Steven Bauman, Kevin Ho,
Derrrick Salmon
Rev: 16 8-31-2010

NOTE: During Integration of the Electric motors with the Geartrain the Brevini gearbox was found to have the incorrect gear reduction. All documentation, manuals, and specification data plates were incorrect. Unfortunetaly all calculations were done with the incorrect gear reduction ratio of 1:140. The worksheet below reflects the corrected calculations.

The correct gear reduction ratio is 1:80. **S.Bauman**

Appendix A 15-03-001 Dome drive requirements document calculations

1. The gear reduction (torque) ratio between the dome wheel (24 " diameter) and the dome track.

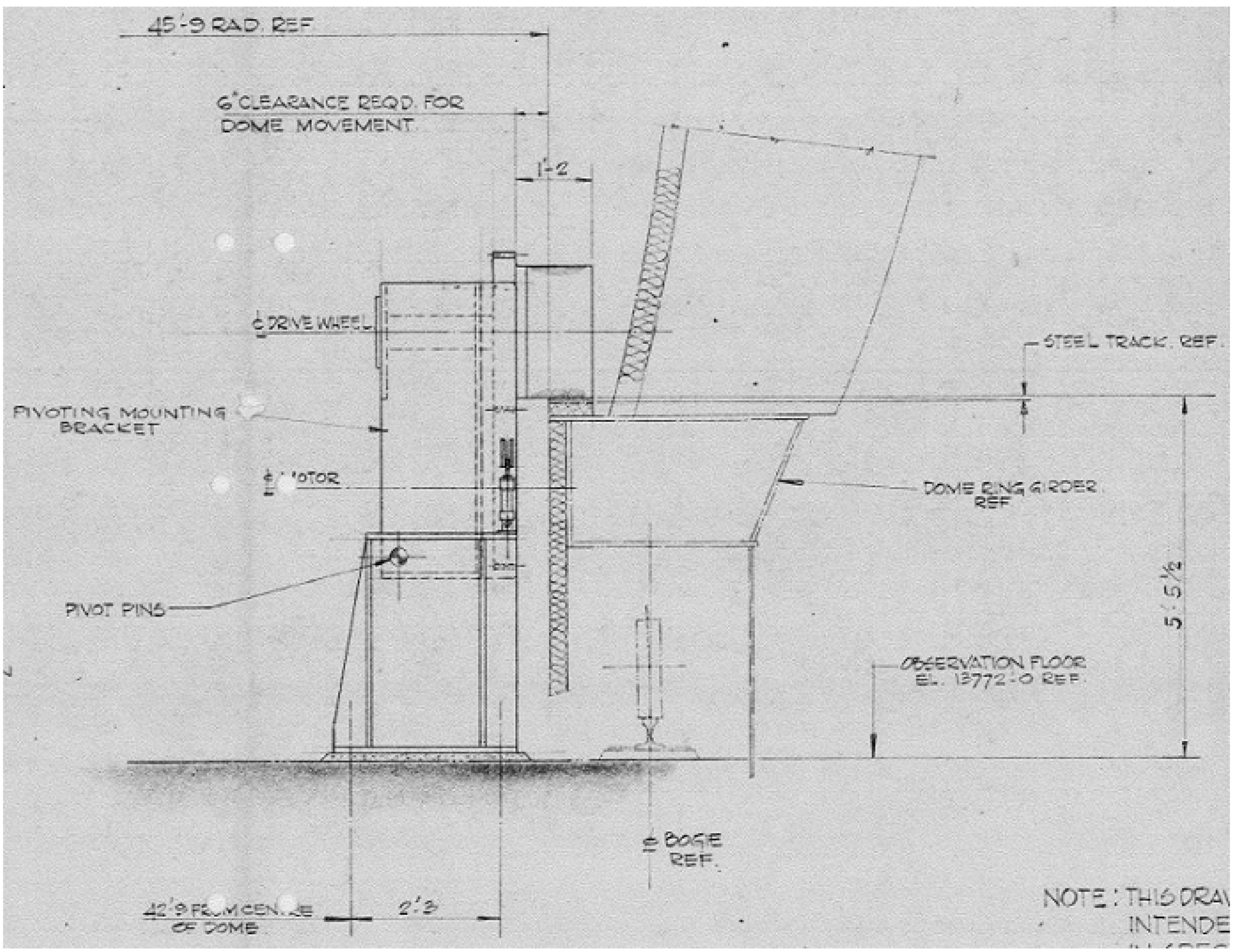

Figure 1: Reference information for the dome drive wheel to track interface, see drawing SNC 21 above or Appendix J.

- The distance from the center point or axis of the dome to the centerline of the dome wheel is 45' 9".

Radius of the dome to the track:

$R\_dome \equiv 45.75 \cdot ft$

$R\_dome = 13.945 \cdot m$

The circumference of the dome:

$C\_dome := 2 \cdot \pi \cdot R\_dome$

$C\_dome = 87.617 \cdot m$

Radius of the wheel:

$R\_wheel \equiv 12 \cdot in$

$C\_wheel := 2 \cdot \pi \cdot R\_wheel$

The circumference of the wheel:

$C\_wheel = 1.915 \cdot m$

The gear reduction ratio at the dome drive wheel to the dome building track interface:

$$MV\_Gear\_reduction\_dome\_wheel := \frac{C\_dome}{C\_wheel}$$

<u>MV, Mechanical advantage of the dome-wheel interface: 1:45.75</u>

$MV\_Gear\_reduction\_dome\_wheel = 45.75$

2. The gear reduction (torque) ratio for the Brevini EC 3090 right angle speed reducer (gear box).

- The gear reduction (torque) ratio of the Brevinvi EC 3090 is 1:80 <u>not</u> 1:140

See Appendix K for Brevini Specification sheet

$MV\_Gear\_reduction\_box \equiv 80$

<u>MV, Mechanical advantage of the gear box: 1:80</u>

3. The total (overall) gear reduction (torque) ratio or mechanical advantage (MV) for the drive train (gearbox and dome wheel to track interface).

The Total gear reduction (torque) ratio or mechanical advantage (MV) ratio :

$$\text{MV\_Total\_Gear\_Reduction} := \text{MV\_Gear\_reduction\_box} \cdot \text{MV\_Gear\_reduction\_dome\_wheel}$$

<u>MV, Mechanical advantage of the drive train: 1:3660</u>

$$\text{MV\_Total\_Gear\_Reduction} = 3.66 \times 10^{3}$$

4. The position (control) resolution for TCS and Manual control.

- To determine the arc distance for small angles the following formula can be used.

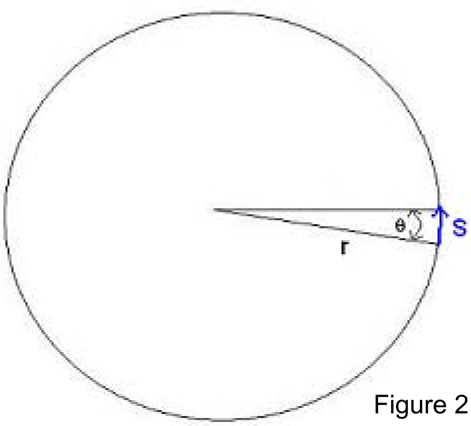

Figure 2: Photo is for visual purposes and is not to scale

S (arc distance) = radius (r) x angle (θ)

$$r \equiv \text{R\_dome}$$

$$\theta \equiv 1 \cdot \text{deg}$$

$$\theta = 0.017 \cdot \text{rad}$$

$$S \equiv r \cdot \theta$$

<u>S, Arc distance is 0.243m per degree rotation of the dome.</u>

$$S = 0.243 \cdot \text{m}$$

- TCS and Manual dome control requirements referenced document: 15-03 Dome Drive Electric upgrade requirements-, section 28.2.

Under TCS control the dome must be able to control the position of the dome:

$$\text{TCS\_input\_requirment} \equiv 0.1044\text{deg}$$

$$\text{TCS\_input\_requirment} = 1.822 \times 10^{-3} \cdot \text{rad}$$

The TCS position control resolution:

$$\text{TCS\_input\_requirment} = 0.104 \cdot \text{deg}$$

$$\text{R\_dome} = 13.945\,\text{m}$$

$$\text{TCS\_positon\_control\_resolution} := \text{TCS\_input\_requirment} \cdot \text{R\_dome}$$

_ _

$$\text{TCS\_positon\_control\_resolution} = 25.409 \cdot \text{mm}$$

_ _

Under Manual control the dome must be able to control the positon of the dome:

$$\text{Manual\_input\_requirment} \equiv 0.1044\text{deg}$$

$$\text{Manual\_input\_requirment} = 1.822 \times 10^{-3} \cdot \text{rad}$$

The Manual position control resolution:

$$\text{Manual\_input\_requirment} = 0.104 \cdot \text{deg}$$

$$\text{R\_dome} = 13.945\,\text{m}$$

$$\text{Manual\_positon\_control\_resolution} := \text{Manual\_input\_requirment} \cdot \text{R\_dome}$$

_ _

$$\text{Manual\_positon\_control\_resolution} = 25.409 \cdot \text{mm}$$

_ _

5. Hydraulic motor rotation Speed:

See Appendix B for 15-03-002 Dome Drive data 8-4-2010

Using the Data from the Appendix:

- The speed of the dome when operating at full continous speed is 60 deg/min or 1 deg/sec.

Angular Velocity (speed) of the dome: ω

at t=0, rest

$$\omega dome0 \equiv 0 \ \frac{rad}{sec}$$

at t~10sec at full speed

$$\omega dome1 \equiv 0.01745329252 \ \frac{rad}{sec}$$

- Therefore the angular velocity (speed) of the hydraulic motor at full continuous speed is:

$$\omega motor := MV\_Total\_Gear\_Reduction \cdot \omega dome1$$

$$\omega motor = 63.879 \ \frac{rad}{sec}$$

$$\omega motor\_rpm := \omega motor \cdot \left(\frac{60}{1 \cdot min}\right) \cdot \left(\frac{1 \cdot rev}{2 \cdot \pi}\right)$$

$$\omega motor\_rpm = 610 \cdot rpm$$

- The speed of the dome when operating at slowest continous operating speed is 13 deg/min or 0.216 deg/sec.

Angular Velocity (speed) of the dome:

at t=0, rest

$$\omega dome0 \equiv 0 \quad \frac{rad}{sec}$$

at t~10sec at slowest speed

$$\omega dome1\_slow \equiv 0.00378$$

- Therefore the angular velocity (speed) of the hydraulic motor at slowest continuous operating speed is:

$$\omega motor\_slow := MV\_Total\_Gear\_Reduction \cdot \omega dome1\_slow$$

$$\omega motor\_slow = 13.835 \quad \frac{rad}{sec}$$

$$\omega motor\_rpm\_slow := \omega motor\_slow \cdot \left(\frac{60}{1 \cdot min}\right) \cdot \left(\frac{1 \cdot rev}{2 \cdot \pi}\right)$$

$$\omega motor\_rpm\_slow = 132.113 \cdot rpm$$

The calculation below is to determine the goal fast speed for the upgrade

- The speed of the dome when operating at a goal fast operating speed is 72 deg/min or 1.2 deg/sec.

Angular Velocity (speed) of the dome:

at t=0, rest

$$\omega dome0 \equiv 0 \quad \frac{rad}{sec}$$

at t~10sec at fastest speed

$$\omega dome1\_fast \equiv 0.02094$$

- Therefore the angular velocity (speed) of the hydraulic motor at slowest continuous operating speed is:

$$\omega motor\_fast := MV\_Total\_Gear\_Reduction \cdot \omega dome1\_fast$$

$$\omega motor\_fast = 76.64$$

$$\omega motor\_rpm\_fast := \omega motor\_fast \cdot \left(\frac{60}{1 \cdot min}\right) \cdot \left(\frac{1 \cdot rev}{2 \cdot \pi}\right) \qquad \omega motor\_rpm\_fast = 731.862 \cdot rpm$$

6. Determine the running Power output of the hydraulic motors and the Power input of the Hydraulic Power unit:

6.1 Determine the power output of the hydraulic motors:

See Appendix B for 15-03-003 Dome Drive data 2-17-2010, maximum running hydraulic motor power

Using the Data from the Appendix B:

- The hydraulic motor power output:

$$\text{Maximum\_running\_hydraulic\_motor\_power} \equiv 3.365 \cdot \text{hp}$$

$$\text{Minimum\_running\_hydraulic\_motor\_power} \equiv 1.565 \cdot \text{hp}$$

There are three (3) motors, therefore:

$$\text{Power\_output\_hydraulic\_motors} \equiv 3 \cdot \text{Maximum\_running\_hydraulic\_motor\_power}$$

$$\text{Power\_output\_hydraulic\_motors} = 7.528 \times 10^{3}\,\text{W}$$

$$\text{Power\_output\_hydraulic\_motors} = 10.095 \cdot \text{hp}$$

- The maximum running hydraulic motor output power will be one basis for later calculations.

Note: The value above for the running hydraulic motor maximum speed is the largest value (best case) that can be achieved by the hydraulic motor from the hydrostatic transfer of energy, i.e. the highest pressure and flow rates that can be delivered.

6.2 The electrical power consumption of the hydraulic power unit:

See Appendix C for 15-03 Dome Drive power consumption 2-17-2010.

Using the Data from the Appendix C:

Power input = 31,41.68 Watts

$$\text{Power\_input\_hydraulic\_powerunit} \equiv 31410.68 \cdot \text{W}$$

6.3 The mechanical efficentcy of the hydraulic system:

- To find the efficiency of the hydraulic system we can use the hydraulic motor power output and the True power measured at the hydraulic power unit:

The mechanical efficiency of the system is:

Mechaincal Efficiency = Power Out / Power input

$$\text{mechanical\_efficiency\_hydraulic\_system} := \frac{\text{Power\_output\_hydraulic\_motors}}{\text{Power\_input\_hydraulic\_powerunit}}$$

The mechanical efficiency of the hydraulic system is 24%

$$\text{mechanical\_efficiency\_hydraulic\_system} = 0.24$$

7. The mass Moment of inertia for the Dome

Assumptions:

- Assume that the dome is a perfect hollow half sphere to streamline calculations.

Mass of the building: 565 tons

Mass of the upper end instrument 12 tons

$$\text{mass\_dome} \equiv 577 \cdot \text{ton}$$

$$\text{mass\_dome} = 5.234 \times 10^{5} \cdot \text{kg}$$

The Moment of Inertia for a hollow sphere:

I = [2 x Mass dome x (Radius dome)^2 / 3] / 2 (half of the sphere)

$$I \equiv \left[\left(\frac{2}{3}\right) \cdot \left[\text{mass\_dome} \cdot (\text{R\_dome})^{2}\right]\right] \cdot \left(\frac{1}{2}\right)$$

mass moment of interia for the dome

$$I = 3.393 \times 10^{7} \cdot \text{kg} \cdot \text{m}^{2}$$

8. The mass Moment of inertia of the dome per drive unit

Since there are three (3) drive units we divide the mass moment of inertia of the dome by 3:

$$I\_per\_drive := \frac{I}{3}$$

<u>mass momemnt of inertia per drive unit</u>

$$I\_per\_drive = 1.131 \times 10^{7} \cdot kg \cdot m^{2}$$

9. The static friction force from the dome bogie to track steel friction (track misalignment and roller stiction)

Ref: Physical measurments were taken by T. Arruda and S. Bauman on 9-25-09 to determine the approximate friction force and corresponding friction breakaway torque needed to rotates the dome with all three (3) dome drive units dis-engaged (all dome drive wheels removed) from the track. The forklift was attached to one of the dome bogie main drive wheel assemblies by a chain. The forklift pulled the chain attached to the dome (main large bracket on the dome bogie wheel assembly) which approximated the force required to begin rotating the dome. The force was measured at 2000 lbf with a 12' (144") chain attached from the forklift to the dome. The approximate distance measured from the forklift to the track was about 40".

To find the angle θ, the values were input into Autocad where the interesecting circle diameters provide the geometry needed to find the angle between P and Py.

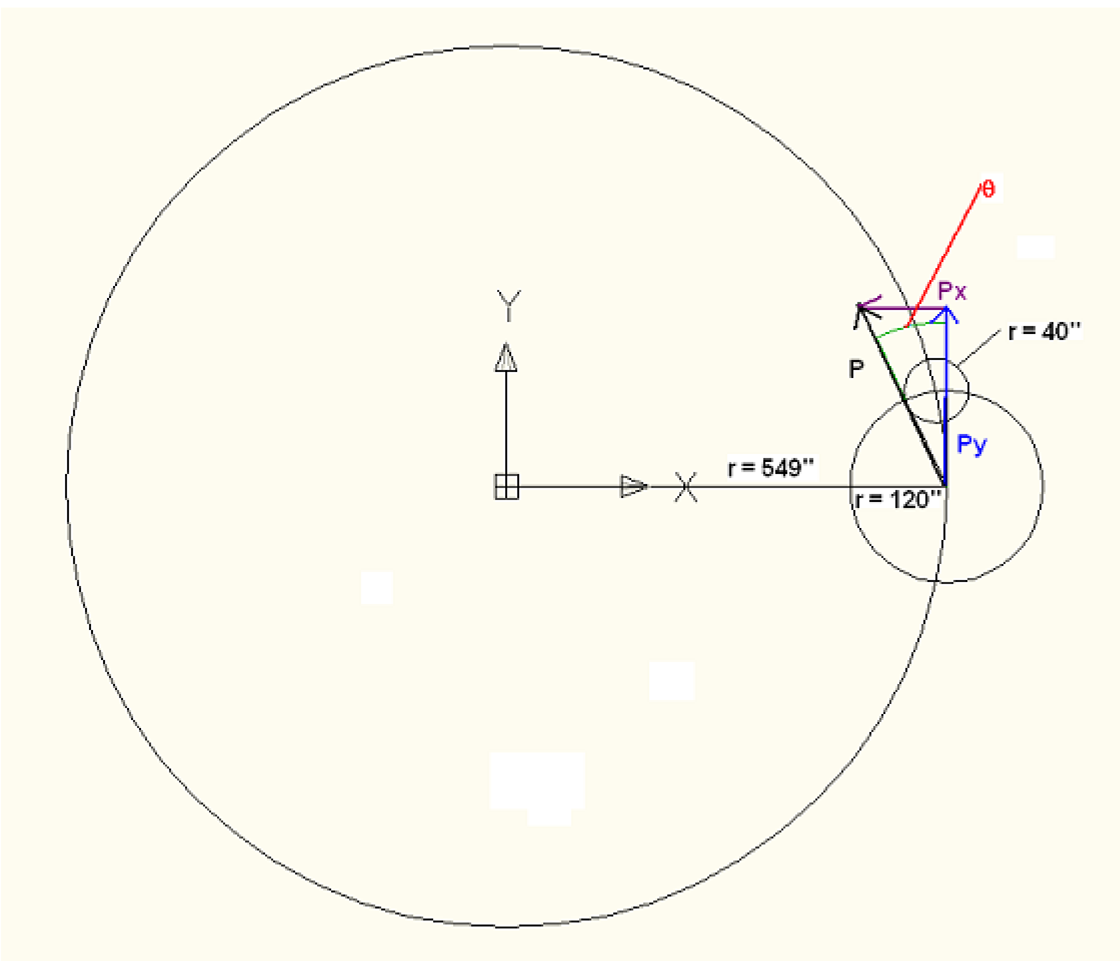

Figure 3: Photo is for visual purposes and is not to scale

$\theta \equiv 23.503\text{deg}$

$P \equiv 2000 \cdot \text{lbf}$

$P = 8.896 \times 10^3 \text{N}$

$Py := P \cdot \cos(\theta)$

<u>Static friction force</u>

$Py = 8.158 \times 10^3 \cdot \text{N}$

$Py = 1.834 \times 10^3 \cdot \text{lbf}$

10. The torque needed to overcome the dome bogie to track steel friction (track misalignment and roller stiction)

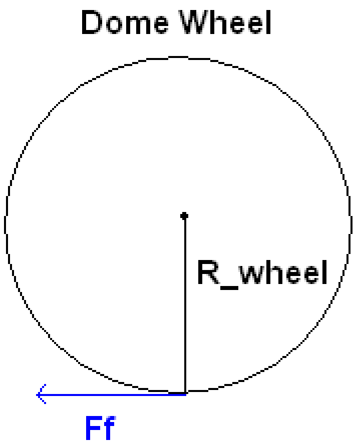

Figure 4: Photo is for visual purposes and is not to scale

Dome Wheel radius:

$$R\_wheel = 0.305 \cdot m$$

Friction force:

Ff = Friction force = Py

Torque needed to overcome the friction force:

$$Torque\_needed\_overcome\_friction := Py \cdot R\_wheel$$

<u>Drive wheel torque needed to overcome friction using a single dome wheel for rotation</u>

$$Torque\_needed\_overcome\_friction = 2.487 \times 10^{3} \cdot N \cdot m$$

Shared load for three (3) wheels would be 1/3 that value.

11. The torque needed per drive unit to overcome the dome bogie to track steel friction (track misalignment and roller stickton)

Since there are three drive units we need to divide the torque by 3:

$$Torque\_needed\_overcome\_friction\_per\_drive\_wheel := \frac{Torque\_needed\_overcome\_friction}{3}$$

<u>Torque to over come friction per drive wheel</u>

$$Torque\_needed\_overcome\_friction\_per\_drive\_wheel = 828.892 \cdot N \cdot m$$

$$Torque\_needed\_overcome\_friction\_per\_drive\_wheel = 611.359 \cdot lbf \cdot ft$$

11.1 The physical slow speed measurments made at the hydraulic motor will be used as a comparison and check with the physical friction force measured in section 11

Refrence Appendix B 15-03-002 Dome Drive Data 8-4-2010

for the slow speed values taken at the hydraulic motor

The dome rotates at 13 deg/min at slow speed

$$T\_hydraulic\_motor\_at\_slow\_speed\_per\_drive \equiv 4.5 \cdot lbf \cdot ft$$

$$T\_hydraulic\_motor\_at\_slow\_speed\_per\_drive = 6.101 \cdot N \cdot m$$

The torque at the drive wheel is equal to the torque at the drive motor times the mechaincal advantage of the gearbox

$$T\_wheel := T\_hydraulic\_motor\_at\_slow\_speed\_per\_drive \cdot MV\_Gear\_reduction\_box$$

$$T\_wheel = 488.094 \cdot N \cdot m$$

The radius of the drive wheel is:

$$R\_wheel = 0.305\,m$$

The force at the wheel is the torque at the wheel divided by the radius of the wheel:

$$F\_wheel := \frac{T\_wheel}{R\_wheel}$$

$$F\_wheel = 1.601 \times 10^{3} \cdot N$$

$$F\_wheel = 360 \cdot lbf$$

This value compares well with the static friction force calculated earlier

$$\frac{Py}{3} = 2.719 \times 10^{3}\,N$$

$$\frac{Py}{3} = 611.359 \cdot lbf$$

12. Determine the angular acceleration of the rotating dome at rest to full speed in 10 secs

- See section 5 above or *See Appendix B for 15-03-003 Dome Drive data 2-17-2010* for angular velocity (speed) of the dome

Angular Velocity (speed) of the dome:

at full speed

$\omega dome1 \equiv 0.01745$

- To calculate the Angular acceleration of the domefrom rest to full speed in t=10secs:

at t=0, rest

$\omega dome0 \equiv 0 \quad \frac{rad}{sec}$

It takes the dome 10 seconds to accelerate to full operating speed.

at full t=10 full speed

at t~10sec

$\Delta t \equiv 10 \cdot sec$

$\Delta\omega \equiv .01745 \cdot \frac{rad}{sec}$

$\alpha dome1 \equiv \frac{\Delta\omega}{\Delta t}$

Angular acceleration of the dome

$\alpha dome1 = 1.745 \times 10^{-3} \frac{1}{s^2} \quad rad$

13. The torque needed to accelerate the inertial dome mass per drive wheel

ΣF(forces) = M (mass) * A (acceleration)

which is closely related to

ΣT(torques) = I (inertia) * α (angular acceleration)

- The Torque needed to accelerate the dome interial mass:

T = I * α

$$I\_per\_drive = 1.131 \times 10^{7}\,m^{2}\cdot kg$$

$$\alpha dome1 = 1.745 \times 10^{-3}\,\frac{1}{s^{2}}$$

$$Torque\_dome\_inertia\_per\_drive\_wheel := I\_per\_drive \cdot \alpha dome1$$

Torque needed to accelerate the inertial dome mass per drive wheel

$$Torque\_dome\_inertia\_per\_drive\_wheel = 1.973 \times 10^{4}\cdot N\cdot m$$

14. The torque needed to rotate the dome

- Next we have to add the torque needed to overcome the friction per drive unit and the torque needed to accelerate the inertial mass of the dome per drive unit:

Therefore the total torque needed to accelerate the dome and overcome static friction:

$$Torque\_rotate\_dome\_per\_drive\_wheel := Torque\_dome\_inertia\_per\_drive\_wheel + Torque\_needed\_overcome\_friction\_per\_drive\_wheel$$

Torque needed to rotate the dome mass per drive wheel

$$Torque\_rotate\_dome\_per\_drive\_wheel = 2.056 \times 10^{4}\cdot N\cdot m$$

$$Torque\_rotate\_dome\_per\_drive\_wheel = 1.517 \times 10^{4}\cdot lbf\cdot ft$$

15. The torque needed by the motor to rotate the dome

- In order to find the torque needed by the motor to rotate the dome we have to add the torque needed to rotate the inertial mass of the dome and torque needed to overcome the friction:
- *See Appendix K for Brevini gear box efficiency*

$$Gear\_box\_Efficiency \equiv 0.87$$

$$Torque\_motor := \frac{Torque\_rotate\_dome\_per\_drive\_wheel}{MV\_Total\_Gear\_Reduction \cdot Gear\_box\_Efficiency}$$

$$MV\_Total\_Gear\_Reduction = 3.66 \times 10^{3}$$

Torque needed by the motor to rotate the dome

$$Torque\_motor = 6.458 \cdot N \cdot m$$

$$Torque\_motor = 4.763 \cdot lbf \cdot ft$$

16. The Power needed by the motor (motor size) to overcome the breakaway (torque) from static friction and the inertia of the dome at full speed.

- Power = Torque x angular velocity

P = T * ω

$$\omega motor = 63.879$$

$$Power\_motor\_overcome\_static\_friction := Torque\_motor \cdot \omega motor$$

Motor power (motor size) needed over come static friction and begin accelerating the dome

$$Power\_motor\_overcome\_static\_friction = 412.537 \cdot N \cdot m$$

$$Power\_motor\_overcome\_static\_friction = 304.272 \cdot lbf \cdot ft$$

NOTE :

There is quite a descrepancy between this value and the minumum and maximum running hydraulic motor power output which was physically measured and calculated above in section 6.1.

$$Minimum\_running\_hydraulic\_motor\_power = 1.565 \cdot hp$$

$$Maximum\_running\_hydraulic\_motor\_power = 3.365 \cdot hp$$

Some reasons for the descreapancy are the physical measurments of the pressure and flow used to determine the power do not take into account many of the loses and efficiencies within the system.

1. The total pump efficency (ηtotal) affects the power ouput, Power input = Power output / η total were ηtotal = volumetric effiecency (ηvol) x hyyromechanical efficiency (ηhm).
2. The hydraulic motor, lines, elbows, and fittings also have individual efficiencies which affect the overall system effeciency.
3. The pressure drop in the hydraulic lines and valves also affect the overall system efficiecny.
4. The overall system efficiency is also affected by the energy lost from heat dissipated in the oil coolers (heat exchangers) to keep the hydraulic fluid cool during operation. This results in losses of input power feed to the hydraulic power unit.

- In conclusion we need to equip each dome drive unit with an electric motor that at least matches the

$$\text{Maximum\_running\_hydraulic\_motor\_power} = 3.365 \cdot \text{hp}$$

to be capable of rotating the dome at a full speed of 60 deg/sec.

- When determining the motor specifications the following equation must be greater or equal to the HP above:

HP = E x I x %eff x PF x 1.73 / 746

Where E is the volts the motor draws, I is the current draw in amperes, %eff is the motor efficiency rating, PF is the power factor, and 1.73 is the sqrt of the phase.

The derating of the motor at an altitude of 14,000ft needs to be incorporated into the motor size and specifications as well, please see section 18. for de-rating calculations.

17. The reflected inertia thru the drive train or total gear reduction (torque) ratio

- Since this is a gear reduction application and there is mechaincal linkages between the dome load and the motor, the load paramteres must be reflected back to the motor shaft.

$$\text{total\_motor\_reflected\_load\_inertia} := \frac{\text{I\_per\_drive}}{(\text{MV\_Total\_Gear\_Reduction})^2}$$

Therefore the reflected load moment of inertia which each motor would see is:

Reflected inertia from dome on the motor shaft

$$\text{total\_motor\_reflected\_load\_inertia} = 0.844 \cdot \text{kg} \cdot \text{m}^2$$

Note : To obtain better servo performance of the system, it is benefical to keep the motor and load intertias low. Since inertia is inversely proportional to the resonance frequency, a lower interia value pushes the resonance frequency out which enables a higher bandwidth and hence stiffer servo system.

See Appendix M

- From page 5, the WK^2 value (motor rotor inertia):

From page 5, the WK^2 value (motor rotor inertia):

$$\text{Motor\_rotor\_inertia} \equiv 1.92 \cdot \text{lb} \cdot \text{ft}^2$$

The torsional stiffness of the motor coupling:

$$CT \equiv 100 \cdot \frac{\text{N} \cdot \text{m}}{\text{rad}}$$

The resonance frequency

$$Fe := \left(\frac{1}{2 \cdot \pi}\right) \cdot \sqrt{CT \cdot \left[\left(\frac{1}{\text{Motor\_rotor\_inertia}}\right) + \left(\frac{1}{\text{total\_motor\_reflected\_load\_inertia}}\right)\right]}$$

$$Fe = 5.857 \cdot \text{Hz}$$

- In Conclussion the resonance frequency responce of the system provides a resonable servo system to control the dome.

18. Determine the motor and motor controller de-rating due to high altitude:

- Information on motor controllers and de-rating for elevation provided by Krieg Richards at Baldor

  - **Altitude de-rating.** Up to 3300 feet (1000 meters) no de-rating required. Above 3300 ft, derate the continuous and peak output current by 2% for each 1000 ft.

The calculation is 2% for every 1000 feet above 3300 feet.

Mauna Kea is at 14,000ft. Therefore

Altitude 14,000 –3,300/1000 x 2 = 21.4%

More conservative method:

The calculation is 1% for every 330 feet above 3300 feet.

Mauna Kea is at 14,000ft. Therefore

Altitude 14,000 –3,300 = 10,700 / 330 = 32.42%

More conservative De-rating of Baldor Reliance ZD22HLine Reactor motor controllers and RPM AC motors

$$\text{Motor\_controller\_Altitude\_Derating} \equiv .3242$$

Continuous and Peak amps rating on the motor and drive will have to be derated by 32.4%.
Operated at no more than 67.6% of sea level rating.

19. Determine the power loss (heat production) in the motors:

Information on motor efficiency provided by Bill Colten at Baldor

Efficiency of Baldor Reliance ZDFRPM21204C 20HP RPM-AC finned frame variable speed alternating current motor series

$$\text{RPM\_motor\_Efficiency} \equiv .94$$

- This inefficiency will also dictate the additional current that will need to be supplied to the motors to get the actual motor shaft torque (due to the loss) from the inefficiency.

$$\text{RPM\_motor\_inefficiency} \equiv 0.06$$

- For a 20HP motor with NO LOAD on the motor
- i.e. No mechanical load on the motor shaft but the shaft is spinning.

Appendix M, page 5: the load performance at base speed specifications for Power factor and NO LOAD amperage.

- Motor voltage

$$E \equiv 460 \cdot V$$

- Motor current

$$\text{motor\_amps\_no\_load} \equiv 14.4\text{amp}$$

$$\text{I\_no\_load} \equiv \text{motor\_amps\_no\_load}$$

- Motor Phase

$$\text{I\_no\_load} = 14.4 \cdot A$$

$$\text{Motor\_Phase} \equiv 3$$

- Motor power factor

$$\text{PF\_no\_load} := 0.066$$

$$\text{Motor\_power\_loss\_no\_load} := \left(\sqrt{\text{Motor\_Phase}}\right) \cdot E \cdot \text{I\_no\_load} \cdot \text{PF\_no\_load} \cdot \text{RPM\_motor\_inefficiency}$$

<u>Motor Power loss (heat production) at NO (mechanical) LOAD</u>

$$\text{Motor\_power\_loss\_no\_load} = 45.433 \cdot W$$

- For a 20HP motor with FULL LOAD on the motor
  i.e. Maximum mechanical load coupled to the motor shaft rotating at full speed.

Appendix M, page 5: the load performance at base speed specifications for Power factor and NO LOAD amperage.

- Motor voltage

$$E \equiv 460 \cdot V$$

- Motor current

$$motor\_amps\_full\_load \equiv 26.9amp$$

$$I\_full\_load \equiv motor\_amps\_full\_load$$

$$I\_full\_load = 26.9\ A$$

- Motor Phase

$$Motor\_Phase \equiv 3$$

- Motor power factor

$$PF\_Full\_load := 0.777$$

$$Motor\_power\_loss\_full\_load := \left(\sqrt{Motor\_Phase}\right) \cdot E \cdot I\_full\_load \cdot PF\_Full\_load \cdot (RPM\_motor\_inefficiency)$$

<u>Motor Power loss (heat production) at FULL LOAD</u>

$$Motor\_power\_loss\_full\_load = 999.178\ W$$

- A 1/6 load on the motor represents the estimated running motor load a 20HP motor will see when running at full speed using the calculated hydraulic maximum motor power value.

$$\text{Maximum\_running\_hydraulic\_motor\_power} = 3.365 \cdot \text{hp}$$

$$\text{MOTOR\_LOAD} \equiv \frac{\text{Maximum\_running\_hydraulic\_motor\_power}}{20 \cdot \text{hp}}$$

Load a 20 hp motor running at full speed will endure

$$\text{MOTOR\_LOAD} = 0.168$$

- For a 20HP motor rotating the dome at 60deg/sec rusults in a 1/6 LOAD on the motor

Appendix M, page 5: the load performance at base speed specifications for Power factor and 1/6 LOAD amperage.

- Motor voltage

$$E \equiv 460 \cdot V$$

- Motor current

Value found from linear interpolation

$$\text{motor\_amps\_sixth\_load} \equiv 15.375\text{amp}$$

$$\text{I\_sixth\_load} \equiv \text{motor\_amps\_sixth\_load}$$

$$\text{I\_sixth\_load} = 15.375\,\text{A}$$

- Motor Phase

$$\text{Motor\_Phase} \equiv 3$$

- Motor power factor

Value found from linear interpolation

$$\text{PF\_sixth\_load} := .1295$$

$$\text{Motor\_power\_loss\_running\_load} := \left(\sqrt{\text{Motor\_Phase}}\right) \cdot E \cdot \text{I\_sixth\_load} \cdot \text{PF\_sixth\_load} \cdot (\text{RPM\_motor\_inefficiency})$$

Motor Power loss (heat production) at 1/6 LOAD

$$\text{Motor\_power\_loss\_running\_load} = 95.182\,\text{W}$$

20. Determine the power loss (heat production) in the motor controller drive units:

See Appendix G

- Use Page 13 (Table 3-1) and Page 93 (Series 22H Vector Control Ratings) from Baldor Series 22H Line Regen AC Flux Vector Control Installation & Operating Manual MN722

See Appendix M

- Use page 5 (AC performance Data) from Baldor part information packet ZDFRPM21204C
- 20HP, 1750RPM, 3PH, 60HZ, 2162C, TEFC, FOOT

  - Calculate the heat losses while the motor controller is enabled and at magnetizing current (No motor Load):

    Information on motor controller calculations provided by Krieg Richards at Baldor:

- For a 25HP motor controller and a 20HP motor with NO LOAD on the motor

From page 5, the load performance at base speed specifies that the amps at NO LOAD is:

$$\text{motor\_amps\_no\_load} \equiv 14.4\text{amp}$$

From page 93, the Continuous output current (IC) in amps is:

$$\text{motor\_controller\_amps\_no\_load} \equiv 34\text{amp}$$

From page 13, the STD PWM CONV & INT (this encompasses all of the losses at no load) in watts is:

$$\text{motor\_controller\_total\_losses} \equiv 544\text{W}$$

$$\text{Motor\_controller\_power\_loss\_NO\_LOAD} := \left(\frac{\text{motor\_controller\_total\_losses}}{\text{motor\_controller\_amps\_no\_load}}\right)\cdot \text{motor\_amps\_no\_load}$$

Motor controller Power loss (heat production) at NO LOAD $\text{Motor\_controller\_power\_loss\_NO\_LOAD} = 230.4\ \text{W}$

- For a 25HP motor controller and a 20HP motor with FULL LOAD on the motor

  From page 5, the load performance at base speed specifies that the amps at FULL LOAD is:

$$\text{motor\_amps\_full\_load} \equiv 26.9\text{amp}$$

  From page 93, the Continuous output current (IC) in amps is:

$$\text{motor\_controller\_amps\_full\_load} \equiv 58\text{amp}$$

  From page 13, the STD PWM total losses (this encompasses all of the losses at full load) in watts is:

$$\text{motor\_controller\_total\_losses} \equiv 834\text{W}$$

$$\text{Motor\_controller\_power\_loss\_FULL\_LOAD} := \left(\frac{\text{motor\_controller\_total\_losses}}{\text{motor\_controller\_amps\_full\_load}}\right)\cdot\text{motor\_amps\_full\_load}$$

<u>Motor controller Power loss (heat production) at FULL LOAD</u>

$$\text{Motor\_controller\_power\_loss\_FULL\_LOAD} = 386.803\ \text{W}$$

21. Determine the Maximum motor input torque allowed to not exceed the drive train rating and capacity.

21.1 Brevini Gear Reduction Box

See Appendix K

- Reference document: Brevinvi-EC3090 Specifications sheet

The Maximum output torque capable on the EC 3090 gear reduction box is:

$$T2 \equiv 15000 \cdot N \cdot m$$

$$Maximum\_gearbox\_output\_torque := T2$$

After the gear box reduction (torque) ratio the torque is:

$$Maximum\_motor\_input\_torque := \frac{Maximum\_gearbox\_output\_torque}{MV\_Gear\_reduction\_box}$$

<u>Maximum motor input torque-capacity limitation at 60deg/sec continuous spped</u>

$$Maximum\_motor\_input\_torque = 187.5 \cdot N \cdot m$$

$$Maximum\_motor\_input\_torque = 138.293 \cdot lbf \cdot ft$$

<u>Note: The maximum output torque from a 20hp motor is 40.6 lbf x ft at 14,000ft altitude, see Appendix AA, therefore the 20hp motor at full torque would not be able to damage the drive train.</u>

21.2 Determine the SHP main drive shaft capacity and rating for the dome drive unit.

See Appendix E

- Reference document: SHP Original Equipment Manufacturer (OEM) specifications- Dome drive
- SHP 5075-F pg 3 of 4, Transmission and wheel assembly

The drive shaft is an Ultimo 4, HTSR (heat treated stress relieved shaft). 6" diameter x 44.125" long

22. Determine the SHP Wheel Assembly capacity and rating for the dome drive unit.

- In an email from Gilles Dionne (Directeur Régional) who consulted with Emile Mortier (Consultant technique) at Bosch Rexroth Canada Corp, who purchased SHP some years back. Can be referenced below:

The SHP main drive shaft and SHP wheel assembly are very old and nobody is left at the company to consult with on the design, ratings, or capacity of these components. However the SHP main drive shaft and wheel assembly are way over designed for the dome drive application. The weakest point in the drive train is the Brevini Gear Reduction box, therefore the motor for driving the system should be chosen with the max input torque for the gear reduction box in mind.

23. Determine the wind load required to move the dome building when all three dome wheels are disengaged from the track, i.e. allowed to rotate freely on the track with only friction of the track counter-acting the motion.

- On January 4th, 2010 a test was performed at the summit by S.Bauman and R. Taroma with the dome due to the large winds that were present at the summit. The winds ranged from 35-45 knots most of the day and were prevalent out of the S and W.
  - The Dome building was rotated to 0.3 deg and stopped. This position was verified with the TCS console. Then all three (3) of the dome drive wheels were lifted off the track to allow the dome building to rotate freely.
    - A 44 knot wind was coming out of the S-SW. The building did not rotate any during the 10 minute test. The movement was verified and confirmed by the TCS console.
- An additional test was performed where the Dome building was rotated to 270 deg and stopped. This position was verified with the TCS console. Then all three (3) of the dome drive wheels were lifted off the track to allow the dome building to rotate freely.
  - A 45 knot wind was coming out of the W-SW. The building did not rotate any amount during the 10 minute test. The movement was verified and confirmed by the TCS console.

- In conclusion the dome will not rotate from a wind load of less 50 knots with shutter closed. The observatory shutdown wind limits and directions specify that the telescope dome shall not be open if sustained winds exceed any of the following speeds:

---

1. Sustained wind speed of 50 knots or 58mph
2. Gust speed of 65 knots or 75mph (gust)

<u>A useful experiment would be to perform a similar test with the dome shutter open and see how the opening and wind loading on the dome at various locations would influence and possibly change the dome drive horsepower needed by each dome drive unit.</u>

24. The mass Moment of inertia for the Dome with a 50.8mm (2.0") thick layer of Ice on the outside skin of the dome

Assumptions:

- Assume that the dome is a perfect hollow half sphere to streamline calculations.

Mass of the dome: 565 tons

Mass of the upper end instrument 12 tons

$$mass\_dome \equiv 577 \cdot ton$$

$$mass\_dome = 5.234 \times 10^{5} \cdot kg$$

Mass of the layer of Ice on the dome:

- Density = Mass / volume

Therefore

- Mass = density x volume

The density of water @ 40 degrees C

$$\rho\_water \equiv 999.8395 \cdot \frac{kg}{m^{3}}$$

The Radius of the dome and the Radius of the Ice layer

$$R\_dome = 13.945\,m$$

$$R\_Ice \equiv 13.9954 \cdot m$$

Thickness of the Ice layer

$$Thickness\_Ice := R\_Ice - R\_dome$$

$$Thickness\_Ice = 50.8 \cdot mm$$

$$Thickness\_Ice = 2 \cdot in$$

Volume of the Ice layer

$$V\_Ice\_layer := \frac{4}{3} \cdot \pi \cdot \left(R\_Ice^{3} - R\_dome^{3}\right)$$

$$V\_Ice\_layer = 124.585 \cdot m^{3}$$

Mass of the Ice layer

$$mass\_of\_ice\_dome := \rho\_water \cdot V\_Ice\_layer$$

$$mass\_of\_ice\_dome = 137.31 \cdot ton$$

$$mass\_of\_ice\_dome = 1.246 \times 10^{5}\, kg$$

The Moment of Inertia for the dome with a 50.8mm ice layer:

$$mass\_dome\_and\_ice\_layer \equiv mass\_dome + 137.31 \cdot ton$$

$$mass\_dome\_and\_ice\_layer$$

$$mass\_dome\_and\_ice\_layer = 6.48 \times 10^{5}\, kg$$

I = [2 x Mass dome x (Radius dome)^2 / 3] / 2 (half of the sphere)

$$I\_dome\_ice \equiv \left[\left(\frac{2}{3}\right) \cdot \left[(mass\_dome\_and\_ice\_layer) \cdot (R\_dome)^{2}\right]\right] \cdot \left(\frac{1}{2}\right)$$

mass moment of inertia for the dome with a 50.8mm ice layer

$$I\_dome\_ice = 4.2 \times 10^{7} \cdot kg \cdot m^{2}$$

25. The mass Moment of inertia for the Dome with a 50.8mm (2.0") thick layer of Ice on the outside skin of the dome per drive wheel

Since there are three (3) drive units (wheels) we divide the mass moment of inertia of the dome by 3:

$$I\_per\_drive\_wheel\_ice := \frac{I\_dome\_ice}{3}$$

Mass moment of inertia per drive wheel (unit) with a 50.8mm ice layer

$$I\_per\_drive\_wheel\_ice = 1.4 \times 10^{7} \cdot kg \cdot m^{2}$$

26. The torque needed to accelerate the inertial dome mass with a 50.8mm (2.0") thick layer of Ice per drive wheel (unit)

ΣF(forces) = M (mass) * A (acceleration)

which is closely related to

ΣT(torques) = I (inertia) * α (angular acceleration)

- The Torque needed to accelerate the dome inertial mass with ice layer:

T = I * α

$$I\_per\_drive\_wheel\_ice = 1.4 \times 10^{7}\ m^{2} \cdot kg$$

$$\alpha dome1 = 1.745 \times 10^{-3}\ \frac{1}{s^{2}}$$

$$Torque\_dome\_inertia\_per\_drive\_wheel\_ice := I\_per\_drive\_wheel\_ice \cdot \alpha dome1$$

<u>Torque needed to accelerate the inertial dome and ice mass per drive wheel (unit)</u>

$$Torque\_dome\_inertia\_per\_drive\_wheel\_ice = 2.443 \times 10^{4} \cdot N \cdot m$$

27. The torque needed to rotate the dome with a 50.8mm (2.0") thick layer of Ice

- Next we have to add the torque needed to overcome the friction per drive unit and the torque needed to accelerate the inertial mass of the dome with ice per drive wheel (unit):

Therefore the total torque needed to accelerate the dome with ice and overcome static friction:

$$Torque\_rotate\_dome\_per\_drive\_wheel\_ice := Torque\_dome\_inertia\_per\_drive\_wheel\_ice + Torque\_needed\_overcome\_friction\_per\_drive\_whe$$

<u>Torque needed to rotate the dome mass with ice per drive wheel (unit)</u>

$$Torque\_rotate\_dome\_per\_drive\_wheel\_ice = 2.526 \times 10^{4} \cdot N \cdot m$$

$$Torque\_rotate\_dome\_per\_drive\_wheel\_ice = 1.863 \times 10^{4} \cdot lbf \cdot ft$$

28. The torque needed by the motor to rotate the dome

- In order to find the torque needed by the motor to rotate the dome we have to add the torque needed to rotate the inertial mass of the dome with ice and the torque needed to overcome the friction:

$$Gear\_box\_Efficiency \equiv 0.87$$

$$Torque\_motor\_ice := \frac{Torque\_rotate\_dome\_per\_drive\_wheel\_ice}{MV\_Total\_Gear\_Reduction \cdot Gear\_box\_Efficiency}$$

$$MV\_Total\_Gear\_Reduction = 3.66 \times 10^{3}$$

Torque needed by the motor to rotate the dome with ice

$$Torque\_motor\_ice = 7.933 \cdot N \cdot m$$

29. The Power needed by the motor (motor size) to overcome the breakaway (torque) from friction and the inertia of the building with ice at full speed.

- Power = Torque x angular velocity

P = T * ω

$$\omega motor = 63.879 \ \frac{rad}{sec}$$

$$Power\_motor\_ice := Torque\_motor\_ice \cdot \omega motor$$

Motor power (motor size) needed per drive wheel to rotate the building with ice layer

$$Power\_motor\_ice = 373.761 \cdot lbf \cdot ft$$

$$Power\_motor\_ice = 506.752 \cdot N \cdot m$$