NRC MOV Course

Design Basis Operation Continued

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Required Torque

The analysis determines the required torque for the valve's design basis, including

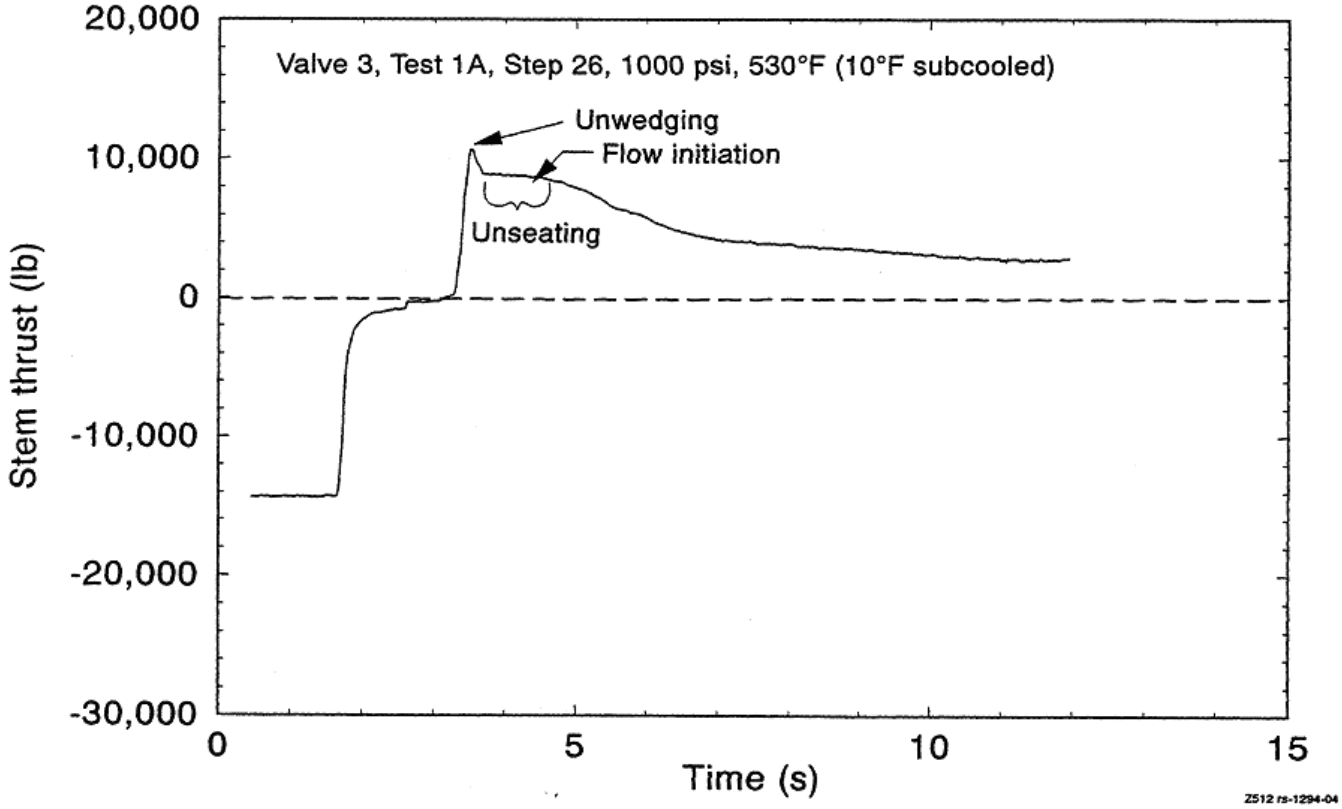
- *Packing load*
- *Stem rejection load*
- *Differential pressure (flow, seat friction, and guide friction)*
- *Stem factor (conversion of torque to thrust, including stem thread friction)*

Basic Stem Thrust Equation

$$F_{Stem} = F_{Packing} \pm F_{Stem\ rejection} + F_{Disc}$$

- *Note the sign for the stem rejection load. This appears because the stem rejection load is always out of the valve body, thus it resists closure and assists opening.*

Typical Opening Stroke Stem Thrust Measurement



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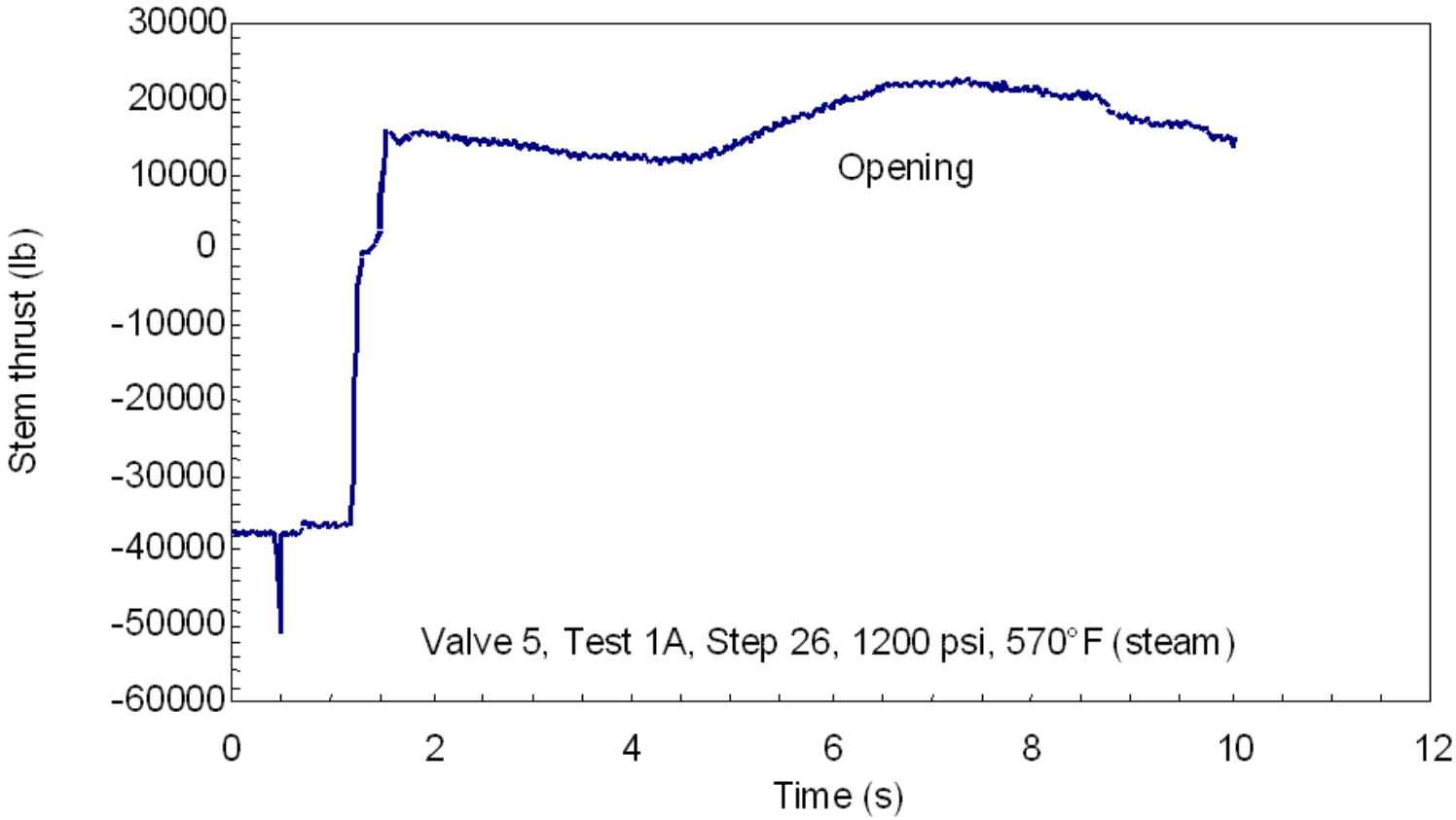
Atypical Opening Stroke Stem Thrust Measurement



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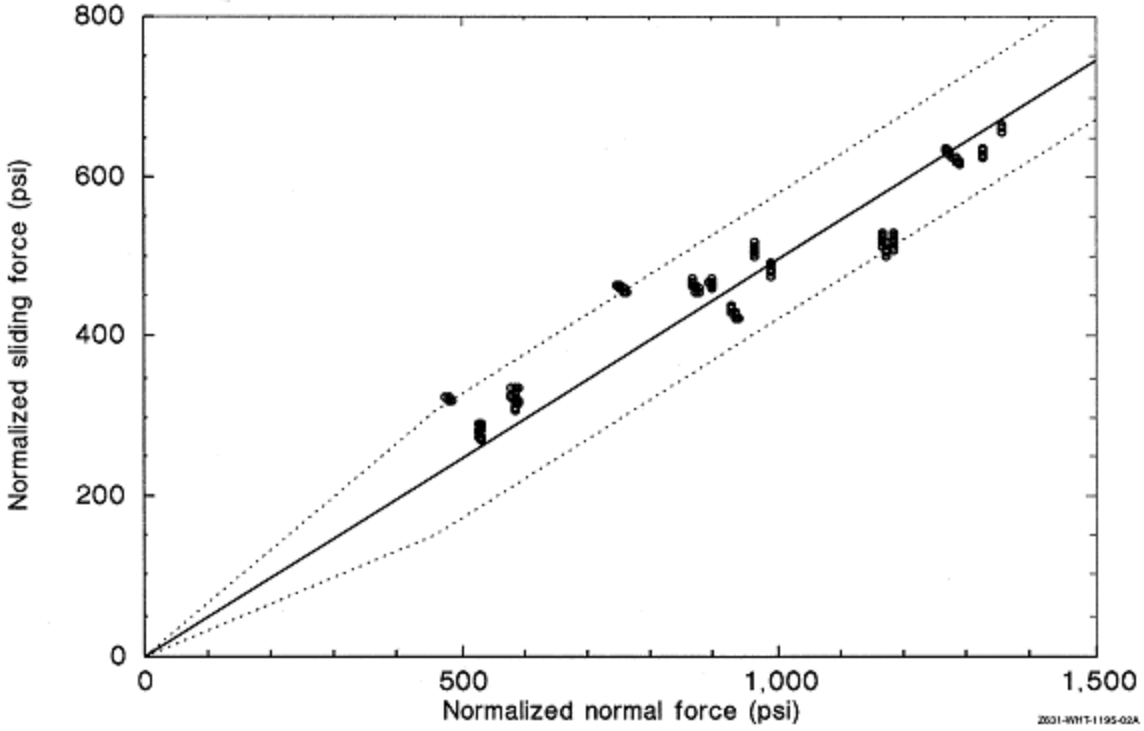
INL Correlation Bounds For Predicting Net Stem Thrust



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INL Gate Valve Opening Correlations

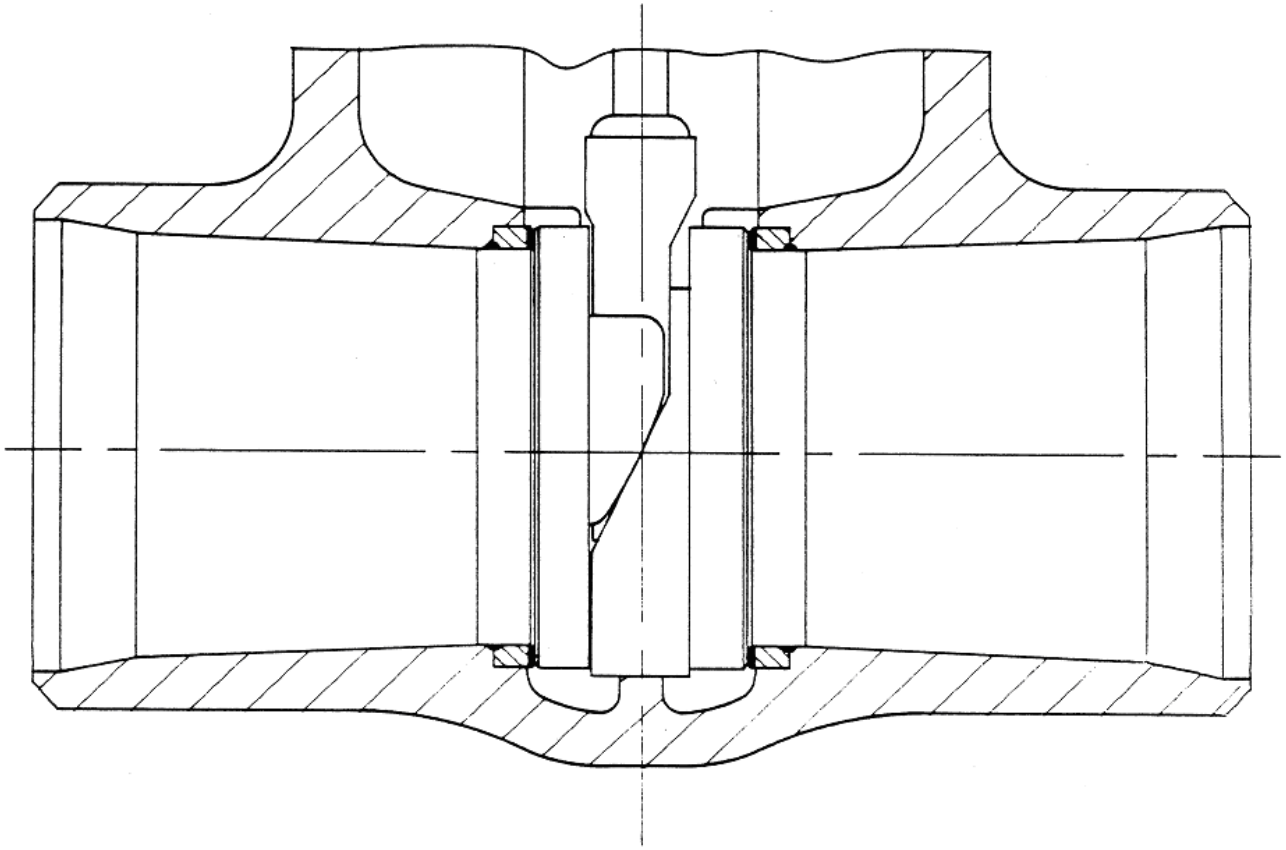
For DP > 415 psi

$$F_{stem} = F_{pack} - F_{stem\,rej} + F_{elps} + \frac{\Delta P A_{disc}\left(f\cos\alpha + \sin\alpha\right) \pm 80 A_{disc}}{\cos\alpha - f\sin\alpha}$$

For DP < 415 psi

$$F_{stem} = F_{pack} - F_{stem\,rej} + F_{elps} + \frac{\Delta P A_{disc}\left(f\cos\alpha\left(1.0 \pm 0.35\right) + \sin\alpha\right)}{\cos\alpha - f\sin\alpha\left(1.0 \pm 0.35\right)}$$

Anchor/Darling Parallel Disc Gate Valve



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Parallel Disc Gate Valve Equations

For parallel disc gate valves, the disc angle is zero. Therefore, the previous INL equations become

For Closing:

$$F_{stem} = F_{pack} + F_{stem\ rej} + \Delta P A_{disc} f$$

For Opening:

$$F_{stem} = F_{pack} - F_{stem\ rej} + \Delta P A_{disc} f$$

Typical Industry Methods and Alternative Methods

- *Since the mid-1980s, valve researchers, utilities, and utility organizations have developed several new analytical tools for evaluating MOVs.*
- *Though the tools are varied in certain details, they represent basically the same formula.*
- *We refer to this as the standard industry equation.*

Standard Industry Equation

$$F_{stem} = F_{pack} \pm F_{stem\ rej} + \Delta P A_{disc} \mu_d$$

Where

F_{stem}	=	*total stem load*
F_{pack}	=	*stem packing load*
$F_{stem\ rej}$	=	*stem rejection load*
ΔP	=	*differential pressure across the disc*
A_{disc}	=	*disc area*
μ_d	=	*disc factor*

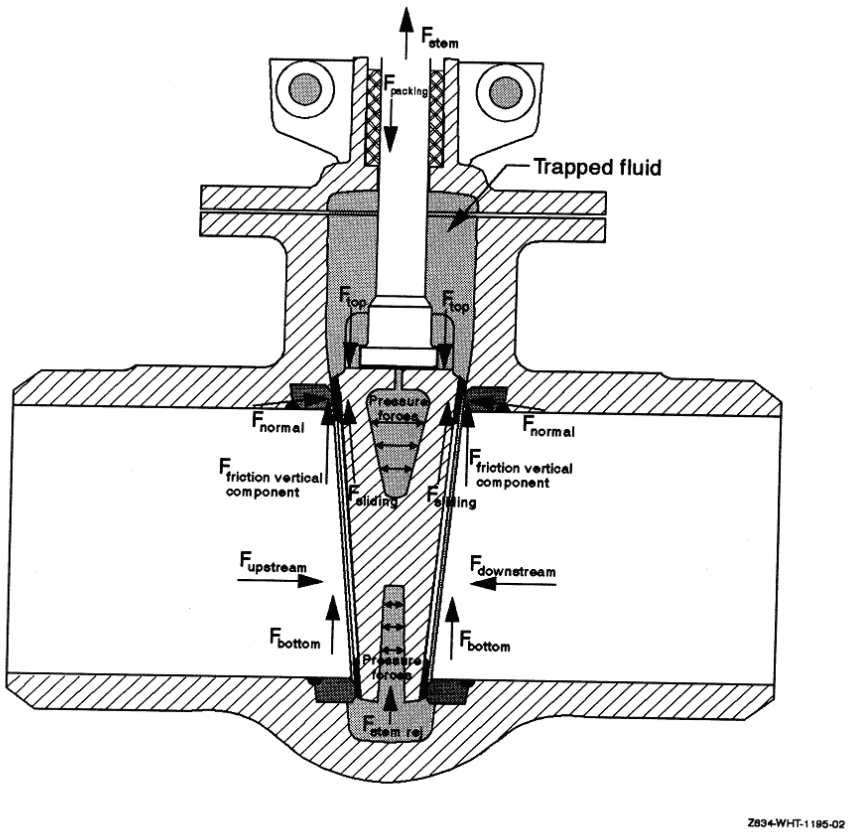
Standard Industry Equation Deficiencies

- *The disc factor of 0.3 was far too low; in some instances a disc factor of 0.5 was too low*
- *It failed to consistently specify the mean seat diameter as the basis for determining the disc area*
- *It failed to account for the elliptical pressure load on the top of the disc*
- *It failed to isolate the disc friction, instead including it in the disc factor along with other unknown or unspecified variables*

Pressure Locking

- *Pressure locking occurs when the valve bonnet pressure is higher than both the upstream and downstream pressures.*
- *The effect is that the pressure of the fluid between the discs acts on both the upstream and the downstream discs, introducing resistance to motion at both disc/seat interfaces rather than just one.*
- *This adds to the total force necessary to unwedge/unseat the valve disc.*
- *At its worst, pressure locking causes the valve to be locked in the closed position.*

Pressure Locked Gate Valve



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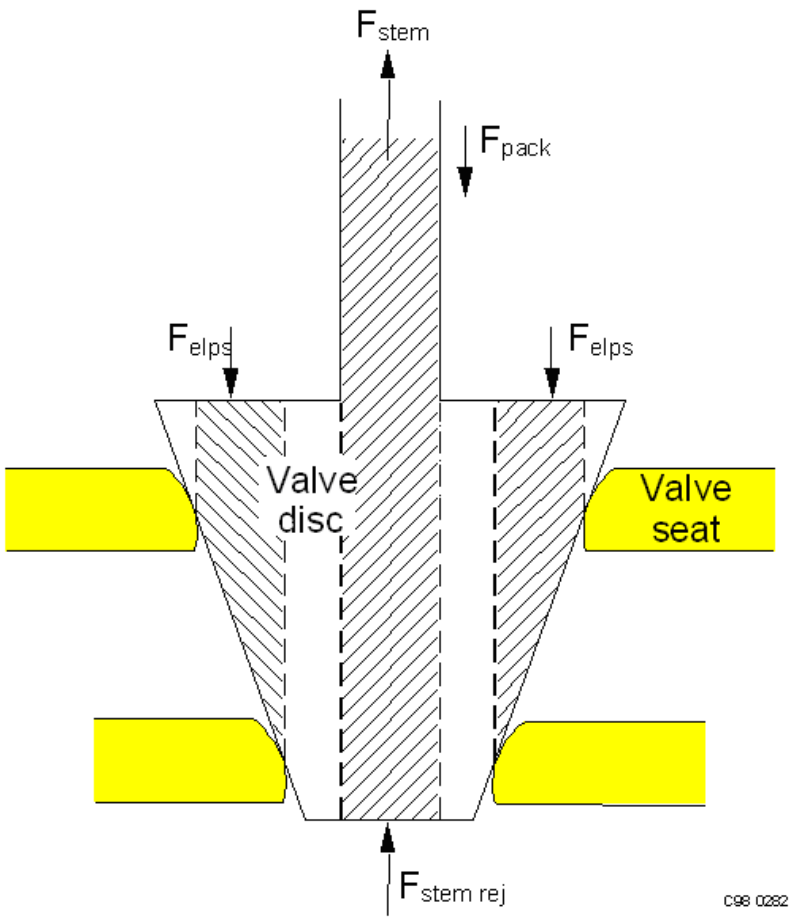
Pressure Locking Load Forces



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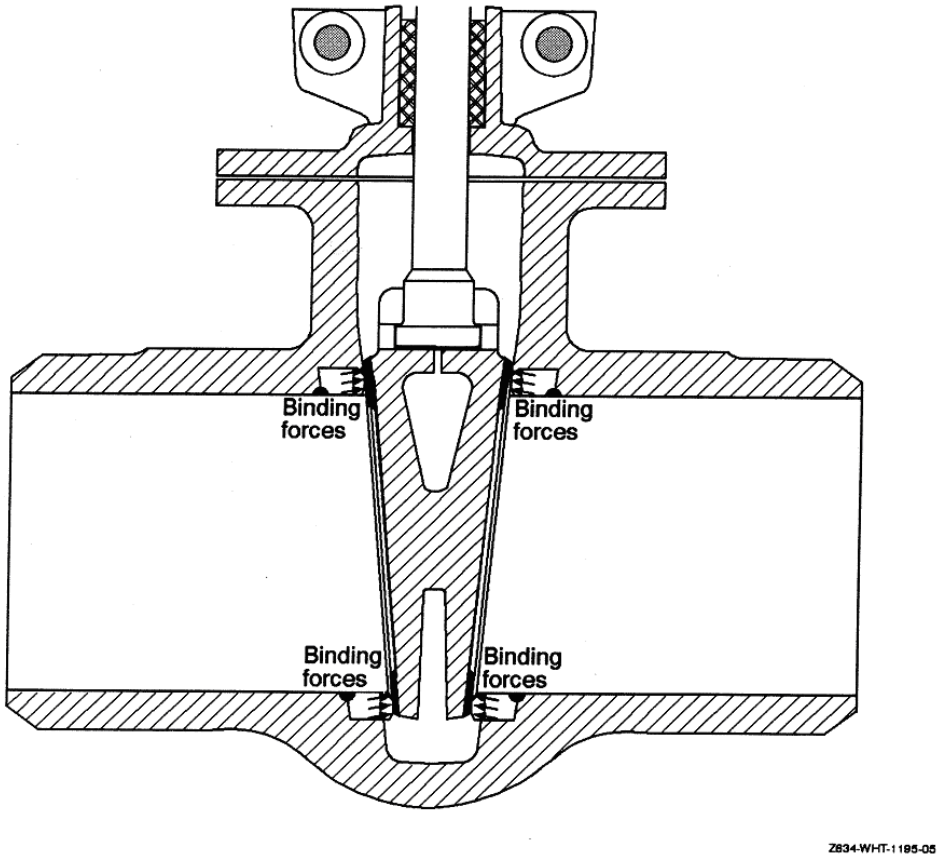
Pressure Locking Research Results

- *Disc factor during a pressure locking event is slightly higher than during a hydro test.*
- *Leakage can reduce the pressure loads associated with thermally-induced pressure locking, but may not prevent pressure locking.*
- *Entrapped air can delay the onset of thermally-induced pressure locking, but will not prevent it.*
- *Flow through the valve will gradually remove entrapped air pockets.*
- *Pressure locking loads can increase opening stem force by a factor of 1.1 to 1.5*

Thermal Binding

- *Thermal binding describes the effects of heatup and cooldown on differential expansion and contraction of the valve internals.*
- *Valves closed in the hot condition might experience thermal binding loads when the seat rings contract against the disc after cooldown.*
- *These binding loads may be minor, or they may be so large that the valve must be reheated in order to free the disc.*

Thermal Binding Forces



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Thermal Binding

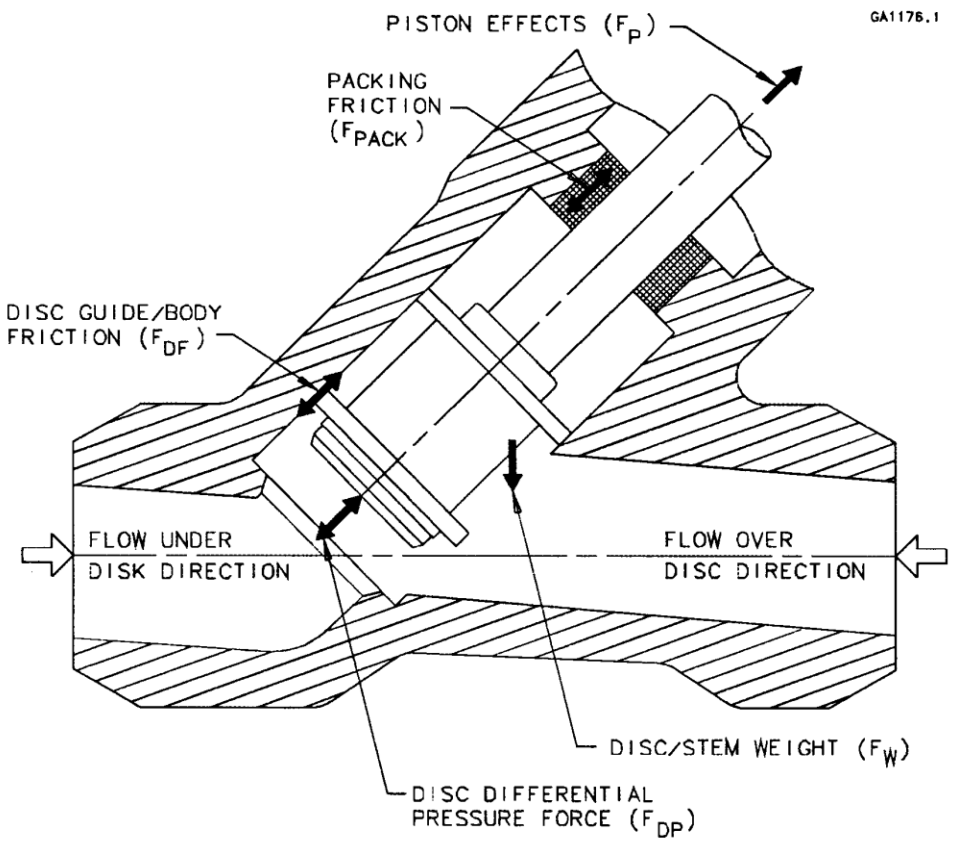
- *We know of no proven method for estimating the thermal binding loads*
- *A few three-dimensional finite-element models have been used to study this phenomena*
- *Industry controls this problem by cycling susceptible valves during heatup or cooldown.*

Globe Valve Stem Thrust

Globe Valve Stem Thrust

- Differential pressure force - can be positive or negative depending on flow direction.
- Friction force between the disc and guide - resists valve disc movement, positive for opening and negative for closing.
- Stem rejection force - pressure trying to expel the stem from the valve body - always negative.
- Disc and stem weight - can be positive or negative depending on valve orientation.
- Stem packing force - resists valve stem movement, positive for opening and negative for closing.
- Torque reaction force - prevents stem rotation, positive for opening and negative for closing.

Y Pattern Globe Valve Stem Thrust Components



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Globe Valve Closing Equation

$$F_{stem} = \pm F_{disc} - F_{stem\ rej} \pm F_{pack}$$

Where

F_{stem}	=	*total stem load*
F_{disc}	=	*disc differential pressure load*
$F_{stem\ rej}$	=	*stem rejection load*
F_{pack}	=	*stem packing load*

Note: Negative sign convention indicates compression in the valve stem or self-opening.

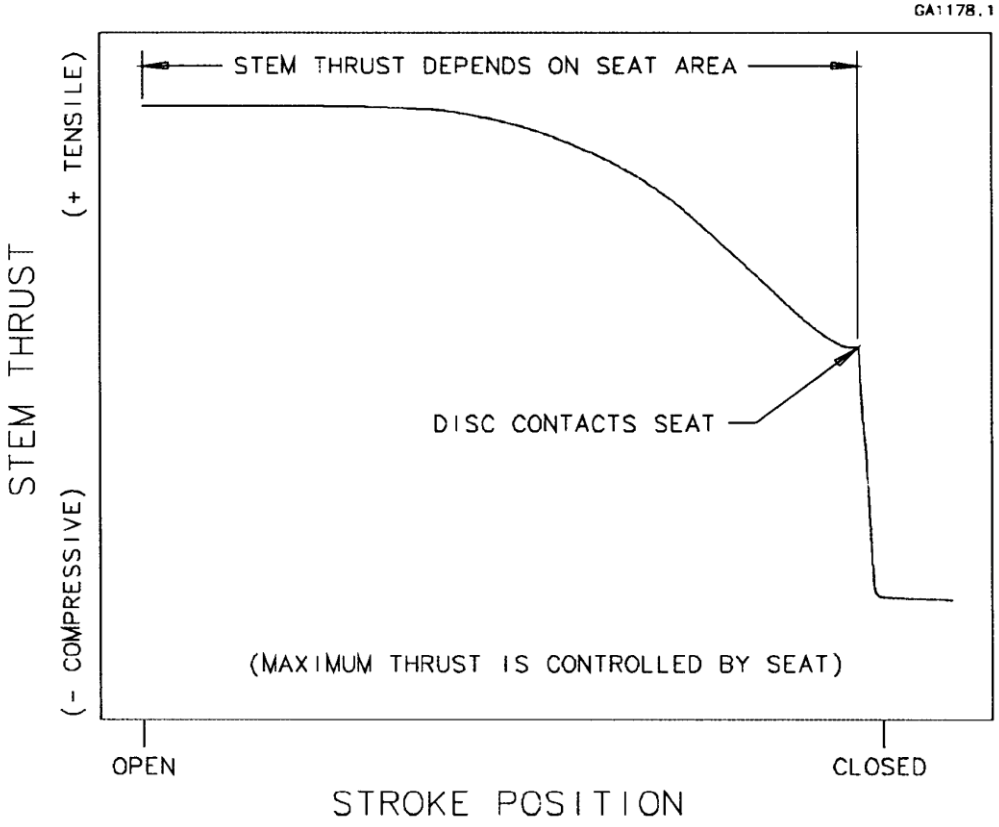
Globe Valve Closing Equation – Disc Differential Pressure Load

$$F_{disc} = f\left(P_{up} - P_{down}\right)A_{DP}$$

Where

F_{disc}	=	*disc differential pressure load*
f	=	*disc factor (1.1 is often used)*
P_{up}	=	*upstream pressure*
P_{down}	=	*downstream pressure*
A_{DP}	=	*area for differential pressure calculation (seat or guide)*

Typical Stem Thrust vs Stroke For Closing a Seat-Based Globe Valve



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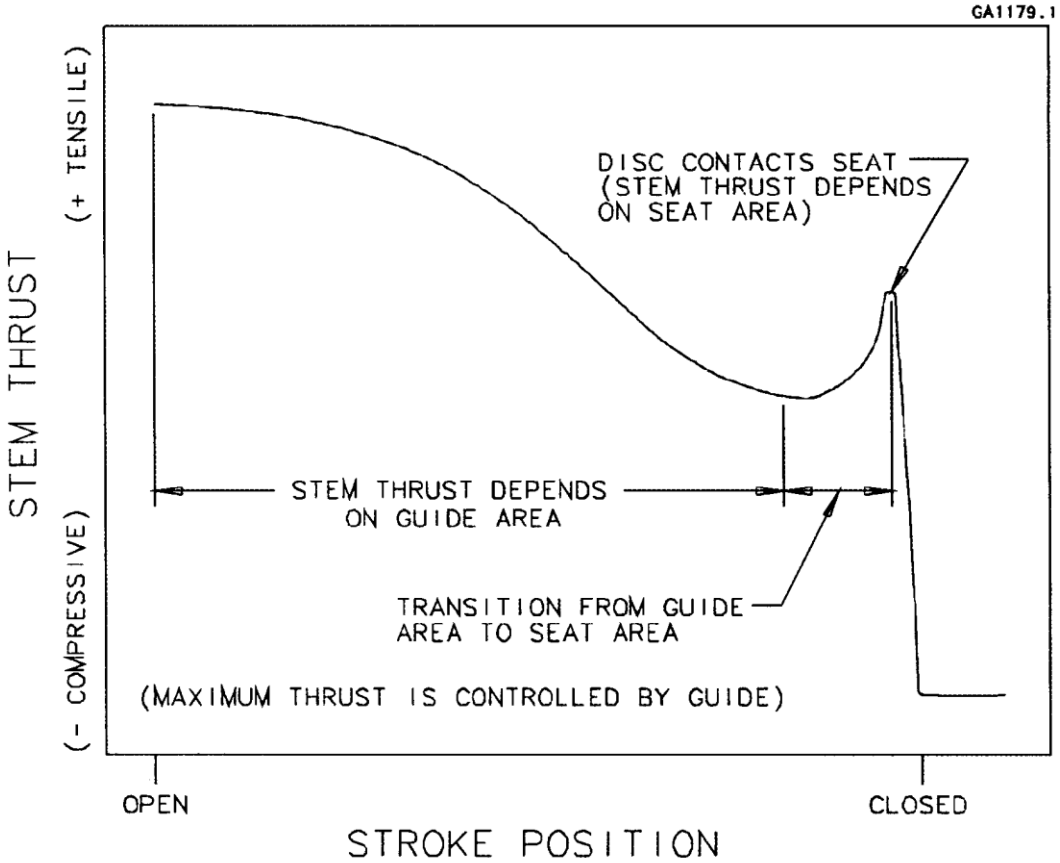
Typical Stem Thrust vs Stroke For Closing a Guide-Based Globe Valve



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Design Basis Valve Stem Torque

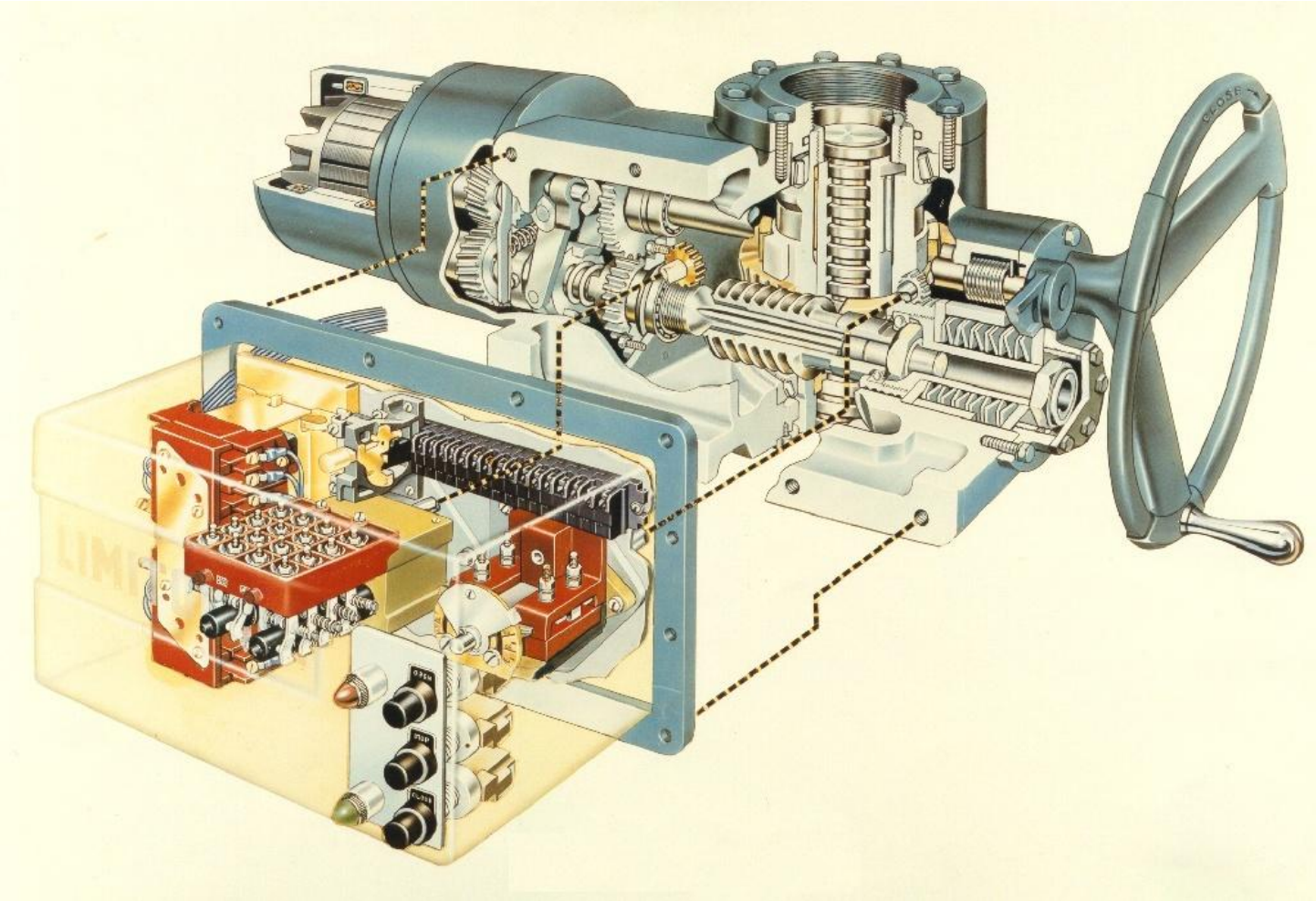
Rising-Stem Valve Stem Torque – Stem Factor

Stem Factor

- *In the previous section we determined the required stem thrust load.*

- *However, motor actuators do not directly produce thrust.*
  - *Motor actuators deliver torque to the stem nut*
  - *The control switches control output torque, not thrust.*

Limitorque SMB Actuator



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Stem Factor

- *The stem factor represents the mechanical process where the rotational motion of the stem nut is converted to linear motion in the stem.*

- *Mathematically, stem factor is the torque divided by the thrust.*

- *This relationship is evaluated in terms of the power thread equation for Acme threads.*

Acme Power Thread Equation

$$StemFactor = \frac{Tq_{output}}{Th_{stem}} = \frac{d\left(.96815\tan a + \mu\right)}{24\left(.96815 - \mu\tan a\right)}$$

Where

Tq_{output} = *output torque of the actuator*

Th_{stem} = *stem thrust*

d = *outside diameter of the stem minus 1/2 pitch*

tan a = $\frac{Lead}{\pi d}$

μ = *stem friction coefficient*

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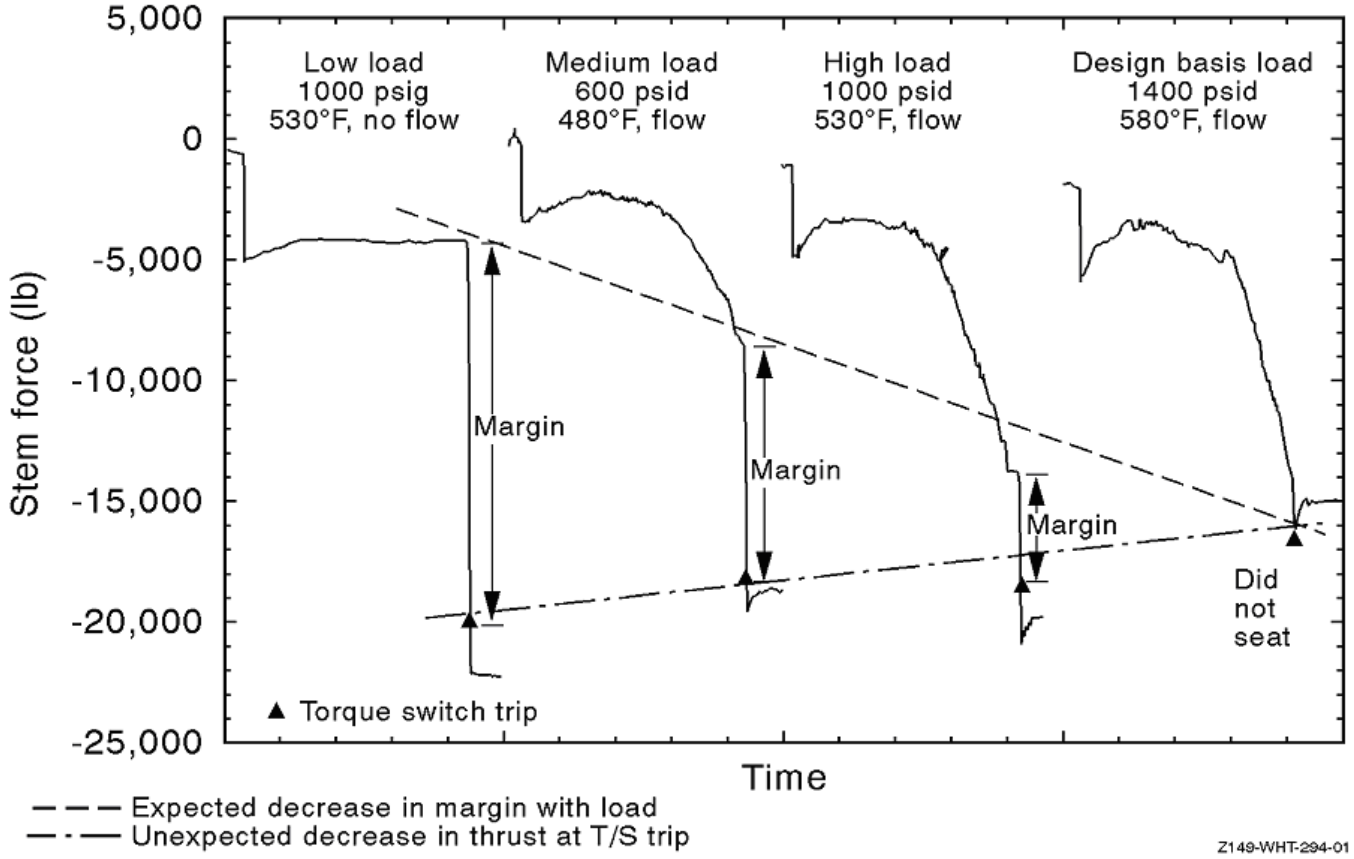
Required Actuator Torque

- *The required actuator torque is one of two parameters that define the valve's operating margin.*
- *The other parameter is the available actuator torque (described in later discussions).*
- *For any specific valve stem and stem nut combination, the only variable is the stem friction.*

Four Stem Thrust Traces



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Stem Friction Coefficient

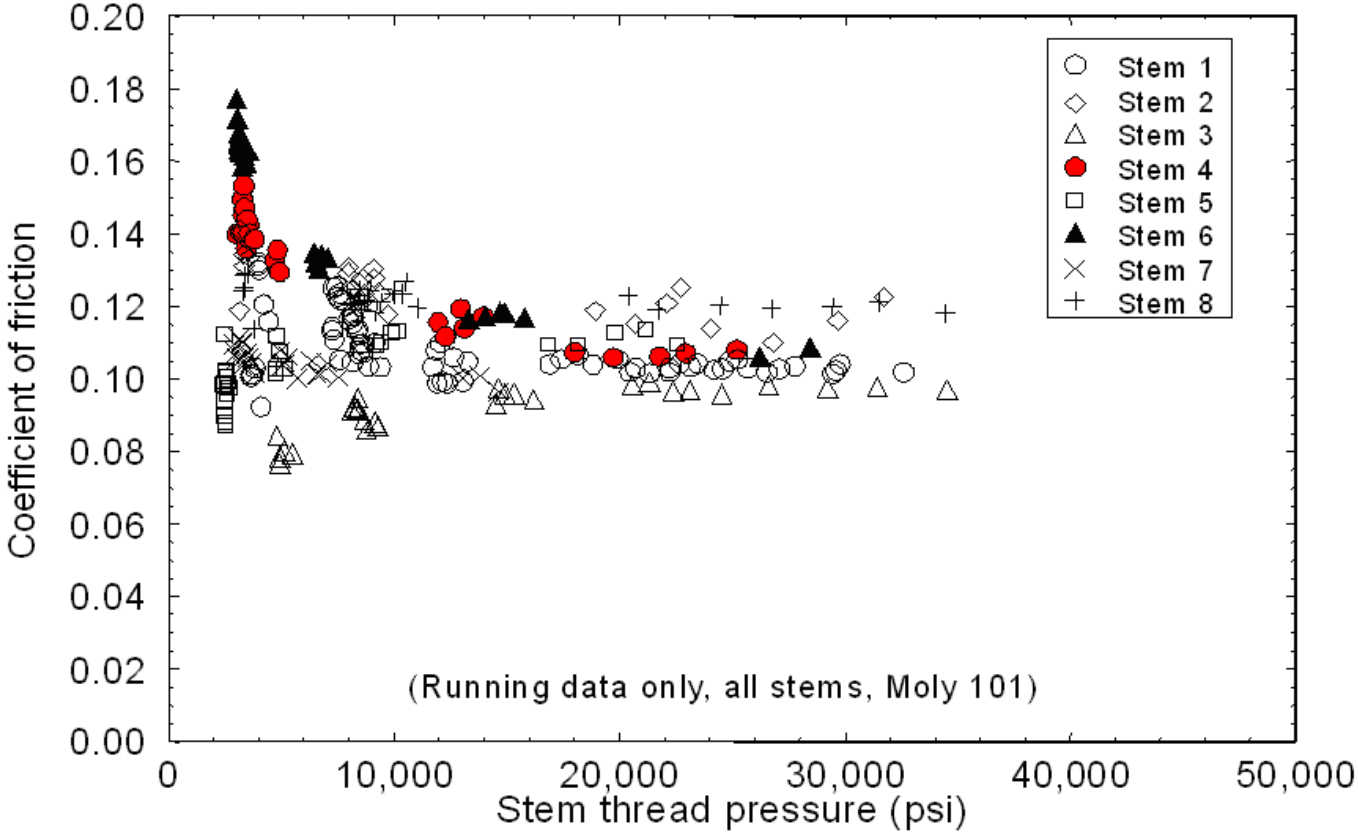
Results from NRC/INL testing show that the stem friction can vary significantly, depending on

- *The particular stem and stem nut configuration.*
- *The lubricant being used.*
- *Lubricant aging.*
- *Load magnitude.*
- *Load profile (load-sensitive behavior or rate-of-loading).*

Two Methods to Determine Stem Friction Coefficient

- *Threshold Method – Above a threshold thread pressure, stem friction reaches a plateau and does not significantly change with increasing thread pressure*
- *Fold Line Method – Used for valves where in-plant testing without flow or pressure is available. An envelope is determined from the observed load-sensitive behavior at the end of the stroke.*

Stem Friction Before Seating



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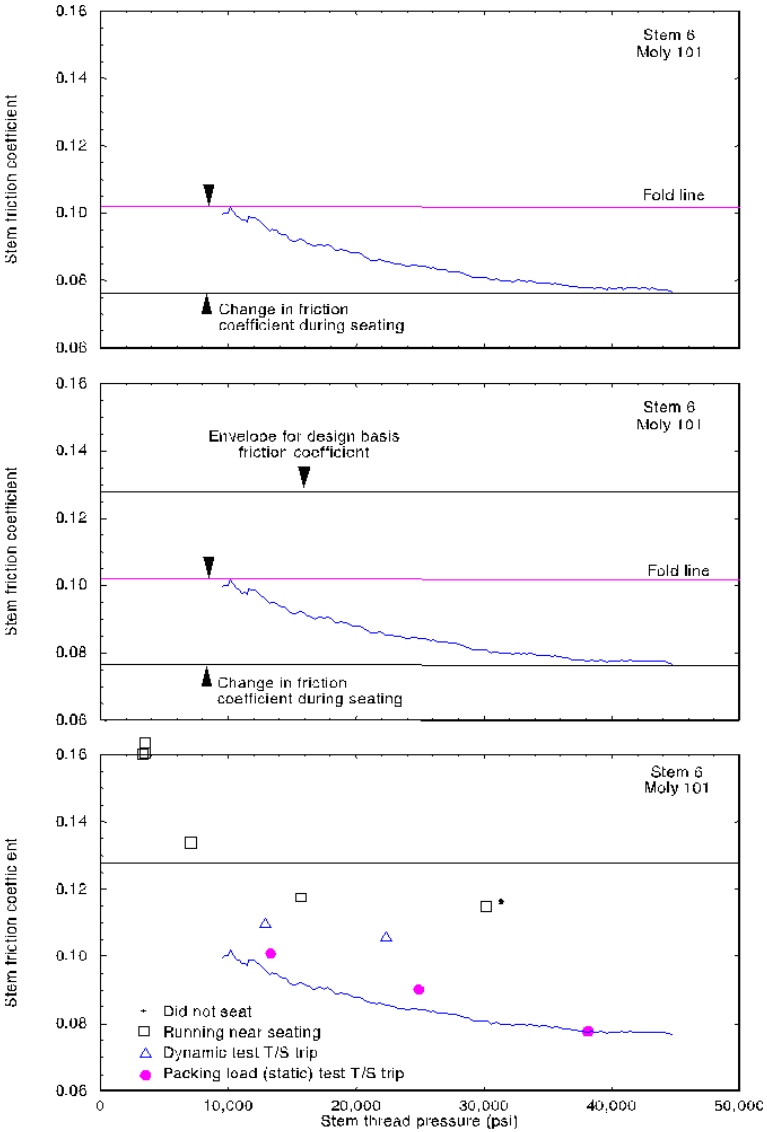
Fold Line Method of Determining Stem Friction



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Stem Lubrication and Friction

- The effectiveness of the lubricant used on the threaded portion of the valve stem can greatly impact the thrust output of the valve actuator and reduce the margin for ensuring MOV performance at design basis.
- Recent testing indicates that an elevated temperature environment can lead to significant increases in the friction coefficient at the stem/stem nut interface.
- Lubricant aging is another phenomenon that can have a deleterious effect on the thrust output of the actuator

Elevated Temperature Performance

- The physical characteristics of each lubricant change at elevated temperature.
- Operation at elevated temperature can have a significant effect on the stem coefficient of friction.
- Stem friction repeatability depends upon the unique stem, stem nut, and lubricant combination.
- The value and the direction of change in the end of stroke friction behavior is highly dependent on the stem/stem nut and lubricant being tested.
- Each individual stem and stem/nut combination has unique elevated temperature performance.

Stem Nut Wear – N5000 Antiseize

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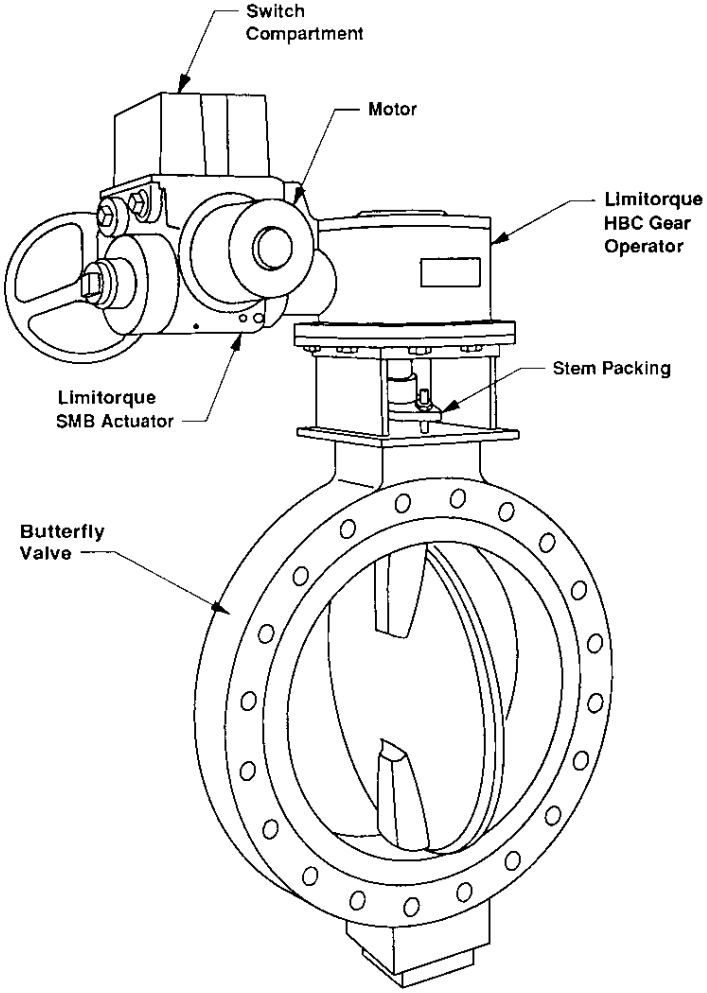
Butterfly Valve Stem Torque

Motor-Operated Butterfly Valve Assembly



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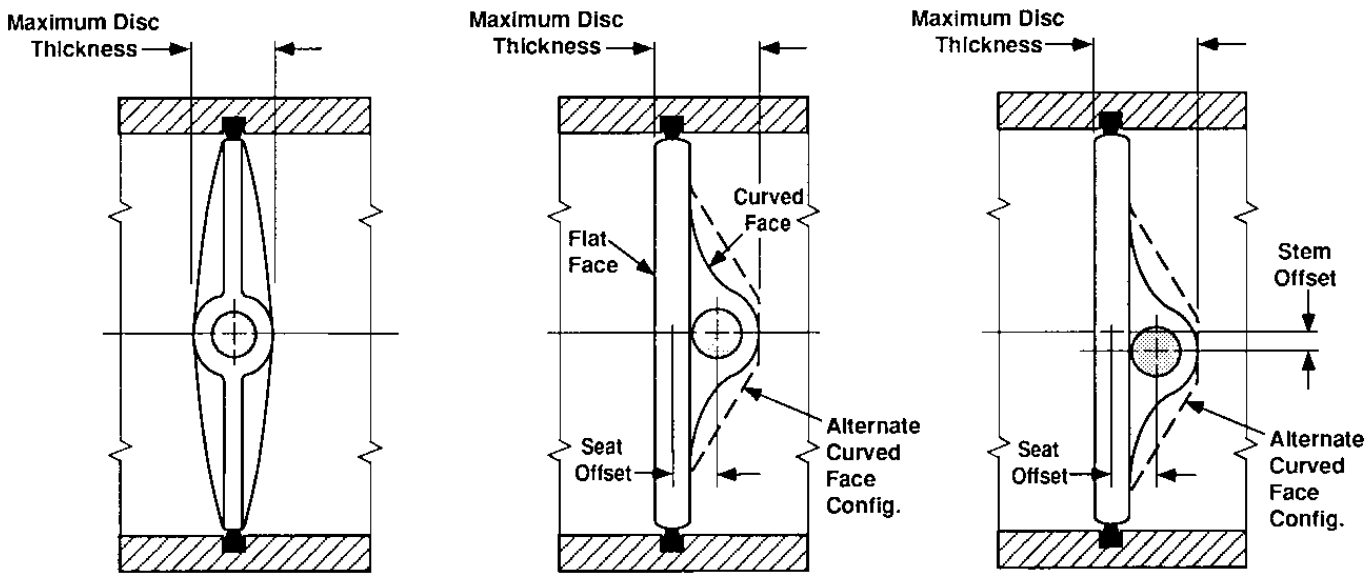
Butterfly Valve Stem Torque

- *The operating torque requirements of a butterfly valve are quite different from other MOVs.*
- *The maximum torque may be dictated by the seating/unseating torque or by the dynamic torque and some intermediate position.*
- *The magnitude of the dynamic torque is strongly dependent upon valve size, disc design, pressure drop, and mass flow.*

Butterfly Valve Disc Designs

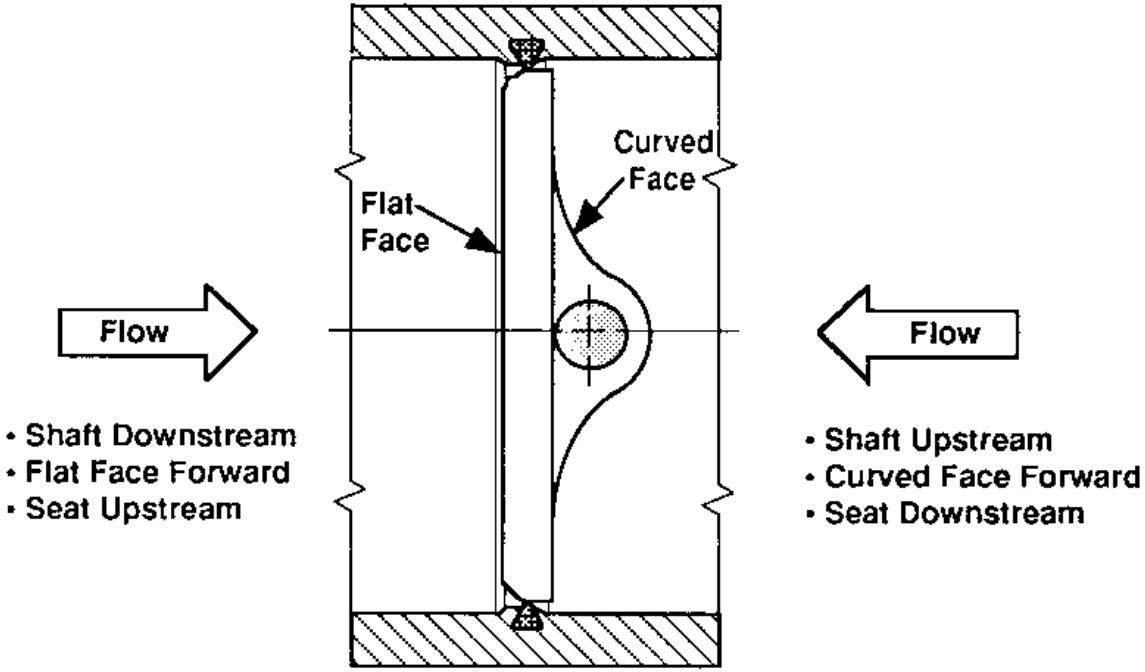


(a) Symmetric Disc Design

(b) Single Offset Design

(c) Double Offset Design

Valve Orientation to Flow

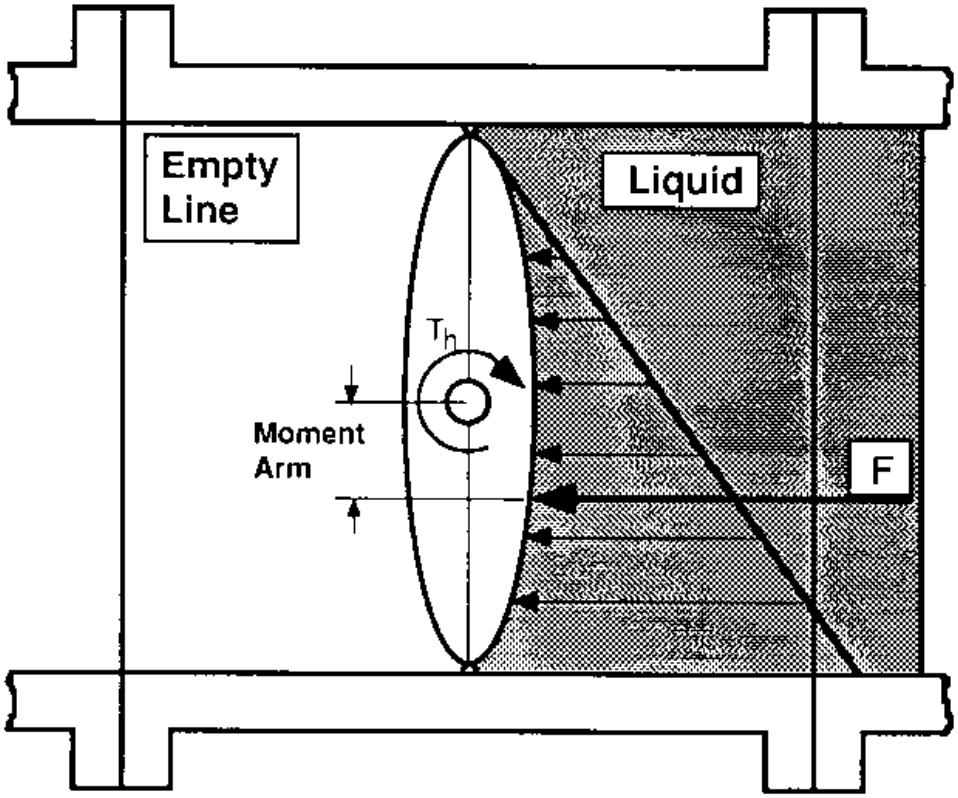


Total Seating/Unseating Torque

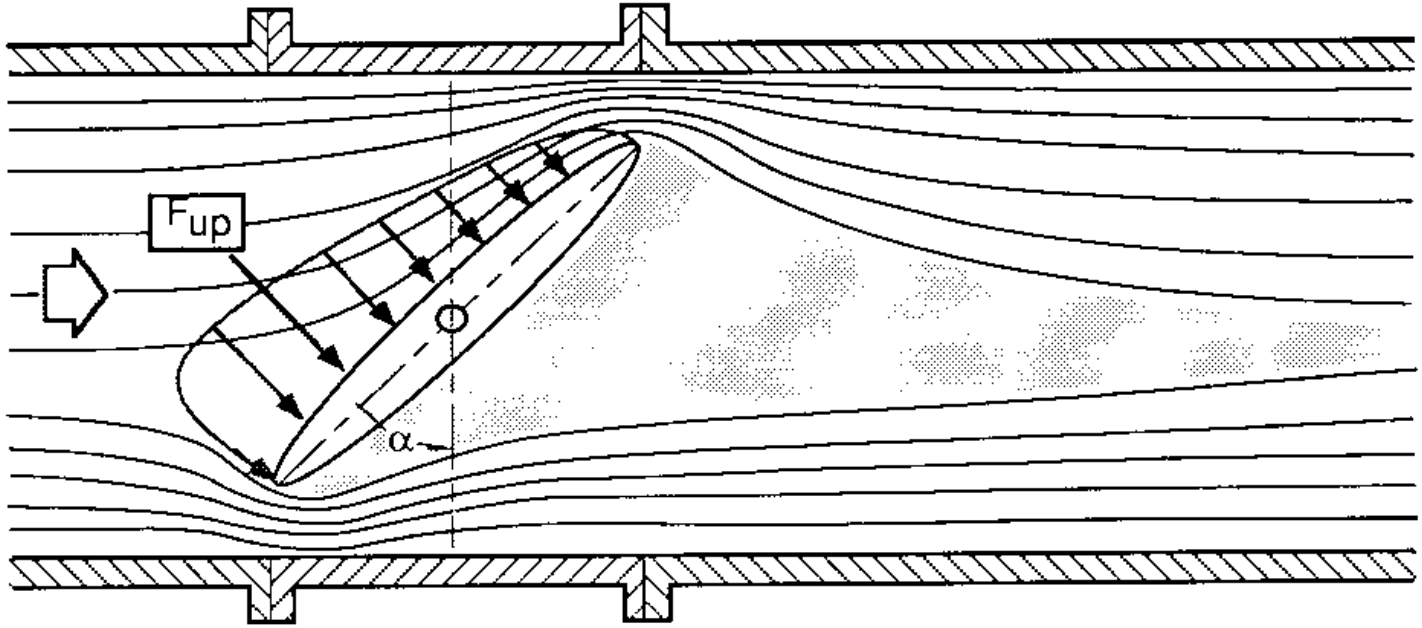
$$T_{TS} = T_{Seat} + T_{Bearing} + T_{Packing} + T_{Hydrostatic}$$

- The seat torque (T_{Seat}) depends on the specific details of the valve seat design.
- The bearing torque ($T_{Bearing}$) is proportional to the differential pressure across the disc.
- The packing torque ($T_{Packing}$) is normally small compared to the total required torque.
- The hydrostatic torque ($T_{Hydrostatic}$) results from the fluid pressure acting on the valve disc to produce a torque load on the valve stem.

Hydrostatic Torque Load



Resultant Force From Nonuniform Pressure Distribution



Total Dynamic Torque

$$T_{TD} = T_{Bearing} + T_{Packing} + T_{Hydrodynamic}$$

- The bearing torque ($T_{Bearing}$) is proportional to the differential pressure across the disc.
- The packing torque ($T_{Packing}$) is normally small compared to the total required torque.
- The hydrodynamic static torque ($T_{Hydrodynamic}$) can be in either direction depending on the disc design, valve orientation to the flow stream, and even the nearby piping configuration.

Hydrodynamic Torque

- *The major characteristic of the hydrostatic torque is that it always acts in a particular direction, regardless or the direction of stem rotation.*
  - self-closing for symmetric discs and for nonsymmetric discs installed with the shaft upstream (curved face forward),
  - self-opening over a portion of the stroke or the entire stroke for nonsymmetric discs installed with the shaft downstream (flat face forward).

Hydrodynamic Torque

$$T_{Hyrodynamic} = C_t d^3 \Delta P$$

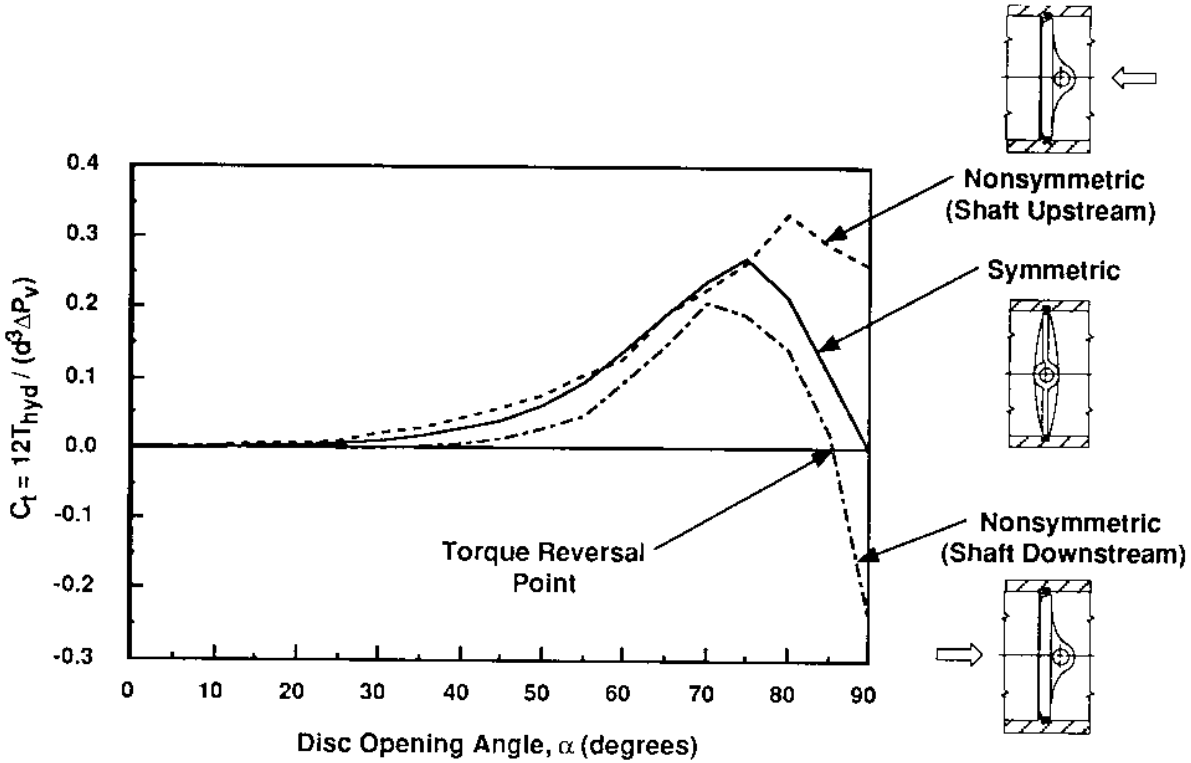
Where

C_t = hydrodynamic torque coefficient (dimensionless)

d = valve nominal diameter

ΔP = differential pressure across the valve

Typical Incompressible Flow Coefficients for Various Disc Designs



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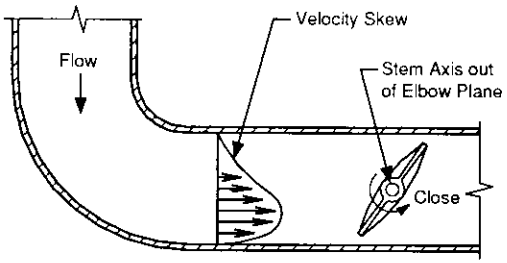
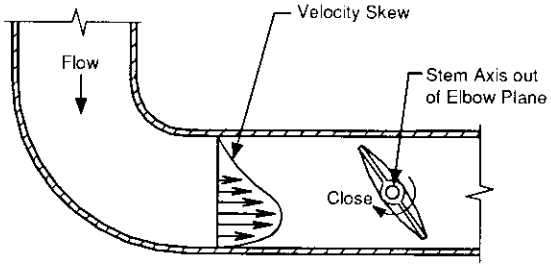
Upstream Disturbances

- *The hydrodynamic torque characteristics discussed earlier are based on uniform approach velocity*
- *Any flow disturbance can significantly affect the magnitude of the hydrodynamic torque.*

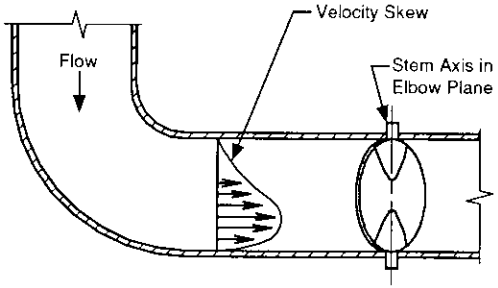
Velocity Profiles With Upstream Flow Disturbance



Configuration 1: Hydrodynamic Torque Increases
(Velocity Skew Assists Self-closing Torque)



Configuration 2: Hydrodynamic Torque Decreases
(Velocity Skew Opposes Self-closing Torque)



Configuration 3: Hydrodynamic Torque Negligibly Affected
(Velocity Skew is Symmetric about Stem Axis)

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Hydrodynamic Torque – With Flow Disturbance

$$T'_{Hyrodynamic} = C_{up} T_{Hyrodynamic}$$

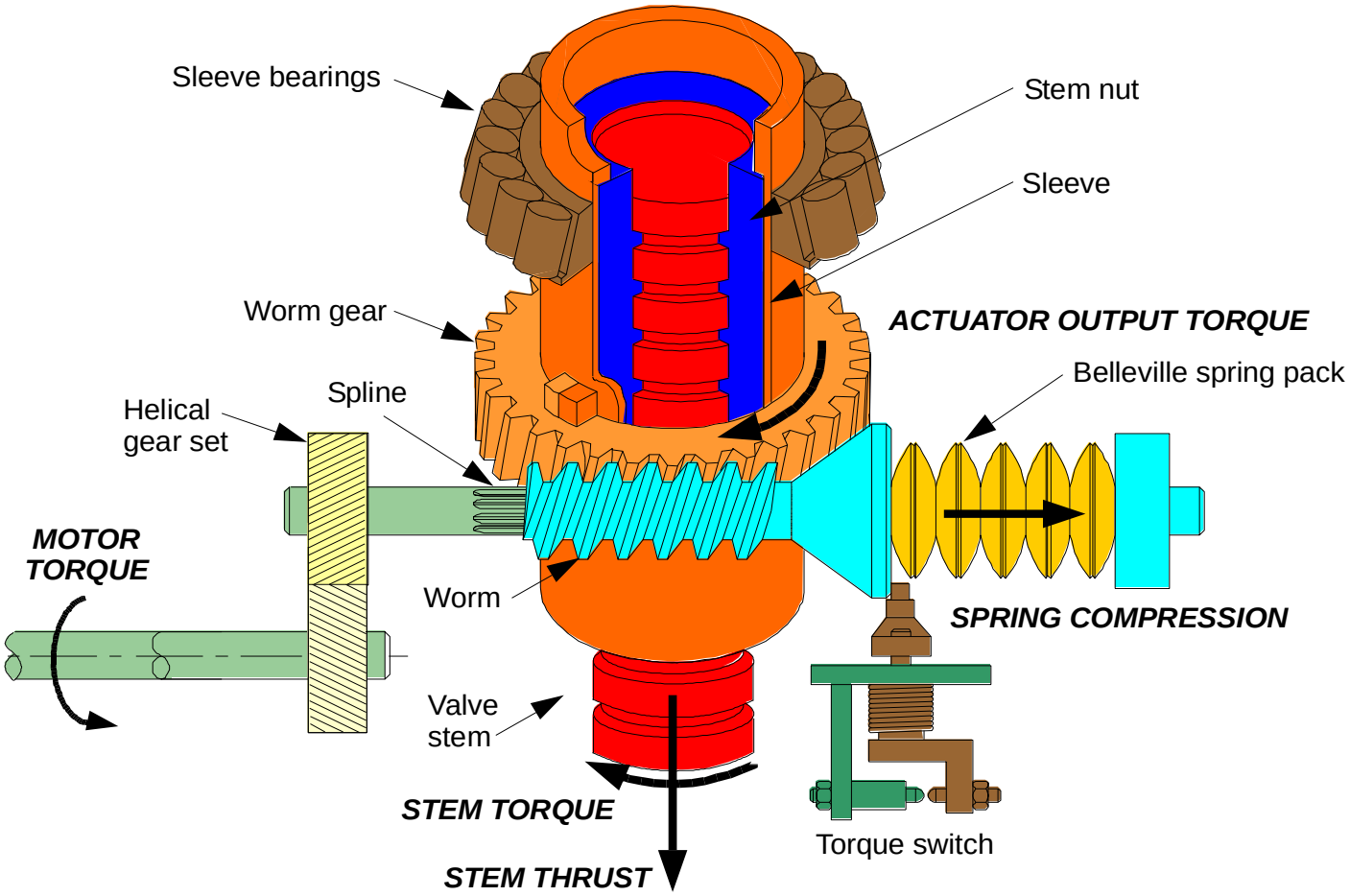
Where

C_{up} = factor to account for the affect of an upstream disturbance

- For compressible flow, a factor of 1.5 has been identified for upstream elbows
- For incompressible flow, factors vary from 1.3 to 1.5

Motor Actuator Output Capability

Diagram of An Actuator Gearbox



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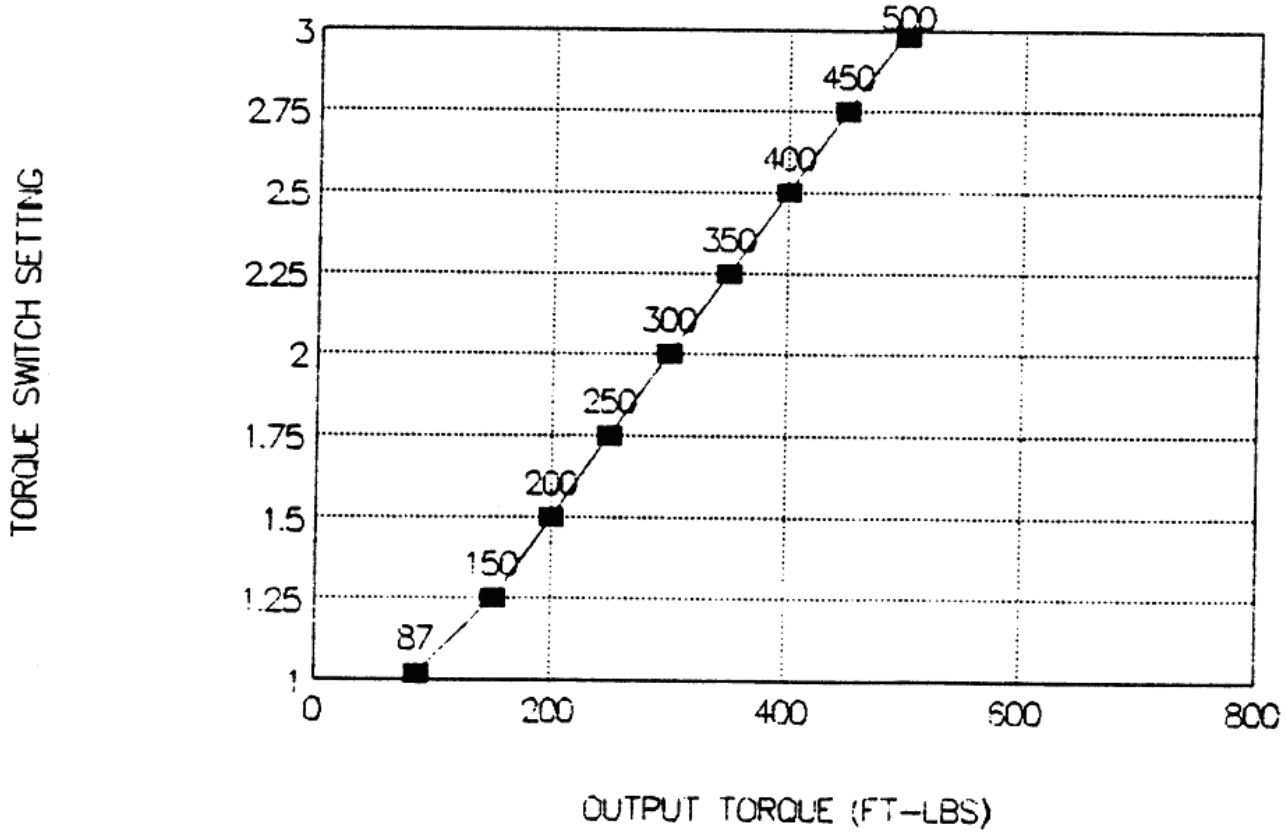
Limitorque Torque Switch Chart For 0501-184 Spring Pack



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The Motor-Operated Valve Load Simulator (MOVLS)

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Actuator Motor Performance

The following formula is typically used to predict the output torque of an actuator motor

$$Tq_{motor} = Tq_{rated}\left(\frac{V_{act}}{V_{rat}}\right)^{n} F_{temp} F_{app}$$

Tq_{motor} = *actual motor torque*

Tq_{rated} = *rated motor torque*

V_{act} = *actual voltage*

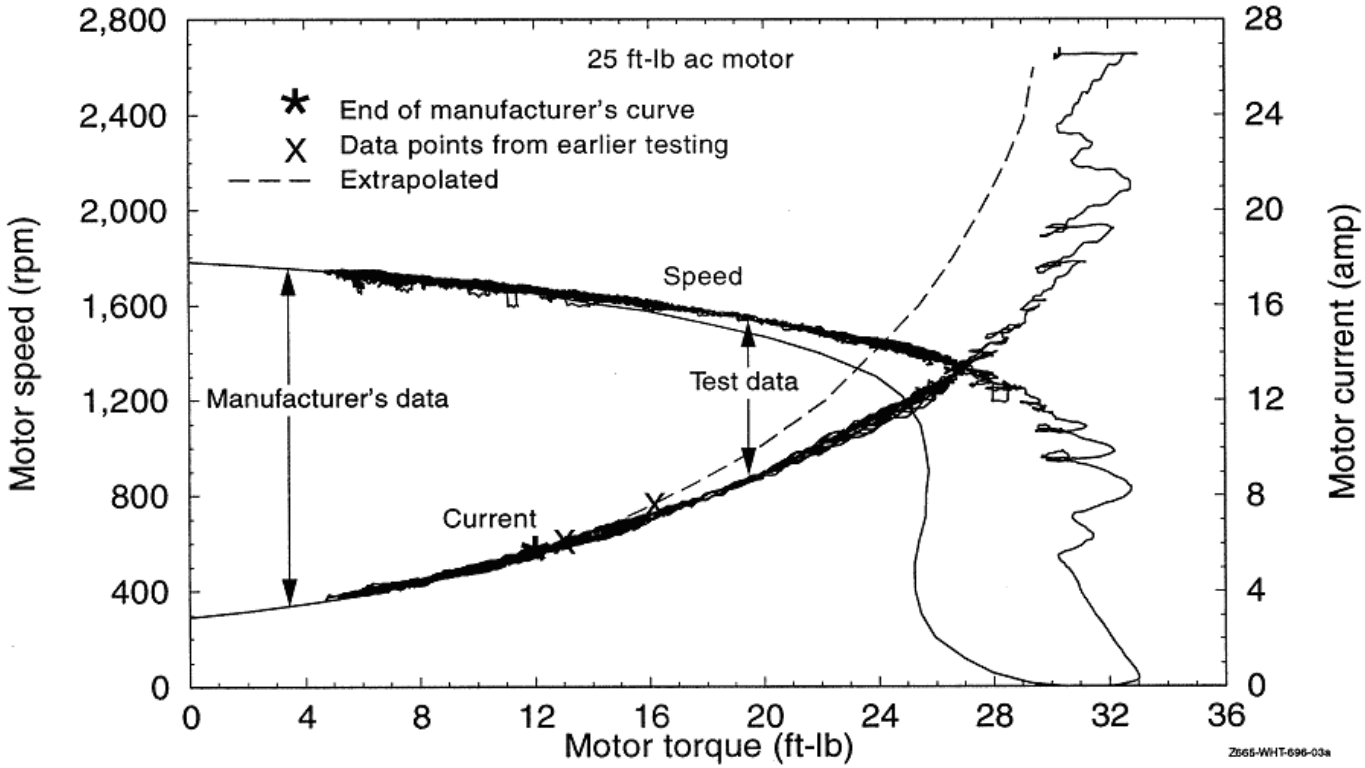
V_{rat} = *rated voltage.*

n = *usually 2 for ac motors and 1 for dc*

F_{temp} = *factor to account for motor heating*

F_{app} = *application factor*

25 ft-lb ac Motor Torque Curve



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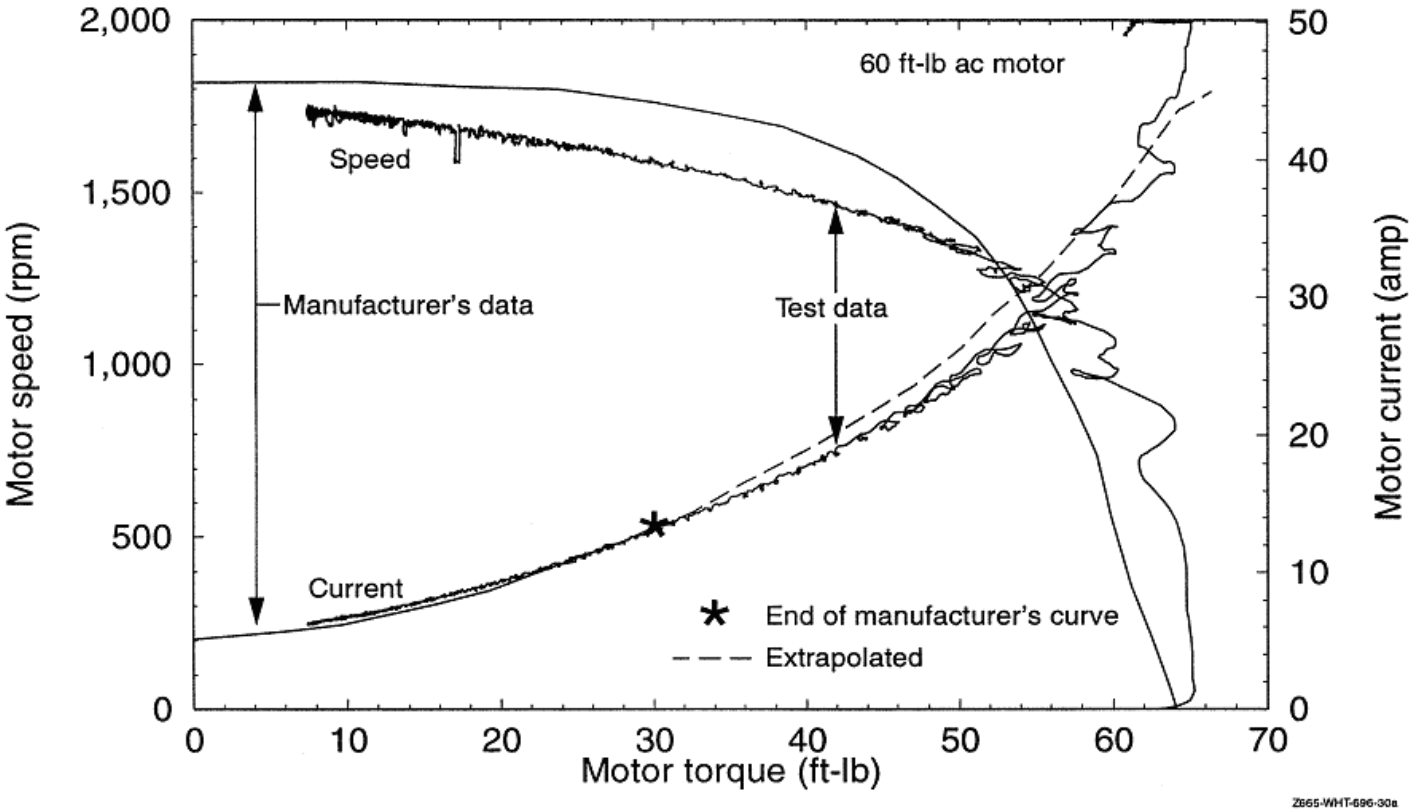
60 ft-lb ac Motor Torque Curve



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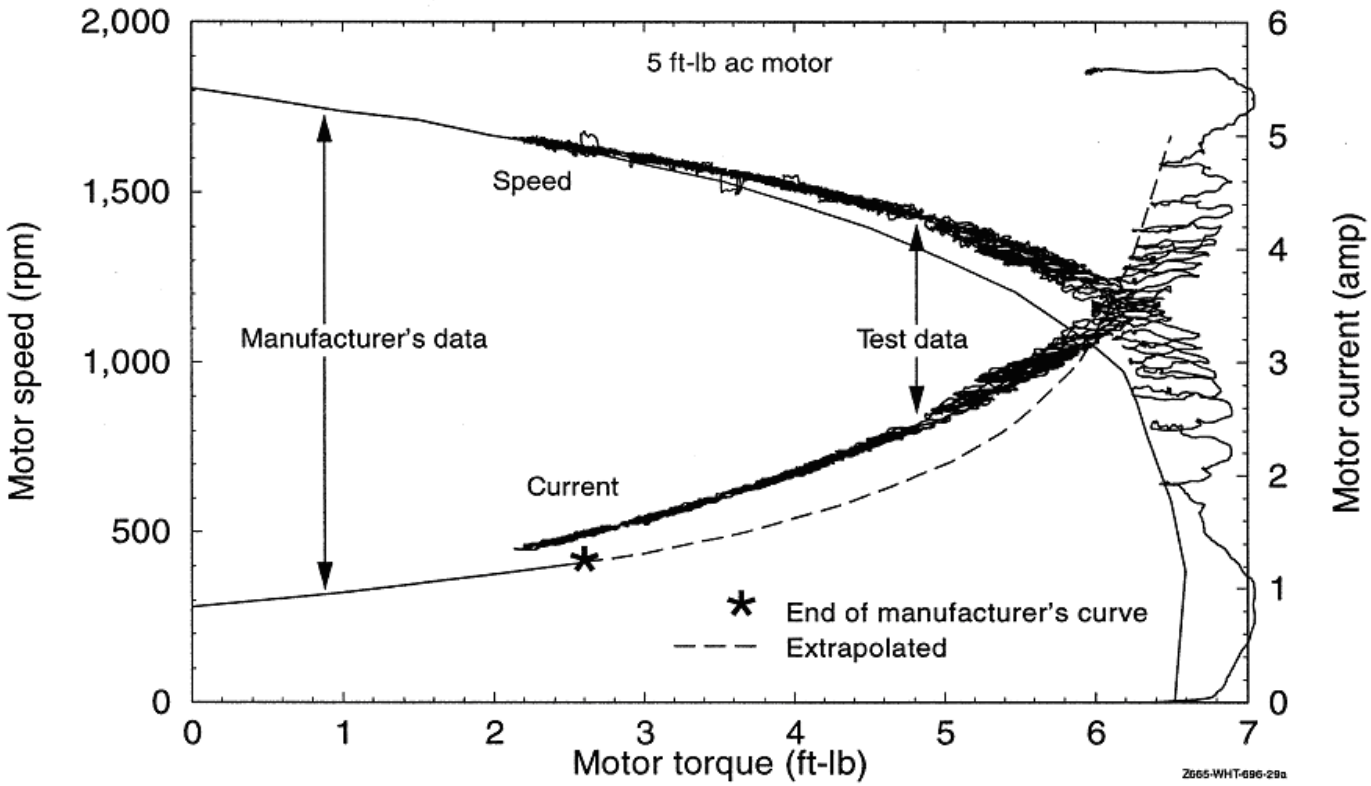
5 ft-lb ac Motor Torque Curve



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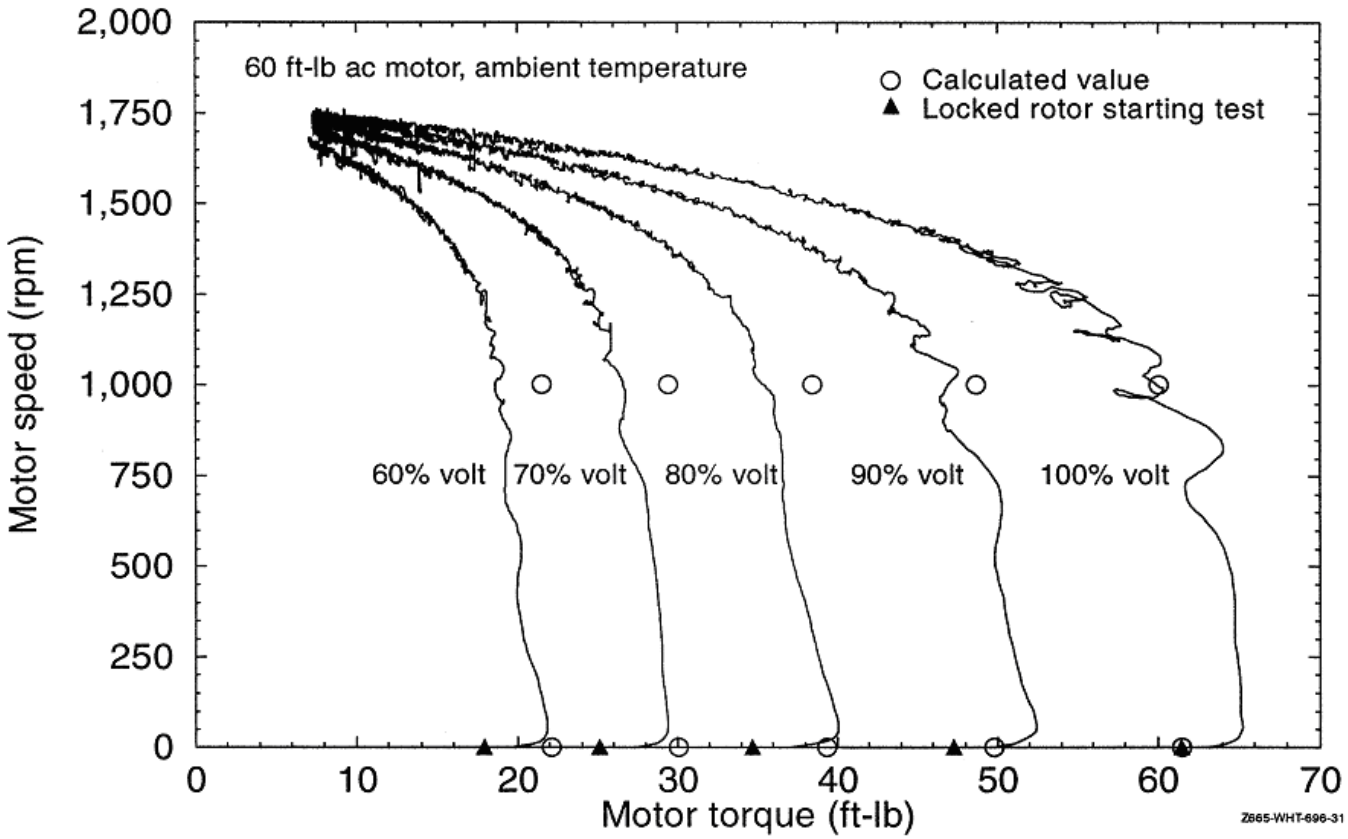
Degraded Voltage Test Results For The 60 ft-lb ac Motor



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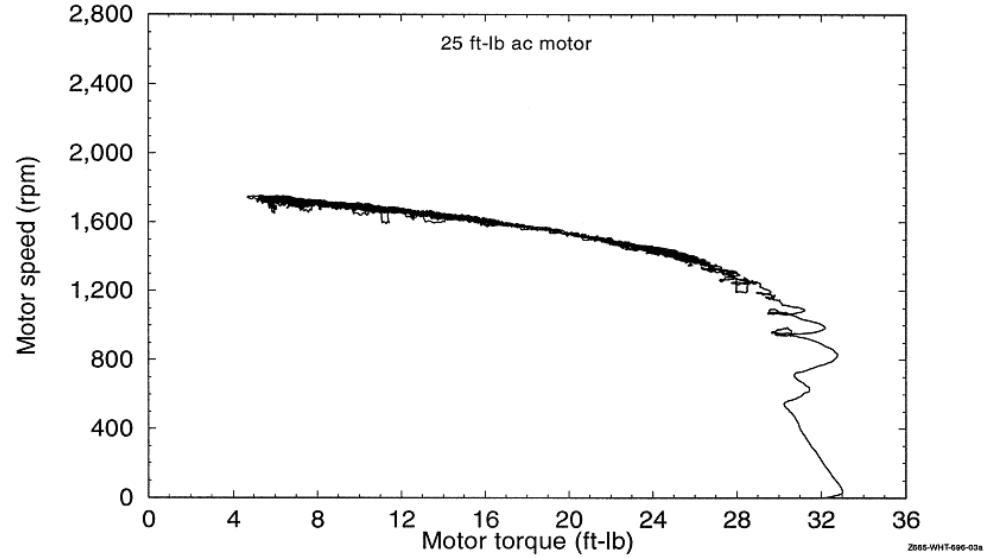
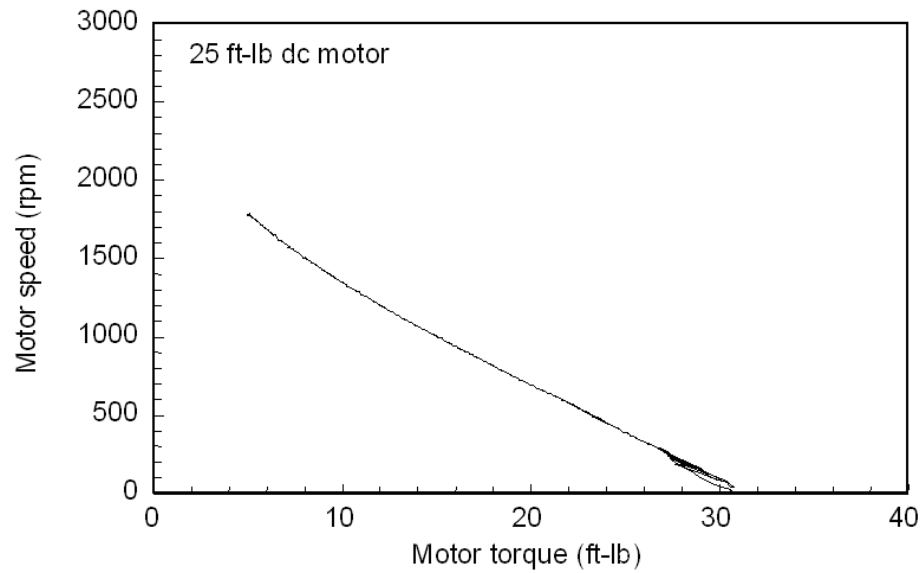
Motor Torque Curves For ac and dc Motors





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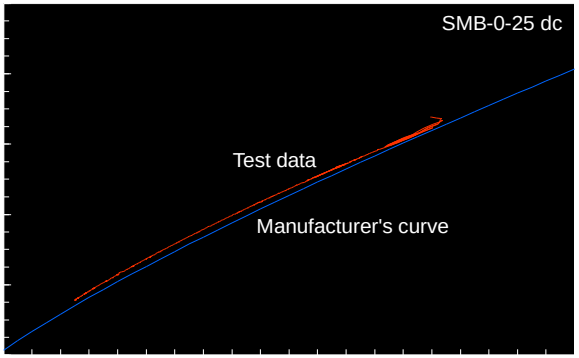
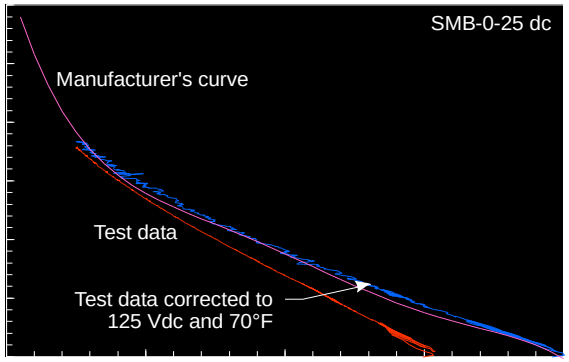
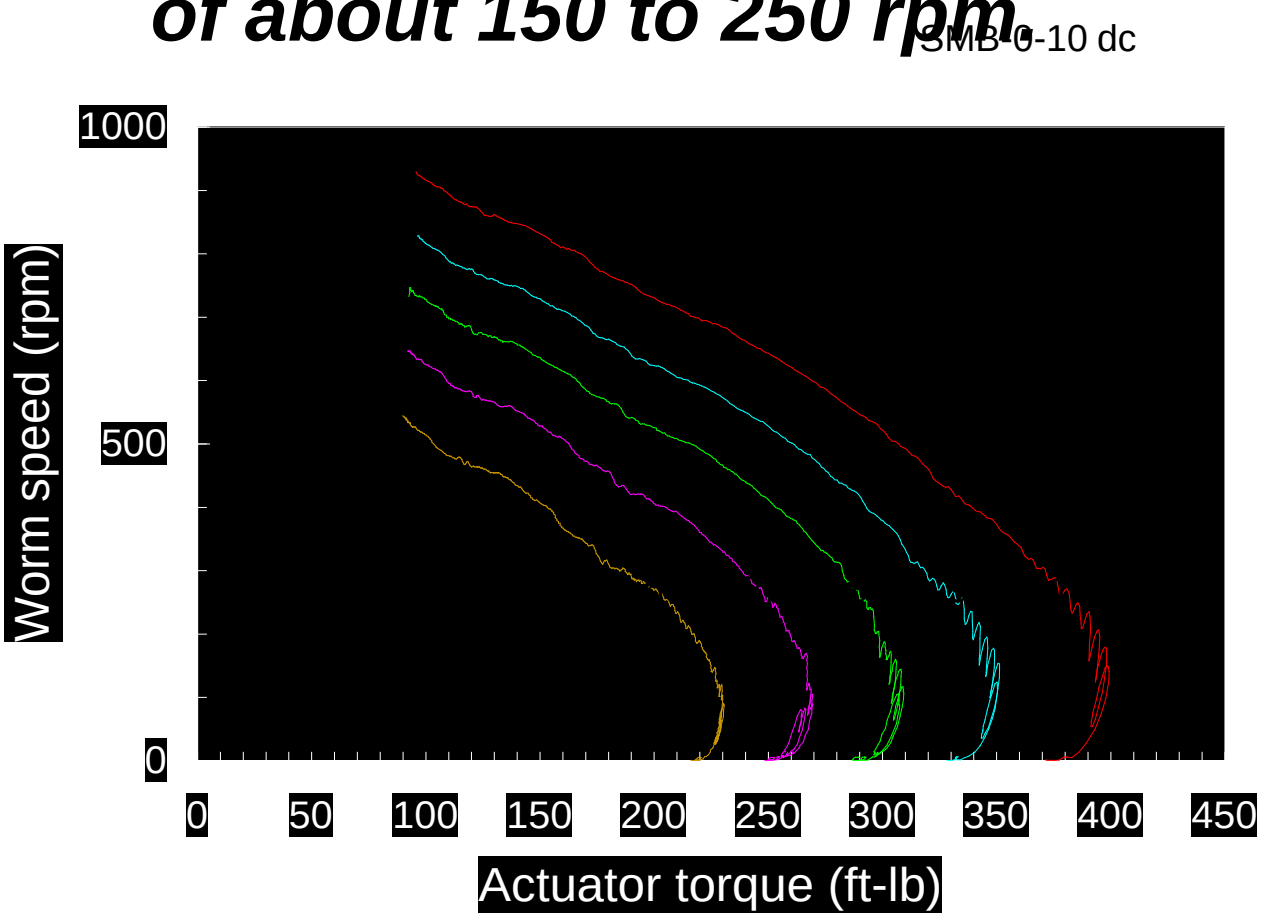
Actual performance was lower that the manufacturers curve due to voltage loss and heatup.





No additional valve stem torque can be expected below a worm shaft speed threshold of about 150 to 250 rpm



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Degraded Voltage Testing

The following formula is typically used to account for reduced dc motor output at degraded voltage:

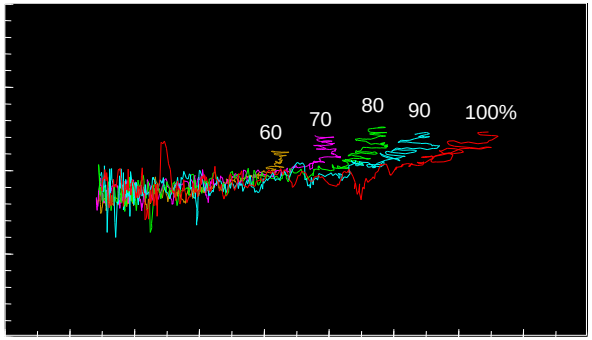
$$Tq_{act} = Tq_{rat}\left(\frac{V_{act}}{V_{rat}}\right)^{1}$$

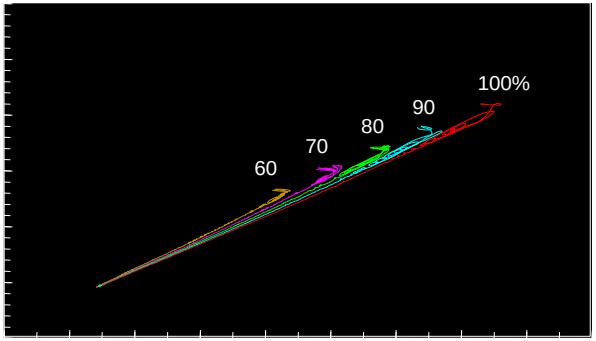
Tq_{act}	=	*actual motor torque*
Tq_{rat}	=	*rated motor torque*
V_{act}	=	*actual voltage*
V_{rat}	=	*rated voltage.*

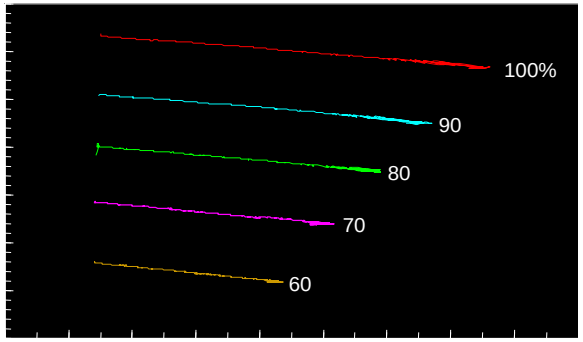
This formula is identical to the V^2 calculation used for ac motors, except that the exponent is 1 instead of 2.

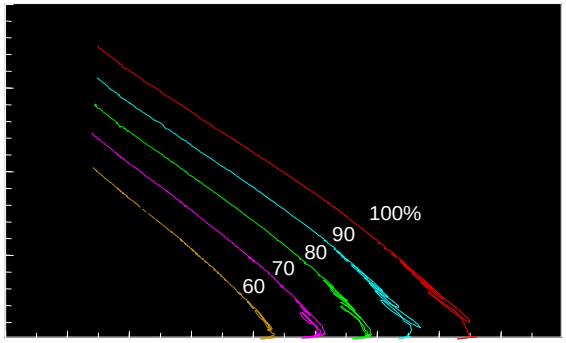
Motor temperature, current, voltage, and speed versus torque during degraded voltage testing of the 10-ft-lb dc motor.

SMB-0-10 dc









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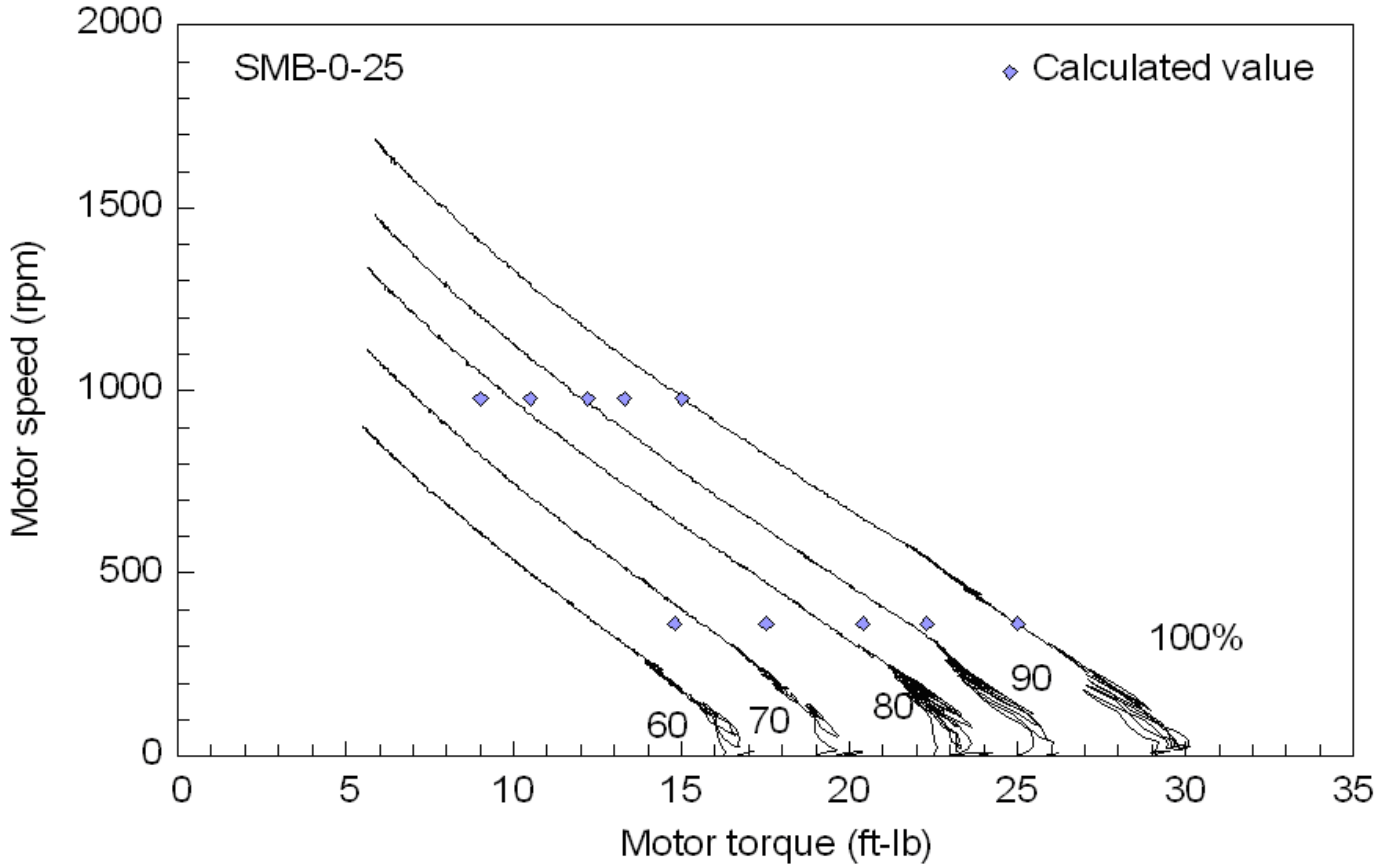
Dynamometer-type Test Results



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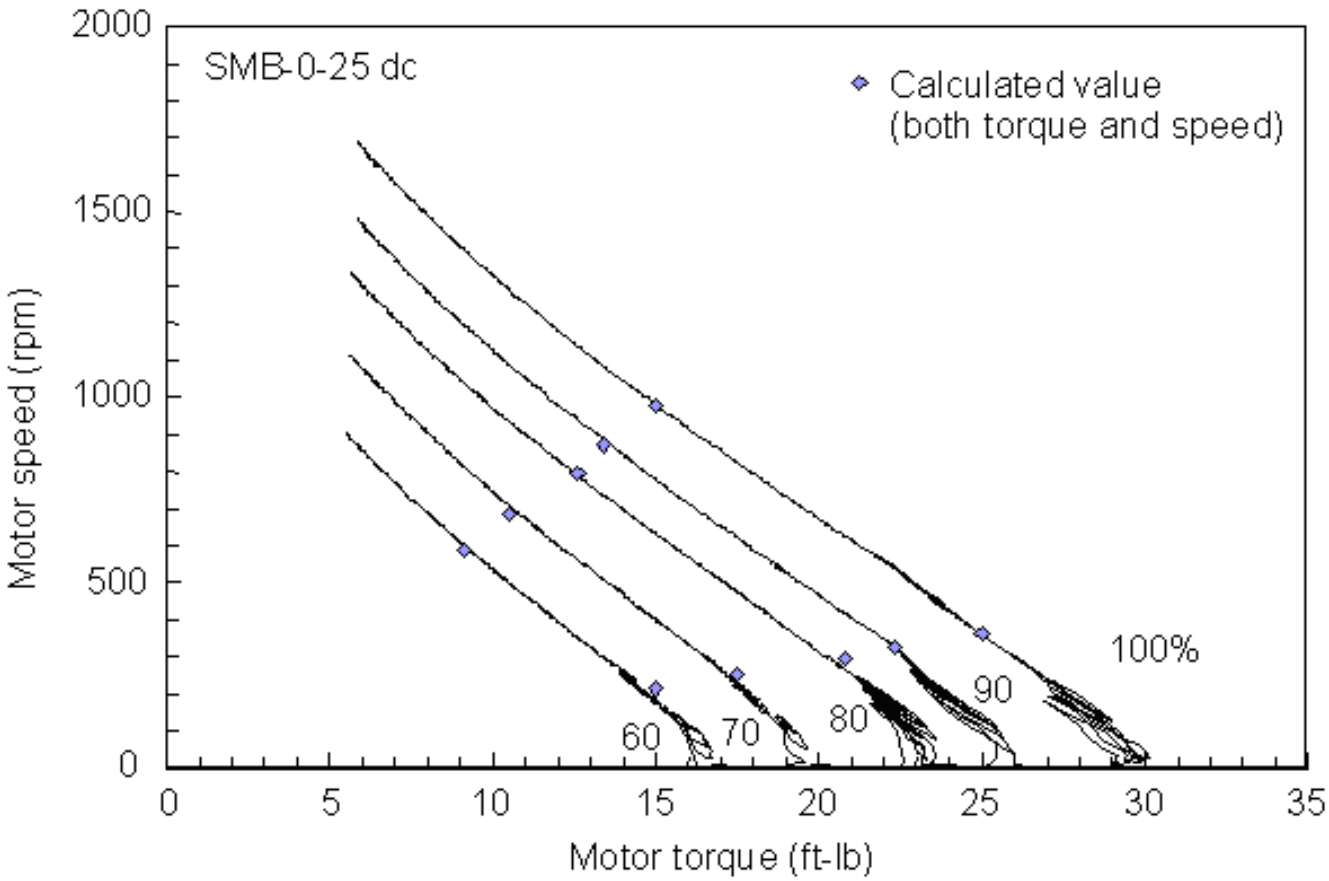
INEEL Predictive Method Results



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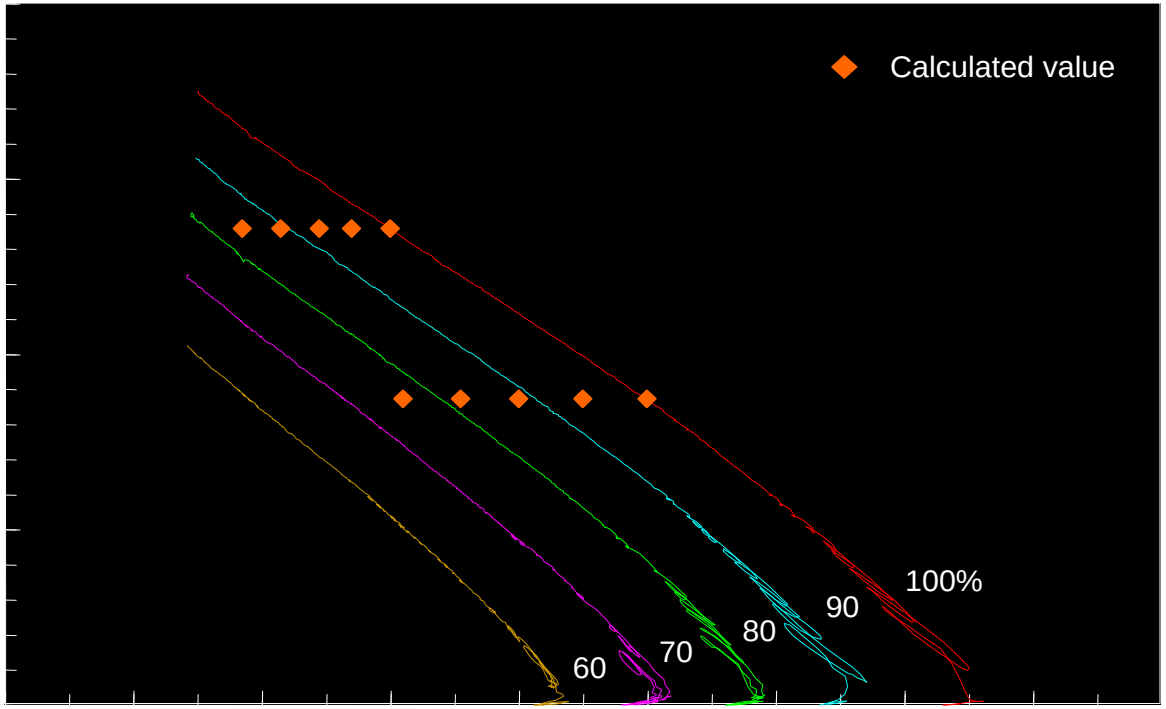
The actual torque losses were greater than that predicted



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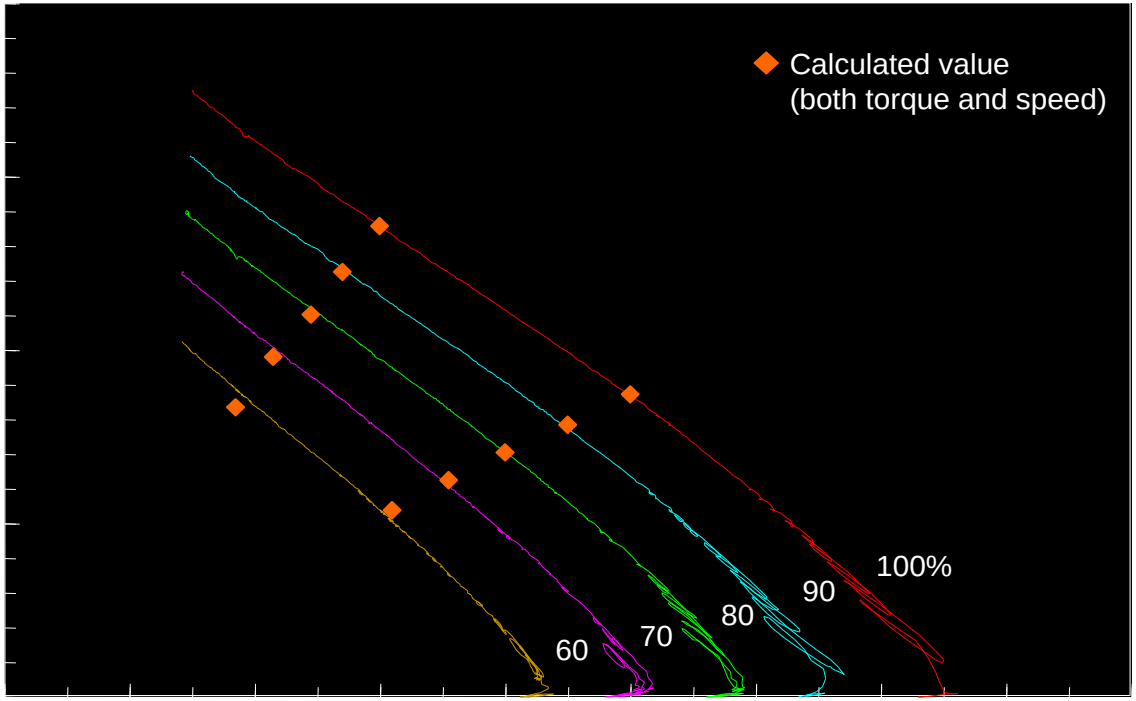
Reduced voltage causes a reduction in motor speed as well as motor torque.



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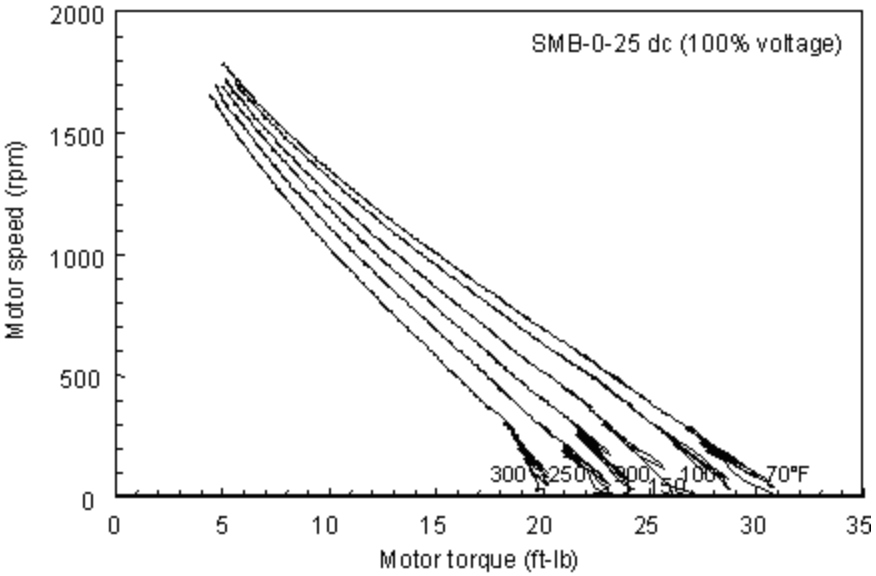
Motor Speed at Reduced Voltage

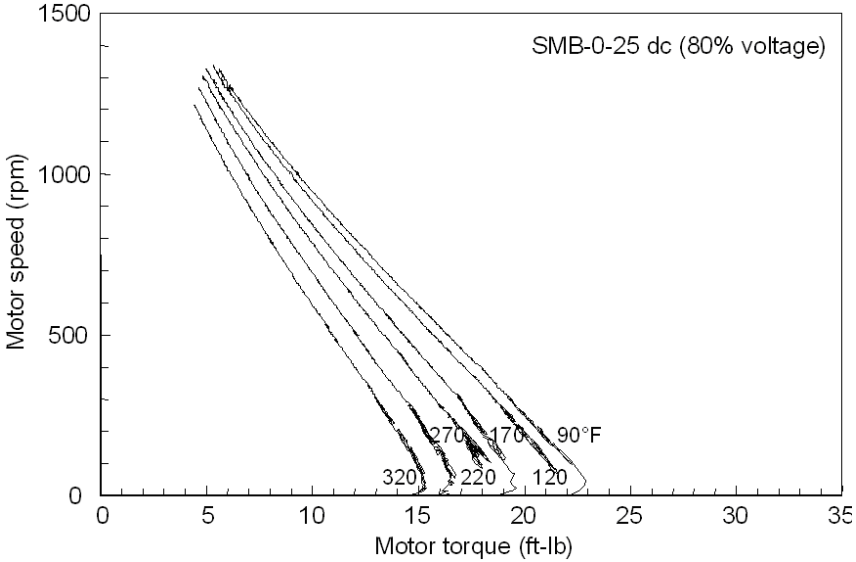
We therefore applied a linear relationship [similar to the torque relationship in Equation (2)] to the motor speed, as follows:

$$S_{act} = S_{rat}\left(\frac{V_{act}}{V_{rat}}\right)$$

S_{act} = *actual motor speed*
S_{rat} = *rated motor speed*
V_{act} = *actual voltage*
V_{rat} = *rated voltage.*

Elevated Temperature Results





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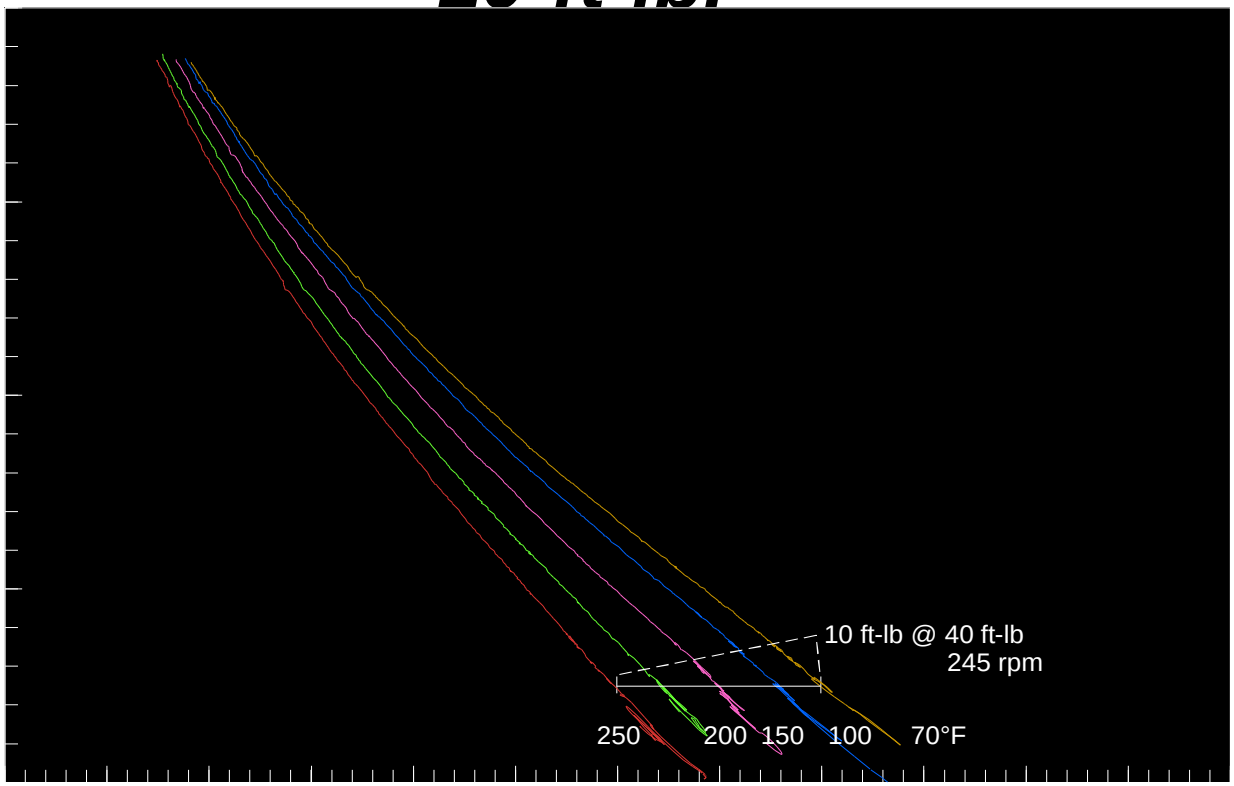
At 100% voltage, the increase from 70 to 250°F reduced the 40-ft-lb motor's torque by 10-ft-lb.



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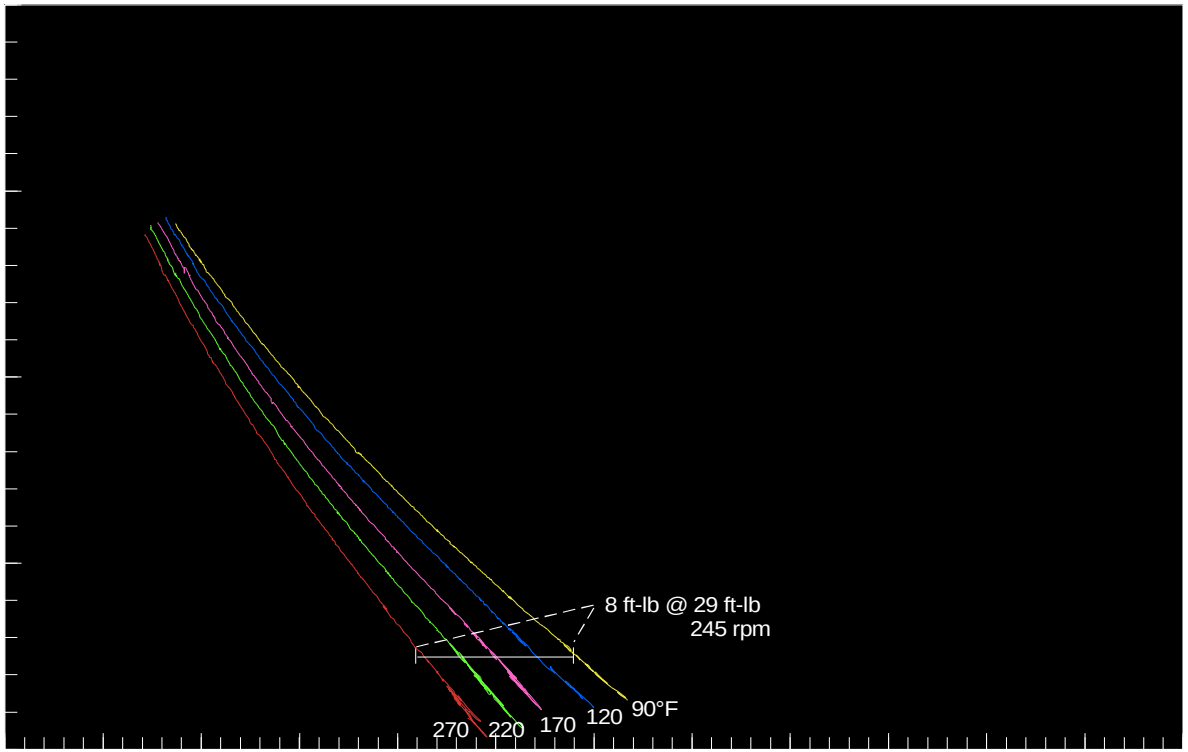
At 80% voltage, the increase from 70 to 250°F reduced the 40-ft-lb motor's torque by 8-ft-lb.



Temperature Effect

- *Temperature has a linear effect on a dc motor's output torque, similar to the temperature effect on the resistance of copper wire.*
- *We therefore applied a linear relationship to estimate the actual torque at elevated temperature.*

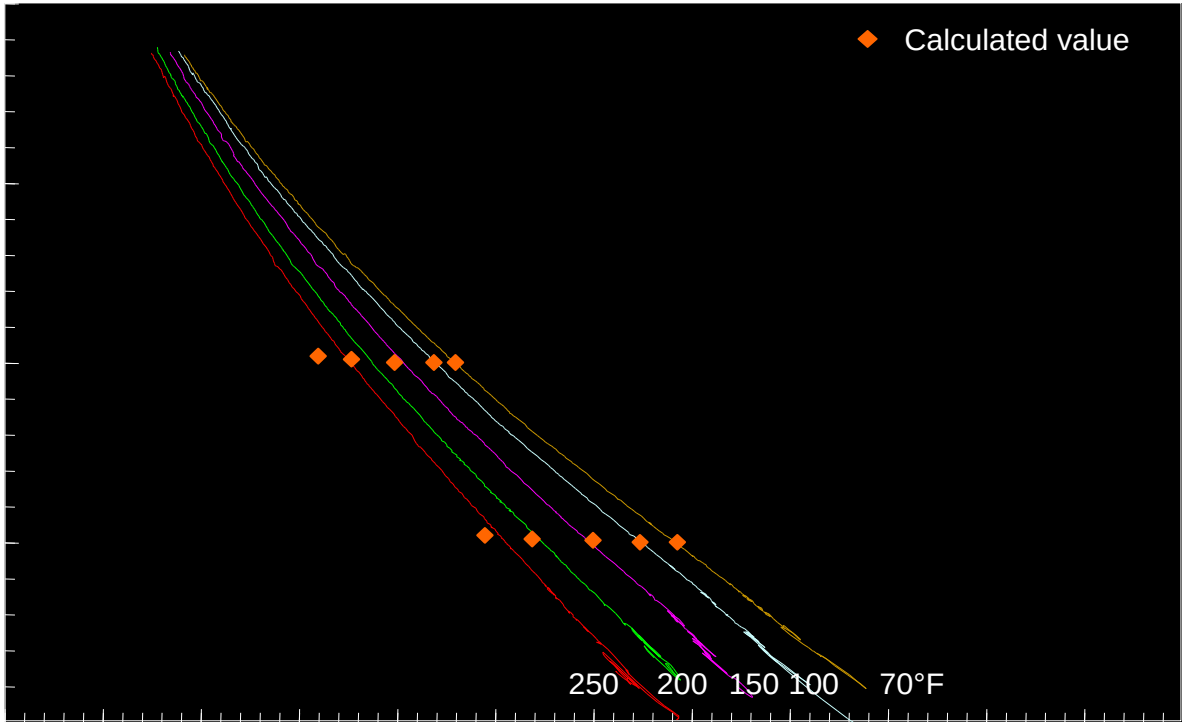
Temperature Effect (continued)

$$Tq_{act} = Tq_{rat}\left(1 - \frac{T_e - T_a}{T_a - T_z}\right)$$

where:

Tq_{act}	=	*actual motor torque*
Tq_{rat}	=	*rated motor torque*
T_e	=	*elevated temperature*
T_a	=	*ambient temperature (room temperature of about 70°F)*
T_z	=	*absolute zero (-273.15°C or -459.67°F).*

Elevated temperature predictions for the 40-ft-lb dc motor at 100% voltage based on Equation 4.



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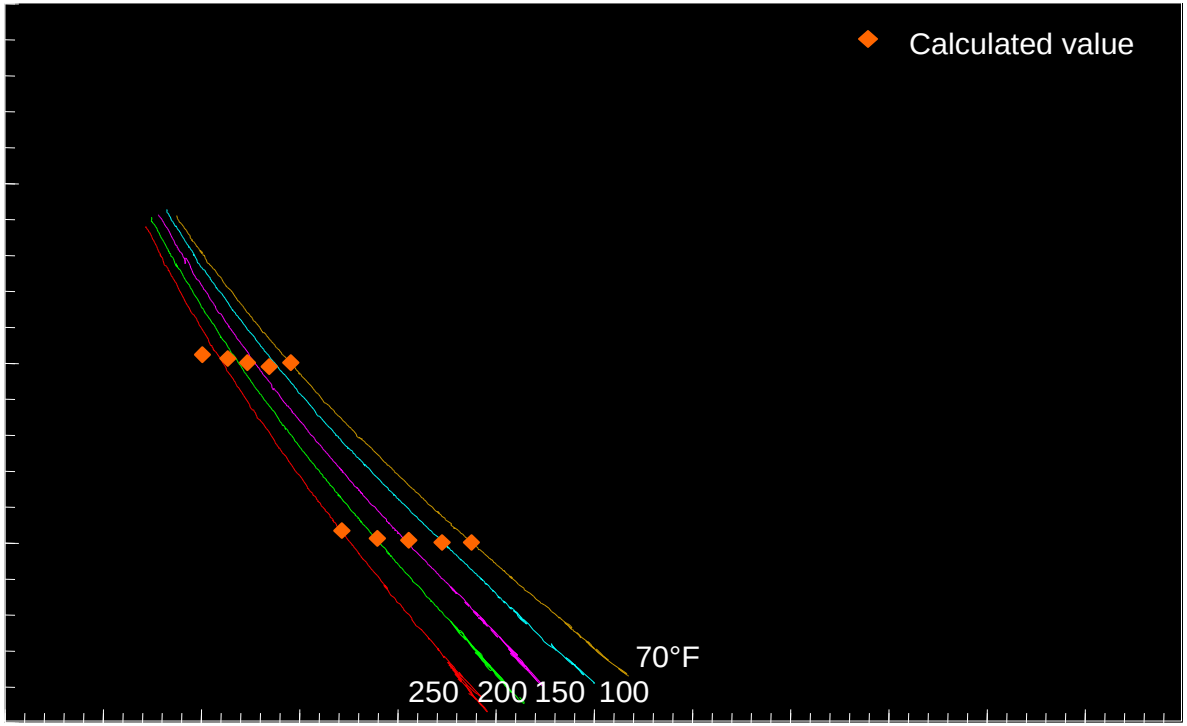
Elevated temperature predictions for the 40-ft-lb dc motor at 80% voltage based on Equation 4.



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Actuator Gearbox Performance

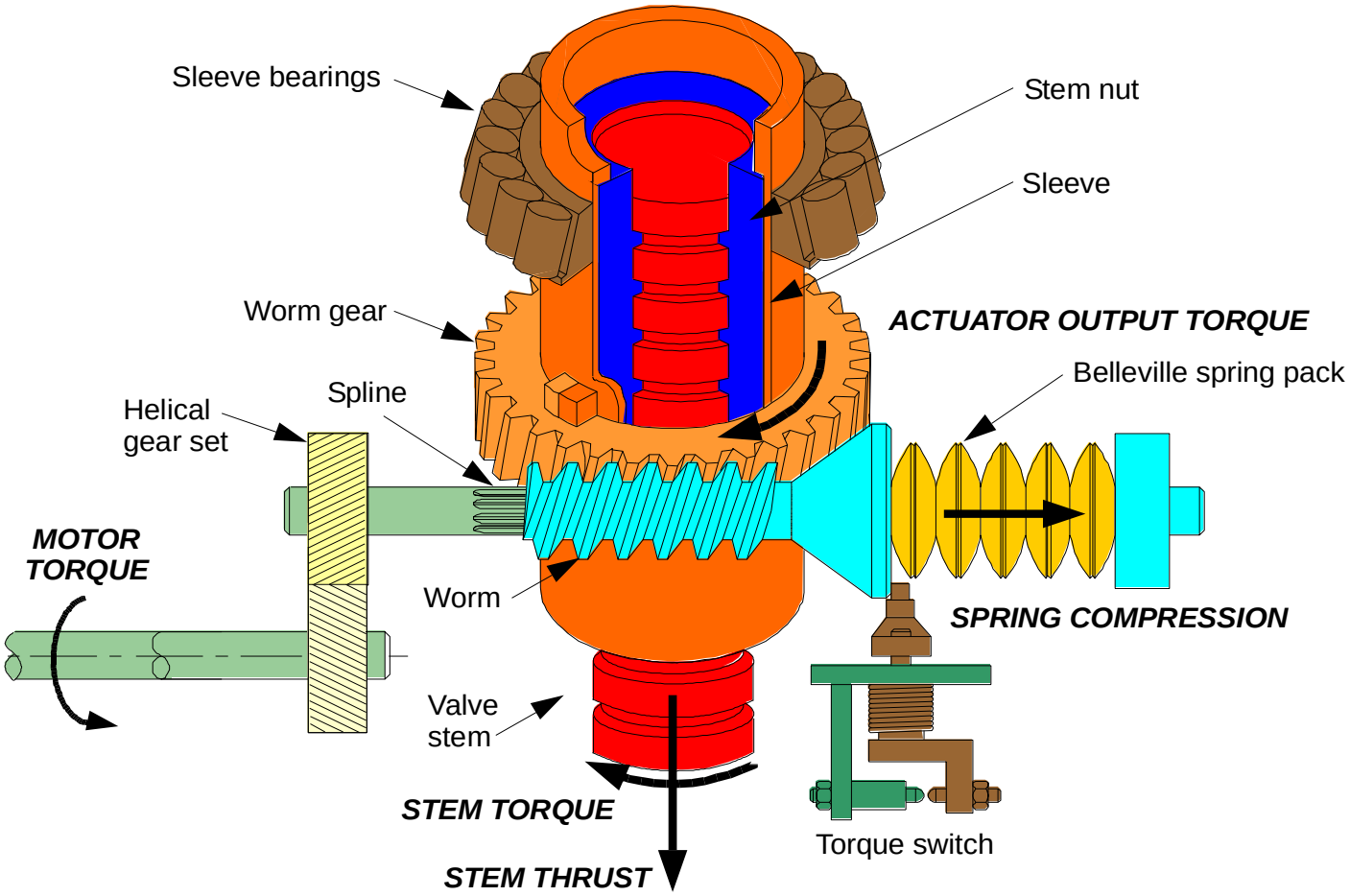
Gearbox output torque can be represented by the following equation:

$$Tq_{output} = Tq_{input} Eff_{gearbox} OAR$$

where

Tq_{output}	=	*output torque of the actuator*
Tq_{input}	=	*input torque (motor torque)*
$Eff_{gearbox}$	=	*efficiency of the gearbox*
OAR	=	*overall gear ratio.*

Diagram of An Actuator Gearbox



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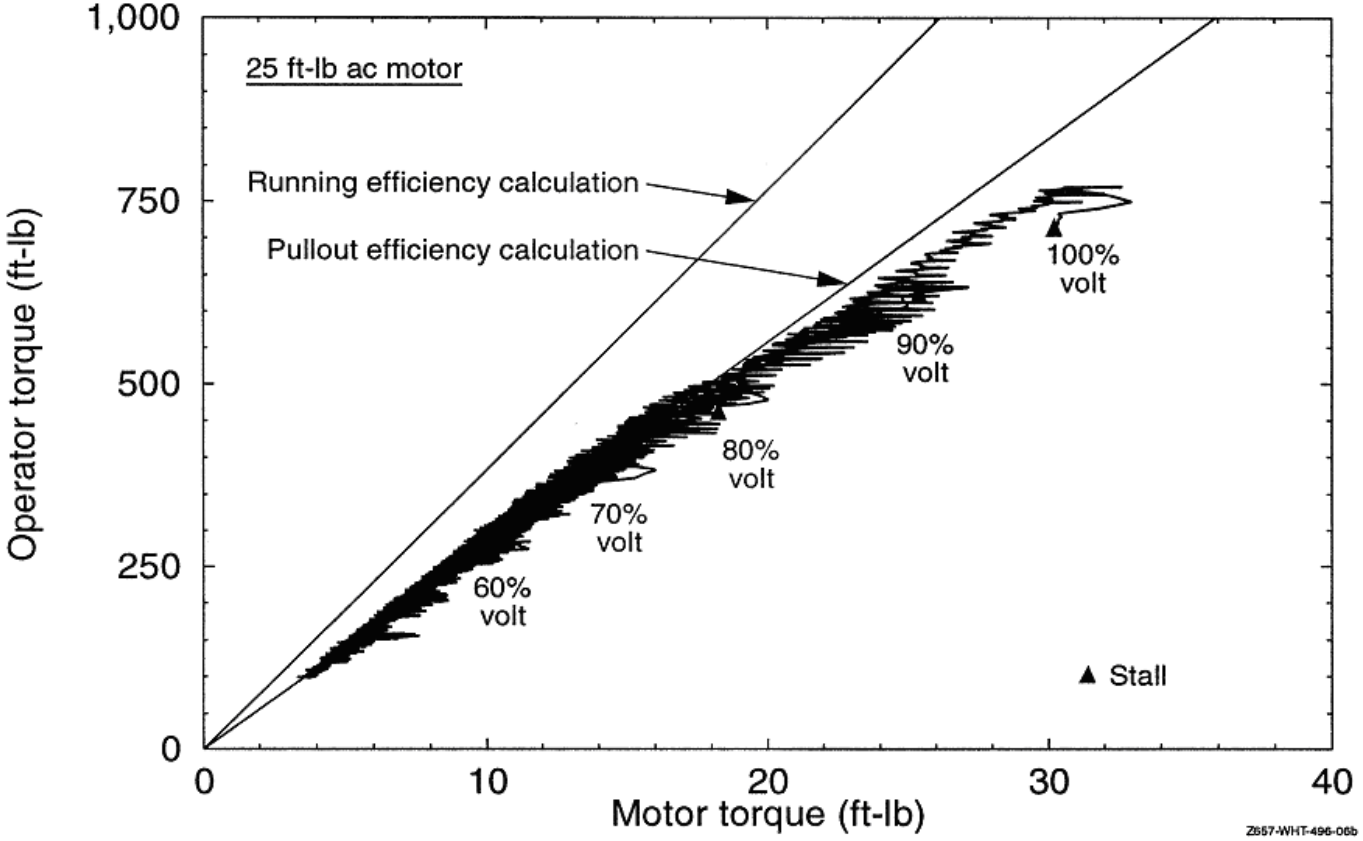
Actuator Torque vs Motor Torque



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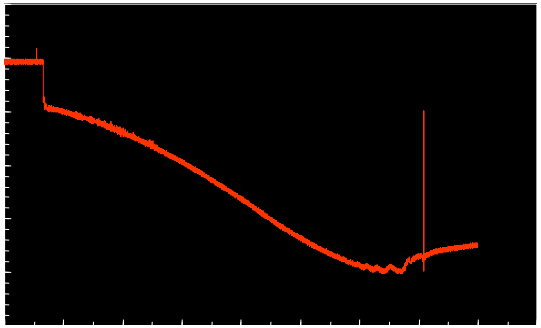
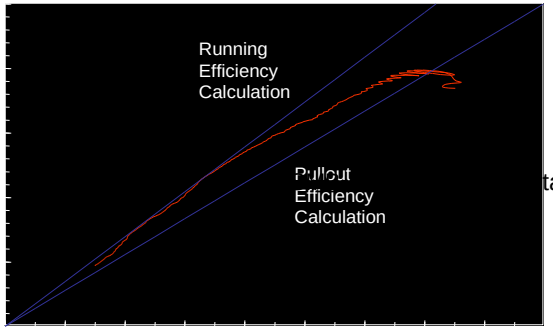
At higher loads the actual gearbox efficiency drops below pullout efficiency.

SMB-0-10 dc (100% voltage)





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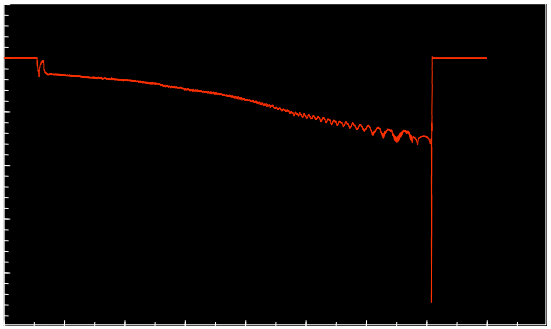

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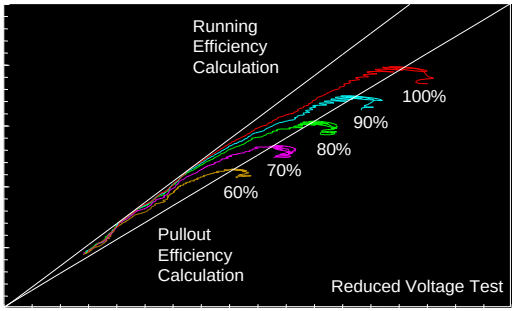
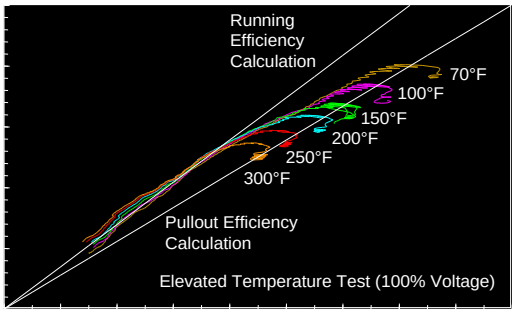
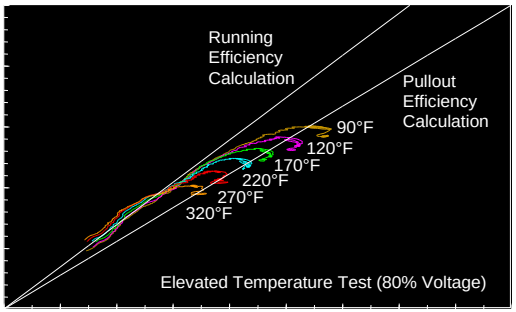
As the motor speed decreases, the efficiency of the gearbox decreases.







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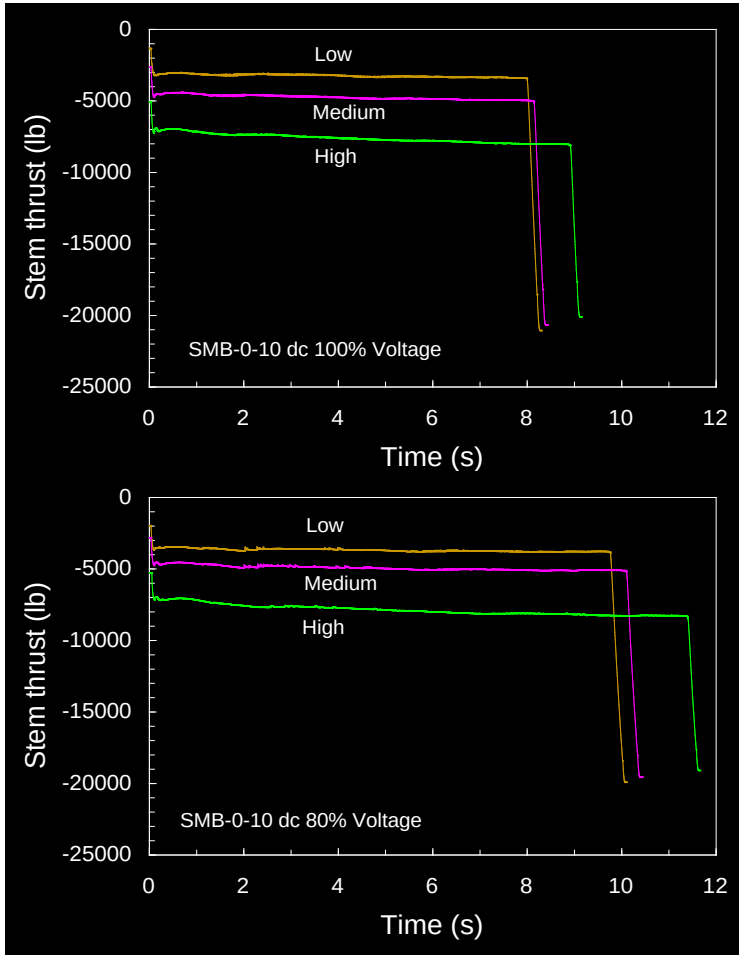
Effects of Load on Stroke Time and Motor Heating

- *In these tests, the MOVLS was set up to create a fairly constant load for the entire stroke until the hydraulic cylinder bottomed out, simulating valve wedging.*
- *Three tests were performed with loads nominally designated low, medium, and high at 100% voltage.*
- *Three tests were performed with the same loads at 80% voltage.*

Changes in running load and voltage can change stroke time significantly.



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