blades as the rotor so as to produce maximum amplitude discrete tones at a lower blade passage frequency and its harmonics. This kind of configuration is not seen in practice as these loud, low frequency tones would produce greater levels of noise pollution, but as the aim of this experiment is to identify a change in the overall sound signature, a loud baseline is beneficial in detecting any significant reduction. In addition, the distance between the stator trailing edge and the rotor leading edge is half the chord length of the stator blades, or 1.27 cm (0.5 in). The smaller the distance between the stator and rotor, the less the viscous wakes shed by the stator dissipate, the greater the velocity deficit, and the greater the forces acting on the rotor [2]. Decreasing the distance between stator and rotor will increase the blade rate tones of the rotor, and will therefore a change in these discrete tones will be more detectible.

**Summary of Computational Predictions**

The Unsteady Forces and Moments script was run for both a counter-rotating and co-rotating stator rpm that was 10% of the rotor rpm, with the stationary stator configuration serving as the baseline for comparison purposes. Figure 7 displays the resultant force coefficients for each stator configuration. A more negative value for the force coefficient means a smaller force is impinging on the rotor, i.e. a co-rotating stator is predicted to reduce the average force acting on the rotor, while a counter-rotating stator is predicted to increase this force, in comparison to a stationary stator.
These forces impinging on the rotor manifest themselves as sound, and as such, UFAM predicts not only a decrease in forces for a co-rotating rotor, but also a decrease in the radiated noise. UFAM predicts an increase and decrease of 0.69 dB for the counter-rotating stator and co-rotating stator, respectively. This value may not be a significant amount for this configuration, but it could manifest in larger decreases for different geometries and inputs.
Experimental Design

The inspiration for this initial proof-of-concept experiment is to find a reduction in the discrete tones produced in a single stage of a turbomachine. Because of its scope, the direct application of the findings of this experiment are limited. This research studies the changes in blade rate tones between a system with a stationary stator versus a slowly rotating stator and does not measure absolute values; i.e. the exact inflow velocity to the stator, stator geometry, etc. is not necessary in detecting the change in blade rate tones, as long as these parameters are kept constant as the stator rpm is varied. Further experimentation must be conducted to determine the effect on efficiency and specifically which kind of engine configuration is optimal for this type of system.

To model a single stage of a turbomachine, two fans were to be placed within a duct, where one fan would act as the IGV and the other as the rotor. A cruciform would be placed upstream of the stator-rotor stage to model a structural support in the duct. To more accurately model the slightly cambered airfoil shape of a typical stator, one had to be customized with compatibility to the rotor and duct.

Custom Stator

The slightly-cambered airfoil shape of the stator blade in a typical turbomachine is designed to reduce the swirl of the flow, inflicted by the upstream rotor, and guide it into the downstream rotor. In any turbomachine, the orientation of the stator blade is an axial reflection of the rotor’s, which serves to counteract the swirl imparted to the flow by the rotor, by guiding the fluid in the direction opposite that of the rotor velocity, towards the axis. As the ducted fans purchased did not resemble this form, the stator to be used in the experiment had to be constructed with blades capable of guiding the flow. The simplest and cheapest method of doing so was ultimately determined to be drawing/constructing the stator in SolidWorks and 3D printing the sketch using Penn State University’s free basic 3D printing service. Two stator geometries were constructed, each with the same simple asymmetrical airfoil shape, one with a 5.77° stagger angle, and the other with a 20° stagger angle. These stators guide the flow into the rotor and impart swirl. This in effect imposes a different relative velocity into the downstream rotor. The stator with the 5.77° stagger angle was only used for the first test series, where the rotor was rotated at 41.8 Hz (2508 rpm). In order to impart more swirl into the flow and further alter the relative velocity into the rotor, a stator with a stagger angle of 20° was constructed and used for all subsequent test series.

The design/geometry of the stator is significant in that it affects the relative velocity entering the rotor, and thus contributes to the unsteady forces impinging on the rotor. As is discussed in the Theoretical Modeling section, the relative velocity into the rotor is a function of the stator exit angle, which itself is a function of the stagger angle and the camber line. The airfoil blade of the stator is shown in Figure 8. The complete 5-bladed stator with a blade stagger angle of 20° is shown in Figure 9.
Figure 8. Customized stator blade (top and isometric views)

Figure 9. Stator with 20º stagger angle airfoils.
Rotor

The rotor is a 15.24 cm (6 in) axial inductor fan. This fan was selected for this experiment due to its availability and inexpensive cost. In order to accurately model the rotor geometry in UFAM, the chord as a function of radius must be specified. These parameters were found manually by measuring the chord distance at twenty-one different span-wise locations of one of the blades, as shown in Figure 10. The chart containing the chord length as a function of radius can be found in Appendix A.

![Rotor Geometry](image)

Figure 10. Rotor Geometry

The blades have a slightly non-linear camber line, but for simplicity in modeling, this was neglected. The blades are oriented at an 18° stagger angle, and the rake and skew are modeled in UFAM as zero for simplicity.

Cruciform

A simple crossbar was used to model an upstream disturbance in the flow. While it was found that the wakes from the cruciform dissipated before interacting with the downstream blade row, and did not have an effect on the blade rate tones, the cruciform was included to model actual application. For example, a support structure in an air-conditioning duct.

Motors

The duct fans purchased came equipped with 7.62 cm x 7.62 cm (3x3-inch), C-Frame two-pole Class B motors which would impose too great an amount of blockage in a 15.24 cm (6-inch) duct to attain viable aerodynamic data. The motors were removed from the duct and replaced with Maxon DC motors with diameters of 2.197 cm (0.865 in). Mounting these new motors in the duct posed a challenge, as the mechanism had been altered. The mounting bar connected to the walls of the duct would remain in place, serving as a means to which the “sandwiched” motor configuration would be connected. The original motors had each come with two caps to hold the rotating piece in place. The bottom caps were kept, and in each, a straight line cut was made 0.51
cm (0.201 in) from the edge to accommodate the wires extending from the bottom of the DC motors. To ensure the top cap would not inflict friction to the rotating shaft, specialized mounting caps were drawn in SolidWorks and 3D printed. As each cap was connected directly to the motor and did not come into contact with the shaft, the motor shaft could rotate freely regardless of how tightly the clamping mechanism was fastened to the mounting bar. Two threaded rods and six nuts were used to secure the motor between the two caps, with the excess length of the rods used to connect the entire configuration to the mounting bar with a set of nuts and washers for each. Figure 11 and Figure 12 display the mounting mechanisms for the stator and rotor motors, respectively, inside the duct.

Figure 11. Mounting mechanism for rotor motor.

Figure 12. Mounting mechanism for stator motor.
Turbulence Grid

For fully developed flow to exist within the “test section” of the duct, a certain entry length of duct must be present. This length for a turbulent flow profile is a function of the Reynolds number and the diameter of the duct as related by the following equation for turbulent flow,

\[ l_e = 4.4Re^{\frac{1}{6}}D \]  

(11)

where D is the diameter of the duct [7]. In order to determine the entry length needed for fully developed flow, the maximum velocity of air through the duct must be found. The ducted fan has a maximum CFM of 180 for free air, which translates to a volume flow rate of 0.0849 m³/s. The 15.24 cm (6-inch) diameter duct has a cross sectional area of 0.0182 m² (0.196 ft²). Dividing the volume flow rate by this area yields a maximum air velocity through the duct of 4.655 m/s. The maximum CFM for the duct fan is based on the original motor with a maximum rpm of 3000.

The inlet free stream velocity into the stator is predicted by measuring the air velocity at the exit of the duct via a digital anemometer. The measurements were made at the various rotor rpm with the stator stationary in the duct. Although the air velocity measurements were made 10.16 cm (4 in) downstream of the rotor, the axial free stream velocity remains fairly constant, and therefore can be used as a relatively accurate approximation of the inlet velocity into the stator. The probe was placed at mid-span (11.43 cm or 4.5 in from the bottom of the duct, along the diameter), outside the wake of the hub and motor mounts. Measurements taken at mid-span are also considered an average of the parameters along the entire span.

Using the maximum velocity of air the rotor fan can draw, 4.655 m/s (15.27 ft/s), a diameter of 15.24 cm (6 in) for the characteristic length, and a kinematic viscosity of 1.511 x 10⁻⁵ m²/s, the Reynolds number is determined to be 46950.5, and the entry length needed for fully developed flow is 4.028 m (13.22 ft). For the scope of this experiment, this length is impractical. Using the idea of a turbulence grid, the entry area of the duct was filed with straws, resembling honeycomb, to decrease the effective area of the duct and, consequently, the entry length required for fully developed flow. Using a straw diameter of 6.0 mm (0.236 in), the Reynolds number becomes 1848.44 and the entry length, using Equation 6 becomes 9.25 cm (3.64 in). The honeycomb configuration of the 15.24 cm (6-inch) long straws is displayed in Figure 13.
The honeycomb produces blockage in the flow, which causes a decrease in the air velocity. A lower velocity requires less entry duct length, and as the maximum possible velocity was used in the calculation of this parameter, there is high confidence that fully developed flow will exist. Once an anemometer was obtained, actual measurements of the velocity through the duct could be found to be used in UFAM. Although the original motors and fan were rated for a maximum of 180 CFM at 3000 rpm, the new configuration was run at 4134 rpm, but produced an air velocity of only 2.74 m/s (9 ft/s) (measured using a digital anemometer), which can be attributed to the blockage caused by the straws at the entry of the duct. The rotor could then be run at any rpm greater than 3000, as long as the measured velocity does not exceed 4.655 m/s. However, due to the high temperature the motor reached at 4134 rpm, the rotor motor was run below 3000 rpm after this series of tests were taken.

Duct

The entire duct is 1.016 m (40 in) long, with the cruciform 10.16 cm (4 in) from the end of the straw configuration, the stator leading edge 30.5 cm downstream from the cruciform, and the rotor leading edge 1.27 cm (0.5 in) (0.5c) downstream from the stator trailing edge. The stator and rotor are placed within one chord length to simulate the stage of an actual air-breathing engine and to reduce the magnitude of stator wake dissipation [2]. The distance between stator trailing edge and rotor leading edge is not necessarily one chord length; however, a shorter distance amplifies the tones; it increases the signal-to-noise ratio and therefore any changes in tones due to stator rotation are more readily detectable. Each experimental component and its location in the duct is given in Figure 14.
The actual components within the duct are shown in Figure 15.

Microphone Placement

The placement of the measuring microphone downstream of the duct is important to be noted for the sound pressure levels reported in this thesis. Typically, a standard distance of 1 m from a source is used, in order to ensure that the measurement is made in the far field. The far field of a source is the region in space where the sound pressure level obeys the Inverse Square Law. The Inverse Square Law is when the sound pressure level decreases 6 dB with each doubling of distance from the source. While this experiment was performed in an anechoic chamber to reduce reflected/background noise, it was still important to minimize the distance between the source and the measuring microphone in order to maximize the signal-to-noise ratio. Therefore, the distance to the far field needed to be calculated. This Fraunhofer Distance is a function of the duct diameter and source wavelength as follows:

\[ d_f = \frac{2D^2}{\lambda} \]  

(12)

where \( D \) is the diameter of the duct, \( d_f \) is the Fraunhofer Distance, and \( \lambda \) is the source wavelength, which can be calculated from the expected frequency of the tones [8]. Higher frequencies correspond to shorter wavelengths. Since the relationship between the distance to the far field
varies inversely with wavelength, the desired Fraunhofer distance must relate proportionally to the highest frequency in the target measurement range. The motor for the rotor in this experiment has the capability of running at 8000 rpm or 836 Hz, so this is the maximum frequency used to determine the far-field distance. For safety concerns, the rotor motor will be run at no more than 3000 rpm.

\[
\lambda = \frac{\text{speed of sound at } 27^\circ \text{C (} \frac{m}{s} \text{)}}{\text{frequency of noise}} = \frac{347.87 \text{ m/s}}{838 \text{ Hz}} = 0.4151 \text{ m (1.362 ft)} \quad (13)
\]

\[
d_f = \frac{2(0.1524 \text{ m})^2}{0.4151 \text{ m}} = 0.112 \text{ m (0.367 ft)} \quad (14)
\]

The Fraunhofer Distance is determined to be 0.112 m (0.367 ft); any length greater than this is outside the near field and in the far-field region. For reasons of simplicity, the microphone was placed at a distance of 0.3048 m (1.0 ft) from the edge of the duct, directly in line with the rotor and stator’s axis of rotation, as seen in Figure 16.

![Figure 16. Axial microphone is 1-ft away from edge of duct.](image)
Experimental Procedure

Using the TENMA DC power supply, the current dial was turned to the maximum setting (approximately 3.10 amps), and the voltage dial was set to approximately 26 Volts. The exact voltage value is not absolute and can be varied between 20 and 30 Volts, but the current should be set at its maximum value, without exceeding 5 amps. If the current is set any lower, the motor will have insufficient power to run without fluctuating. The power supply with its voltage and current settings before power is output to the motors is displayed in Figure 17.

![Figure 17. Current and voltage settings on power supply.](image17)

The rpm of both the stator and rotor motor were initially found by means of a voltmeter. One lead was connected to the hall sensor of the controller and the other to the ground port of the controller, as seen in Figure 18. Leads of voltmeter connected to ground and hall sensors. The setting on the voltmeter was set to Hertz and read out the shaft rate. The issue with the voltmeter, which was discovered during the first round of data acquisition, is that it is not capable of measuring frequencies below approximately 40 Hz, and since the stator is rotating at very low frequencies, the voltmeter was unable to accurately measure its frequency. The voltmeter, however, was capable of actually reading the rotor shaft rate, as can be seen in Figure 19.

![Figure 18. Leads of voltmeter connected to ground and hall sensors.](image18)
The rotor and stator rotational frequencies were measured with a universal counter, which has the capability of measuring frequency from 0 to 100 MHz to 7-decimal place accuracy. While the voltmeter was capable of reading the rotor shaft rate, both the stator and rotor shaft rates were measured with a universal counter for its increased accuracy and continuity purposes. The signal input port to the universal counter is a BNC, but using a simple adapter, the banana plugs that had formerly been plugged into the voltmeter, could then be inserted into the counter. While the universal counter provides a more accurate measure of the frequency than the voltmeter, it also read out a frequency that was far too great to be the true shaft rate of the stator. For this reason, it was deduced that the hall-effect sensor in the motor controller of the stator was reading out some unknown multiple of the actual shaft rate. In order to find its multiplier, a marker was tied to one of the five stator blades, and the amount of time taken for stator to complete 40 revolutions was measured. The multiplied shaft rate was then taken across the motor controller with the universal counter. These measurements were taken at several different stator frequencies, and plotted against one another. Referencing Figure 20, the relationship between true stator shaft rate and the frequency given by the hall effect sensor is highly linear (R² value of 0.9999), where the slope of this curve represents the multiplier.
Test Matrix

At each rotor rpm, the effect of different stator rpms was analyzed using a calibrated microphone to measure the sound pressure level emanated from the end of the duct containing the cruciform, the stator, and the rotor. The data runs collected are summarized in Table 1.

Table 1. Test Matrix

<table>
<thead>
<tr>
<th>Test</th>
<th>Configuration</th>
<th>Measurements</th>
<th>Stator Stagger</th>
<th>Stator RPM</th>
<th>Rotor RPM</th>
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<tbody>
<tr>
<td>0</td>
<td>Microphones calibration</td>
<td>n/a</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>Ambient background noise</td>
<td>SPL</td>
<td>5.77°, 20°</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>Ducted rotor with cruciform only</td>
<td>SPL, RPM/shaft rate</td>
<td>5.77°</td>
<td>0</td>
<td>2508</td>
</tr>
<tr>
<td>3</td>
<td>Ducted rotor with cruciform only</td>
<td>SPL, RPM/shaft rate</td>
<td>20°</td>
<td>0</td>
<td>1044, 2038, 2502, 2976, 4134</td>
</tr>
<tr>
<td>4</td>
<td>Ducted rotor with cruciform and stator</td>
<td>SPL, RPM/shaft rate</td>
<td>5.77°</td>
<td>25.3</td>
<td>2508</td>
</tr>
<tr>
<td>5</td>
<td>Ducted rotor with cruciform and stator</td>
<td>SPL, RPM/shaft rate</td>
<td>20°</td>
<td>25.3</td>
<td>1044, 2038, 2502, 2976, 4134</td>
</tr>
</tbody>
</table>
At each rotor rpm, the stator was first kept stationary to serve as the baseline for comparison purposes. The next stator rpm for a given rotor rpm is the lowest rpm achievable by the motor controlling the stator, or 25.3 rpm. All subsequent tests at a given rotor rpm were taken at a stator rpm increased by approximately 6 rpm from the previous test until the stator rpm was approximately 9% of the rotor rpm; this is denoted by the 25.3 + 6n in the farthest column of Test 6 and 7, where n equals 1 through a number contingent upon the final stator rpm to rotor rpm ratio. The stator rpm was not increased beyond approximately 9% of the rotor rpm because the overall noise signature, in addition to blade rate tones, began to increase when this ratio was exceeded.

<table>
<thead>
<tr>
<th>Test</th>
<th>Description</th>
<th>SPL, RPM/shaft rate</th>
<th>5.77°</th>
<th>25.3 + 6n</th>
<th>20°</th>
<th>25.3 + 6n</th>
<th>1044, 2038, 2502, 2976, 4134</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>Ducted rotor with cruciform and stator</td>
<td>SPL, RPM/shaft rate</td>
<td>5.77°</td>
<td>25.3 + 6n</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Ducted rotor with cruciform and stator</td>
<td>SPL, RPM/shaft rate</td>
<td>20°</td>
<td>25.3 + 6n</td>
<td></td>
<td></td>
<td>1044, 2038, 2502, 2976, 4134</td>
</tr>
</tbody>
</table>
Results

Each data set was taken over a period of five minutes at a sampling rate of 4096 Hz. The data was taken on several different dates to ensure repeatability. With these long runs, the frequency resolution of the data was high enough to capture the discrete tones expected. Using the fast Fourier transform in MATLAB, the signals were converted into the frequency domain. Changes in the blade rate tones between a stationary stator versus a rotating stator were determined by taking the difference between the threshold noise and the tone peaks.

As discussed in the theoretical modeling section, the presence of a counter-rotating stator in the duct is predicted to decrease the blade rate tones at the blade passage frequency in comparison to a stationary stator configuration, but by less than a co-rotating stator. During preliminary data acquisition, tests were conducted for both a co-rotating and counter-rotating stator configuration at a certain rotor shaft rate, over the course of two separate testing phases. The first test phase rotated the rotor at 41.8 Hz (2508 rpm) with the stator blades having a stagger angle of 5.77°, and the second, for reasons of imparting greater swirl into the flow, with a stator blade stator angle of 20°. A range of stator shaft rates (0.411-5.09 Hz (24.7 – 305.4 rpm)) were tested for each rotor shaft rate. Measurements for both co-rotating and counter-rotating stator configurations were taken at each stator shaft rate. The measurements that were taken at the eleven different shaft rates yielded the same general trend in noise level. In other words, the results were consistent for a range of shaft rates. The peaks at the blade rate tones either increased or decreased, depending upon stator frequency and harmonic, and/or shifted in frequency. Rotating the stator in the direction opposite to the rotor shifts the blade rate frequencies lower, while rotating the stator and rotor in the same direction shifts the blade rate frequencies higher. This is demonstrated in Figure 21 for an example configuration.
The co-rotating stator generally yielded a greater decrease in blade rate tones than the counter-rotating stator, and for these reasons, any notions of decreasing discrete tones by means of a counter-rotating stator were abandoned. All subsequent tests were conducted with a co-rotating stator.

**Co-Rotating Stator**

One of the tests which displayed a significant decrease in blade rate tones was taken at a constant rotor shaft rate of 41.7 Hz (2502 rpm). As both the stator and the rotor have 5 blades, the discrete tones for a non-rotating stator are seen at multiples of the shaft rate multiplied by the blade number, i.e. 209 Hz, 418 Hz, 626 Hz, and 834 Hz. As displayed in Figure 22, implementing a co-rotating stator at 2.06% of the rotor rpm decreases the tones; at 209 Hz there is a net decrease of 2.85 dB. This tone also shifts in frequency to 213 Hz, as a result of the change in relative velocity discussed earlier. As seen in Figure 22, in addition to the primary blade rate tone being shifted, a tone greatly reduced in amplitude remains at the original frequency of 209 Hz.

*Figure 21. Autospectrum of microphone measurements.*
For the second harmonic at 418 Hz, the stationary stator yielded a blade rate tone 7.7 dB above the threshold noise level. The counter-rotating and co-rotating stator decreased the tone and shifted the blade passage frequency to 6.2 dB, 409 Hz and 3.9 dB, 426 Hz, respectively, above the threshold, for a decrease of 1.5 dB and 3.8 dB from the stationary stator configuration. In addition, the blade rate tone which remains at the original frequency is reduced by 5.2 dB and 5.3 dB for the counter-rotating and co-rotating stator, respectively. As lower frequencies have the capability of traveling greater distances without dissipating than higher frequencies, the changes in the blade rate tones of the first and second harmonics are more important in this analysis than the third and fourth harmonics.

Figure 22 displays only one of the stator-rotor configurations which were tested in this experiment. While this particular ratio of stator to rotor shaft rate yields a decrease in blade rate tone at the first and second harmonic for a co-rotating stator, other stator to rotor ratios caused either an increase or decrease in the blade rate tones at either the first or second harmonic. Figure 23 shows the qualitative change in the blade rate tones at the first harmonic for each stator and rotor shaft rate. The red and blue circles depict a decrease and increase, respectively, in the blade rate tone for a certain ratio of the stator to rotor shaft rate when compared to a ratio of zero, i.e. a stationary stator. The size of the circle represents the amount the acoustic level changed, with a larger circle corresponding to a larger increase or decrease in magnitude.
Although an optimal stator to rotor shaft rate ratio cannot be deduced from this data, a few general trends can be observed. As stated previously, above a ratio of approximately 0.09, the role of the co-rotating stator becomes counterproductive and actually increases the blade rate tone at a given frequency. In the tests where the rotor was rotated at a shaft rate of 17.4 Hz, the maximum stator shaft rate to rotor shaft rate was 0.11, and as can be seen in Figure 22, all ratios above 0.09 produced an increase in the blade rate tone. It appears from the data, that a greater rotor shaft rate corresponds to more consistent decreases in the blade rate tone at the first harmonic, regardless of the exact shaft rate (as long as it does not exceed a certain percentage of the rotor rate). While this data presents such a trend, a more comprehensive experiment at several more rotor shaft rates must be taken before any conclusions can be made. In a more sense, however, a decrease in the blade rate tones was observed at all rotor shaft rates, depending upon the stator shaft rate, verifying this method as a potential way to reduce noise due to blade rate tones.

The same qualitative analysis was conducted for the same stator-rotor configurations at the second harmonic of the blade rate tones. Figure 24 displays the change in the blade rate tones at the second harmonic for each stator and rotor shaft rate.
The most notable difference between the data at the first and second harmonics occurs when the rotor is rotating at 68.9 Hz (4143 rpm). In contrast to the blade rate tones at the first harmonic, where every stator shaft rate produced a decrease in acoustic level, the introduction of a rotating stator increased the tones. Otherwise, the results are promising in that a decrease in the blade rate tones occurs for all stator shaft rates when the rotor is rotated at 34.0, 41.8, and 49.6 Hz (2040 rpm, 2508 rpm, and 2975 rpm, respectively), with the exception of when the stator is rotated at 5.13 Hz (10.3% rotor shaft rate), which again, is above the stator to rotor shaft rate ratio.

A more qualitative analysis is shown in Figure 25, where the change in the blade rate tones at the first and second harmonic are plotted against the stator and rotor shaft rates.
Any negative change in the acoustic level is a decrease in the blade rate tone, and vice versa. The maximum tone decrease of the first harmonic occurs at a rotor shaft rate of 49.6 Hz and has a value of −5.7 dB. A rotating stator does not necessarily consistently decrease the sound pressure level of the blade rate tones, but for certain stator-rotor shaft rate ratios, when a decrease does occur, it can be significant, as demonstrated by Figure 25.

While UFAM predicted a general decrease in decibel level for a co-rotating and an increase for a counter-rotating stator, the results vary depending on rotor rpm and the stator rpm as a percentage of rotor rpm. UFAM predicts an idealized case, when in reality, blockage exists in the duct, which could introduce radial components into the flow, and decrease the overall axial velocity. The additional components of velocity in the flow produce more complex interactions in the flow between the stator and rotor than the code predicts. This, coupled with the assumptions made in the theoretical prediction, i.e. for use of 2D strip theory, and zero rake and skew in the geometry of the rotor, produce discrepancies between the theoretical predictions and the experimental results.

While a few stator-rotor configurations exceeded the theoretical threshold of the stator rotating at a maximum of 10% the rotor rpm, most tests were conducted with the stator running at a lower percentage than this. UFAM predicted a decrease of 0.69 dB for a co-rotating stator rotating at
10% the rotor rpm, yet when the stator was rotated at only 6.3% the rotor rpm (2976 rpm), the decrease in the shifted tone was 5.7 dB, which is far greater than UFAM predicted. For other stator-rotor rpm ratios, a co-rotating actually increased the blade rate tone by a small amount (< 1 dB). The exact cause for these variations is unknown and would need to be further investigated.
Conclusions and Recommendations for Future Work

The objective of this thesis was to reduce turbomachinery noise produced by unsteady stator/rotor interaction tones by rotating the stator at some small fraction of the rotor rpm. This method was investigated computationally and validated experimentally. Theoretical modeling was conducted in MATLAB using the Unsteady Forces and Moments script, which calculates the unsteady lift using a modified Sears function and 2D strip theory. UFAM was modified to account for both a co-rotating and counter-rotating stator, as opposed to one that is stationary, by adjusting the relative velocity into the rotor. Theoretical calculations were made using a generic unsteady stator gust and rotor geometry, but for the same blade numbers and rotation rates to be modeled experimentally. This is because this thesis is measuring the change in blade rate tones and not the overall decibel level of these tones, and so the assumptions made in the calculations will not affect the overall trends. It was predicted by theory that a slowly co-rotating stator will yield a decrease in the discrete tones at the blade passage frequency of a rotor, in contrast to a conventionally stationary stator.

An experiment was designed in order to validate the results predicted by theory. This experiment consisted of a ducted stator and rotor, with an upstream cruciform. The flow into the duct was conditioned using straws. The noise produced by this turbomachine was measured using a microphone in line with the axial axis of the duct, placed downstream of the duct in the far-field. Measurements were made in an anechoic chamber in order to increase the signal-to-noise ratio. Additionally, measurements were made on different days in order to show repeatability. Data was taken at several different rotor rpms, while varying the stator rpm at different low percentages of the rotor rpm. Once all of the data was obtained, the time data was read into MATLAB and converted into the frequency domain. Each configuration was analyzed, and the sound pressure level of the discrete tones produced were tabulated. In most cases, a counter-rotating stator-rotor configuration resulted in an increase in blade rate tones and a shift in the blade passage frequency to a lower frequency, while a co-rotating stator/rotor configuration resulted in a decrease in the discrete tones and a shift in blade passage frequency to a higher frequency. The greatest decreases in blade rate tone were observed for a stator rotating in the same direction as the rotor with an rpm below 10% that of the rotor.

Because this was a proof-of-concept experiment, only changes in the blade rate tones were calculated and analyzed, to validate the idea that a slowly-rotating stator would reduce blade rate tones. In order for this method to be applicable to turbomachines, experiments need to be conducted using infrastructure that more closely resembles the stator and rotor stages of an axial compressor. These would then need to be modeled more precisely in the computational calculation of forces and moments. In this experiment, the skew and rake of the rotor geometry file were assumed to be zero and the stagger angle was assumed constant, which will not be the case for actual compressors blades. In addition, in terms of the inflow velocity to the rotor, the radial velocity was assumed to be zero and the relative velocity into the rotor was modeled as a constant scalar value, which, even for this experiment, is not necessarily the case. In order to accurately model the flow into the rotor caused by a rotating stator, a more complex inflow velocity file must be computed.
Additionally, noise level and performance efficiency are typically two dependent quantities, i.e. increasing one could increase the other and vice versa. This method of slowly rotating a stator proves effective in reducing blade rate tones; however, the effect of such a configuration on efficiency was not investigated and should be in future experiments to determine whether or not it has enough merit to be put into practice.

A possible avenue for future research is investigating different stator and rotor geometries and which ones work together to produce the optimum decrease in blade rate tones. As aforementioned, the efficiency of these configurations should also be investigated. In collecting data for this type of experiment, the static and dynamic pressures should be measured along the duct in order to calculate pressure ratios through the stages.
Bibliography


### Appendix A: Rotor Geometry

*Table 2. Rotor Radius and Chord Non-dimensionalized by Blade Radius (0.54 cm or 0.216 in)*

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<tr>
<th>Radius</th>
<th>Chord</th>
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Academic Vitae

Kerstyn M. Auman  
Email: kma5477@psu.edu

Education
The Pennsylvania State University, University Park, PA  
Schreyer Honors College  
B.S. in Aerospace Engineering (focus in astronautics)  
Expected Graduation Date: May 2017

Relevant Work Experience
Applied Research Laboratory at Penn State (Distinguished Undergraduate Research Position, 6/8/2015-present)  
Supervisor: Joseph Welz, Margalit Goldschmidt  
• Designed and conducted experiments to investigate a new concept to reduce unsteady forces/noise on rotor blades in turbomachinery and submarine propulsion  
• Designed and constructed the apparatus to conduct aforementioned experiments (including 3D-printed components)  
• Wrote a supplemental algorithm to predict theoretical forces and moments in UFAM (Unsteady Forces and Moments) code with a slowly rotating “stator” as compared to a stationary stator  
• Collected sound pressure level data at various stator and rotor shaft rates and computed decibel level difference between a stationary and rotating “stator” scenarios  
• Demonstrated a decrease of several dB in the blade rate tones of the system at certain stator/rotor shaft rates

Technical Experience
Aero Design Club (Spring 2016-Present) (Electronics Subsystem Lead and Secretary)  
• Leading the team to design and build the electronics system of a 40-lb RC airplane  
• Competing in 2017 SAE International Collegiate Design Competition  
Engineering Design Study Abroad in Singapore (Summer 2015)  
• Designed and built a working prototype of a device intended to improve the local workers’ cleaning efficiency as a team of NUS, PSU, and BYU students

Relevant Coursework
Space Dynamics, Spacecraft Design, System Dynamics and Controls, Aerospace Structures, Advanced Aerospace Structures, Space Propulsion, Advanced Orbital Mechanics, Space Astronomy

Skills
- Proficient in MATLAB, LabView, SolidWorks/CAD modeling program  
- Competent in Microsoft Word, PowerPoint, Excel, C++  
- Experienced with 3D printing  
- Knowledgeable in the basics of aero-acoustics  
- Fluent in German

Leadership
Co-founder and Secretary of Penn State Music Service Club (a club which plays live music for the elderly and mentally disabled (Fall 2014-Present)  
• Oversee distribution of event information  
• Coordinate events and outings

Schreyer Honors College Mentor to Incoming Freshmen (Fall 2014)  
• Co-organized various orientation events  
• Mentored a group of ten incoming freshmen for a three-day orientation

Awards
Sigma Gamma Tau (Aerospace Engineering Honors Society) (2016)  
Tau Beta Pi Engineering Honors Society (2015)  
Penn State Provost Scholarship (2013-2015)  
President’s Freshman Award (Fall 2013)  
Dean’s List (All semesters)