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# Precision Planetary Servo Gearheads

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American Gear Manufacturers Association

TECHNICAL PAPER

# Precision Planetary Servo Gearheads

**Gerhard G. Antony, Neugart USA LP and Arthur Pantelides, mG miniGears North America**

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

## Abstract

Modern automated machineries are increasingly using flexible high dynamic servomotors because of their ability to speed up and flexibly automate complex motions these machineries need to perform. Planetary gearheads are used frequently in conjunction with servomotors to match the inertias, lower the motor speed, boost the torque, and at the same time provide a sturdy mechanical interface for pulleys, cams, drums and other mechanical components.

This paper addressing following topics:

- main reasons why the planetary (epicyclical gear systems) are the preferred choice for ”servo applications” (applications using servo motors);
- what influencing the positioning accuracy repeatability of a planetary servo gear;
- helical gears in planetary systems;
- the rating practices establishing a transparent ”comparability” of different torque listings;
- introduction of a simple reliable method to the required gearbox torque rating for a servo-application based on the selected motor toque data.

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# Precision Planetary Servo Gearheads

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Because of their versatility as well as their ability to speed-up and automate a wide-range of highly-complex motion sequence programs required in many of today's industries, computer-controlled, programmable, highly dynamic-capable servo motors are increasingly being used in modern machinery required in complex automation applications. Precision planetary gearheads are frequently used in conjunction with such servo motors in order to: balance inertial loading conditions seen during frequent speed cycling sequences, decrease motor speeds, and boost torque, while at the same time, provide a robust mechanical interface for pulleys, cams, drums, and other mechanical transmission components.

This paper shall present a foundation and fundamental approach for understanding why the planetary system is the preferred design choice for servo gearheads; clear up some misconceptions about planetary servo gearheads; compare rating practices by establishing a transparent comparability of different torque listings; and introduce a simple and reliable method of determining the required gearbox torque rating for a selected servo motor/gearbox application.

Main topics covered in the paper shall be:

- The planetary (epicyclic) gear system as the "*system of choice*" for servo gearheads;
- The best "balanced" planetary ratio from a torque density point of view;
- The gearhead design influence on positioning-accuracy and repeatability;
- Typical dynamic servo applications and servo gearhead torque ratings;
- How to establish comparative torque ratings;
- Sizing/selection of servo gearheads for matching AC servo motors in automation applications.

## 1. The Planetary (Epicyclical) Gear System as the "System of Choice" for Servo Gearheads

Frequent misconceptions regarding planetary gears systems involve backlash: *planetary systems are used for servo gearheads because of their inherent low backlash*; *low backlash is the main characteristic requirement for a servo gearboxes*; *backlash is a measure of the precision of the planetary gearbox*.

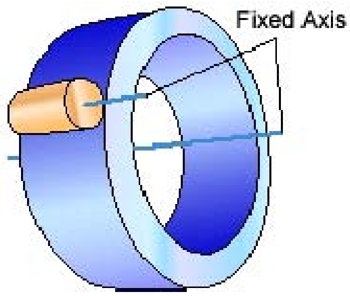
The fact is, fixed-axis, standard, "spur" gear arrangement systems can be designed and built just as easily for low backlash requirements. Furthermore, low backlash is not an absolute requirement for servo-based automation applications. A *moderately*-low backlash is advisable (in applications with very high start/stop, forward/reverse cycles) to avoid internal shock loads in the gear mesh. That said, with today's high-resolution motor-feedback devices and associated motion controllers it is easy to compensate for backlash anytime there is a change in the rotation or torque-load direction.

If, for the moment, we discount backlash, then what are the reasons for selecting a more expensive, seemingly more complex planetary systems for servo gearheads? What advantages do planetary gears offer?

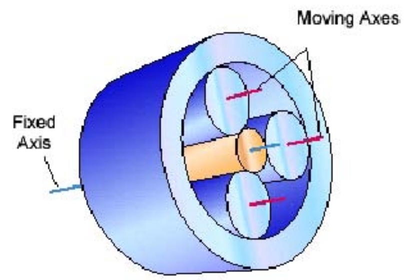
### High Torque Density - Compact Design

An important requirement for automation applications is high torque capability in a compact and light package. This high torque density requirement (a high torque/volume or torque/weight ratio) is important for automation applications with changing high dynamic loads in order to avoid additional system inertia.

Depending upon the number of planets, planetary systems distribute the transferred torque through multiple gear mesh points. This means a planetary gear with say three planets can transfer three times the torque of a similar sized fixed-axis "standard" spur gear system. Reference Figure 1.




Fixed axis “standard” gear




Planetary “Epicyclical” gear system

**Figure 1.**

### Rotational Stiffness/Elasticity

High rotational stiffness, or minimal elastic windup, is important for applications with elevated positioning accuracy and repeatability requirements; especially under fluctuating loading conditions. The load distribution unto multiple gear mesh points means the load is supported by *N* contacts (where *N* = number of planet gears) increasing the torsional stiffness of the gearbox by factor *N.* This means it considerably lowers the lost motion compared to a similar size standard gearbox; and this is what is desired.

### Low Inertia

Added inertia results in an additional torque/energy requirement for both acceleration and deceleration. The smaller gears in planetary system result in lower inertia. Compared to a same torque rating standard gearbox, it is a fair approximation to say that the planetary gearbox inertia is smaller by the square of the number of planets. Again, this advantage is rooted in the distribution or “*branching*” of the load into multiple gear mesh locations.

### High Speeds

Servomotors run at high rpm, hence a servo gearbox must also operate in a reliable manner at high input speeds. For servomotors 3,000 rpm is practically the standard and in fact speeds are constantly increasing in order to optimize more and more complex application requirements. Servomotors running at speeds in excess of 10,000 rpm are not uncommon. From a rating point of view with increased speed the power density of the motor increases proportionally without any real size increase of the motor or electronic drive. Thus Amp rating stays about the same while only the Voltage must be increased. An additional, important factor is in regards to lubrication and operating speed. Fixed-axis spur gears will exhibit lubrication “starvation” and quickly fail if running at high speeds because the lubricant is slung away. Only special means such as expensive pressurized forced lubrication systems can solve this problem. On the other hand, grease lubrication is impractical because of a its “tunneling effect,” in which the grease, over time, is pushed away and cannot flow back into the mesh.

In planetary systems the lubricant *cannot escape* — it is continuously redistributed, “*pushed and pulled*” or “*mixed*” into the gear contacts, ensuring safe lubrication practically in any mounting position and at any speed. Furthermore, planetary gearboxes can be grease lubricated. This feature is inherent in planetary gearing because of the relative motion between the different gears making up the arrangement.

## 2. The Best “Balanced” Planetary Ratio from a Torque Density Point of View

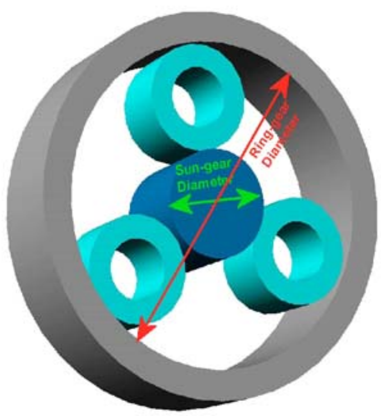
For easier computation it is preferred that the planetary gearbox ratio is an exact integer (3, 4, 6 ...). Since we are so used to the decimal system, we tend to use 10:1 even though this has no practical advantaged for the computer/servo/motion controller. Actually, as we will see, 10:1 or higher ratios are the weakest, using the least “balanced” size gears and hence have the lowest torque rating.

This paper addresses “simple planetary” gear arrangements, meaning all gears are engaging in the same plane. The vast majority of the epicyclical gears used in servo applications are of this “simple planetary” design. Figure 2a illustrates a cross-section of such a planetary gear arrangement with its central sun gear, multiple planets (3), and the ring-gear. The definition of the ratio of a planetary gearbox shown in the figure is obtained directly from the unique kinematics of the system. It is obvious that a 2:1 ratio is not possible in a simple planetary gear system, since to satisfy the above equation for a ratio of 2:1 the sun gear would need to have the same diameter as the ring-gear. Figure 2b shows the sun gear size for different ratios. With increased ratio the sun gear diameter (size) is decreasing.

Since gear size effects loadability the ratio is a strong and direct influence factor for the torque rating. Figure 3a below shows the gears in a 3:1, 4:1, and 10:1 simple system. At 3:1 ratio, the sun gear is large and the planets are small. The planets are becoming "thin walled" thus limiting the space for the planet bearings and carrier pins, hence limiting the loadability. The 4:1 ratio is an well-balance ratio, with sun and planets having the same size. 5:1 and 6:1 ratios still yield fairly good balanced gear sizes between planets and sun. With higher ratios approaching 10:1, the small sun-gear becomes a strong limiting factor for the transferable torque. Simple planetary designs with 10:1 ratios have very small sun-gears, which sharply limits torque rating.

Adding more planets can increase the torque density of the arrangement. To this effect we see that with lower ratios additional planet gears can be used; but for higher ratios, such as 10:1, multiple gears beyond say 3 planets, would cause interference.

This is illustrated in Figure 3b.




$$\text{Ratio} = \frac{\text{Ring gear Diameter}}{\text{Sun gear Diameter}} + 1$$

$$\text{Ratio} = \frac{\text{Ring gear Number of Teeth}}{\text{Sun gear Number of Teeth}} + 1$$

**Figure 2a. Definition of the (Reduction) Ratio for a simple planetary gear arrangement having a stationary ring gear. The input is at the sun-gear and output at the planet carrier shaft.**

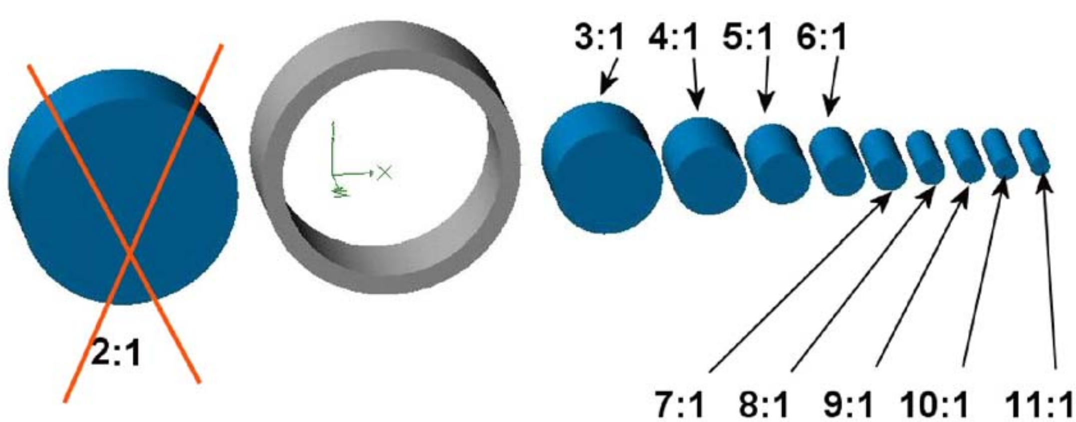



**Figure 2b. Sun gear size for different ratios**

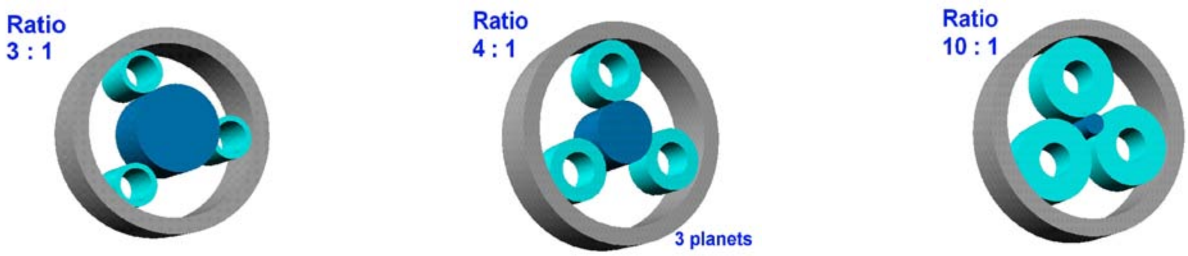
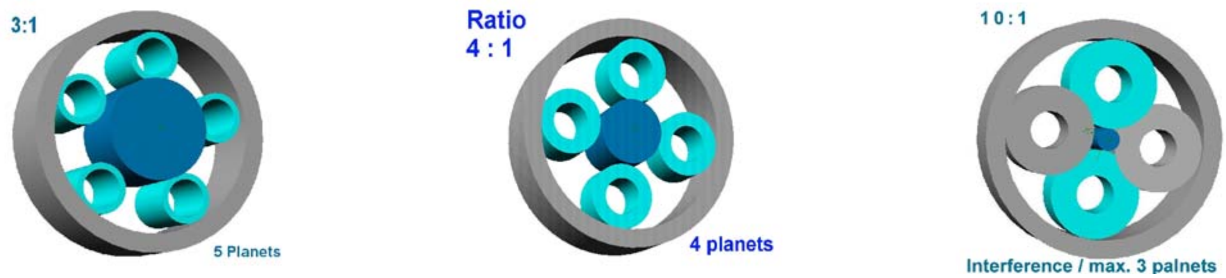



**Figure 3a. Planetary gear ratios and the relationship between sun/planet size**




Ratio 3:1 with 5 planets | Ratio 4:1 with 4 planets | Ratio 10:1 max. 3 planets possible

**Figure 3b. 10:1 ratios should be avoided unless absolutely necessary from a technical point of view. If such ratios are used additional consideration must be given to arrangement/size vs. rating**

## 3. How Positioning-Accuracy & Repeatability is Effected by the Precision and Quality Class of the Servo Gearhead

As previously mentioned, it is a general misconception that the backlash of a gearbox is a measure of the quality or precision. The fact is that the backlash has practically nothing to do with the quality or precision of a gear. Only the ***consistency*** of the backlash can be considered, up to certain degree, a form of measure of gear quality. From the application point of view the relevant question is "*what gear properties are influencing the precision of the motion?*"

**Positioning Accuracy** – is a measure of how exact a desired position is reached. In a closed loop system the prime determining/influencing factor of the positioning accuracy is the accuracy and resolution of the feedback device and where the position is measured. If the position is measured at the final output of the actuator the influence of the mechanical components can be practically eliminated. (Direct position measurement is used mainly in very high precision applications such as machine tools). In applications with a lower positioning accuracy requirement the feedback signal is generated by a feedback devise (resolver, encoder) in the motor. In this case auxiliary mechanical components attached to the motor such as a gearbox, couplings, pulleys, belts, etc., will influence the positioning accuracy.

**Influence of the Gearbox:**

**Gearbox Stiffness/Elasticity -** The elastic deformation or the "wind-up" of the components under load can significantly affect the positioning accuracy; and the positioning error is load-dependant since the wind-up of course depends on the load.

**Backlash -** the clearance between mechanical components - (such as the backlash of a gearbox) can contribute to the positioning error if the sense of the rotation or torque is changed during the positioning move. The overall rotational backlash of a gearbox is determined not only by the clearance between the gear teeth in mesh, it is influenced also by the other components of the gearbox such as the housing, bearings, shafts, and shaft/hub connection to name a few.

**Transmission Error – (TE)** can be also described as "the fluctuation of the theoretical reduction ratio" the output does not follow the input rotation exactly at the theoretical reduction ratio but fluctuates (+/-) a certain angle during the rotation, due to the inconsistencies of the gears (gear errors). These include pitch, lead, profile error, general eccentricity due to non-optimum positioning/placement, and others. The TE of a gear is directly dependent to the gear precision or gear class of the particular gear in question. And, just like the backlash, the overall gearbox TE is influenced by the other components of the gearbox.

**Example 1:**

Given:

- Gearhead PLS115
- 4:1 ratio
- worst case backlash 3 min
- rotational stiffness = 20 Nm/arcmin
- Torque rating 200 Nm
- Neugart 115 measured gearbox transmission error (TE) approximately +/- 1.25 arc min

1. What is the worst case positioning error from the TE?

   ➧ **1.25 arc min**

2. What is the worst case positioning error due to the Backlash in a motion cycle with motion direction reversal at rated torque load?

   ➧ **3 arc min**

3. What is the worst case positioning error due to the Stiffness or "wind-up" in a motion cycle with motion direction reversal at nominal torque load?

   ➧ 200 [Nm] x 2/20 [Nm/arc min] = **20 arc min**

**Gear Precision Class –** a number of geometrical measurements and associated tolerances and deviations for these determine gear precision class. Various national and international organizations have established standards which define various gear precision class levels; these include AGMA, ISO, DIN, JIS, and others. It is a frequent misconception that the low backlash is a prerequisite for high precision. The fact is that the gear precision class has little to no influence on the backlash. On the other hand it has a determining influence on the TE as indicated in the example above. Also, as shown, the stiffness can have considerably higher influence on the positioning error as opposed to either the Backlash or the TE, again shown above.

**Repeatability –** a measure of how exact a certain position is reached when a positioning motion cycle is repeated a number of times.

**Influence of the Gearbox Stiffness –** if the repeated motion cycle is performed with different loads the stiffness of the system has a significant influence on the repeatability.

**Influence of the Gearbox Backlash –** If the motion cycle is exactly repeated the backlash of the gearbox has theoretically ***NO*** influence on the repeatability; not even at fluctuating loads.

## 4. Torque Rating of Servo Gearheads in Automation, Motion Control, and Robotics Applications

The basic limiting factor for electrical devices is the temperature; more specifically, the instantaneous, or gradual breakdown of the insulation of the device due to temperature, ultimately resulting in a failure condition. Other than the strength of the magnets used in a particular motor design, or the currents, and demagnetization characteristics, the torque rating of an AC servo motor is mainly determined by its "thermal loadability." In a motor the generated heat is proportional to $R \times I^2 \times t$ and the rating limitation is the RMS value of the current (I) and its duration (t). Therefore RMS Torque is proportional to the current.

On the other hand from a mechanical point of view, the basic limiting factors of mechanical devices are the mechanical stresses, tension, compression, bending, shear, and Hertzian Pressure. The thermal loading is secondary.

In virtually all automation applications, frequently changing loads created by duty cycles with multiple starts/stops and accelerations/decelerations are present and very common in the servo drives associated with these applications. Even if the external loads are constant, all major components of a gearhead are subjected to cyclic mechanical stresses.

As an example, we calculate the number of peak load cycles being subjected to the sun gear in a moderate/low cycle automation application. In the case of a 5:1 ratio planetary gearbox, with 3 planets, driven at 3000 rpm input, 2 peak load cycles of 1 sec duration, the sun-gear must withstand: **$2.304 \times 10^6$ peak load cycles in 10 days of operation!**

See Figure 4/Example 2. The stress cycle rate on the planetary gears are lower, but they are subjected to reverse bending which fatigues the material even more.

**Example 2:**

Moderate/Low Cycle Rate Automation Application

- 2 load cycles per minute (acceleration/deceleration)
- 16 hours/day operation
- 1 second (1/60 min) - peak load duration/cycle
- 3000 rpm motor speed during the peak load period
- Unit characteristics: ratio 5:1, with 3 planet gears

Number of load cycles on one sun gear tooth after 10 days of operation:

**3** gear planets
**x** [ 3000 rpms – (3000/5) relative rpms ]
**x 1/60** (rps/rpm)
**x 1** sec (peak load duration)
**x 2** (peak load cycles/min)
**x 60** (min/hr)
**x 16** (hours per day operation) **x 10** (days)
= 2,304,000 or $2.304 \times 10^6$
Peak loads on each sun-gear tooth every 10 days

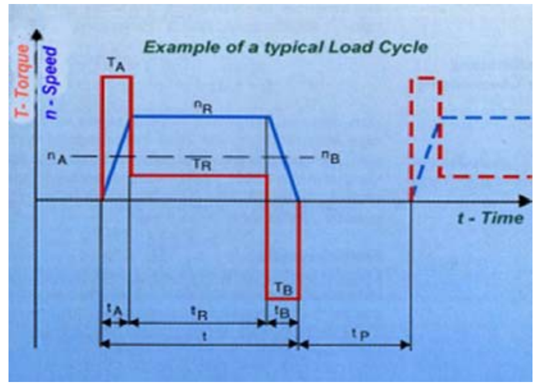



**Figure 4/Example 2.**

## Fatigue Behavior of Components Made of Ferrous Materials

Fatigue phenomena of metallic ferrous components is well known. Parts subjected to cyclic loads will fail after a certain finite number of load cycles even though the magnitude of the stress load is considerably lower than the static strength, which the part can endure without any damage. If the magnitude of the cyclic stress load is decreased, one can observe that at a certain stress level the part can endure unlimited load cycles. This stress level is called the ***"endurance limit."*** This behavior is well represented graphically by S-N Curves ( ***S***tress over the ***N***umber of Cycles ). We can conclude, components loaded at or below the endurance limit will endure virtually unlimited load cycles.

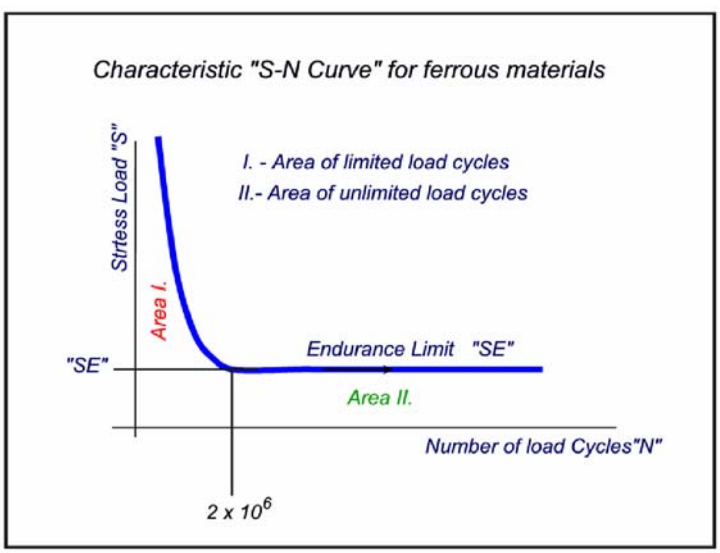



**Figure 5.**

Countless tests have confirmed that for ferrous materials the progressive exponential relationship between the stress **S** and the Number of endured load cycles **N** without damage levels-off into the "***endurance limit"*** (the constant horizontal part of the S-N curve) at about $2x10^6 \sim 2x10^7$ cycles. This is valid if the part is subjected to bending, shear, tension, or compression. Based on this we can distinguish 2 characteristic areas of the S-N relationship, namely:

I. The "***area of limited load cycles"*** (limited life) – essentially the exponential relationship between S and N. This area is mathematically described as:

   $S = 1/N^{1/E} \Rightarrow S = N^{-1/E}$ or $N = 1/S^E$

   In its logarithmic form:
   $\log S = -\ 1/E \times \log N$ or $\log N = -E \times \log S$

II. The "***area of unlimited load cycles"*** - the horizontal portion of the S-N curve.

In the above equations "E" is an empirically-determined factor which relates material properties, heat treat processes, and type of loading. Graphically it represents the slope of the line defining Area I. It should be expected for E to have a wide range of values since there is a wide range of materials being heat treated under various processes. Normally $E = 6 \sim 80$. Note that if we plot on a logarithmic scale, E demonstrates a linear relationship. See Figure 6.

The above relationships allow us to calculate life expectancy at a certain stress load level. If we expand to estimate cumulative impact under a range or set of different stress levels and frequency of stress level occurrence, that is, the repetition of various loadings, the analysis could then be done using various "Damage Accumulation" calculation methods which combine individual analysis into a comprehensive estimation. One such well-known method is "Miner's rule."

## 5. Establishing The Torque Rating of a Gearbox

The majority of real-world applications subject gear components to a significantly higher cycle loading than just $2 \times 10^6$ load cycles. Therefore, the recommendation of practically all gear rating standards ( AGMA, ISO, DIN, etc. ) is to determine and list the torque rating of a gearbox based on the endurance limit and on a certain minimum bearing life; for example for industrial gear boxes AGMA recommends 5,000 ~ 10,000 hours.

From a purely technical point of view, there is only one "*true rated-torque*" for a gearbox and that is, the rated-torque under **continuous duty** conditions. Unfortunately, for various non-technical reasons, often times additional ill-defined torques such as acceleration torque, peak torque, emergency-stop torque, etc., are listed and taken under consideration but without reference to the number of load cycles.

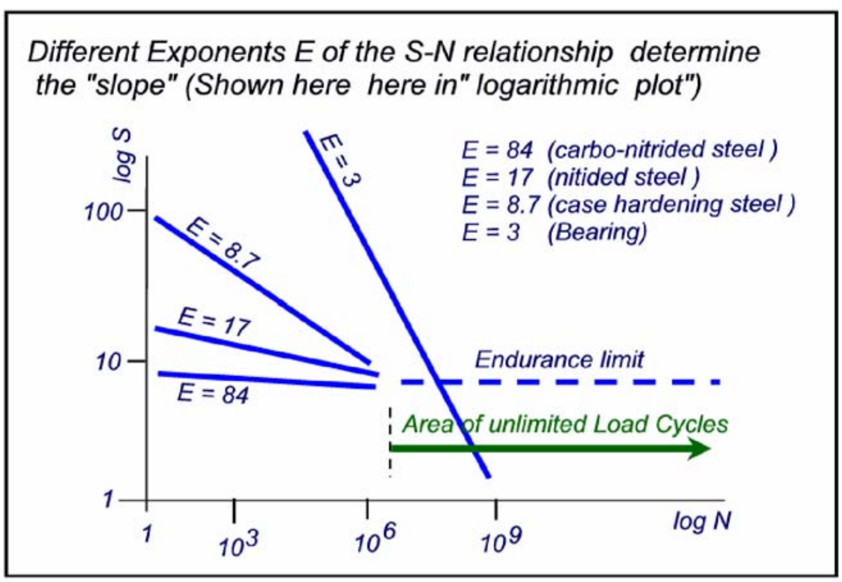



**Figure 6.**

It is possible to establish a comparison to the "*artificial*" torque ratings listed for a certain gearbox's true rated-torque, based on the endurance limit, using the similar basic concept used in Miner's rule and S-N curves.

Let us consider the following relationship for 2 gearboxes:

$$(T_{NG2}/T_{AG2})^{E2} = (T_{NG1}/T_{AG1})^{E1}$$

where:

$T_{NG1}$ = true rated torque (based on endurance limit) of gearbox 1

$T_{NG2}$ = true rated torque (based on endurance limit) of gearbox 2

$T_{AG1}$ = artificial torque rating of gearbox 1

$T_{AG2}$ = artificial torque rating of gearbox 1

E1, E2 = exponents/S-N slopes for gearbox 1 and 2 respectively

Using the above, an equivalent to artificial torque rating of a gearbox 2 can be determined using the following:

$$T_{AG2} = (T_{AG1}/T_{NG1})^{E1/E2} \times T_{NG2}$$

The fact remains that only in applications where the number of load cycles is below $2x10^6$ are higher loads than the rated load permissible. However, the majority of real world automation applications reach this number of load cycles in just a few days, weeks, or at best, after some months of operation. Therefore for a safe servo gearhead selection the rule must be:

> *If the peak load cycle is part of the standard (designed) working duty cycle of the machinery, the peak load should not be higher than the rated torque unless the machinery is only working a very limited time "say an hour a day" or if the user and OEM does not expect an extended long maintenance-free life from the machinery. To predict the life in such applications a detailed analysis of the load spectrum is required.*

The familiar and frequently used method based on the "Root Mean Cube" (RMC) torque value,

$$T_{RMC} = \left[\frac{\left(N_1 t_1 T_1^3 + N_2 t_2 T_2^3 + \cdots + N_i t_i T_i^3\right)}{\left(N_1 t_1 + N_2 t_2 + \cdots + N_i t_i\right)}\right]^{1/3}$$

is strictly only applicable to the bearings and not the other vital components of the gearbox such as the gear teeth or shafts. A correct calculation requires the application of unique exponents, for the individual components, made of different materials, with different heat treatment, and loading conditions, as shown in previous Figure 6. ***Thus we desire to establish a torque rating based on a similar method/algorithm as that of Miner's rule***.

## 6. A Reliable and Realistic Selection Method for Gearheads for Applications with Repeated Dynamic Loading

Automation, motion control and positioning applications are characterized by repeated acceleration and deceleration cycles associated with frequent starts/stops and direction reversals. The characteristic load cycle of an application usually consist of a number of load peaks of different intensity and duration. The exact load cycle of a machine is not easy to predict; however, it can be measured and a statistically representative load characteristic cycle can be generated.

In most cases, the exact load cycle is not known and the task is to size/select a gearbox for a given servomotor. With basic load related data, namely the load inertia, the magnitude of the maximum non-dynamic load (such as friction load) and the available motor data a reliable and realistic estimation of the required gearbox torque rating is possible.

For a given servomotor, the maximum dynamic (acceleration) peak torque is known from the motor data sheet. To assume that the gearbox is exposed to the maximum motor peak torque can easily lead to over-sizing and associated cost increases. In the vast majority of real applications a substantial portion of the motor peak acceleration torque is not transmitted through the gearbox because a significant part of it is "consumed" in order to accelerate the motor rotor i.e. only a part of the motor peak torque will "travel" through the gearbox.

### The Torque Magnitude Through the Gearbox in a System with Cyclic Dynamic Loading, Such as Start/Stop.

We shall now derive the general governing equation, which will allow us to determine the magnitude of the peak torque of a gearbox. Knowing the magnitude of the peak torque traveling through the gearbox there is a straight forward method which can be used to easily select an appropriate gearbox size.

Consider the system consisting of Motor, Gearbox and Load, shown in Figure 7. All system parameters are reflected to the motor axis i.e., "$J_{LR} = J_L/i^2$" and $T_{LR} = T_{Ld}/i$ with "$i$" being the reduction Ratio, $J_L$ the load inertia at the output, and $T_L$ the true torque at the output. The $T_{LR}$ load torque is the friction torque and/or a gravity-born torque present in the system under steady state condition as well as during the acceleration phases of the motion cycle. $T_L$ represents the ***non-inertial*** resistance forces the drive has to overcome.

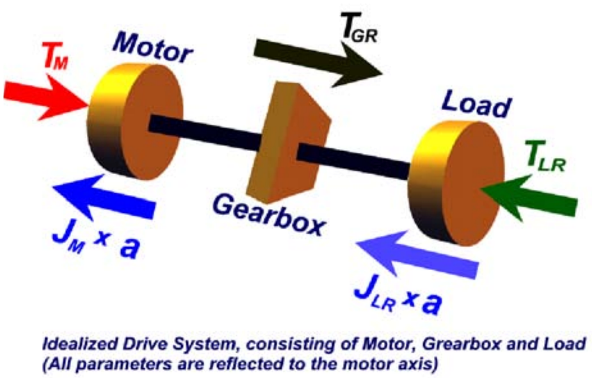



**Figure 7.**

The torque equilibrium equation during the acceleration, with the maximum available motor peak torque "$T_M$", motor rotor inertia "$J_M$", and "$a$" as the angular acceleration, can be written as:

$$T_M - T_{LR} - J_M \times a - J_{LR} \times a = 0 \qquad (1)$$

Solving for angular acceleration:

$$a = \frac{(T_M - T_{LR})}{(J_M + J_{LR})} \qquad (1a)$$

The torque traveling through the gearbox is:

$$T_{GR} = T_M - J_M \times a \qquad (2)$$

Similarly we have:

$$T_{GR} = T_{LR} + J_{LR} \times a \qquad (2a)$$

Substituting Equation 1a, into Equation 2, for the acceleration, we can derive the following equation:

$$T_{GR} = T_M \times \left( \frac{1 - J_M}{(J_{LR} + J_M)} \right) + \frac{T_{LR} \times J_M}{(J_{LR} + J_M)}$$

Introducing the "*inertia parameter*" $k = J_M/(J_{LR}+J_M)$, which is a function of the system inertias only, the torque through the gearbox can be written in the form:

$$T_{GR} = (T_M - T_{LR}) \times (1 - k) + T_{LR}$$

This is an easy to use formula and one which is valid for all motor inertial and frictional torque combinations.

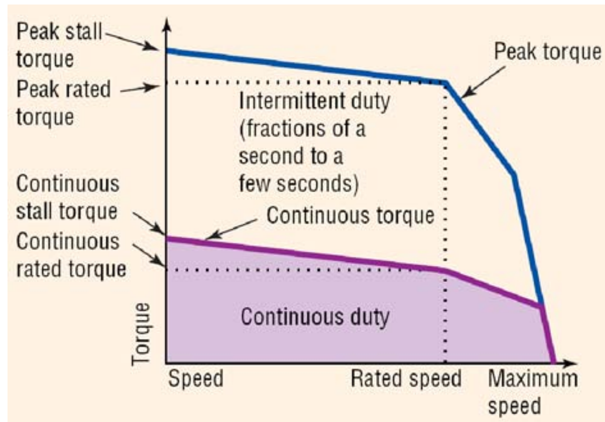



**Figure 8. Typical characteristic torque/speed curve of an AC servo motor**

To have a long, maintenance-free gearbox life, the calculated gearbox load "$T_{GR} \times i$" should be equal to or less than the gearbox torque rating. The inertia parameter defined above, $k = J_M/(J_{LR} + J_M)$, is closely related to the ***Inertia Match Ratio*** which is Load Inertia/Motor Inertia, $IMR = J_{LR}/J_M$.

With some substitutions above we have: $k = 1/(IMR+1)$. With the IMR being a widely- used basic characteristic of a servo-driven system. It shows how well the system is balanced from the control as well as the overall economic point of view. Systems with "*balanced*" inertias, meaning IMR of approximately ≈ 1:1, have good responsive control characteristics.

Consider a servo system with matched inertias of 1:1. If we assume we have only dynamic inertial loads, i.e. $(T_{LR}=0)$, with $IMR = J_{LR}/J_M = 1$ the inertia parameter k becomes:

$$k = \frac{1}{(1 + 1)} = 0.5$$

which will result in:

$$T_{GR} = T_M \times 0.5$$

This means, in a system with 1:1 inertia ratio, 50% of the motor peak torque will be "consumed" or expended to accelerate the motor rotor and only the other 50% of it will "travel through" the gearbox to accelerate the load. Using the above equation, an appropriate gearbox can be selected based on this calculated maximum output torque requirement of **"$T_{GR}$ x i"** instead of selecting one based on the full peak motor torque, i.e. based on **"$T_M$ x i"**.

Based on the proposed method developed here, one can visualize the torque through the gearhead in terms of percent of the motor peak torque. In Figure 9 the torque through the gearbox is depicted as a function of IMR (Inertia Match Ratio). The non–dynamic torques such as friction torque, again, relative to the motor peak torque, is used as a parameter.

### High Cycle Rate Applications

The peak torque of the servo motor can be utilized only within very limited time parameters in the motion cycle, i.e. in high cycle rate applications, motor full peak torque is not available for a rapid sequence of accelerations/decelerations, and the available motor torque is considerably less than the peak torque, if not, the motor can burn out.

For high cycle rate applications repeating many times in a minute or even multiple times in a second, it is practically a must to have a low IMR close to, or even less than unity, IMR ≦ 1.

On the other hand, low cycle rate applications certainly can utilize the peak torque since it does not occur with high frequency over the motion cycle, here we can have a larger inertial mismatch which means high torques will be transmitted through the gearbox.

Furthermore, contrary to the common belief that high cyclic operations are tough on gearheads, high cyclic automation application, with <u>matched inertias</u> using the methods outlined in this paper, in fact can utilize lower torque rating (smaller size) gearheads; compared to low cycle rate operation, with <u>high inertial mismatch</u>, where a gearbox with larger torque rating is needed .

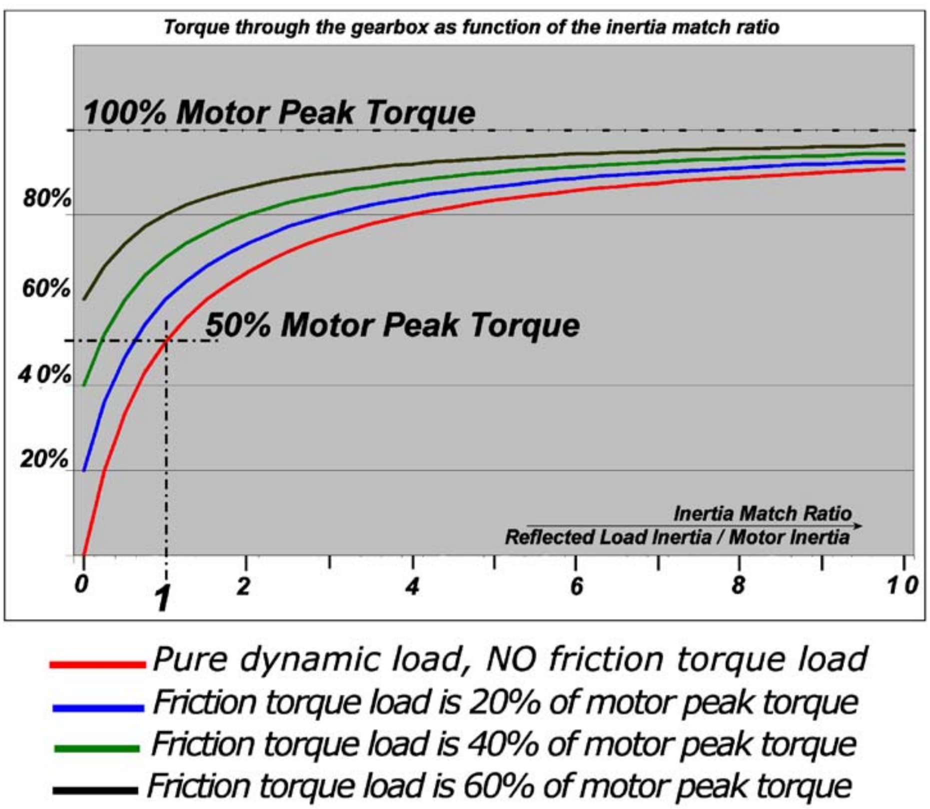



**Figure 9**

The described selection method is generally applicable and can be expanded for use with all gearboxes and gear motors as long as the gearbox torque rating is a "true rating" and the application does not have frequent external heavy shock loading, such as sudden jam or frequent emergency stops.

## Conclusion

This paper presented a general foundation for the further approach and understanding of planetary gearing and why such type of gearing system is the preferred design choice for servo gearheads. This was done within the context and discussion of torque density, rotational stiffness, inertia, speed, positioning accuracy, and other important factors and requirements in demanding applications such as automation, motion control and robotics.

The paper also attempted to clear up some misconceptions about planetary servo gearheads; it laid the groundwork for a unique rating method by starting with a discussion of fatigue behavior, and the expansion of well known concepts into a rating method of a system with cyclic dynamic loading, with the concept of inertial matching or balancing in order to optimize planetary gearhead/servo motor selection.

It is the intention of the author to stimulate further research, thought, and open discussion on the potential use, further rating refinement, and general optimization of precision planetary servo gearheads in various demanding applications for which they are best suited.

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