James Webb Space Telescope Deployment Brushless DC Motor Characteristics Analysis

Anh N. Tran*

Abstract

A DC motor's performance is usually characterized by a series of tests, which are conducted by pass/fail criteria. In most cases, these tests are adequate to address the performance characteristics under environmental and loading effects with some uncertainties and decent power/torque margins. However, if the motor performance requirement is very stringent, a better understanding of the motor characteristics is required.

The purpose of this paper is to establish a standard way to extract the torque components of the brushless motor and gear box characteristics of a high gear ratio geared motor from the composite geared motor testing and motor parameter measurement. These torque components include motor magnetic detent torque, Coulomb torque, viscous torque, windage torque, and gear tooth sliding torque.

The Aerospace Corp bearing torque model [1] and MPB torque models [2] are used to predict the Coulomb torque of the motor rotor bearings and to model the viscous components. Gear tooth sliding friction torque is derived from the dynamo geared motor test data. With these torque data, the geared motor mechanical efficiency can be estimated and provide the overall performance of the geared motor versus several motor operating parameters such as speed, temperature, applied current, and transmitted power.

JWST Deployment Mechanism General Description

There are fourteen brushless DC motors that are used for the James Webb Space Telescope deployment mechanisms. Three of these motors are used in the Optical Telescope Elements deployment, which is the study of this paper. Most of these motors demonstrate excellent performance during flight tests. In order to assess the deploying margin on these systems, the geared motor performance characteristics, particularly the power losses, need to be estimated. Unfortunately, most of the tests for these motors are performed at the geared motor level. Therefore, the losses at the motor and the gearbox levels are not available. In order to extract these individual components, a geared motor model is developed. The model, then, is matched with the composite measured data to derive the individual torque components. The overview of the geared motor is shown in Figure 1.

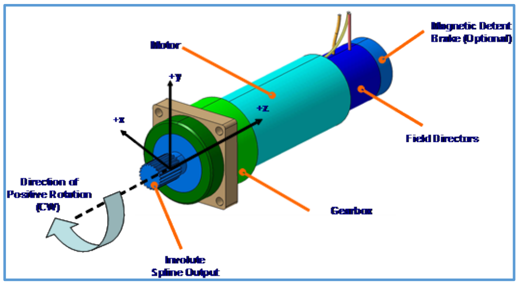

Figure 1. JWST Deployment Brushless DC Geared Motor Overview

* Northrop Grumman Aerospace Systems, Redondo Beach, CA

Proceedings of the 43rd Aerospace Mechanisms Symposium, NASA Ames Research Center, May 4-6, 2016

JWST Deployment Brushless DC Motor Parameters

The geared motor consists of a 3-phase, 4-pole, permanent-magnet brushless DC motor with an add-on resolver for position, and a 4608:1 planetary gearbox (4-stage). The detailed geared motor parameters are shown in Figure 2.

Number of Poles	4
Number of Phases	3
Motor Rotor Inertia	1.E-4 oz-in-s^2 (0.77 µN-m-s^2)
Back EMF Constant (ambient), kb	2.11 mV/rpm
Winding Resistance (leg-to-leg) (ambient)	1.64 ohm
Winding Inductance (leg-to-leg) (ambient)	605 µhenry
Motor Torque Constant, kb	2.85 in-oz/A (0.020 N-m/A)
Operating Gearbox Shaft Speed	0.1 rpm
Gearbox Speed Reduction Ratio	4608
Maximum Applied Current	2 A

Figure 2. PMSA Geared Motor Parameters

Brushless DC Motor General Equations

The simplified brushless DC motor torque governing equations are expressed as follows:

$$V = L\frac{dI}{dt} + RI + k_b\dot{\theta} \tag{1}$$

$$M\ddot{\theta} = \left(k_T I - k_{vm}\dot{\theta}^{.667} - k_{vg}\dot{\theta}^{.667} - k_w\dot{\theta}^2 - T_{cm} - T_{cg} - T_d\right) * \left(1 - k_{sg}\right) - \frac{T_l * 16}{GR} \tag{2}$$

where

V = voltage applied to the motor, volt
L = motor inductance, henry
I = net current through the motor windings, ampere
k_T = motor torque constant, in-oz/A
k_b = motor back emf constant, V/(rpm)
k_{vm} = motor rotor bearing viscous torque constant, in-oz/(rpm)$^{(2/3)}$
k_w = motor rotor windage torque constant, in-oz/(rpm)^2
k_{vg} = gearbox viscous torque constant, in-oz/(rpm)$^{(2/3)}$
T_l = geared motor output torque, in-lb
T_{cm} = motor rotor bearing Coulomb torque, in-oz
T_{cg} = gearbox Coulomb torque, in-oz
k_{sg} = gear tooth sliding coefficient
M = motor rotor moment of inertia, oz-in-sec^2
T_d = motor magnetic cogging or detent torque, in-oz
GR = gearbox gear ratio

In this equation, the motor and the gear box viscous torques are modelled using the MPB ball bearing viscous torque equation. This non-linear viscous torque model showed a remarkable comparison with the measurement data on geared motor spin-down tests reported by Tran and Halpin [3].

There are seven unknowns in the geared motor torque equations, which are listed as follows:

k_{vm} = motor rotor bearing viscous torque constant, in-oz/(rpm)$^{(2/3)}$
k_w = motor rotor windage torque constant, in-oz/(rpm)^2
k_{vg} = gearbox viscous torque constant, in-oz/(rpm)$^{(2/3)}$
T_{cm} = motor rotor bearing Coulomb torque, in-oz
T_{cg} = gearbox Coulomb torque, in-oz
k_{sg} = gear tooth sliding coefficient
T_d = motor magnetic cogging or detent torque, in-oz

There are only five usable sets of motor torque test data available. Two unknowns need to be estimated. Among these unknowns, for this particular application, the motor windage torque appears to be small and is predicted using the model developed by James E. Vkzncik [4] as shown Equation 3.

$$T_w = \pi C_d \rho R^4 \omega^2 L \qquad (3)$$

where

Cd = skin friction coefficient. It can be determined by solving the below equation with a known Reynolds number

$\frac{1}{\sqrt{C_d}} = 2.04 + 1.768\ln(Re\sqrt{C_d}\,)$
Re = Reynolds number = $\frac{Rt\omega}{\vartheta}$
ω = rotor speed, rad/sec
υ = kinematic viscosity of air = 1.36e-4 ft^2/s (12.6 mm^2/s)
ρ = air density = 0.0765 lbm/ft^3 (1.225 kg/m^3)
t = rotor/stator radial gap thickness = 0.002 in (0.05 mm)
R= rotor radius = 0.5 in (13 mm)
L= rotor length = 1 in (25 mm)

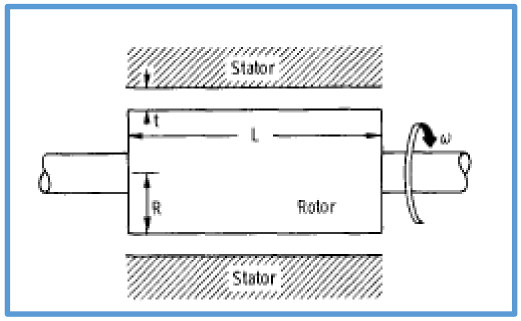

Figure 3. Motor Rotor Parameters

The skin friction coefficient and the windage torque versus rotor speed in air at room temperature and 70% relative humidity are plotted in Figure 4. From Figure 4, it is observed that the windage torque for this particular motor application is very small and can be ignored in the subsequent calculation.

The other unknown, k_{sg}, gear tooth sliding coefficient is derived from the geared motor stalled torque vs. current test and the geared motor dynamo test, which are shown later.

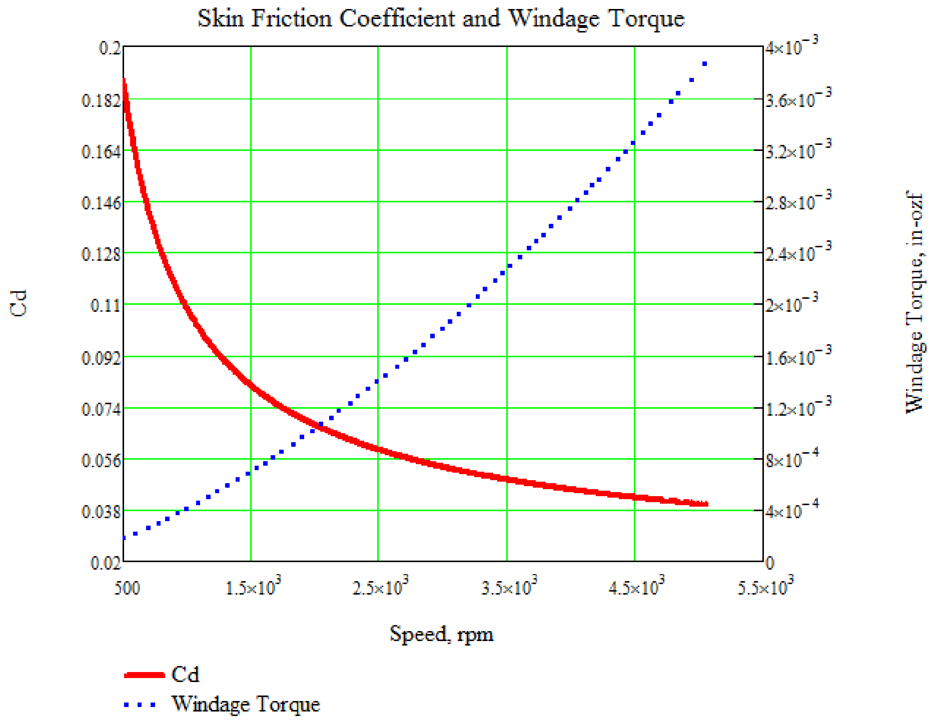

Figure 4. Motor Skin Friction Coefficient and Windage Torque as a Function of Rotor Speed

The rest of the five unknowns are extracted from the series of motor and geared motor tested conducted at the motor vendors and NGC listed as follows:

1. Motor No-Load Synchronous Test
   Parameters:
   Speed = 1800 rpm; no load; motor only; applied voltage, V: 3.0 volts; motor inductance, L=605 µHenry; commutation frequency, f = 1000 Hz; motor winding resistance, R = 1.64 Ω; kt = 2.85 in-oz/A (2.01 N-cm/A); kb=0.0021 V/rpm
   Torque governing equation:
   $$\left(k_T I - T_{cm} - T_d - k_{vm}\dot{\theta}^{.667}\right) = \left(k_T \frac{V - k_b\dot{\theta}}{\sqrt{R^2 + (L*f*2*\pi)^2}} - T_{cm} - T_d - k_{vm}\dot{\theta}^{.667}\right) = 0 \quad (4)$$
2. Motor Dynamo Test
   Parameters:
   Speed = 461 rpm; motor load torque, T_{ml} = 3.0 in-oz (2.1 N-cm); applied current, I = 1.2 A
   Torque governing equation:
   $$\left(k_T I - T_{cm} - T_d - k_{vm}\dot{\theta}^{.667}\right) = T_{ml} \quad (5)$$
3. Geared Motor No-Load Synchronous Test
   Parameters:
   Speed = 1797 rpm; no load; geared motor; applied voltage, V: 3.58 volts; motor inductance, L=605 µHenry; commutation frequency, f = 1000 Hz; motor winding resistance, R = 1.64 Ω; kt = 2.85 in-ozf/A (2.01 N-cm/A); kb=0.0021 V/rpm
   Torque governing equation:
   $$\left(k_T I - T_{cm} - T_{cg} - T_d - k_{vm}\dot{\theta}^{.667} - k_w\dot{\theta}^2 - k_{vg}\dot{\theta}^{.667}\right) = \left(k_T \frac{V - k_b\dot{\theta}}{\sqrt{R^2 + (L*f*2*\pi)^2}} - T_{cm} - T_{cg} - T_d - k_{vm}\dot{\theta}^{.667} - k_{vg}\dot{\theta}^{.667}\right) = 0 \quad (6)$$

4. Geared Motor Stalled Torque from the Stalled Torque vs Current Test in Figure 8
   Parameters:
   Speed = 0 rpm; free current, I = 0.161 A; motor winding resistance, R = 1.64 Ω; kt = 2.85 in-oz/A (2.01 N-cm/A)
   Torque governing equation:
   $$\left(k_T I - T_{cm} - T_{cg} - T_d\right) = 0 \tag{7}$$
5. The motor bearing Coulomb torque is predicted based on the Aerospace Corporation BRGS10C with the friction coefficient selected to be 0.15 for this bearing size and precision class with the measured preload of 2.5 lb (11 N). The bearing parameters and the output are shown in Figure 5. The predicted motor bearing Coulomb torque of 0.028 in-oz (0.20 mN-m) is comparable with the reported range of this motor Coulomb torque of 0.02 to 0.05 in-oz (0.14 to 0.35 mN-m) by the motor vendor [5]

```
                    The number of rows is   2
                    -----   ROW  1  INPUT    -----
FREE CONTACT ANGLE          11.20     BALL DEN LBS/IN^3               0.2830
NO OF BALLS                    8.     IR PRESS FIT                  -.000200
BALL DIAMETER            0.093750     OR PRESS FIT                  -.000300
PITCH DIAMETER           0.437500     SHAFT ID                      0.000000
IR CURVATURE                .5650     BEARING BORE                  0.250000
OR CURVATURE                .5650     INNER RING OD (calculated)    0.381250
IR H/D     (49.8 degree)    .2000     OUTER RING ID (calculated)    0.493750
IR DAM H/D (49.8 degree)    .2000     BEARING OD                    0.625000
OR H/D     (49.8 degree)    .2000     HOUSING OD                    1.000000
OR DAM H/D (49.8 degree)    .2000     INNER RING WIDTH              0.196000
                                      OUTER RING WIDTH              0.196000
MODULUS OF ELASTICITY
  SHAFT                  0.30E+08     PRELOAD =      2.60          CASE = 1
  BEARING RINGS          0.28E+08      1=UMT 2=MTD AT 68f 3=MTD AT TEMP
  BALLS                  0.28E+08     STRADDLE                      0.156000
  HOUSING                0.30E+08     AXIAL SPRING RATE             0.10E+21
POISSON RATIO                         AXIAL SPRING GAP              0.000000
  SHAFT                     .3000     TYPE OF DOUBLE ROW = 1.  -1=DF 0=DT
DB
  BEARING RINGS             .2850     FACE FLUSH ERROR              0.000000
  BALLS                     .2850     FRICTION COEF AT SLOW SPEED       .150
  HOUSING                   .3000
COEF LINEAR EXPAN
  SHAFT                  0.60E-05     IR CLAMPING FORCE(LBS)              0.
  BEARING RINGS          0.56E-05     IR CLAMPING FRICTION COEF         .000
  BALLS                  0.56E-05     OR CLAMPING FORCE(LBS)              0.
  HOUSING                0.60E-05     OR CLAMPING FRICTION COEF         .000
OPERATING TEMPS-F
  SHAFT                      68.0     FRICTION COEFFICIENTS
  INNER RING                 68.0       CAGE TO LAND                    .150
  OUTER RING                 68.0       CAGE TO BALL                    .150
  BALLS                      68.0       BALL TO RACE                    .150
  HOUSING                    68.0     BALL POCKET DIAMTRAL CLR        0.0000

LUBE VISCOSITY                 0.     RADIAL DEF. VECTOR (DELR)     0.000000
LUBE QUANITTY FACTOR         .000     AXIAL  DEF. VECTOR (DELA)     0.000000
                                      FILM PER SURFACE   (FILM)     0.000000
---- LOAD AND SPEED INPUTS ----
THRUST + ---> ON SHAFT         0.
RADIAL + ^ UP ON SHAFT         0.
MOMENT +   CW @ ROW 1          0.
ALPHA  + CW              0.000200
SPEED  IR RPM                  1.
SPEED  OR RPM                  0.
```

```
                        ---- PROBLEM  1   RESULTS  ----
                           UNMOUNTED      MOUNTED      MOUNTED AT
                                        68 DEG. F   TI= 68. TO= 68.

FREE DIAMETRAL PLAY      0.000464       0.000464       0.000464
FREE END PLAY            0.004734       0.004734       0.004734
FREE CONTACT ANGLE         11.200         11.200         11.200

IR  PRESS FIT                          -0.000200      -0.000200
OR  PRESS FIT                          -0.000300      -0.000300

PRELOAD                      2.60          2.60          2.60
AXIAL DEFLECTION         0.000316       0.000316       0.000316
SOL  AXIAL DEF.          0.000316       0.000316       0.000316
CONTRACTION OF IR DIAM.   .000002        .000002        .000002
EXPANSION   OF OR DIAM.   .000003        .000003        .000003
IR MEAN CONTACT STRESS    127296.        127296.        127296.
OR MEAN CONTACT STRESS    105916.        105916.        105916.
CONTACT ANGLE AT PRLD      12.654         12.654         12.654
REQUIRED IR H/D AT PRLD     0.025          0.025          0.025
REQUIRED OR H/D AT PRLD     0.025          0.025          0.025

 --- THE TORQUE AND SPRING RATES BELOW ARE FOR BRG PAIR ---
              ------- BRG PAIR LOAD TORQUES  IN-OZ -------
          TORQUES BASED ON A FRIC. COEF. = 0.150
1. BALL TO RACE SPIN      0.01887        0.01887        0.01887
2. BALL TO RACE ROLL      0.00733        0.00733        0.00733
3. MATERIAL HYSTER.       0.00150        0.00150        0.00150
TOTAL 1+2+3               0.02771        0.02771        0.02771

       ------- BRG PAIR SPRING RATES LBS/IN -------
          CAUTION! ASSUMES A RIGID PRELOAD SPRING
AXIAL                      27511.         27511.         27511.
RADIAL                    269203.        269203.        269203.
```

Figure 5. Motor Rotor Bearing Analysis Using the Aerospace Corporation BRGS10C

With five equations and five unknowns, the system of linear equations can be solved as shown in Figure 6. The result is summarized in Figure 7.

It is important to note that under high speed running conditions, this cogging or detent torque, T_d, has a mean value of zero and, therefore, does not constitute a loss of torque. The variation in the instantaneous value can, however, cause an unacceptable increase in the percentage ripple torque.

Solve a linear system Mx = v

$$MX := \begin{pmatrix} 1 & 0 & 0 & 1800^{.667} & 0 \\ 1 & 0 & 1 & 461^{.667} & 0 \\ 1 & 1 & 0 & 1800^{.667} & 1800^{.667} \\ 1 & 1 & 1 & 0 & 0 \\ 1 & 0 & 0 & 0 & 0 \end{pmatrix} \quad x := \begin{pmatrix} Tcm \\ Tcg \\ Td \\ Kvm \\ Kvg \end{pmatrix} \quad v := \begin{pmatrix} .3115 \\ .42 \\ .8794 \\ .4617 \\ .028 \end{pmatrix} \quad x := \mathrm{lsolve}(MX, v) = \begin{pmatrix} 0.028 \\ 0.156 \\ 0.278 \\ 1.911 \times 10^{-3} \\ 2.777 \times 10^{-3} \end{pmatrix}$$

Figure 6. System of Linear Equations for Motor Torque Test Data

Torque Component or Constant	Value	
Tcm	2.800E-2 in-oz	197.7 µN-m
Tcg	1.560E-1 in-oz	1.102 mN-m
Td	2.780E-1 in-oz	1.963 mN-m
Kvm	1.911E-3 in-oz/rpm$^{0.667}$	13.49 µN-m/rpm$^{0.667}$
Kvg	2.777E-3 in-oz/rpm$^{0.667}$	19.61 µN-m/rpm$^{0.667}$
Kw	1.600E-10 in-oz/rpm^{2}	1.130 pN-m/rpm^{2}

Figure 7: Geared Motor Derived Torque Components

The derived motor viscous torque of 0.001911 in-oz/rpm$^{.667}$ (13.49 µN-m/ rpm$^{.667}$) is also comparable with this motor viscous torque constant, Bv, of 1.33E-4 in-oz/rpm or 0.00162 in-oz/rpm$^{.667}$ reported by the motor vendor [6].

The static gear tooth sliding coefficient is derived based on the geared motor stalled torque versus applied current test data. The geared motor stalled torque versus applied current condition can be modelled as follows:

$$V = L\frac{dI}{dt} + RI \tag{8}$$

$$T_s = (k_T I - T_{cm} - T_{cg} - T_d)(1 - k_{sg}) \tag{9}$$

where k_{sg} is the static gear tooth contact frictional torque loss coefficient, which is the only loss component depending on the transmitted torque.

By applying a linear regression to the stalled torque versus applied current data as shown in Figure 8, the slope and the x-intercept are expressed as follows:

k_{sg} = 1-slope/kt = 1-1.773/2.85 = 1-0.622 = 0.38 or 38% gear tooth loss at static loaded condition.

$T_{cm}+T_{cg}+T_d$ = 2.85*0.162 = 0.462 in-oz (3.26 mN-m)

The dynamic gear tooth sliding coefficient is derived from the geared motor dynamo test, which has the operating parameters listed as follows:

Speed = 4608 rpm; load torque, Tl = 375 in-lb (42.4 N-m) at gear shaft; applied current, I = 1.2 A
Torque governing equation:

$$\left(k_T I - T_{cm} - T_d - k_{vm}\dot{\theta}^{.667} - T_{cg} - k_{vg}\dot{\theta}^{.667}\right)\left(1 - k_{sg}\right) = \frac{T_l * 16}{GR} \tag{10}$$

From the data in Figure 7, the dynamic gear tooth sliding friction coefficient is derived to be 0.33, which is consistent with the static sliding friction coefficient of 0.38 extracted from the stalled torque vs current curve shown in Figure 8. The static and the dynamic gear tooth sliding coefficients are very close. To be conservative, the static gear tooth sliding friction coefficient will be used for the subsequent analysis.

The motor bearing viscous torque model is well described analytically by the MPB torque equation [2]. It's assumed that the gearbox viscous torque is also modelled using the MPB torque equation (Equation 11).

$$T_{visc} = a * 2.4 * 10^{-5} RPM^{0.67}\left(10^{10^{4.354-1.612*log10(T)}} - 0.6\right)^{.67} \tag{11}$$

where T is oil temperature in degree Kelvin, RPM is rotor speed in rev per minute, and a is a scaling factor to match with the measured data, which is derived to be 5.045 to match the derived composite geared motor torque constant. Figure 9 shows the geared motor viscous torque for different speeds and oil temperatures. This model uses the Brayco 815Z oil for both motor bearings and gearbox components.

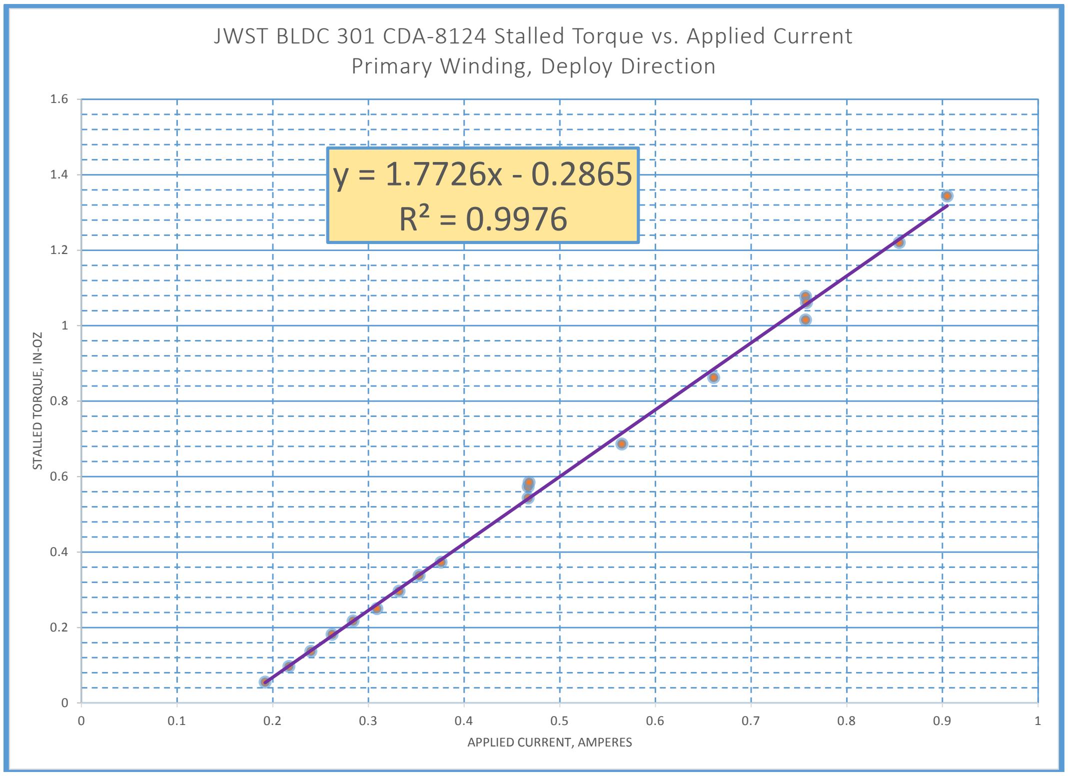

Figure 8. Geared Motor Stalled Torque vs Applied Current Curve

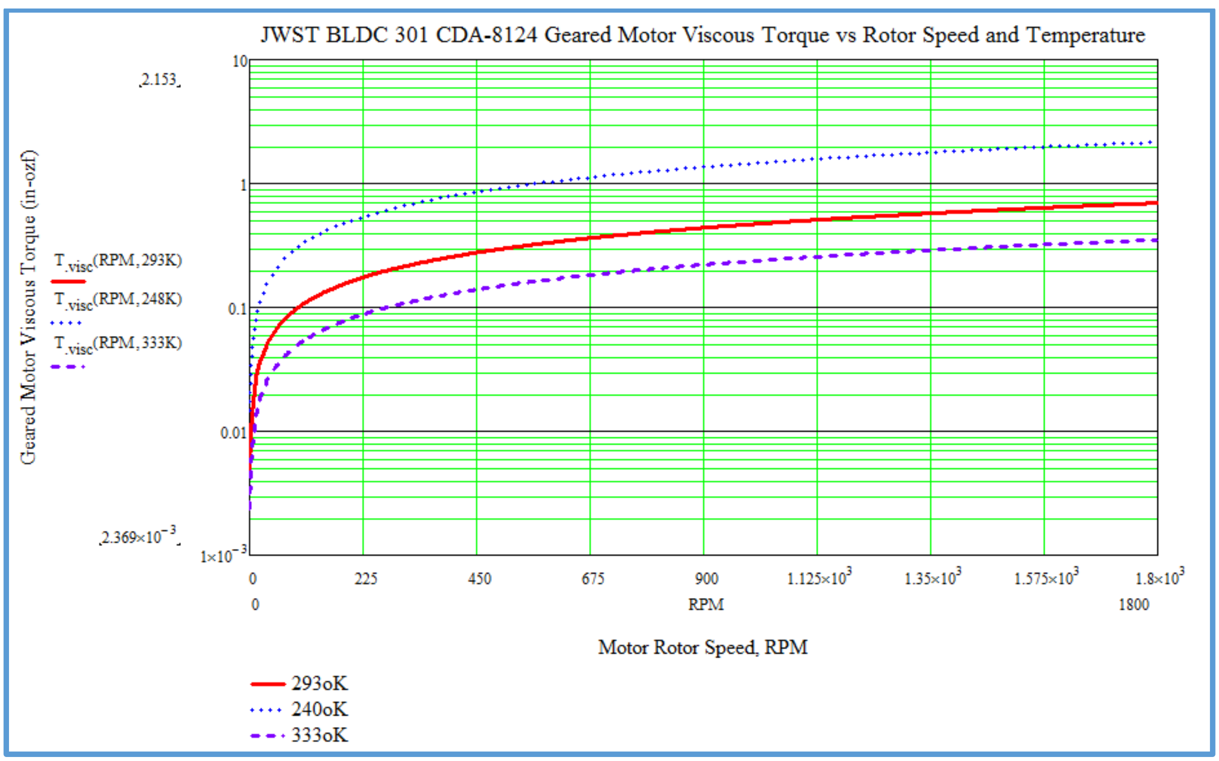

Figure 9. JWST Motor Viscous Torque versus Operating Temperature and Rotor Speed

To increase the fidelity of the geared motor model and its prediction, the model prediction is compared with the no-load current versus motor rotor speed test data. The governing torque model equation is expressed as follows:

$$V = RI + k_b\dot{\theta} \tag{12}$$

$$k_T I = T_{cm} + T_{cg} + T_d + k_{vm}\dot{\theta}^{.667} + k_w\dot{\theta}^2 + k_{vg}\dot{\theta}^{.667} \tag{13}$$

By applying the linear regression analysis from Equation 13 on the measured no-load current versus motor rotor speed as shown in Figure 10, the results from both the regression analysis and the model prediction using system of equations are plotted in Figure 10. From Figure 10, these two curves show a remarkable match.

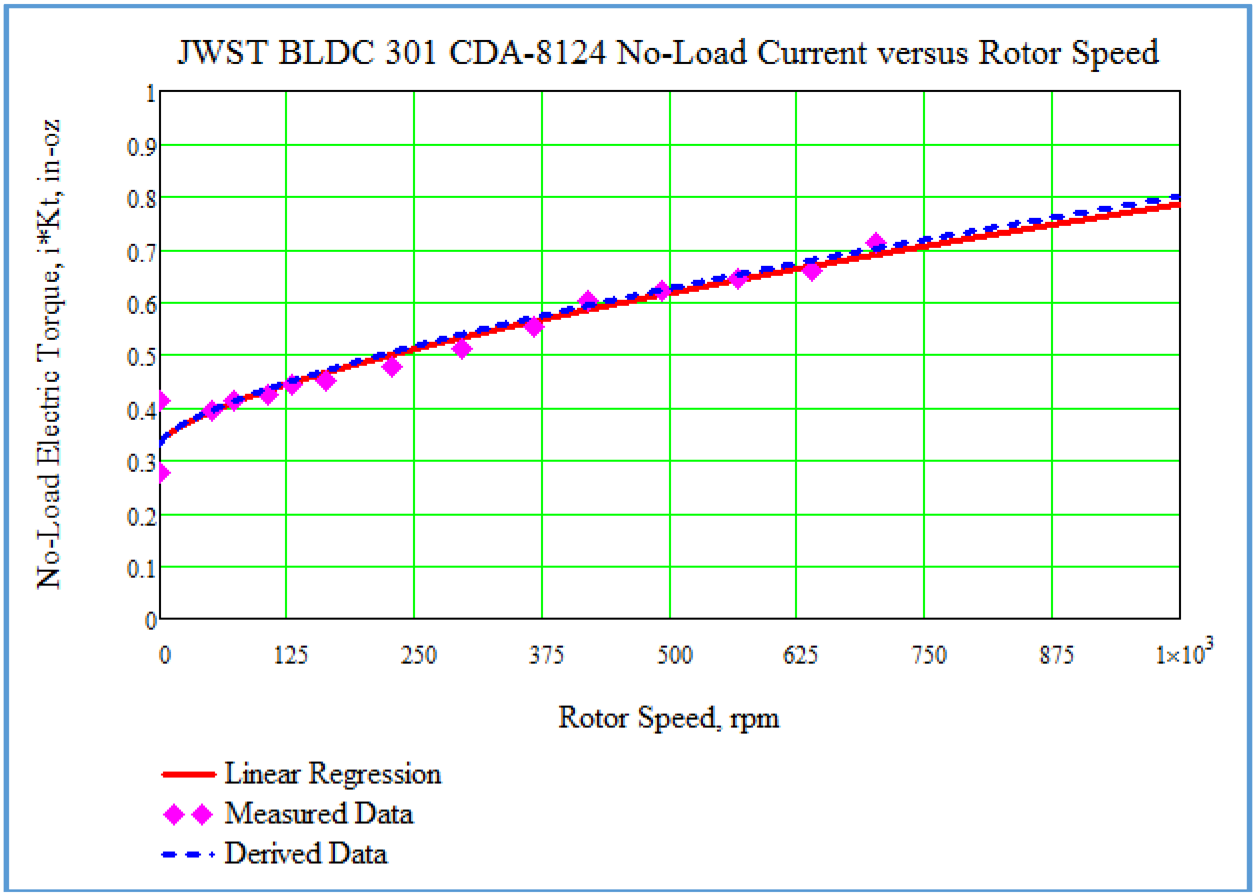

Figure 10. No-Load Applied Current versus Rotor Speed for the Measured and Derived Data

For the deployment, the applied current and the motor rotor speed are controlled with a forward control fashion. There is no force/torque feedback control. The gearbox output torque is expected to responded accordingly to the controlled input within a certain level of uncertainty under the imposed environmental conditions such as operating temperature, vibration induced load, humidity, etc. If the gear output torque exceeds a certain limit, deployment mechanical failure will occur. On the other hand, the gear output torque may not be sufficient to deploy the system. The limit between these values sometime is small.

The effect of motor parameter variation and environmental conditions on the geared motor output torque is evaluated as follows.

Variation of Output Torque from a Given Applied Current

The geared motor output torque equation is expressed as follows:

$$T_{.g\_load\_output}(i,\omega,T) := \left(1-k_{.sg}\right)\cdot\left[k_{.t}\cdot i - T_{.cm} - T_{.cg} - T_{.d} - \left(k_{.vg}+k_{.vm}\right)\cdot\omega^{.667}\cdot\left[\frac{10^{10^{\left[4.354-1.612\cdot\log\left[\frac{T}{(1K)}\right]\right]}}-0.6}{10^{10^{(4.354-1.612\cdot\log(293))}}-0.6}\right]^{\left(\frac{2}{3}\right)}\right]\cdot\frac{GR}{16} \tag{14}$$

Applying the above procedure to the rest of the motors, all motor parameters, test data, and the torque components are calculated and shown in Figure 11. The geared motor output torque versus speed at room temperature and applied current of 1 ampere for three geared motors are plotted in Figure 12.

From Figure 12, at the nominal operating motor rotor speed of 461 rpm, the gear motor output torque can vary as much as 7%. The variation can be higher if the sample size becomes larger or both the primary and the redundant winding data are evaluated.

			Brushless DC Motor ID			Unit
			CDA-8124	CDA-8125	CDA-8126	
Measured Motor Parameters	Kt		2.85	2.9	2.85	in-oz/A
	R		1.64	1.63	1.63	Ohm
	L		605	595	614	mHenry
	Kb		0.00211	0.00214	0.00211	V/rpm
	f		1000	1000	1000	Hz
Test Measured Data	Motor No-Load	Vrms	3.6	3.6	3.6	Volts
		V[1]	0.3115	0.2715	0.3080	in-oz
	Motor Dynamo	Tml	3	3	3	in-oz
		I	1.2	1.2	1.2	Amperes
		V[2]	0.4259	0.4860	0.4259	in-oz
	Geared Motor No-Load	Vrms	4.3	4.3	4.3	Volts
		V[3]	0.8794	0.8581	0.8693	in-oz
	Geared Motor Stalled	X-intercept	0.4617	0.55825	0.5358	in-oz
		V[4]	0.4617	0.55825	0.5358	in-oz
	Motor Coulomb Torque	V[5]	0.028	0.028	0.028	in-oz
Motor Torque/Friction Components	Tcm		0.0280	0.0280	0.0280	in-oz
	Tgm		0.1560	0.1700	0.2230	in-oz
	Td		0.2780	0.3600	0.2850	in-oz
	Kvm		0.0019	0.0016	0.0019	in-oz/rpm^.67
	Kvg		0.0028	0.0028	0.0023	in-oz/rpm^.67
	ksg		0.3780	0.3050	0.3407	unitless

Figure 11. Geared Motor Parameters, Test Data, and Torque Component Values and Constants

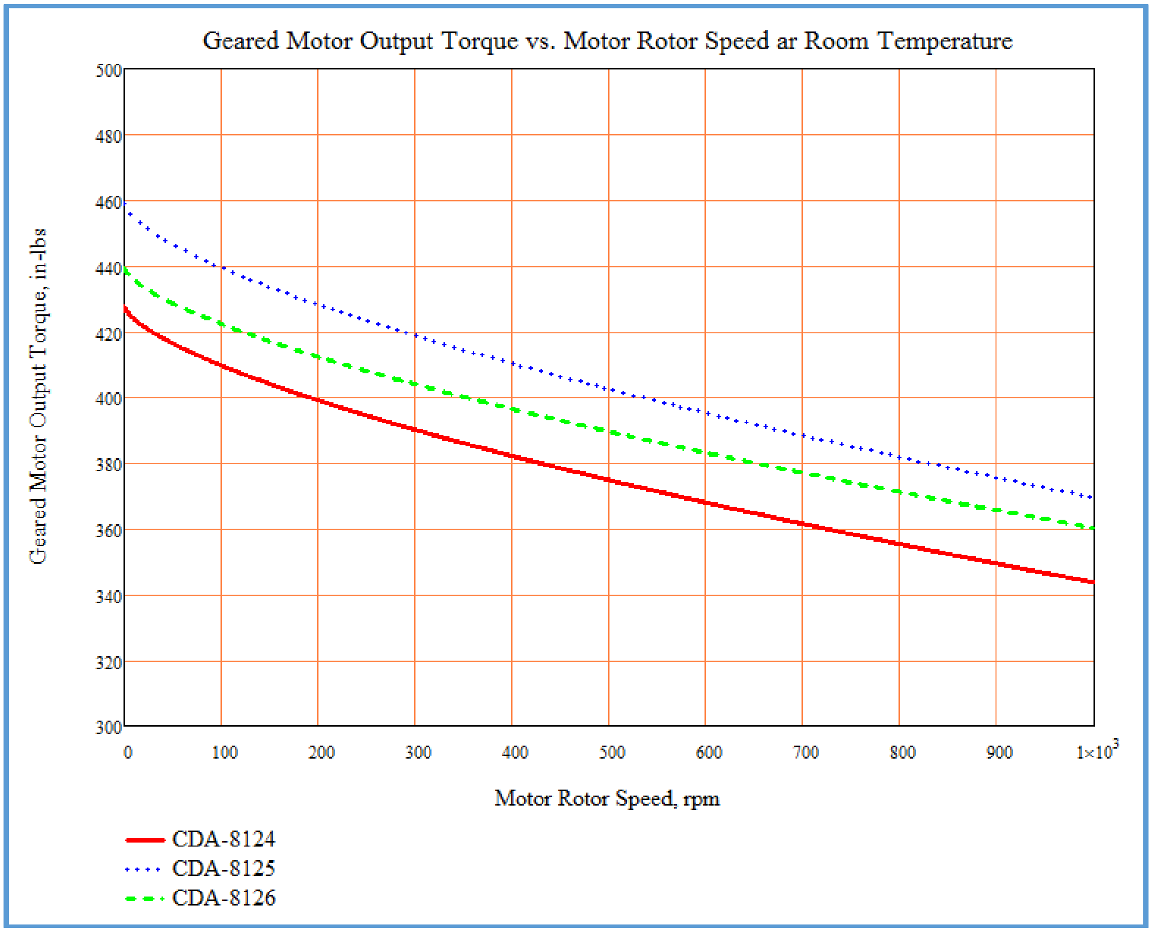

Figure 12. Geared Motor Output Torque vs Speed for Three Motors

Environmental Induced Effects on Gear Output Torque

The environmental effects on the geared motor output torque are numerous. In the scope of this paper, only the temperature effect is evaluated. Applying Equation 14, the geared motor output torque versus operating speed at three operating temperatures: 243°K (cold), 293°K (RT), and 348°K (hot) and the given applied current of 1 ampere is plotted in Figure 13. From Figure 13, at the nominal operating speed of 461 rpm, the output torque variation can be as high as 75%.

It is observed that the combined effect of the motor parameter variation and the temperature can cause the output torque variation to be as much as 82%. This is a conservative prediction due to the small motor sample size (i.e., 3) and the motor parameters are evaluated for the primary winding and the deployment direction only.

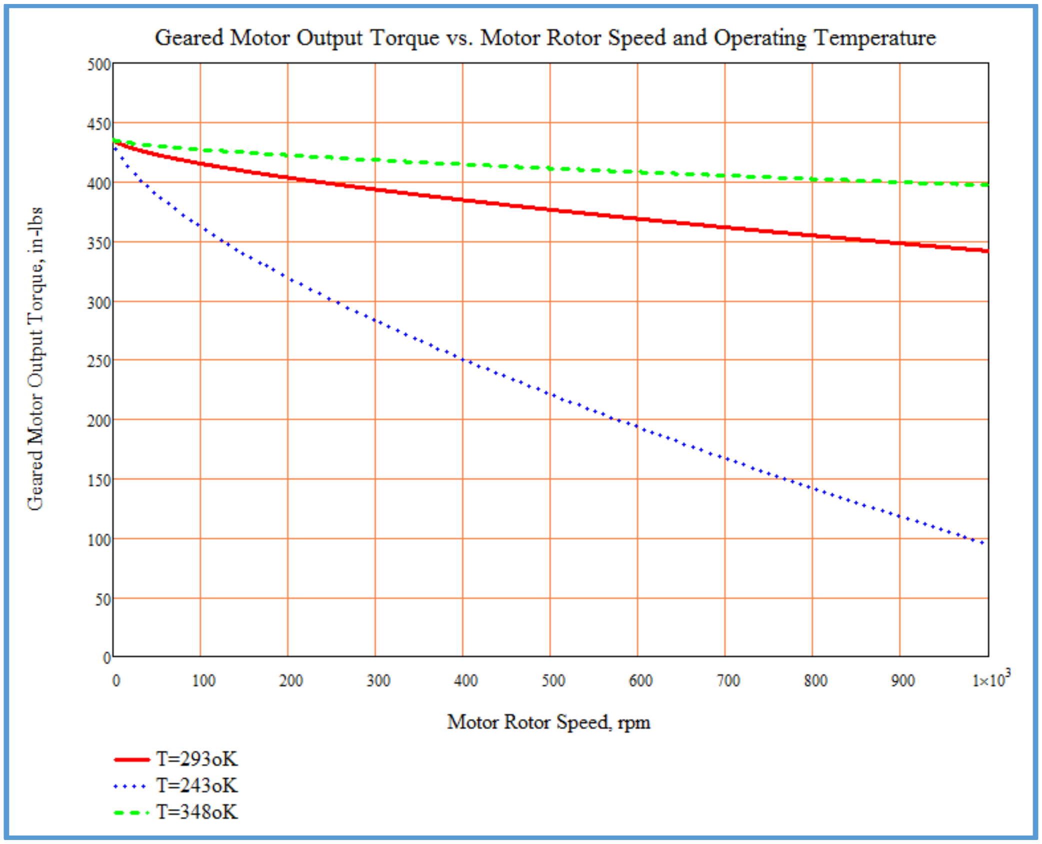

Figure 13. Geared Motor Output Torque vs Speed for CDA-8124 at Different Operating Temperature

Geared Motor and Gearbox Torque Efficiency Calculation

For the motor evaluation, it's useful to show the geared motor and the gearbox torque efficiency. In general, the torque efficiency η of a motor is defined as the ratio between the input electrical power, i.e., the product of voltage times current, and the output mechanical power minus the power loss due to mechanical means, which is expressed as follows:

$$\eta_{motor} = \frac{(k_T i - T_{loss})(1 - k_{sg})}{k_T i} \tag{15}$$

where T_{loss} is the net projected torque loss of the motor and the gearbox to the motor rotor location due to speed, temperature, and windage, and n_{gslide} is the gearbox tooth power efficiency due to applied torque.

Using Equation 2 with motor parameters from Figure 11, the geared motor torque efficiency can be calculated as follows:

$$\text{geared\_motor\_}\eta(i,x,T) := \frac{\left[k_{,t} \cdot i - T_{,cg} - T_{,cm} - T_{,d} - (K_{,vg} + K_{,vm}) \cdot \left[\dfrac{10^{10^{\left[4.354 - 1.612 \cdot \log\left[\frac{T}{(1K)}\right]\right]}} - 0.6}{10^{10^{(4.354 - 1.612 \cdot \log(293))}} - 0.6}\right]^{\left(\frac{2}{3}\right)} \cdot x^{.67}\right] \cdot (1 - 0.378)}{k_{,t} \cdot i} \tag{16}$$

Figure 14 shows the geared motor torque efficiency as a function of applied current and geared motor output torque at room temperature. From Equation 16, it is clear that the mechanical efficiency for the geared motor is a strong function of several motor operating parameters such as motor transmitted torque, operating speed, operating temperature, humidity, atmospheric pressure, etc. Therefore, it is not very specific and should be used as a guideline to motor selection or design/configuration comparison.

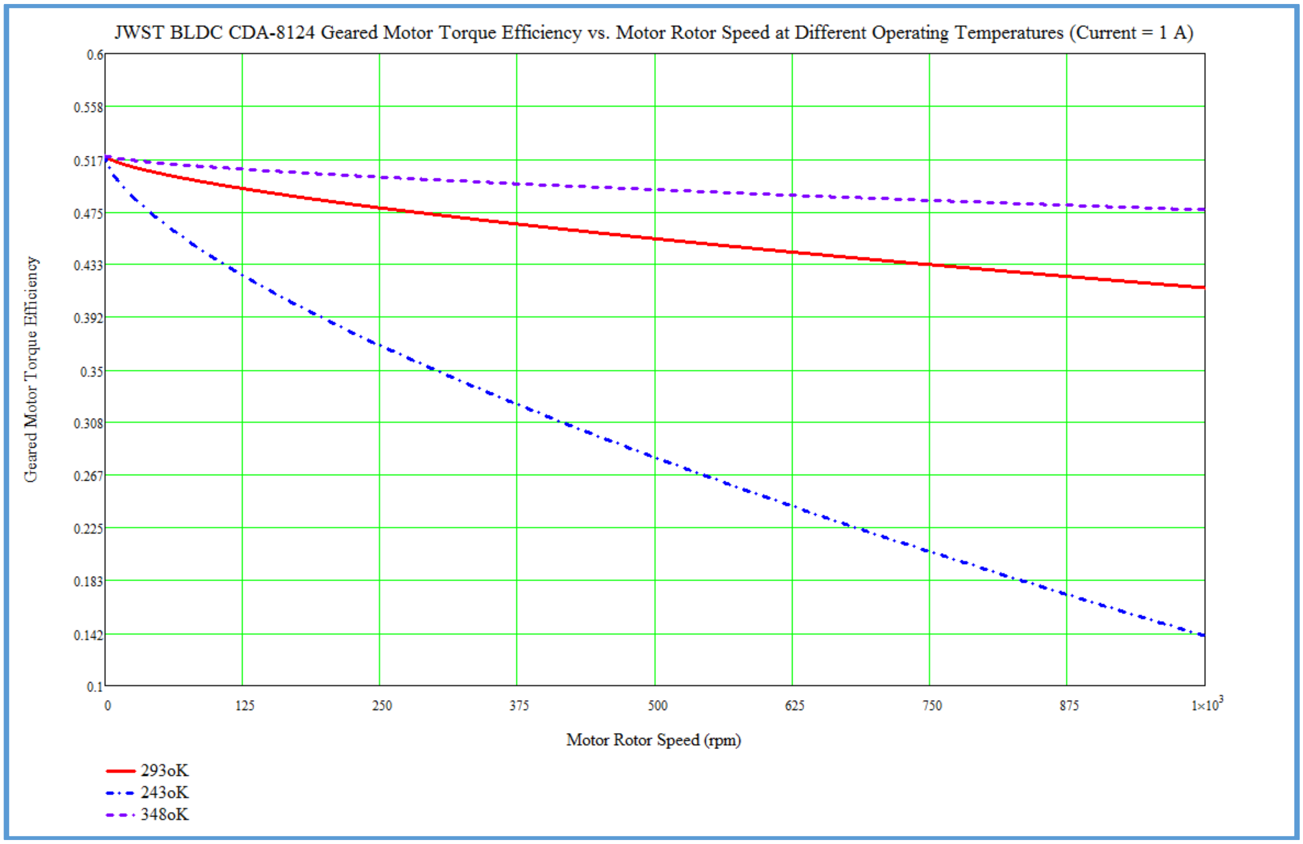

Figure 14. Geared Motor Torque Efficiency vs Motor Rotor Speed at Different Temperatures

The gearbox torque efficiency can be calculated as follows:

$$\text{gearbox\_}\eta(\text{torque},x,T) := \frac{\left[\text{torque} - T_{.cg} - K_{.vg}\cdot\left[\frac{10^{10^{\left[4.354-1.612\cdot\log\left[\frac{T}{(1K)}\right]\right]}} - 0.6}{10^{10^{(4.354-1.612\cdot\log(293))}} - 0.6}\right]^{\left(\frac{2}{3}\right)}\cdot x^{.67}\right]\cdot(1-0.378)}{\text{torque}} \quad (17)$$

Where the values for each parameters for each motor are defined and can be found in Figure 11.

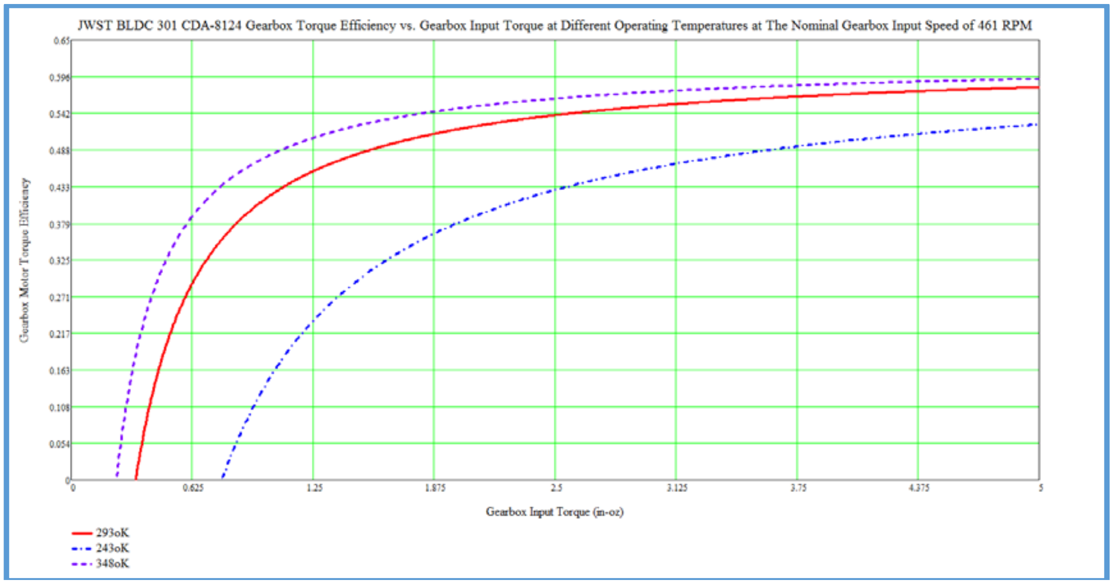

Figure 15. Simulated Power Efficiency vs. Motor Rotor Speed and Operating Temperature at 1-A Applied Current

It is observed that at low input torque, the gearbox torque efficiency is dominated by the Coulomb, viscous, and detent torques. However, at close to full load, the gearbox efficiency is controlled by gear tooth sliding friction coefficient.

Conclusion

A brushless DC motor torque model has been developed in order to extract the geared motor torque component loss for the gearbox and the motor using data from motor measurements and the geared motor testing. The geared motor output torque and mechanical efficiency vary from motor to motor and are a strong function of the transmitted torque, rotor speed, and operating temperature.

If the geared output torque requirement is stringent, the motor characterisation process and the selected operating condition as shown above are a must in order to reduce the uncertainty.

Lessons Learned

1. The geared motor output torque significantly varies from motor to motor and with operating conditions. The output torque variation has been shown to be 82%. If additional factors are accounted for, it can be 100% or higher. By characterizing each motor and selecting optimal operating conditions, the output variation can be significantly reduced.
2. The geared motor tests at the motor vendor should be well defined and the results should be verified at each step to assure that the physics can be understood or explained.
3. The MPB viscous torque model shows a good correlation with the geared motor test data and is proportional to the speed to the power of 0.667. The motor bearing Coulomb torque can be reasonably predicted using the ball bearing torque model in BRGS10C with a friction coefficient of 0.15 to 0.20 for small bearings (OD < 1 in (2.5 cm))
4. The gearbox efficiency is not a constant and a non-linear function throughout operating conditions.

References

1. Ward, Peter, "Effects of Bearing Cleaning and Lube Environment on Bearing Performance", 29th Aerospace Mechanisms Symposium Proceeding, Houston, TX, 1995.
2. Leveille, A., The Aerospace Corporation Bearing Analysis Program, BRGS10C version 10C, 2005
3. Tran, A.N., Halpin, J.D.," JAMES WEBB SPACE TELESCOPE DEPLOYMENT TOWER ASSEMBLY DEPLOYING ANOMALY AND LESSONS LEARNED", 16th European Space Mechanisms and Tribology Symposium 2015, Bilbao, Spain - 23rd to 25th September 2015
4. Vrancik, J.E., "Prediction of Windage Power Loss In Alternators", NASA TN D-4849, Lewis Research Center, Cleveland, Ohio, 1968
5. Personal communication with staff at CDA InterCorp, Florida, USA Dec 23, 2015
6. Brushless Permanent Magnet Motor Engineering Reference Data, CDA InterCorp Catalog

Acknowledgement

The author would like to thank Paul Reynolds, Optical Telescope Elements Mechanisms IPT leader, Dr. Scott Texter, NGC JWST Optical Telescope Elements program manager for their guidance, and my NGC colleagues Shaun O'Neill and Stan Klyza for technical discussion. The technical discussion with CDA InterCorp is also appreciated.