

iSHELL Cross Disperser Geneva Drive Torque Calculations

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05/12/2011

Introduction

The cross disperser geneva drive mechanism has been analyzed to determine the loads on the drive motor as the turret moves from one observing position to another. The goal is to determine the loads generated so that an appropriate stepper motor can be selected to drive the cross disperser turret. Two stepper motors rated for cryogenic, high-vacuum use are being considered: the Phytron VSS 32, and VSS 42 which produce a maximum of 32N-mm & 92N-mm of torque respectively and a maximum speed of 600RPM. Since the cross disperser turret is the largest turret in the iSHELL instrument, this analysis may be considered as a worst case loading scenario and may be used as a guide for selecting motors for the other turret mechanisms in the instrument.

The results indicate that the VSS 42 stepper motor is more than adequate to drive the turret and produces a factor of ~3 more torque than what is needed to drive the turret. Additionally, the VSS 32 motor is sufficient to drive the turret if a 4:1 in-line gearbox is mounted to the motor.

Geneva Drive Design

The cross disperser turret drive consists of a 12 discreet position geneva wheel and crank that is driven by a stepper motor with a 3:1 reduction miter gear set between the motor and the crank. To provide a precise and repeatable positioning accuracy that the geneva/crank alone cannot achieve, a detent mechanism, which consists of a spherical ball mounted to a parallel spring flexure, gets actuated to seat into a vee-groove on the turret, thus precisely and repeatably locating the turret in the observing position. The detent is actuated via a cam lobe mounted to the crank and a follower bearing mounted to the translating detent mechanism. As the crank cycles through a single turn, the detent unseats to allow the turret to rotate to the next position and then seats again to precisely locate the turret.

As the drive cycles the turret from one position to the next, the motor must provide sufficient torque to overcome various loads that exist such as, spring preloads from the detent mechanism, friction, & inertial loads from accelerating masses. Since the turret is assumed to be balanced, gravity loads are neglected.

The following figures describe the design of the drive and the results of the calculations.

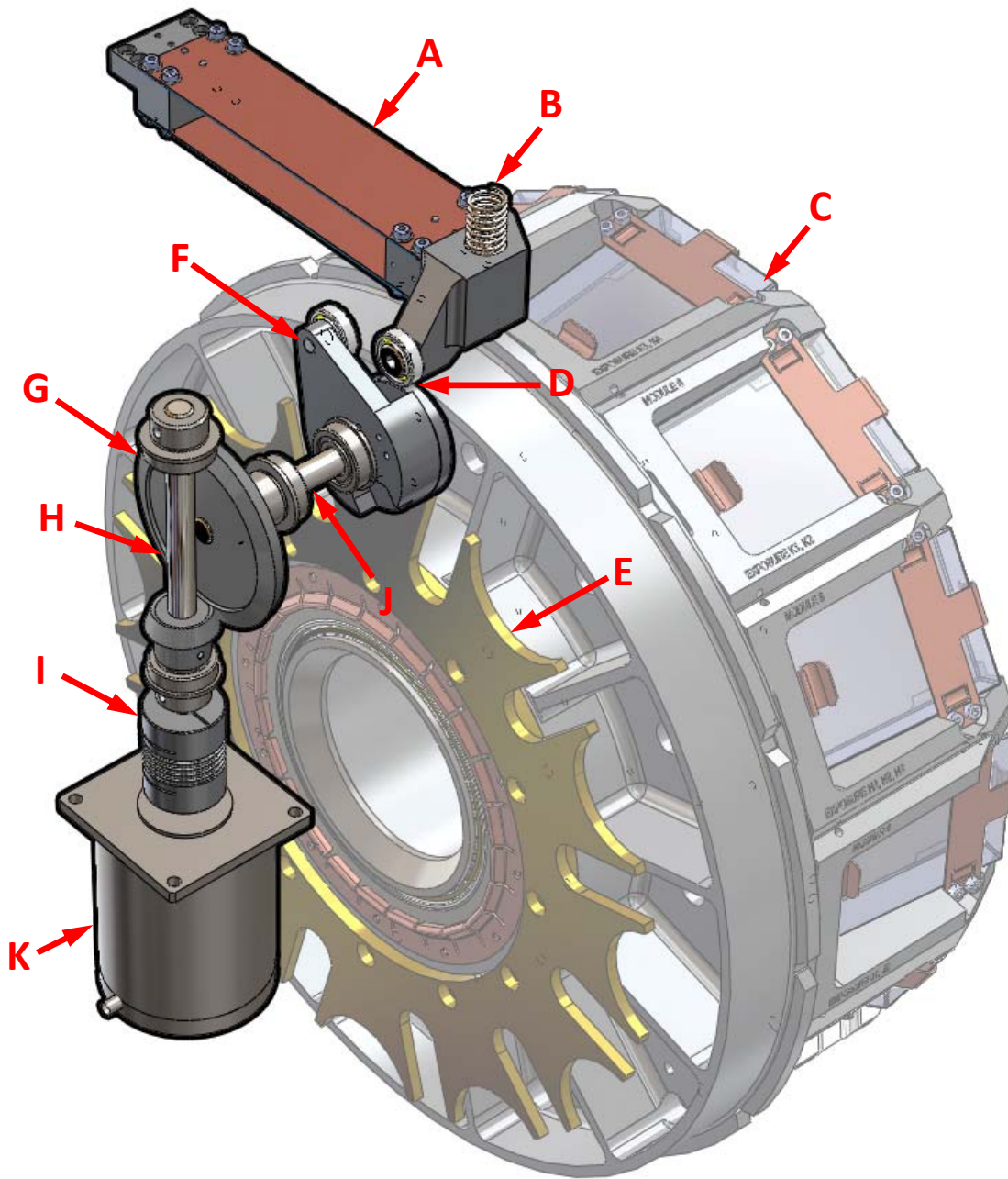


Figure 1. Cross dispenser turret drive design showing the following items: A) parallel spring flexure, B) detent preload compression spring, C) turret, D) cam & follower, E) geneva wheel, F) crank, G) 3:1 ratio miter gear set, H) motor drive shaft, I) motor flex coupling, J) crank shaft, K) stepper motor.

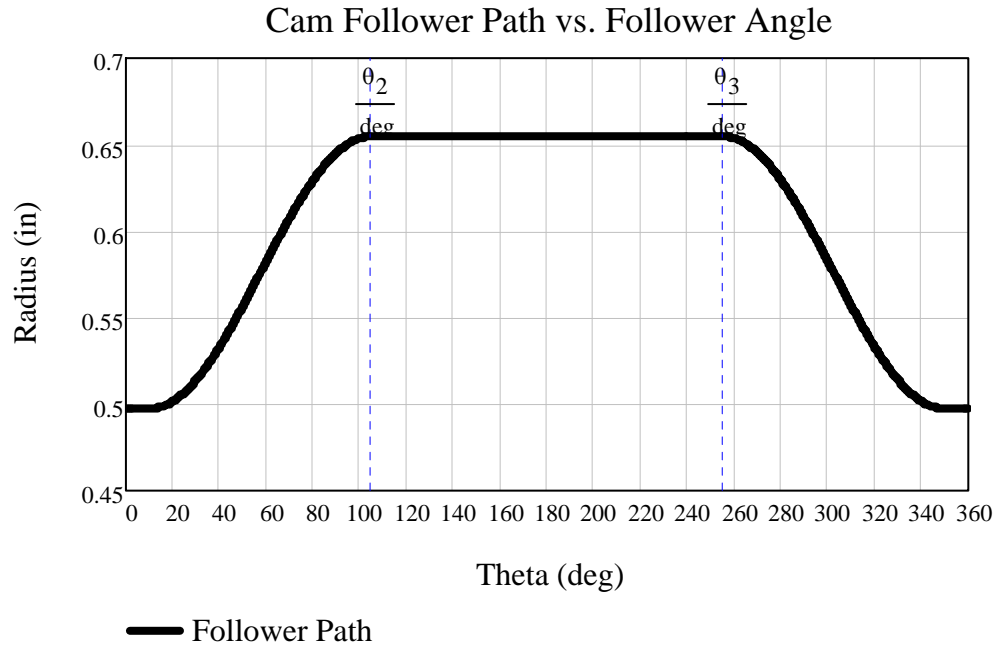


Figure 2. Path of the cam follower through one cycle. As the cam (crank) rotates from the zero degree position, the follower (detent) remains fully seated for 10 degrees, then climbs up the cam until the detent completely clears the turret at 105 degrees. A sinusoidal shaped path was chosen to ensure the follower makes a smooth transition into and out of the detent. From 105 degrees to 255 degrees, the follower remains clear of the turret so that the turret can move next position. From 255 degrees to 350 degrees the follower lowers back down thus seating the detent into the vee-groove. From 350 degrees to 360 degrees, the follower remains seated.

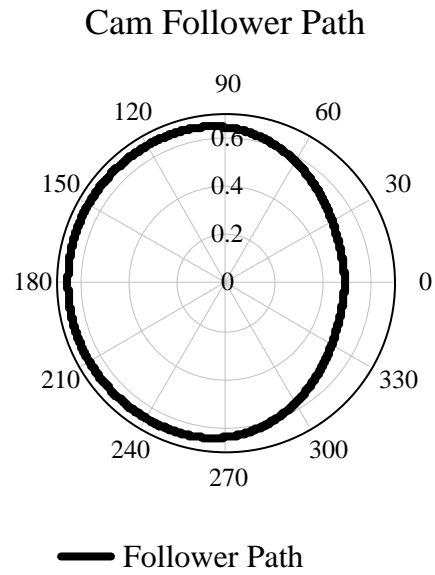


Figure 3. Plot of the follower path (from figure 2) in polar coordinates reveals the actual shape of the cam.

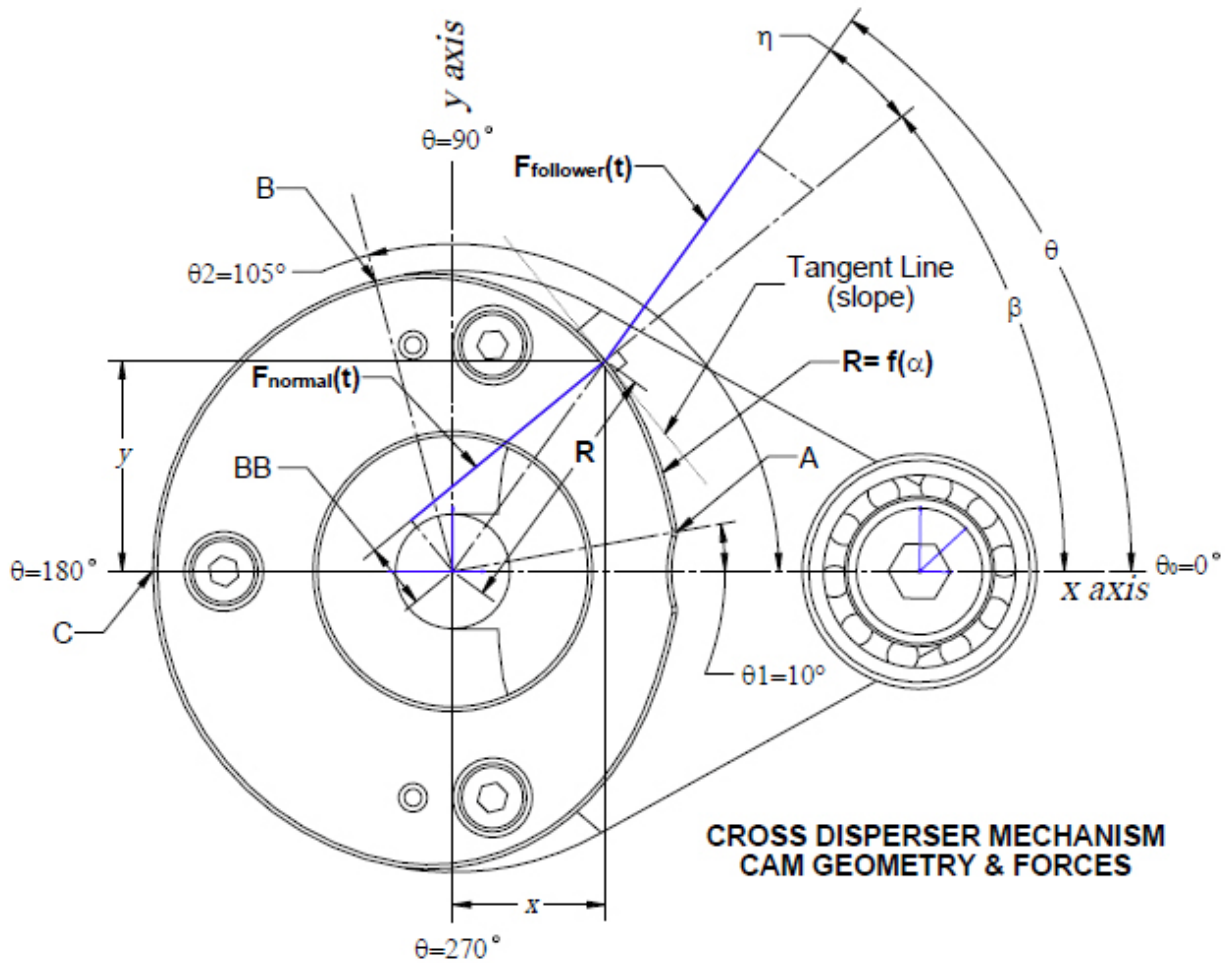


Figure 4. Drawing of the crank & cam with dimensions used in the calculations. Note that there is a cutout in the cam profile that is plus/minus 10 degrees around the zero degree position to ensure that the follower does not make contact with the cam when the detent is fully seated into the vee-groove. From position A to B above, the follower (detent) climbs out of the seated position. From B to C, the follower remains fully disengaged. The shape of the cam is symmetric about the x-axis and thus the motion from C back to the zero degree position is a mirror image of the first half of the motion.

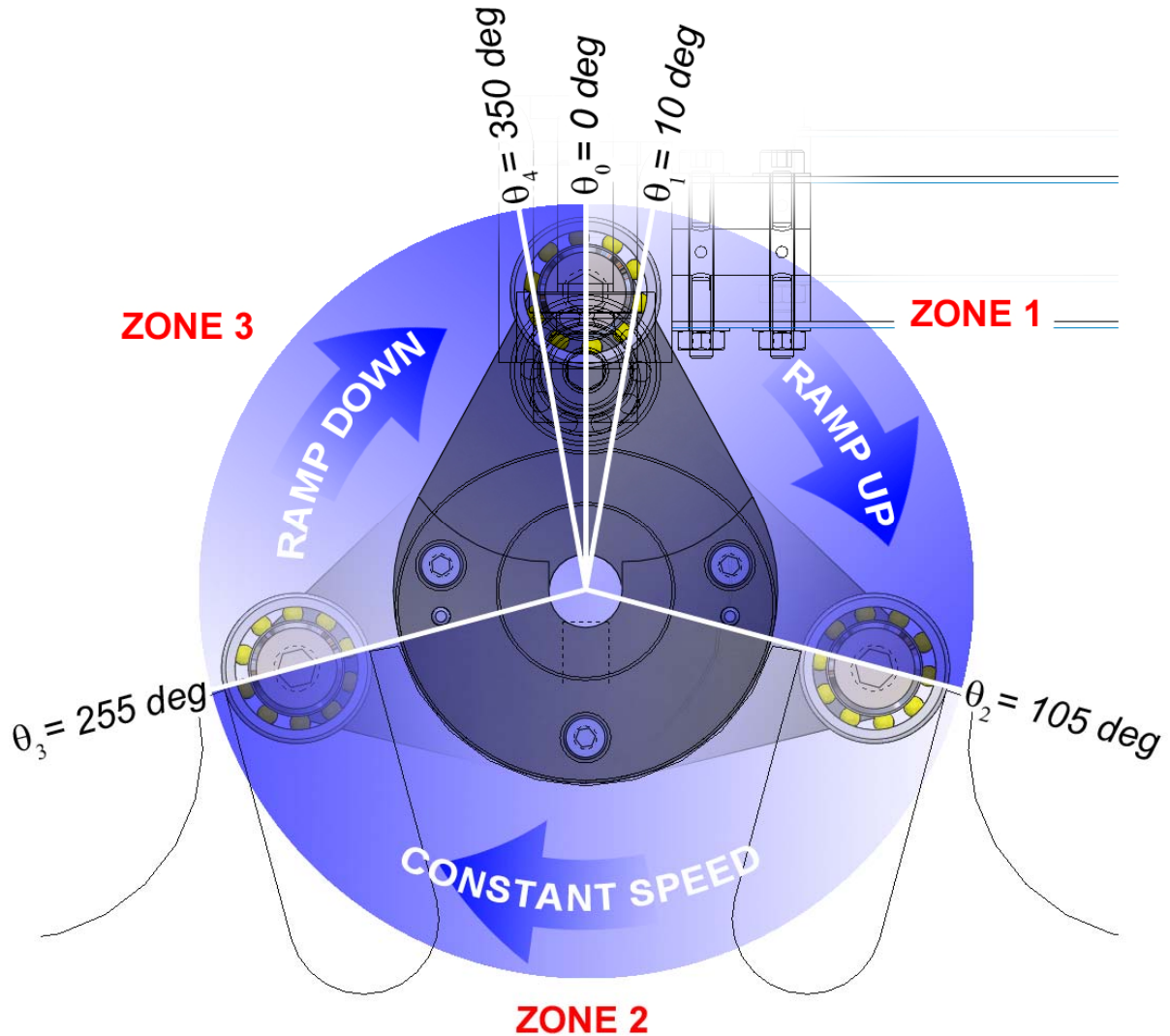


Figure 5. There are three zones in one cycle of the crank. In zone 1, the crank starts at the zero degree position and ramps up to full speed when it reaches the 105 degree position. During that time, the follower travels from the fully seated position to the fully disengaged position in preparation for the turret to move. In zone 2, the crank engages with the geneva slot at 105 degrees and the turret gets moved to the next position where the crank disengages with the geneva slot at 255 degrees. The crank speed remains constant through this portion of the cycle. In zone 3, the crank ramps down to a stop beginning at 255 degrees and ending at 360 degrees. The follower lowers back down and the detent seats in the vee-groove of the next turret position to locate the turret.

Table 1 below lists the parameters that can be adjusted to affect the loads on the drive motor. The parameters that have the most affect on the drive torque are the gear ratio and the compression spring preload. To a lesser extent, the cycle time and bearing frictions affect the loads on the drive. The values in the table are somewhat arbitrary but seem reasonable. The bearing friction values were determined by experimenting with a torque gauge and some guessing based on common sense.

Table 1. Parameters used in drive torque calculations.	
Parameter	Value
Desired cycle time (one cycle of crank)*	5 seconds
Detent preload compression spring	Associated Spring Part No. C0360-035-2500-S
Spring rate	3.582 lb/in (stainless steel)
Spring preload when detent fully seated	3.5 lb
Gear ratio between motor & crank	3:1
In-line motor gearbox ratio	4:1
Crank shaft bearing friction	0.1 in-lbf
Motor shaft bearing friction	0.1 in-lbf
Turret hub bearing friction	1.0 in-lbf

*Note that the maximum distance move of ½ turn of the turret represents 6 grating positions and thus, 6 cycles of the crank. Therefore, the total move time is calculated by multiplying the cycle time by the number of positions moved. In this case, 6 cycles x 5 seconds/cycle = 30 seconds.

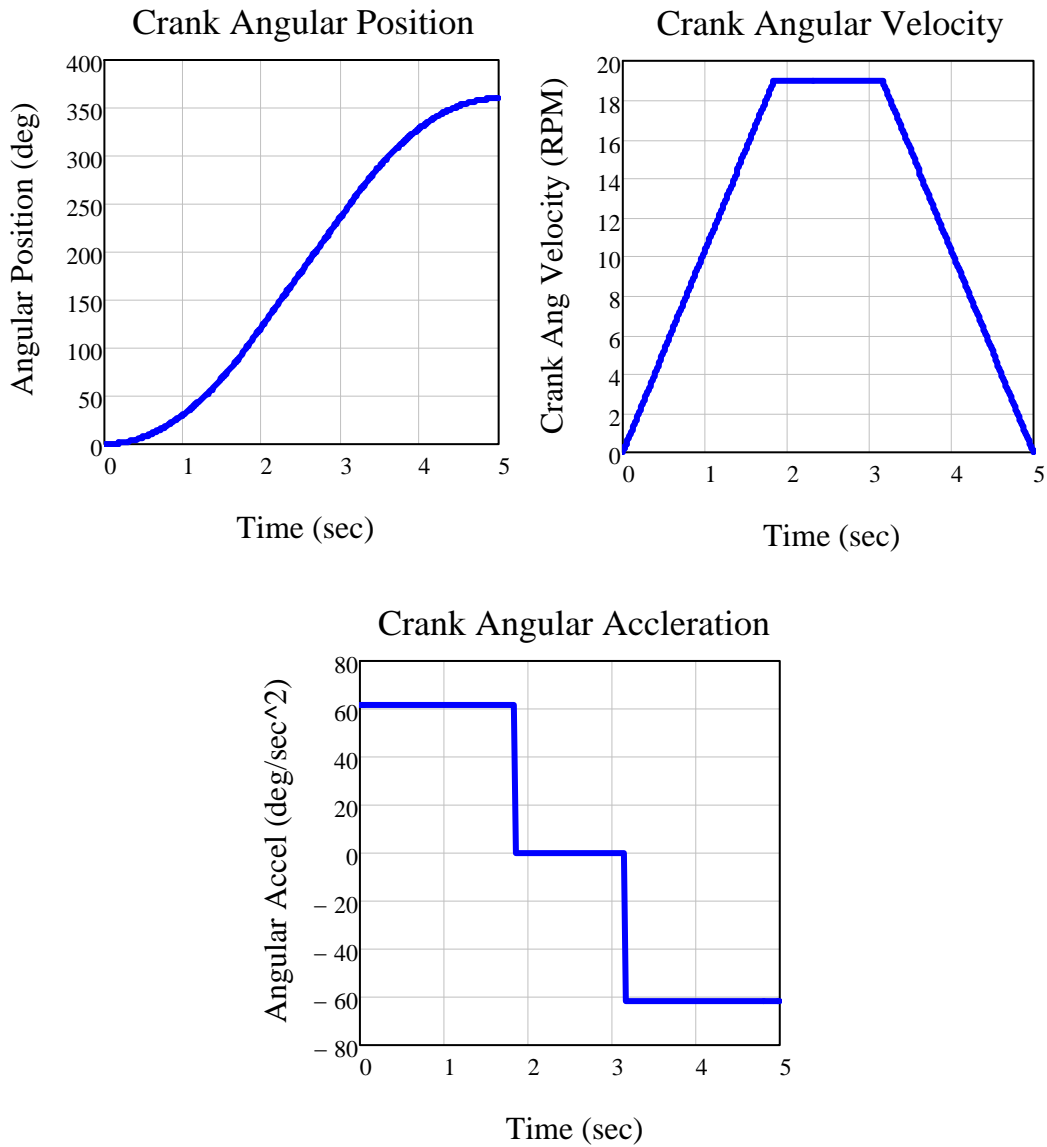


Figure 6. Plots of the crank angular position, velocity, and acceleration as a function of time for one cycle of the crank.

Cross Disperser Turret Drive Inertias & Masses

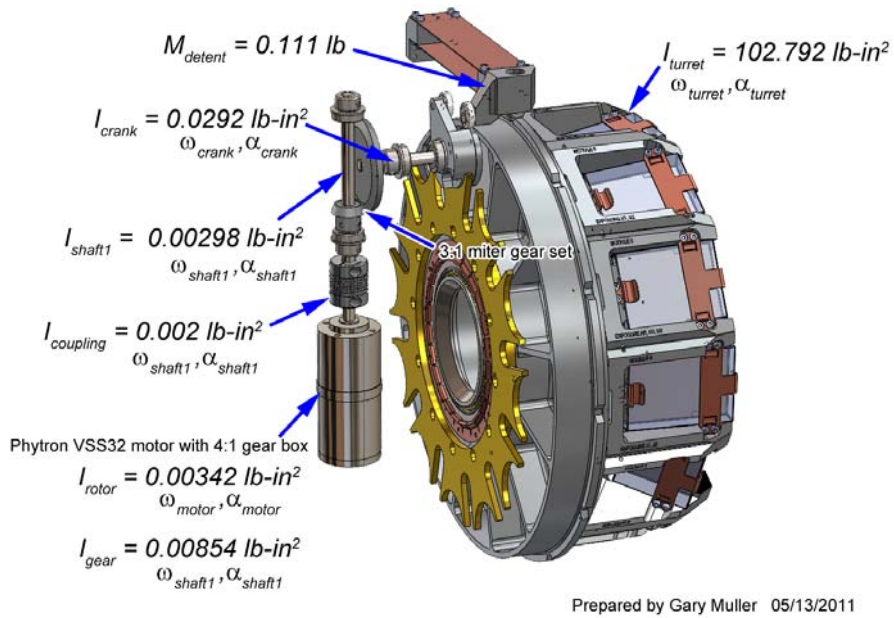


Figure 7. Inertias of the rotating masses used in the calculations.

Detent Parallel Spring Flexure Forces & Mass

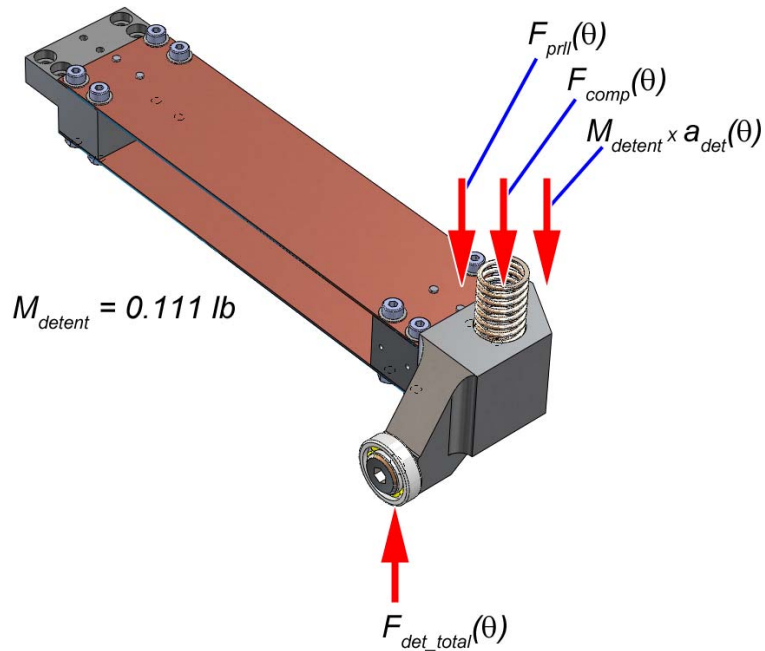


Figure 8. Free body diagram of parallel spring flexure showing force balance used in the calculations.

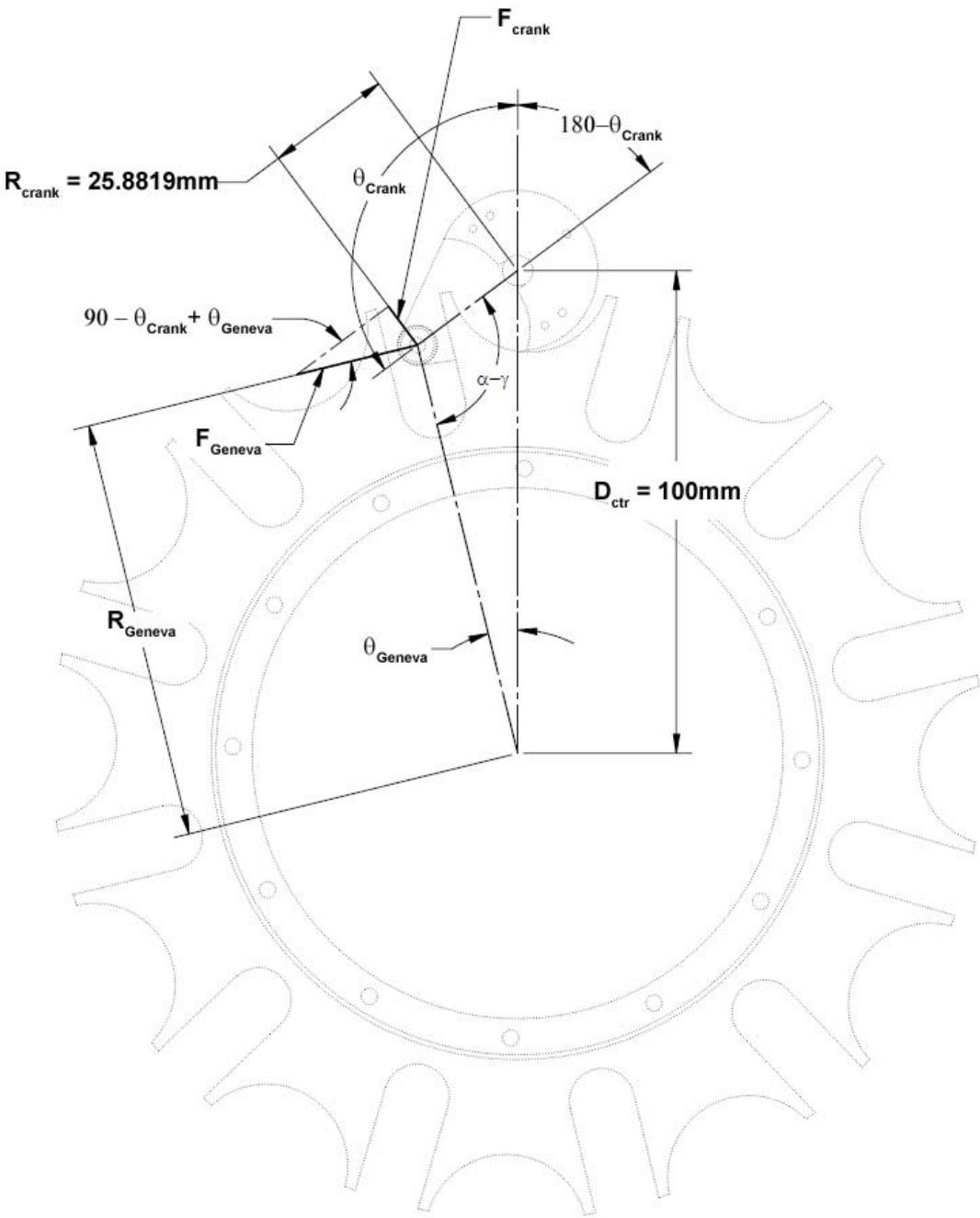


Figure 9. Geneva and crank geometry and dimensions used in calculations.

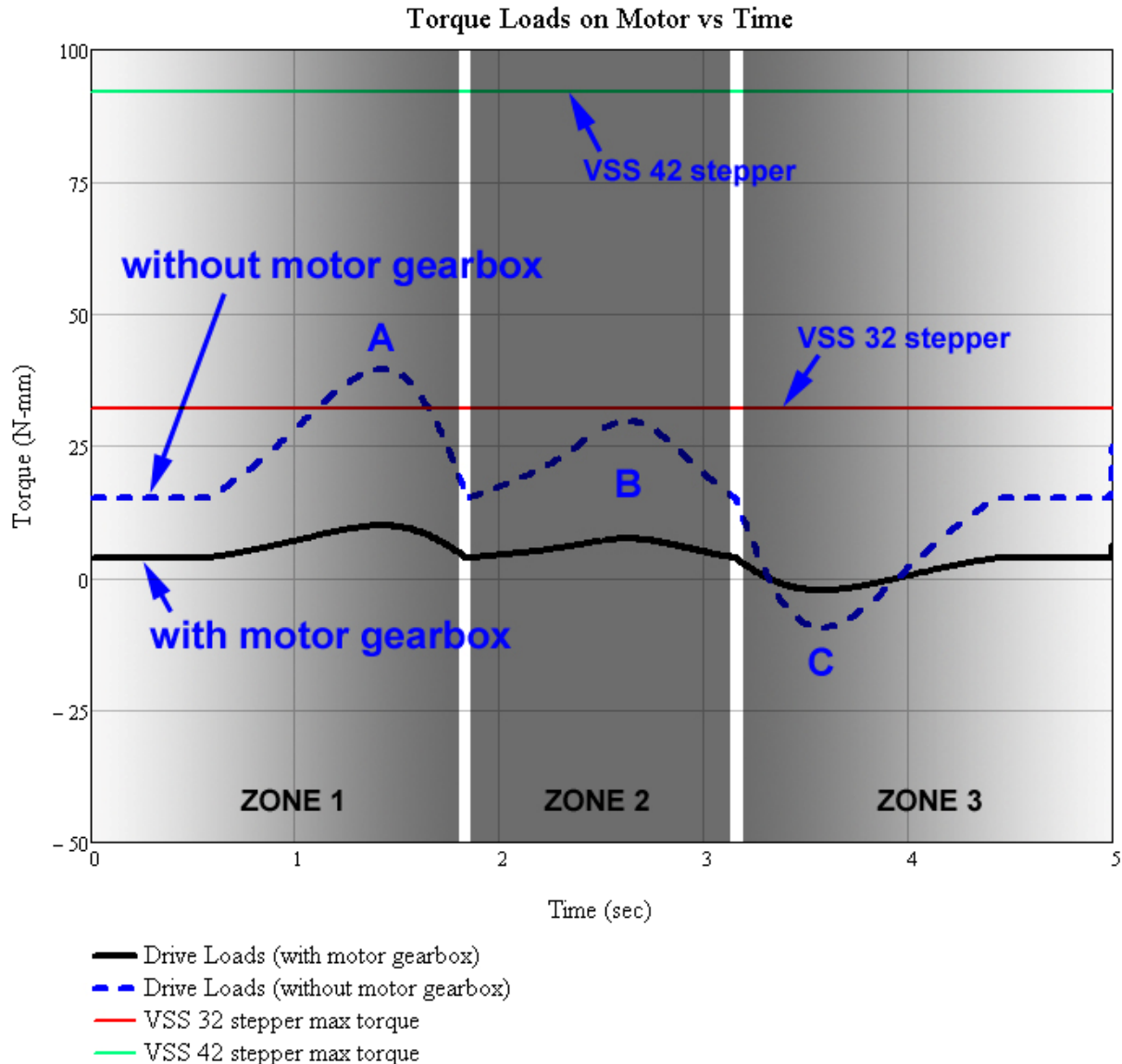


Figure 10. Loads on stepper for one cycle of the geneva mechanism. The solid black curve represents the loads with a 4:1 ratio gearbox in-line with the stepper motor and the dashed blue line represents the loads without a gearbox. In zone 1, the dominant loads that result in the peak at “A” are from the detent mechanism compression spring and parallel spring flexure. The compression spring preload is the significant factor in calculating the maximum torque load on the drive. In zone 2, the peak at “B” is caused by friction and the accelerating mass of the turret as it gets actuated from one position to the next. In zone 3, the peak at “C” is again caused by the spring forces but are acting in to assist the motor rather than fight it.

Results

The results of the calculations indicate that by using the parameters in table 1, the VSS 42 stepper has sufficient torque to drive the cross disperser turret with plenty of margin (92N-mm max torque versus ~39.5N-mm load for the motor without a gearbox). Additionally, the VSS 32 stepper has sufficient torque to drive the turret (32 N-mm max torque versus ~9.9N-mm load for the motor with a gearbox) if a 4:1 gearbox is mounted in-line with the stepper motor. If a 4:1 gearbox is not used, then the VSS32 stepper will stall as it attempts to lift the detent mechanism out of the vee-groove.

For a 5 second cycle time, the maximum motor speed will be 228 RPM if a gearbox is not used (for the VSS 42 stepper only) and 57 RPM if a gearbox is used. The maximum motor speed is 600 RPM so a 5 second move time is reasonable.

Conclusions

The analysis indicates that either stepper motor, the Phytron VSS 42 or VSS 32, may be used to drive the cross disperser turret. However, if the VSS 32 is selected, a 4:1 ratio in-line gear box must be mounted to the stepper to prevent stalling.

Since the cross disperser turret is the largest turret in the iSHELL instrument, this analysis represents the worst case load scenario and thus, the VSS 32 with a 4:2 ratio gear box appears to be a good choice for the turret drives.

Appendix**Calculations**

User inputs that affect the motor torque required to drive the turret.

Enter the desired cycle time. This is the time to move the turret from one position to an adjacent position. The maximum number of cycles for the longest move is 6 cycles.

$t_{\text{cycle}} := 5\text{sec}$ Desired cycle time

Select a desired compression spring for the detent preload force.

The following spring is assumed: Associated Spring Part No. C0360-035-2500-S.

$L_{\text{free}} := 2.50\text{in}$ Free Length of spring $F_{\text{max_comp_spr}} := .833 \cdot 5.73\text{lbf}$ $F_{\text{max_comp_spr}} = 4.773\text{lbf}$
 $k_{\text{spr}} := 4.3 \cdot .833 \frac{\text{lbf}}{\text{in}}$ Spring rate with .833 factor for stainless steel $k_{\text{spr}} = 3.582 \frac{\text{lbf}}{\text{in}}$

Select a desired preload force for seating the detent into the vee-groove.

$F_{\text{pre}} := 3.5\text{lbf}$ Desired initial preload of compression spring when the detent is seated into the vee-groove.

The following two stepper motors are considered for driving the turret.

Motor Parameters for Phytron Steppers

$T_{\text{vss42_max}} := 92\text{N}\cdot\text{mm}$ VSS42 Stepper motor drive (stall) torque
 $T_{\text{vss32_max}} := 32\text{N}\cdot\text{mm}$ VSS 32.200.1.2-VGPL 32 4-UHVC stepper motor with 4:1 reduction gear box (stall) torque
 $\text{Ratio}_{\text{mot_gearbox}} := 4$

Select a desired gear reduction ratio between the motor and the crank of the geneva mechanism (default ratio is 3:1).

$\text{Ratio}_{\text{miter_gear}} := 3$ Miter gear reduction ratio between stepper and crank

Define crank shaft bearing friction (torque).

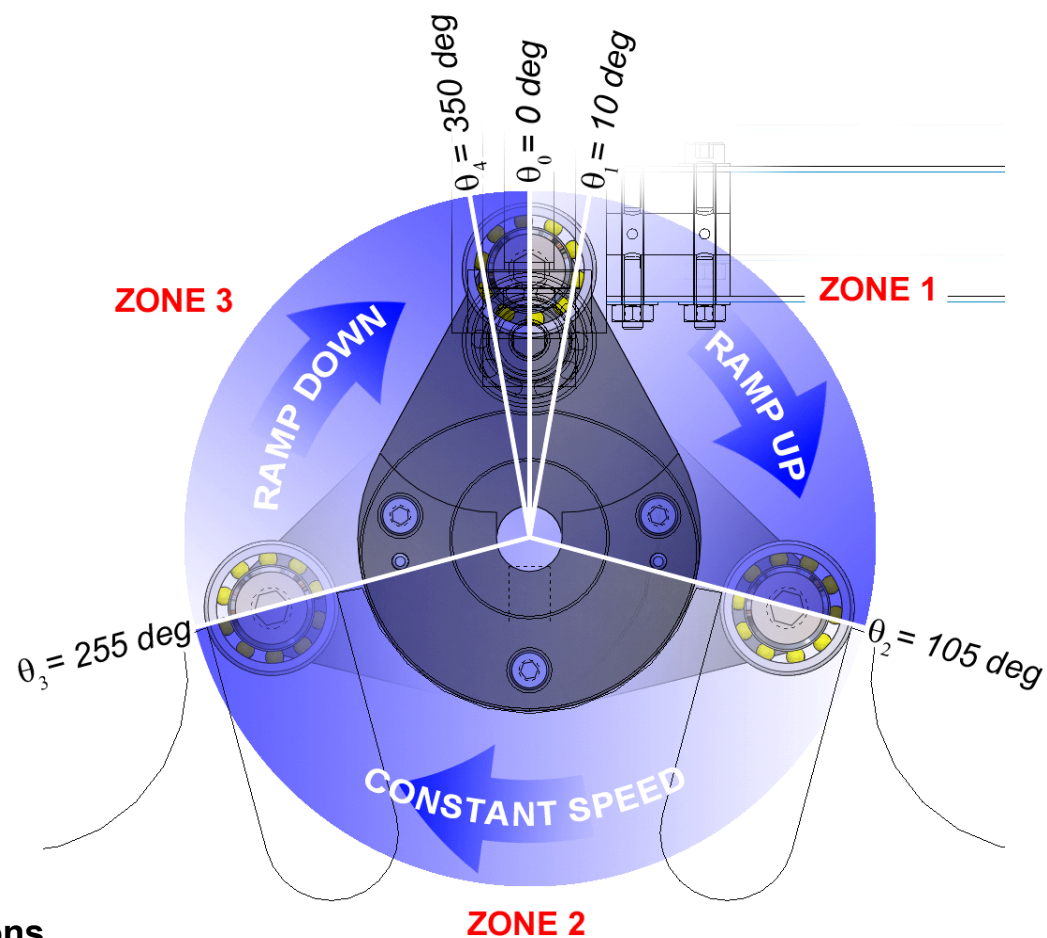
$T_{\text{crank_friction}} := .10\text{in}\cdot\text{lbf}$

Define motor drive shaft bearing friction (torque).

$T_{\text{fric_shaft1}} := T_{\text{crank_friction}}$

Define turret hub bearing friction (torque).

$T_{\text{trt_fric}} := 1\text{in}\cdot\text{lbf}$



Part 1. Cam Move Profile Calculations

This calculations assumes a trapezoidal move profile for the Geneva crank. The crank starts at $\theta_1 = 0 \text{ deg}$, which corresponds to the detent mechanism fully seated in the vee-groove. The crank accelerates from θ_1 to θ_2 with constant acceleration. From θ_2 to θ_3 , the angular velocity is constant. From θ_3 to θ_4 , the crank decelerates back to a stop, completing one cycle of the crank.

$\theta_0 := 0 \text{ deg}$ Starting position of the crank, which corresponds to the detent mechanism seated in the vee-groove.
 $\theta_1 := 10 \text{ deg}$ Cam angle when follower begins to lift detent off of vee-groove.
 $\theta_2 := 105 \text{ deg}$ Position of crank when the detent mechanism completely clears the vee-groove. The crank accelerates to full speed from θ_1 to this position and begin to engage with the geneva wheel to begin cycling the turret to the next position.
 $\theta_3 := 360 \text{ deg} - \theta_2$ $\theta_3 = 255 \cdot \text{deg}$ Position of crank when the geneva has completed a move of the turret to the next position. The crank rotates at constant speed from this position to θ_4 .
 $\theta_4 := 360 \text{ deg} - \theta_1$ $\theta_4 = 350 \cdot \text{deg}$ Cam angle when follower finishes seating the detent onto the vee-groove.
 $\theta_5 := 360 \text{ deg}$ Ending position of the crank, which corresponds to the detent mechanism seated in the vee-groove.
 $\theta := \theta_0..1 \cdot \text{deg}.. \theta_5$ Define range for crank angle.

Calculate the corresponding times for the angular positions above.

$$\omega_{\text{crank_max}} := \frac{1}{t_{\text{cycle}}} (2\theta_5 - \theta_3 + \theta_2) \quad \omega_{\text{crank_max}} = 19 \frac{\text{rev}}{\text{min}} \quad \text{Calculated maximum crank velocity for the desired cycle time}$$

$$t_2 := 2 \cdot \frac{(\theta_2)}{\omega_{\text{crank_max}}} \quad t_2 = 1.842 \text{ s} \quad \text{Calculated time to get to the } \theta_2 \text{ position.}$$

$$t_1 := \sqrt{\frac{2t_2 \cdot \theta_1}{\omega_{\text{crank_max}}}} \quad t_1 = 0.568 \cdot \text{sec}$$

$$t_3 := t_2 + \frac{(\theta_3 - \theta_2)}{\omega_{\text{crank_max}}} \quad t_3 = 3.158 \text{ s} \quad \text{Calculated time to get to the } \theta_3 \text{ position.}$$

$$t_5 := t_3 + 2 \cdot \left(\frac{\theta_5 - \theta_3}{\omega_{\text{crank_max}}} \right) \quad t_5 = 5 \text{ s} \quad \text{Calculated time to get to the } \theta_5 \text{ position.}$$

$$t_4 := t_5 - t_1 \quad t_4 = 4.432 \text{ s}$$

Calculate angular position of crank as a function of time.

$$t := 0s, .01s .. t_5 \quad \text{Define time range}$$

$$\theta_x(t) := \text{if} \left[t < t_2, \frac{\omega_{\text{crank_max}}}{2t_2} \cdot t^2, \frac{\omega_{\text{crank_max}}}{2} \cdot t_2 + \omega_{\text{crank_max}} \cdot (t - t_2) \right]$$

$$\theta_{\text{crank}}(t) := \text{if} \left[t < t_3, \theta_x(t), \frac{\omega_{\text{crank_max}}}{t_5 - t_3} \cdot \left(t_5 \cdot t - \frac{1}{2} \cdot t^2 \right) + \theta_5 - \frac{\omega_{\text{crank_max}}}{t_5 - t_3} \cdot \frac{t_5^2}{2} \right]$$

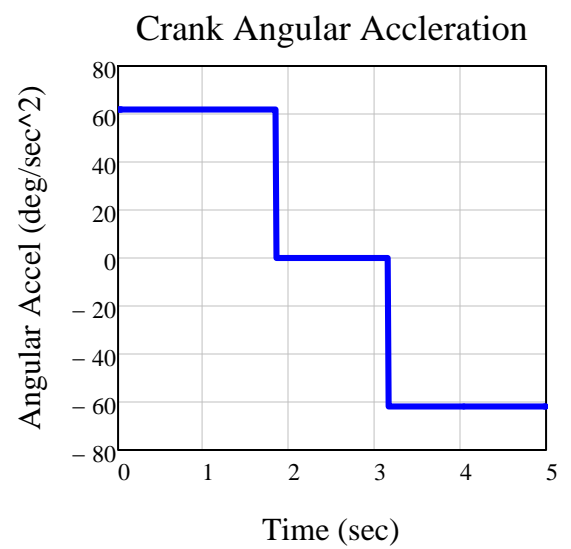
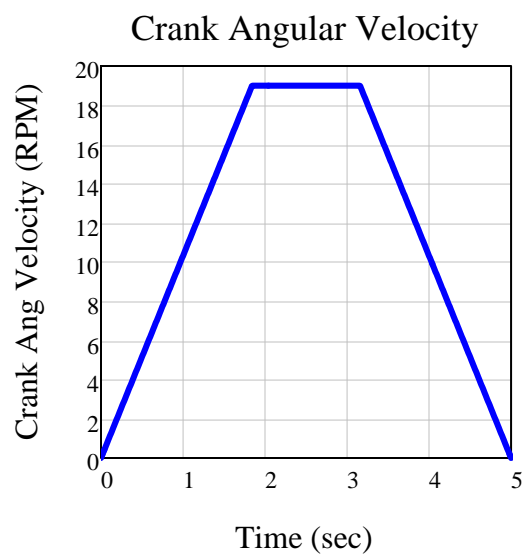
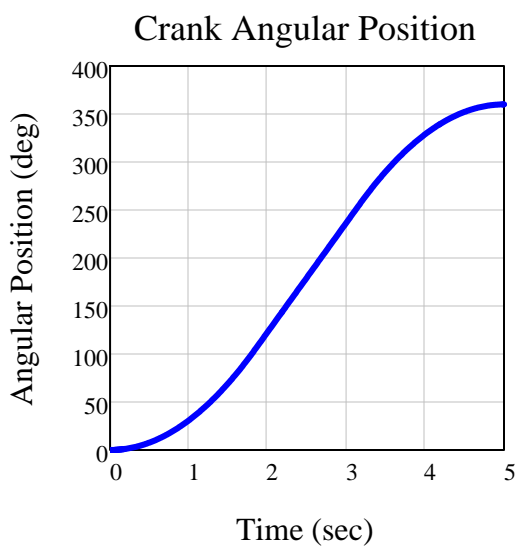
Calculate angular velocity of crank as a function of time.

$$\omega_t(t) := \text{if} \left[t < t_2, \omega_{\text{crank_max}} \cdot \frac{t}{(t_2)}, \omega_{\text{crank_max}} \right]$$

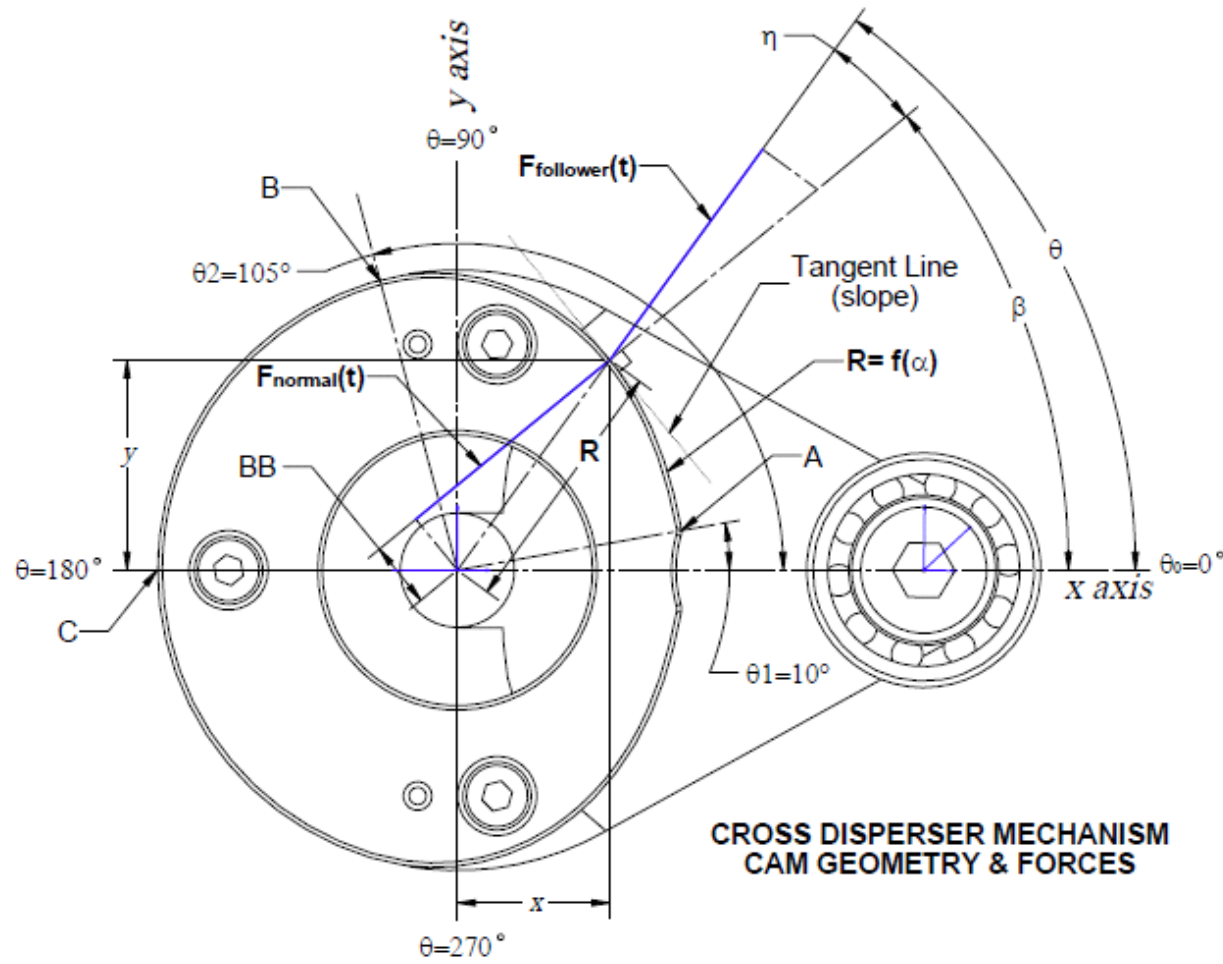
$$\omega_{\text{crank}}(t) := \text{if} \left[t < t_3, \omega_t(t), \omega_{\text{crank_max}} \cdot \frac{(t_5 - t)}{(t_5 - t_3)} \right]$$

Calculate angular acceleration of crank as a function of time.

$$\alpha_{\text{crank}}(t) := \frac{d}{dt} \omega_{\text{crank}}(t) \quad \text{Crank angular acceleration.}$$



Cam Follower Path Calculations



This calculation uses a sinusoidal cam follower path for a smooth transition between the seated to unseated positions of the detent mechanism.

- Lift := 4mm Cam lift (distance from detent fully seated to fully disengaged).
- R₂ := 16.63190451·mm R₂ = 0.654799·in Radius of cam when detent is fully disengaged.
- R₁ := R₂ - Lift R₁ = 0.497319·in Radius of cam when detent is seated.

Define cam radius equation for three segments of cam from 0 deg to theta₂, & theta₂ to 180 deg.

$$R_{zone1}(\theta) := \text{if} \left[\theta < \theta_1, R_1, R_1 + \left(\frac{R_2 - R_1}{2} \right) \cdot \left[1 + \sin \left[\left(\frac{180\text{deg}}{\theta_2 - \theta_1} \right) \cdot (\theta - \theta_1) - 90\text{deg} \right] \right] \right]$$

Radius of cam path in zone 1.

$$R_{zone12}(\theta) := \text{if} (\theta < \theta_2, R_{zone1}(\theta), R_2)$$

Radius of cam path in zones 1 & 2.

$$R_{zone123}(\theta) := \text{if} \left[\theta < \theta_3, R_{zone12}(\theta), R_1 + \left(\frac{R_2 - R_1}{2} \right) \cdot \left[1 + \sin \left[\left(\frac{180\text{deg}}{\theta_4 - \theta_3} \right) \cdot (\theta - \theta_3) + 90\text{deg} \right] \right] \right]$$

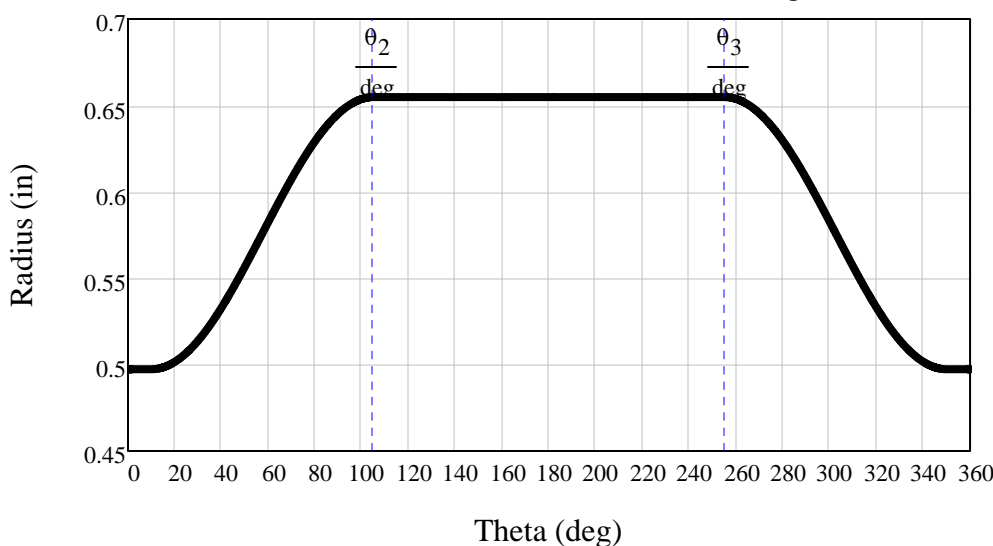
Radius of cam path in zones 1, 2 & 3.

$$R_{zone1234}(\theta) := \text{if} (\theta < \theta_4, R_{zone123}(\theta), R_1)$$

Radius of cam path in zones 1, 2, 3 & 4.

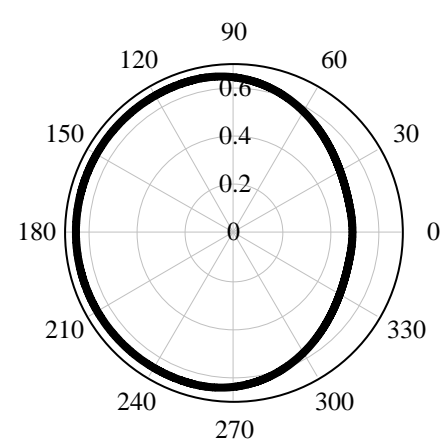
$$R_{cam}(\theta) := R_{zone1234}(\theta)$$

Cam Follower Path vs. Follower Angle



— Follower Path

Cam Follower Path



— Follower Path

$$R_{z1}(t) := \text{if} \left[t < t_1, R_1, R_1 + \left(\frac{R_2 - R_1}{2} \right) \cdot \left[1 + \sin \left[\left(\frac{180\text{deg}}{\theta_2 - \theta_1} \right) \cdot (\theta_{\text{crank}}(t) - \theta_1) - 90\text{deg} \right] \right] \right]$$

Radius of cam path in zone 1.

$$R_{z12}(t) := \text{if} (t < t_2, R_{z1}(t), R_2)$$

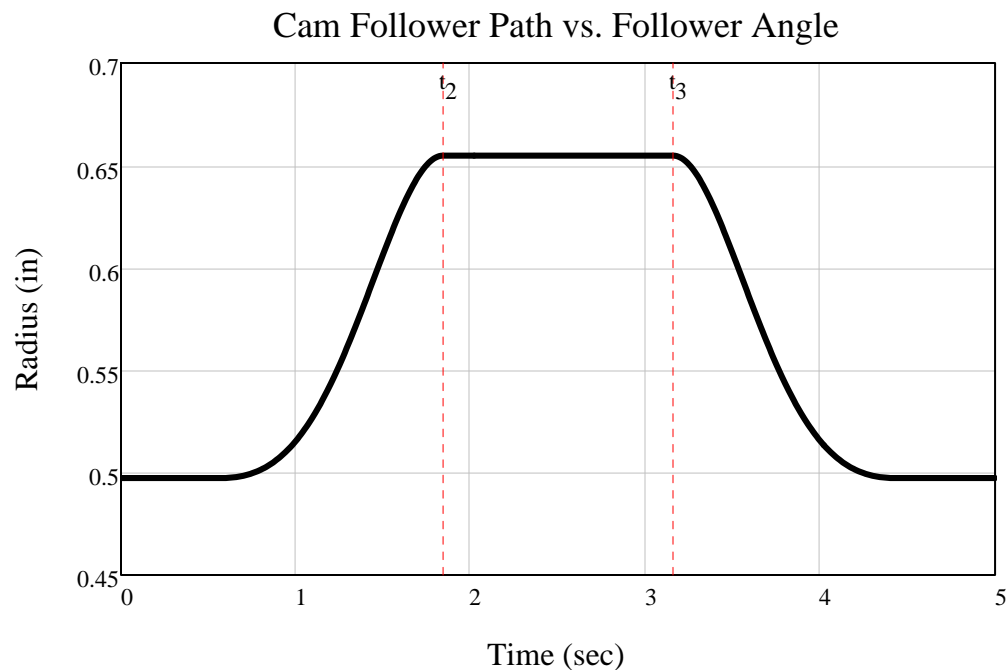
Radius of cam path in zones 1 & 2.

$$R_{z123}(t) := \text{if} \left[t < t_3, R_{z12}(t), R_1 + \left(\frac{R_2 - R_1}{2} \right) \cdot \left[1 + \sin \left[\left(\frac{180\text{deg}}{\theta_4 - \theta_3} \right) \cdot (\theta_{\text{crank}}(t) - \theta_3) + 90\text{deg} \right] \right] \right]$$

Radius of cam path in zones 1, 2 & 3.

$$R_{z1234}(t) := \text{if} (t < t_4, R_{z123}(t), R_1) \quad R_{cm}(t) := R_{z1234}(t)$$

Radius of cam path in zones 1, 2, 3 & 4.



Convert cam follower path to cartesian coordinates

$x(\theta) := R_{cam}(\theta) \cdot \cos(\theta)$ x coordinate
 $y(\theta) := R_{cam}(\theta) \cdot \sin(\theta)$ y coordinate

$x_t(t) := R_{cam}(\theta_{crank}(t)) \cdot \cos(\theta_{crank}(t))$ x coordinate
 $y_t(t) := R_{cam}(\theta_{crank}(t)) \cdot \sin(\theta_{crank}(t))$ y coordinate

R _{cam} (θ) =	·in	x(θ) =	·in	y(θ) =	·in
0.497319		0.497319		0	
0.497319		0.497318		8.679851 · 10 ⁻⁴	
0.497319		0.497316		1.735968 · 10 ⁻³	
0.497319		0.497312		2.603945 · 10 ⁻³	
0.497319		0.497307		3.471914 · 10 ⁻³	
...		

Calculate angle beta. See figure of cam geometry above.

$$\beta(\theta) := \text{if} \left(\theta < \theta_2, 90\text{deg} + \text{atan} \left(\frac{\frac{d}{d\theta} y(\theta)}{\frac{d}{d\theta} x(\theta)} \right), \theta \right) \quad \beta_{1t}(t) := \text{if} \left(t < t_2, 90\text{deg} + \text{atan} \left(\frac{\frac{d}{dt} y_t(t)}{\frac{d}{dt} x_t(t)} \right), \theta_{crank}(t) \right) \quad \beta_t(t) := \text{if} \left(t < t_3, \beta_{1t}(t), 270\text{deg} + \text{atan} \left(\frac{\frac{d}{dt} y_t(t)}{\frac{d}{dt} x_t(t)} \right) \right)$$

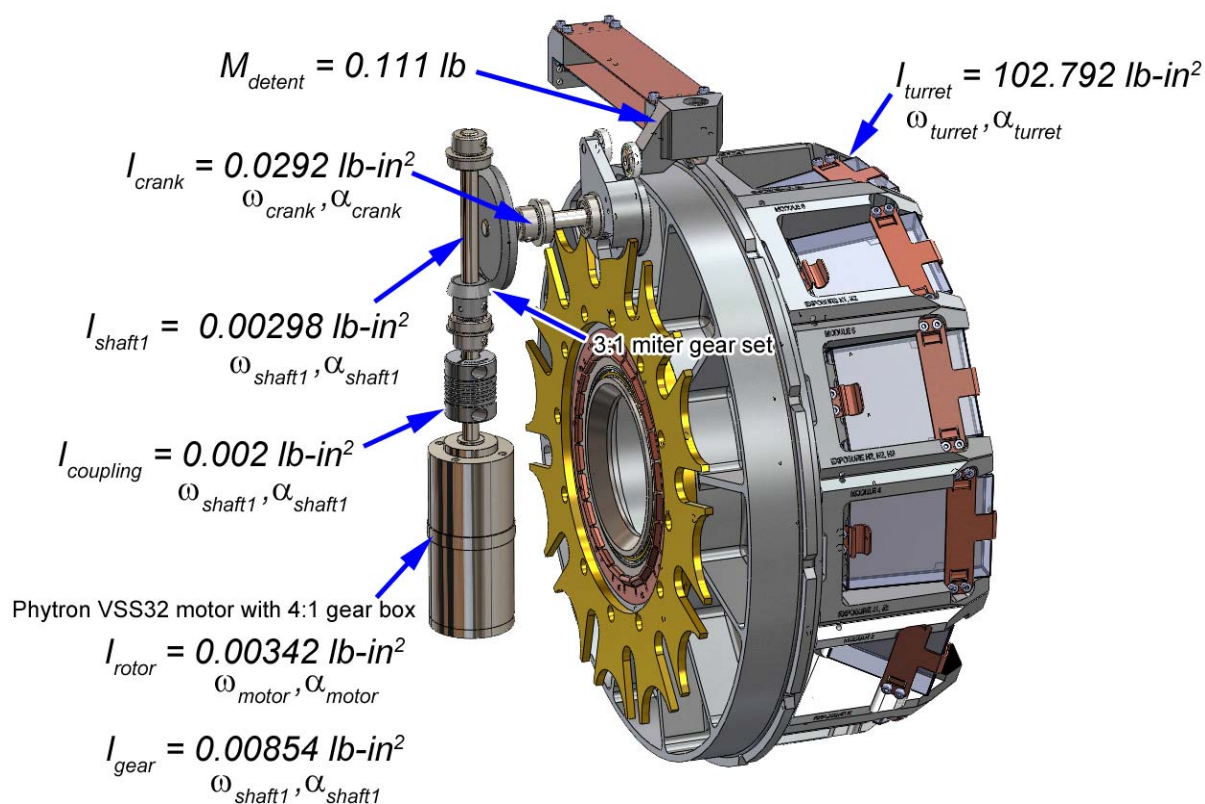
Calculate angle η.

$\eta(\theta) := \theta - \beta(\theta)$ $\eta_t(t) := \theta_{crank}(t) - \beta_t(t)$

Calculate moment arm BB

$BB(\theta) := R_{cam}(\theta) \cdot \sin(\eta(\theta))$ $BB_t(t) := R_{cm}(t) \sin(\eta_t(t))$ Moment arm length.

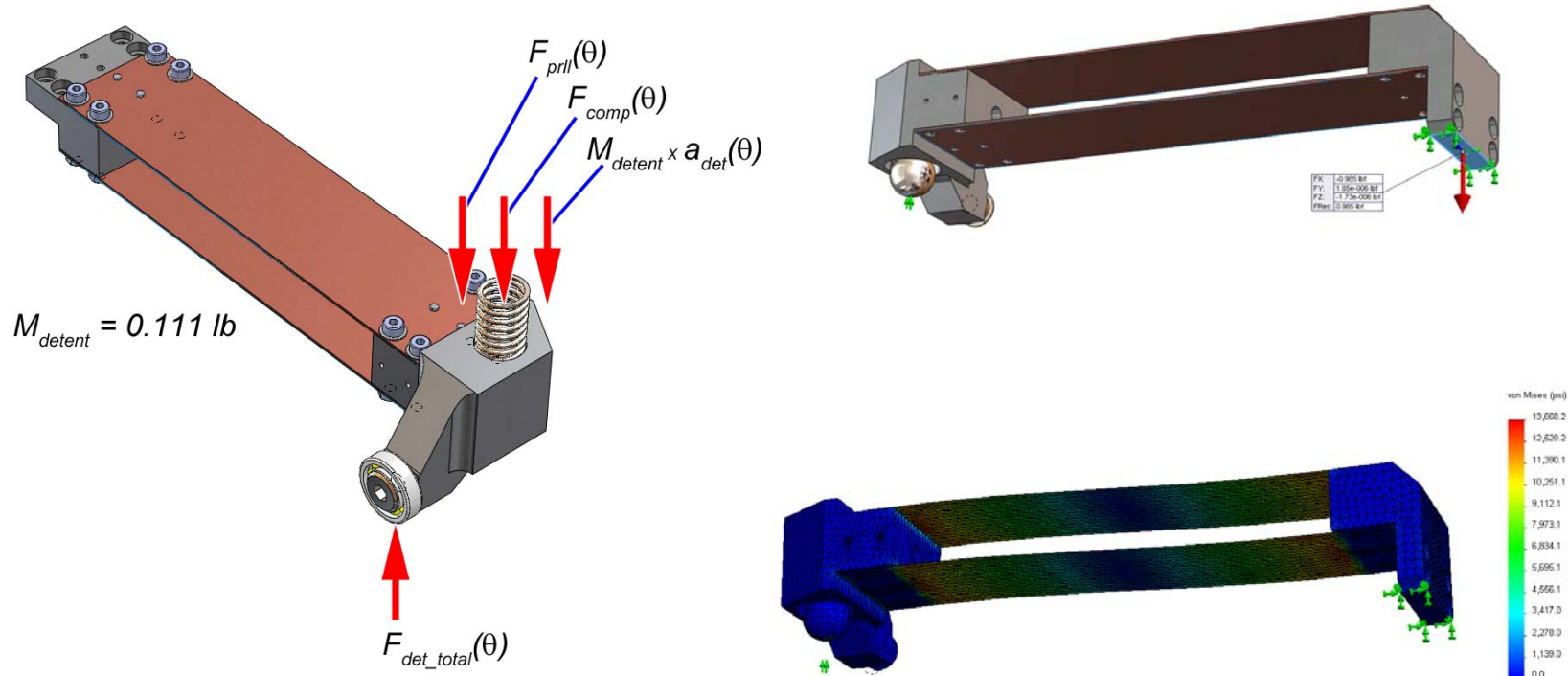
Cross Dispenser Turret Drive Inertias & Masses



Motor & Drive Inertias

$I_{vss32rotor} := .00342 \text{ lb-in}^2$ $I_{mot_gearbox} := .00854 \text{ lb-in}^2$ $I_{coupling} := .002 \cdot (\text{lb-in}^2)$ $I_{shaft1} := .00298 \text{ lb-in}^2$ $I_{crank} := .0292 \text{ lb-in}^2$ $I_{turret} := 102.972 \text{ lb-in}^2$

Detent Parallel Spring Flexure Forces & Mass



Calculate loads on crank in Zone 1 & Zone 3. (Cam angle from 0 degrees to θ_2 and θ_3 to θ_5).

The crank is subjected to the following loads in zones 1 & 3.

1. Parallel spring flexure loads from detent mechanism.
2. Preload force from detent mechanism compression spring.
3. Inertial forces caused by accelerating mass of detent mechanism when crank is moving.
4. Inertial force (torque) on crank shaft from accelerating crank ($I \times \alpha$).
5. Frictional forces (torque) from bearings.

1. Calculate Parallel Spring Flexure Load.

$$k_{prll} := .9985 \frac{\text{lbf}}{\text{Lift}} \quad k_{prll} = 6.34 \frac{\text{lbf}}{\text{in}} \quad \text{The spring rate was determined by finite element analysis to be 1.47 lbf over a 4mm deflection.}$$

$$F_{prll}(t) := k_{prll} (R_{cm}(t) - R_1) \quad \text{Component of detent spring force provided by the parallel spring flexure.}$$

2. Calculate detent mechanism preload compression spring forces.

$$L_{preload} := L_{free} - \frac{F_{pre}}{k_{spr}} \quad L_{preload} = 1.523 \cdot \text{in} \quad \text{Length at initial preload (seated in vee-groove).}$$

$$L_{maxcomp} := L_{preload} - \text{Lift} \quad L_{maxcomp} = 1.365 \cdot \text{in} \quad \text{Length at max compression (unseated)}$$

$$F_{comp}(t) := F_{pre} + (k_{spr}) \cdot (R_{cm}(t) - R_1) \quad \text{Component of detent spring force provided by the compression spring.}$$

3. Calculate the inertial forces caused by accelerating mass of detent mechanism when crank is moving.

$$v_{detent}(t) := \left(\frac{d}{dt} R_{cm}(t) \right) \quad \text{Linear velocity of parallel spring flexure when climbing the cam.}$$

$$a_{detent}(t) := \left(\frac{d^2}{dt^2} R_{cm}(t) \right) \quad \text{Linear acceleration of parallel spring flexure when climbing the cam.}$$

$$M_{detent} := .111 \text{lb} \quad \text{Mass of moving portion of parallel spring flexure.}$$

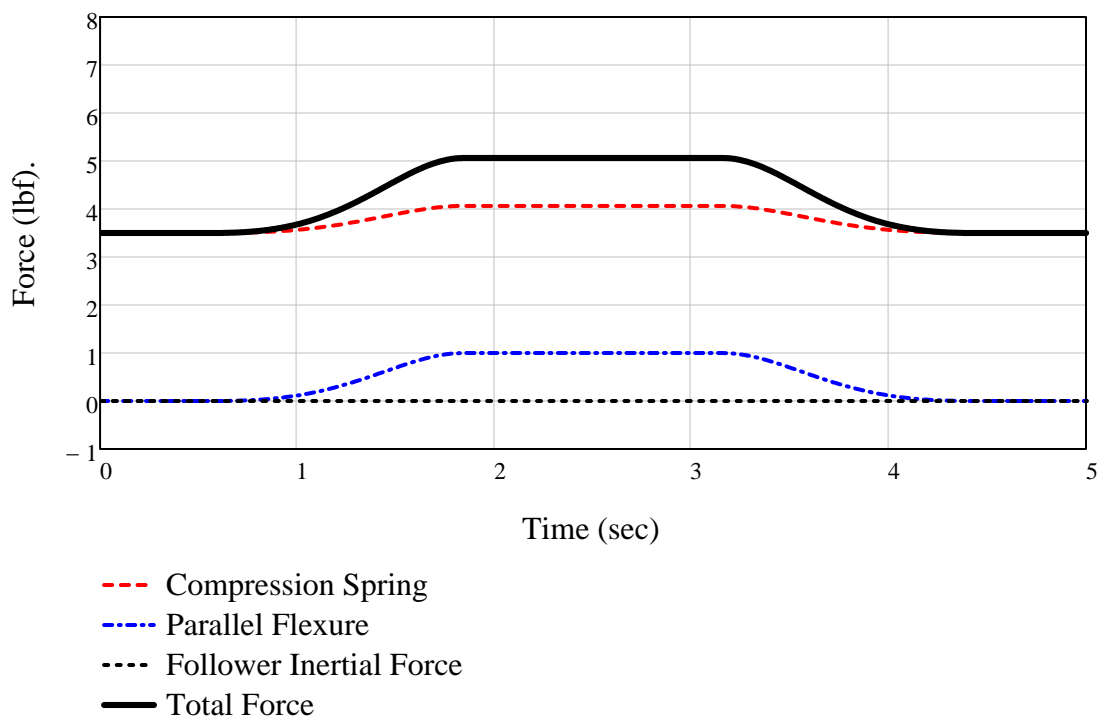
$$F_{detent_inertial}(t) := M_{detent} \cdot a_{detent}(t) \quad \text{Force component of parallel spring flexure due to the acceleration when climbing cam.}$$

Calculate total force on cam from follower loads.

$$F_{follower}(t) := F_{comp}(t) + F_{prll}(t) + F_{detent_inertial}(t) \quad \text{Total detent spring force.}$$

$$F_{normal}(t) := \frac{F_{follower}(t)}{\cos(\eta_t(t))} \quad \text{Reaction force to the spring in the normal direction}$$

Cam Follower Forces vs. Time



Calculate motor speed and acceleration

$$\omega_{\text{shaft1}}(t) := \omega_{\text{crank}}(t) \cdot \text{Ratio}_{\text{mitergear}} \quad \alpha_{\text{shaft1}}(t) := \text{Ratio}_{\text{mitergear}} \cdot \alpha_{\text{crank}}(t)$$

$$\omega_{\text{motor}}(t) := \omega_{\text{shaft1}}(t) \cdot \text{Ratio}_{\text{mot_gearbox}} \quad \alpha_{\text{motor}}(t) := \alpha_{\text{shaft1}}(t) \cdot \text{Ratio}_{\text{mot_gearbox}}$$

**Calculate loads on crank in Zone 2
(Cam angle from θ_2 to θ_3).**

The crank is subjected to the following loads in zone 2.

1. Inertial force (torque) of accelerating turret ($I \times \alpha$) as it cycles from one position to the next.
2. Frictional forces (torque) from turret bearings.

$R_{\text{crank}} := 25.88190451\text{mm}$ Radius of Geneva crank

$D_{\text{ctr}} := 100\text{mm}$ Distance between centers of Geneva wheel and crank

$$\theta_{\text{gen2}}(t) := \text{if} \left(t < t_2, \text{atan} \left(\frac{R_{\text{crank}} \cdot \sin(180\text{deg} - \theta_2)}{D_{\text{ctr}} - R_{\text{crank}} \cdot \cos(180\text{deg} - \theta_2)} \right), \text{atan} \left(\frac{R_{\text{crank}} \cdot \sin(180\text{deg} - \theta_{\text{crank}}(t))}{D_{\text{ctr}} - R_{\text{crank}} \cdot \cos(180\text{deg} - \theta_{\text{crank}}(t))} \right) \right)$$

$$\theta_{\text{geneva}}(t) := \text{if} \left(t < t_3, \theta_{\text{gen2}}(t), \text{atan} \left(\frac{R_{\text{crank}} \cdot \sin(180\text{deg} - \theta_3)}{D_{\text{ctr}} - R_{\text{crank}} \cdot \cos(180\text{deg} - \theta_3)} \right) \right)$$

$R_{\text{geneva}}(t) := \frac{R_{\text{crank}} \cdot \sin(180\text{deg} - \theta_{\text{crank}}(t))}{\sin(\theta_{\text{geneva}}(t))}$ Geneva radius as a function of time

$\omega_{\text{geneva}}(t) := \frac{d}{dt} \theta_{\text{geneva}}(t) \quad \alpha_{\text{geneva}}(t) := \frac{d^2}{dt^2} \theta_{\text{geneva}}(t)$

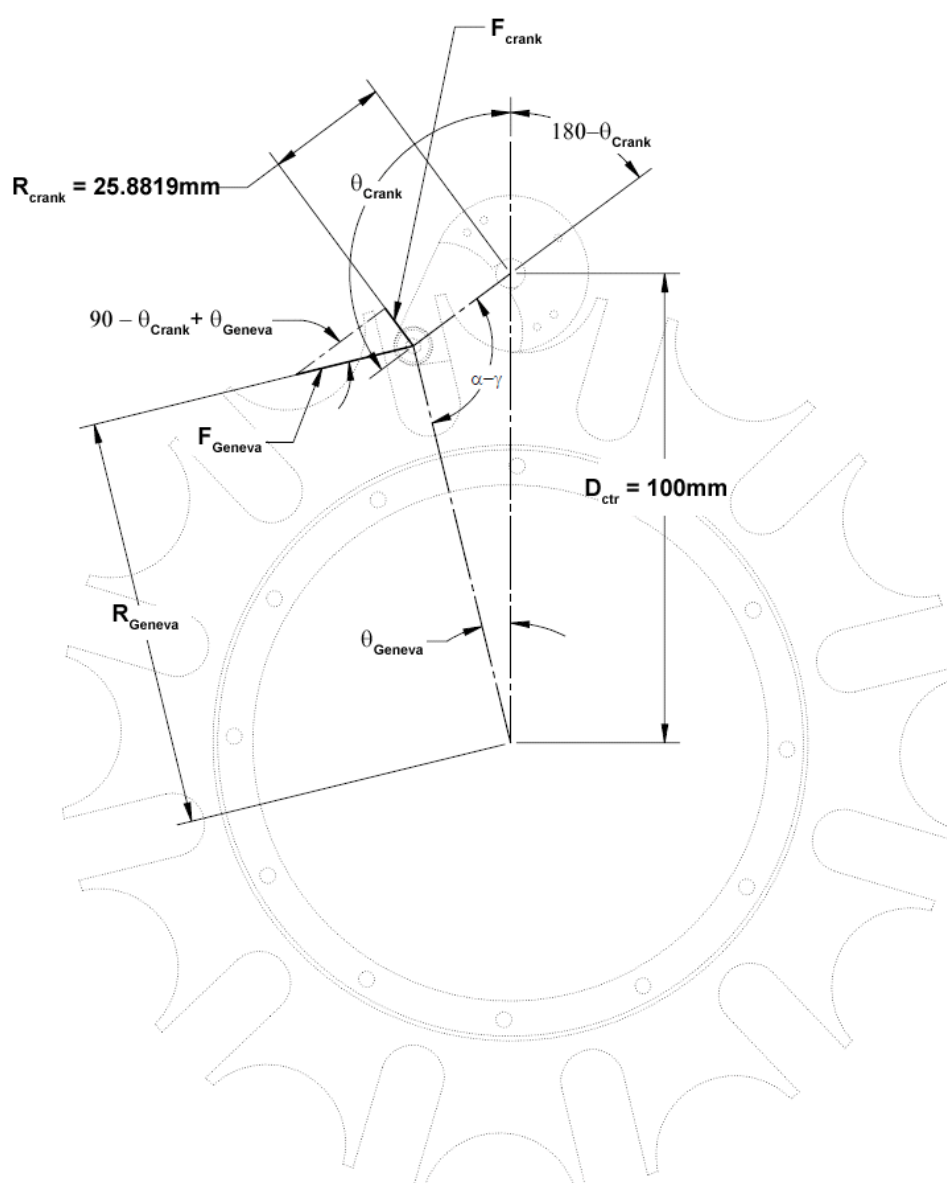
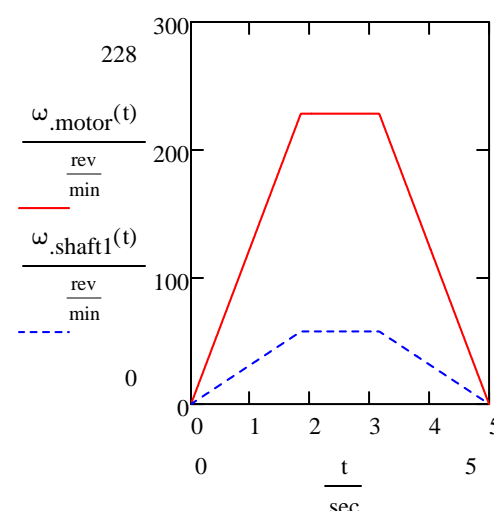
$T_{\text{turret}}(t) := I_{\text{turret}} \cdot \alpha_{\text{geneva}}(t) + T_{\text{trt_fric}}$

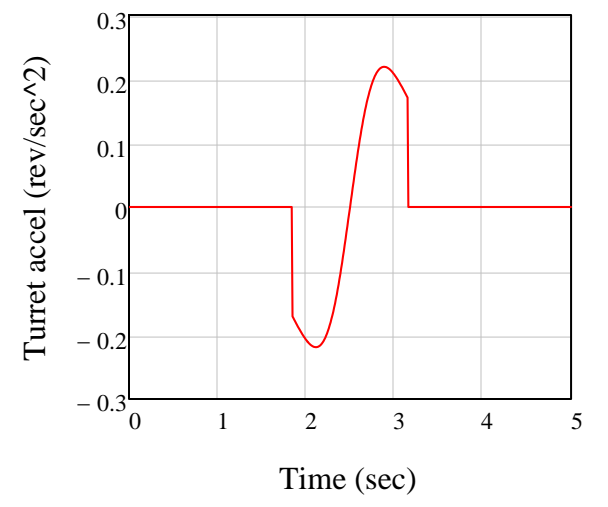
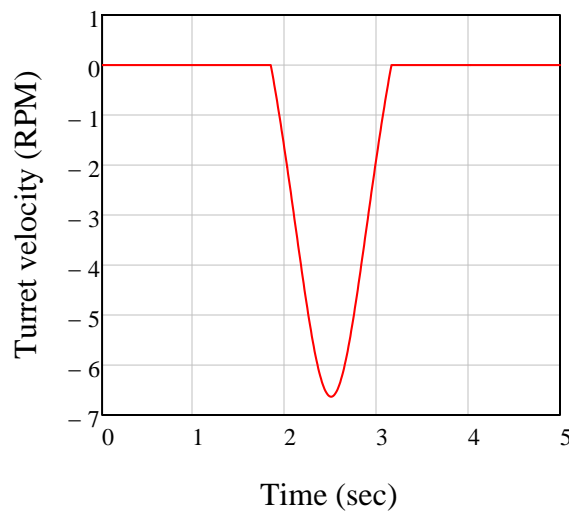
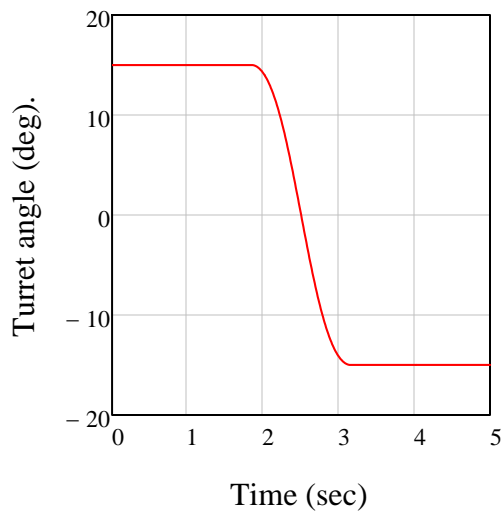
$F_{\text{tan_trt_1}}(t) := \text{if} \left(t < t_2, 0\text{lbf}, \frac{T_{\text{turret}}(t)}{R_{\text{geneva}}(t)} \right)$

$F_{\text{tan_turret}}(t) := \text{if} \left(t < t_3, F_{\text{tan_trt_1}}(t), 0\text{lbf} \right)$

$F_{\text{tan_crank}}(t) := F_{\text{tan_turret}}(t) \cdot \sin(\theta_{\text{crank}}(t) - \theta_{\text{geneva}}(t) - 90\text{deg})$

$T_{\text{crank_z2}}(t) := F_{\text{tan_crank}}(t) \cdot R_{\text{crank}}$



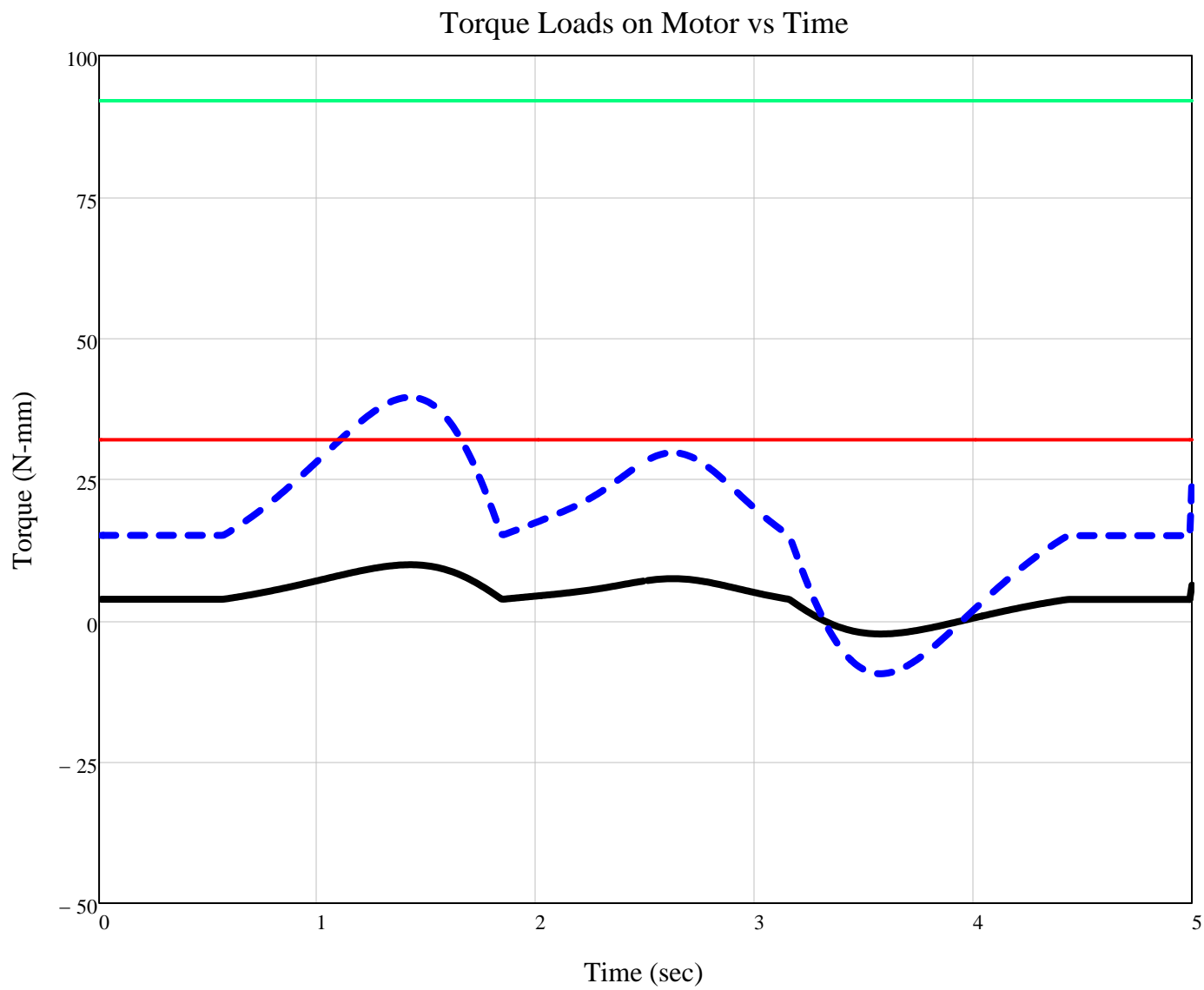


$$T_{crank}(t) := F_{normal}(t) \cdot BB_t(t) + (I_{crank}) \cdot \alpha_{crank}(t) + T_{crank_friction} + F_{tan_crank}(t) \cdot R_{crank}$$

$$T_{shaft1}(t) := (I_{shaft1} + I_{coupling} + I_{mot_gearbox}) \cdot \alpha_{shaft1}(t) + \frac{T_{crank}(t)}{Ratio_{mitergear}} + T_{fric_shaft1}$$

$$T_{mot_gearbox}(t) := I_{vss32rotor} \cdot \alpha_{motor}(t) + \frac{T_{shaft1}(t)}{Ratio_{mot_gearbox}}$$

$$T_{mot_nogrbox}(t) := I_{vss32rotor} \cdot \alpha_{shaft1}(t) + T_{shaft1}(t)$$



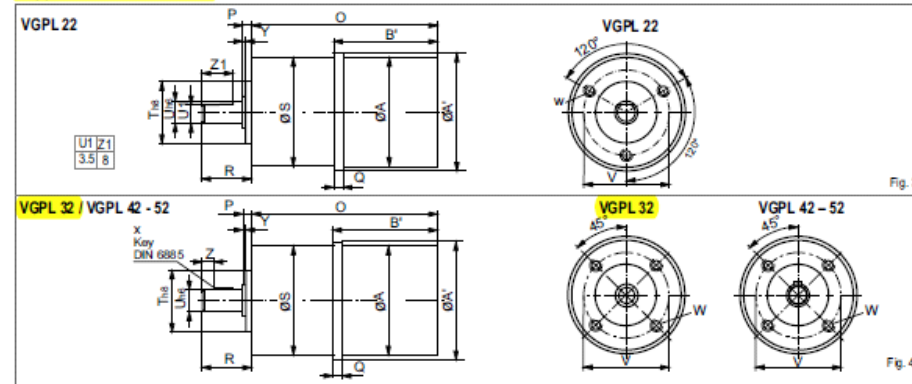
- Drive Loads (with motor gearbox)
- - - Drive Loads (without motor gearbox)
- VSS 32 stepper max torque
- VSS 42 stepper max torque

Size	Standard Part 200 steps/rev.	Electrical Characteristics									Mechanical Characteristics					
		Parallel Windings ⁶⁾ (4 leads)			Series Windings (4 leads)			Unipolar Connection (5 or 6 leads)			4) 5)	5)		Loads		Mass
		1)	2)	3)	1)	2)	3)	1)	2)	3)	Holding Torque	Detent Torque	Rotor Inertia	axial	radial	
		V/Ph	R/Ph	L/Ph	I/Ph	R/Ph	L/Ph	V/Ph	R/Ph	L/Ph						mNm
A	Ω	mH	A	Ω	mH	A	Ω	mH	mNm	mNm	kg cm ²	N	N	kg		
19	VSS 19.200.0.3	0.3	6	2.2	0.15	24	8.8	0.21	12	2.2	3.4	0.9	0.0009	3	3	0.05
	VSS 19.200.0.6	0.6	2.1	0.55	0.3	8.4	2.2	0.42	4.2	0.55						
	VSS 19.200.1.2	1.2	0.625	0.15	0.6	2.5	0.6	0.84	1.25	0.15						
25	VSS 25.200.0.3	0.3	12	6	0.15	48	24	0.21	24	6	12	2	0.002	5	5	0.08
	VSS 25.200.0.6	0.6	3.25	1.5	0.3	13	6	0.42	6.5	1.5						
	VSS 25.200.1.2	1.2	0.95	0.4	0.6	3.8	1.6	0.84	1.9	0.4						
32	VSS 32.200.0.6	0.6	4.6	5.3	0.3	18.4	21.2	0.42	9.2	5.3	45	3	0.01	5	15	0.17
	VSS 32.200.1.2	1.2	1.25	1.2	0.6	5.0	4.8	0.84	2.5	1.2						
	VSS 32.200.2.5	2.5	0.3	0.3	1.25	1.2	1.2	1.75	0.6	0.3						
32	VSS 33.200.0.6	0.6	7.5	9.3	0.3	30	37.2	0.42	15	9.3	68	3.3	0.018	5	15	0.26
	VSS 33.200.1.2	1.2	1.9	2.2	0.6	7.4	8.8	0.84	3.8	2.2						
	VSS 33.200.2.5	2.5	0.47	0.6	1.25	1.88	2.4	1.75	0.94	0.6						
42	VSS 42.200.0.6	0.6	7.25	11	0.3	29	44	0.42	14.5	11	130	5	0.045	20	40	0.35
	VSS 42.200.1.2	1.2	1.7	3	0.6	6.8	12	0.84	3.4	3						
	VSS 42.200.2.5	2.5	0.34	0.7	1.25	1.36	2.8	1.75	0.68	0.7						
42	VSS 43.200.0.6	0.6	9.5	22.9	0.3	38	91.6	0.42	19	22.9	235	7	0.077	20	40	0.52
	VSS 43.200.1.2	1.2	2.6	5.2	0.6	10.4	20.8	0.84	5.2	5.2						
	VSS 43.200.2.5	2.5	0.5	1.2	1.25	2	4.8	1.75	1	1.2						
52	VSS 52.200.1.2	1.2	2.65	7	0.6	10.6	28	0.84	5.3	7	405	13	0.15	25	70	0.72
	VSS 52.200.2.5	2.5	0.6	1.6	1.25	2.4	6.4	1.75	1.2	1.6						
	VSS 52.200.5	5	0.165	0.4	2.5	0.66	1.6	3.5	0.33	0.4						
56	VSS 57.200.1.2	1.2	3.9	9.5	0.6	15.6	38	0.84	7.8	9.5	630	50	0.24	40	80	0.99
	VSS 57.200.2.5	2.5	0.8	2.4	1.25	3.2	9.6	1.75	1.6	2.4						
	VSS 57.200.5.0	5	0.25	0.8	2.5	1	3.2	3.5	0.5	0.8						

1) V/Ph: Phase current
 2) R/Ph: Phase resistance
 3) L/Ph: Phase inductance
 4) Holding torque in bipolar mode with parallel windings, 2 phases ON at rated current
 5) 7 mNm ~ 1 inoz
 6) Standard wiring at delivery (if no wiring mode was given in the order)
 Design voltage 42 V_{DC} (operation with SELV type supply)
Bold letters = Preferred types

Important:
 All values given above were measured at room temperature of 25 °C (77 °F) and atmospheric pressure.

Stepper Motor with VGPL Gear



Gear	Stepper Motor	Dimensions in mm															Mass (motor + gear)				
		stages															stages				
		A	A'	B'	O	P	Q	R	S	T	U	V	W	X	Y	Z	1	2	3		
VGPL 22	VSS 19	19	22	29	50	57	64	3	4.5	15	22	12	4	16	M 2.5x4	-	1	-	0.1	0.13	0.15
	VSS 25	25	25.5	33.5	53.5	60.5	67.5	5	5	15	22	12	4	16	M 2.5x4	-	1	-	0.13	0.15	0.18
VGPL 32	VSS 32	32	33	40.5	69.5	78.5	87.5	4	5	20	32	20	6	26	M 3x5	-	1	-	0.31	0.35	0.42
	VSS 33	32	33	59.5	88.5	97.5	106.5	4	5	20	32	20	6	26	M 3x5	-	1	-	0.39	0.44	0.51
VGPL 42	VSS 42	42	43	58	93	105.5	118	4	7	22.5	42	25	8	32	M 4x8	3x3x4	1	2.25	0.63	0.7	0.8
	VSS 43	42	43	73	108	120.5	133	4	7	22.5	42	25	8	32	M 4x8	3x3x4	1	2.25	0.8	0.85	0.93
VGPL 52	VSS 52	52	53	68.5	109.5	124	138.5	4	6.7	24	52	32	12	40	M 5x8	4x4x6	1	2	1.2	1.4	1.5
	VSS 57	56.4	57	78	119	133.5	148	4	9	24	52	32	12	40	M 5x8	4x4x6	1	2	1.4	1.5	1.6

Gear	Stepper Motor	Stages	Gear 1) backlash arc-min	Max. torque Nm	Gear inertia kg cm ²	Radial load ²⁾ N	Axial load N	Efficiency %	Reduction ratio		
									4:1	5:1	7:1
VGPL 22	VSS 19	1	20'	0.1	0.005			94	4:1	5:1	7:1
	VSS 25	2	35'	0.5	0.015	30	24	85	16:1	20:1	28:1
VGPL 32	VSS 32	1	6'	0.4	0.025	80	60	94	4:1	4.5:1	5.2:1
	VSS 33	2	10'	2	0.04	80	60	85	16:1	18:1	25:1
VGPL 42	VSS 42	1	6'	0.7	0.05	150	120	94	4:1	5:1	6:1
	VSS 43	2	10'	4	0.08	150	120	85	14:1	16:1	20:1
VGPL 52	VSS 52	1	6'	1.5	0.1	250	200	94	4:1	4.5:1	5.2:1
	VSS 57	2	10'	10	0.2	250	200	85	16:1	25:1	29:1

1) no load 2) center of shaft

